APPLIED PROCESS DESIGN

FOR CHEMICAL AND PETROCHEMICAL PLANTS

Volume 1, Third Edition

Emphasizes how to apply techniques of process design and interpret results into mechanical equipment details

Ernest E. Ludwig

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Volume 1:

1. Process Planning, Scheduling, Flowsheet Design
2. Fluid Flow
3. Pumping of Liquids
4. Mechanical Separations
5. Mixing of Liquids
6. Ejectors
7. Process Safety and Pressure-Relieving Devices
   Appendix of Conversion Factors

Volume 2:

8. Distillation
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Volume 3:

10. Heat Transfer
11. Refrigeration Systems
12. Compression Equipment
13. Compression Surge Drums
14. Mechanical Drivers

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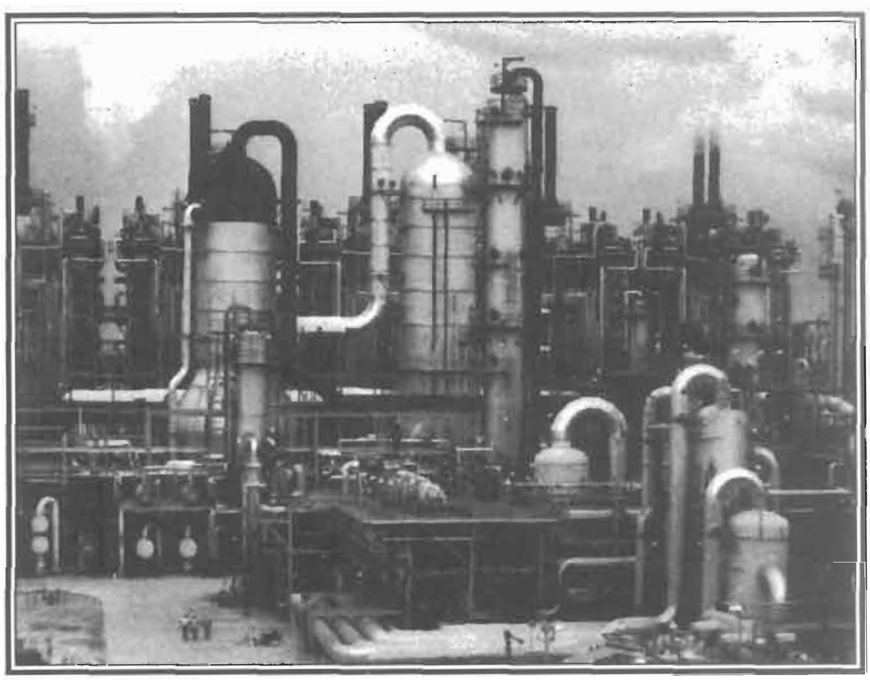

Ernest E. Ludwig

To my wife, Sue, for her
patient encouragement and help

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Contents

Preface to the Third Edition

This volume of *Applied Process Design* is intended to be a chemical engineering process design manual of methods and proven fundamentals with supplemental mechanical and related data and charts (some in the expanded Appendix). It will assist the engineer in examining and analyzing a problem and finding a design method and mechanical specifications to secure the proper mechanical hardware to accomplish a particular process objective. An expanded chapter on safety requirements for chemical plants and equipment design and application stresses the applicable Codes, design methods, and the sources of important new data.

This manual is not intended to be a handbook filled with equations and various data with no explanation of application. Rather, it is a guide for the engineer in applying chemical processes to the properly detailed hardware (equipment), because without properly sized and internally detailed hardware, the process very likely will not accomplish its unique objective. This book does not develop or derive theoretical equations; instead, it provides direct application of sound theory to applied equations useful in the immediate design effort. Most of the recommended equations have been used in actual plant equipment design and are considered to be some of the most reasonable available (excluding proprietary data and design methods) that can be handled by both the inexperienced as well as the experienced engineer. A conscious effort has been made to offer guidelines of judgment, decisions, and selections, and some of this will also be found in the illustrative problems. My experience has shown that this approach at presentation of design information serves well for troubleshooting plant operation problems and equipment/systems performance analysis. This book also can serve as a classroom text for senior and graduate level chemical plant design courses at the university level.

The text material assumes that the reader is an undergraduate engineer with one or two years of engineering fundamentals or a graduate engineer with a sound knowledge of the fundamentals of the profession. This book will provide the reader with design techniques to actually design as well as mechanically detail and specify. It is the author's philosophy that the process engineer has not adequately performed his or her function unless the results of a process calculation for equipment are specified in terms of something that can be economically built or selected from the special designs of manufacturers and can by visual or mental techniques be *mechanically* interpreted to actually perform the process function for which it was designed. Considerable emphasis in this book is placed on the mechanical Codes and some of the requirements that can be so important in the specifications as well as the actual specific design details. Many of the mechanical and metallurgical specifics that are important to good design practice are not usually found in standard mechanical engineering texts.

The chapters are developed by *design function* and not in accordance with previously suggested standards for unit operations. In fact, some of the chapters use the same principles, but require different interpretations that take into account the *process* and the *function* the equipment performs in the process.

Because of the magnitude of the task of preparing the material for this new edition in proper detail, it has been necessary to omit several important topics that were covered in the previous edition. Topics such as corrosion and metallurgy, cost estimating, and economics are now left to the more specialized works of several fine authors. The topic of static electricity, however, is treated in the chapter on process safety, and the topic of mechanical drivers, which includes electric motors, is covered in a separate chapter because many specific items of process equipment require some type of electrical or mechanical driver. Even though some topics cannot be covered here, the author hopes that the designer will find design techniques adaptable to 75 percent to 85+ percent of required applications and problems.

The techniques of applied chemical plant process design continue to improve as the science of chemical engineering develops new and better interpretations of the fundamentals of chemistry, physics, metallurgical, mechanical, and polymer/plastic sciences. Accordingly, this third edition presents additional reliable design methods based on proven techniques developed by individuals and groups considered competent in their subjects and who are supported by pertinent data. Since the first and second editions, much progress has been made in standardizing (which implies a certain amount of improvement) the hardware components that are used in designing process equipment. Much of the important and basic standardization has been incorporated in this latest edition. Every chapter has been expanded and updated with new material.

All of the chapters have been carefully reviewed and older (not necessarily obsolete) material removed and replaced by newer design techniques. It is important to appreciate that not all of the material has been replaced because much of the so-called "older" material is still the best there is today, and still yields good designs. Additional charts and tables have been included to aid in the design methods or explaining the design techniques.

The author is indebted to the many industrial firms that have so generously made available certain valuable design data and information. Thus, credit is acknowledged at the appropriate locations in the text, except for the few cases where a specific request was made to omit this credit.

The author was encouraged to undertake this work by Dr. James Villbrandt and the late Dr. W. A. Cunningham and Dr. John J. McKetta. The latter two as well as the late Dr. K. A. Kobe offered many suggestions to help establish the usefulness of the material to the broadest group of engineers and as a teaching text.

In addition, the author is deeply appreciative of the courtesy of The Dow Chemical Co. for the use of certain noncredited materials and their release for publication. In this regard, particular thanks is given to the late N. D. Griswold and Mr. J. E. Ross. The valuable contribution of associates in checking material and making suggestions is gratefully acknowledged to H. F. Hasenbeck, L. T. McBeth, E. R. Ketchum, J. D. Hajek, W. J. Evers, and D. A. Gibson. The courtesy of the Rexall Chemical Co. to encourage completion of the work is also gratefully appreciated.

Ernest E. Ludwig, P.E.

Chapter 1

Process Planning, Scheduling and Flowsheet Design

Process engineering design is the application of chemical, mechanical, petroleum, gas and other engineering talents to the process-related development, planning, designs and decisions required for economical and effective completion of a process project [7]. Although process design engineers are organizationally located in research, technical service, economic evaluation, as well as other specific departments, the usual arrangement is to have them available to the engineering groups concerned with developing the engineering details of a project. This is in order to provide process details as well as to evaluate bids for the various items of equipment. Process design is usually a much more specific group responsibility in engineering contractor organizations than in a chemical or petrochemical production company, and the degree of distinction varies with the size of the organization.

The average process engineer has the following responsibilities:

1. Prepares studies of process cycles and systems for various product production or improvements or changes in existing production units; prepares material and heat balances.
2. Prepares economic studies associated with process performance.
3. Designs and/or specifies items of equipment required to define the process flowsheet or flow system; specifies corrosion resistant materials of construction.
4. Evaluates competitive bids for equipment.
5. Evaluates operating data for existing or test equipment.
6. Guides flowsheet draftsmen in detailed flowsheet preparation.

The process engineer also develops tests and interprets data and information from the research pilot plant. He aids in scaling-up the research type flow cycle to one of commercial feasibility.

The process engineer must understand the interrelationship between the various research, engineering, purchasing, expediting, construction and operational functions of a project. He must appreciate that each function may and often does affect or influence the process design decisions. For example, it is foolish to waste time designing or calculating in detail, when the basic components of the design cannot be economically fabricated, or if capable of being fabricated, cannot possibly be delivered by the construction schedule for the project. Some specific phases of a project that require process understanding include plant layout, materials of construction for corrosion as well as strength, start-up operations, trouble-shooting, maintenance, performance testing and the like.

Organizational Structure

The process design function may be placed in any one of several workable locations in an organization. These locations will be influenced by the primary function of the overall company, i.e., chemical production, engineering, engineering sales, design and manufacture of packaged or specific equipment manufacture, etc. For best efficiency, regardless of the business nature of the company, the process design being a specialty type operation, works best when specifically identified and given the necessary freedom of contact within and without the company to maintain a high level of practical, yet thorough direction.

A typical working arrangement is shown in Figure 1-1 [7].

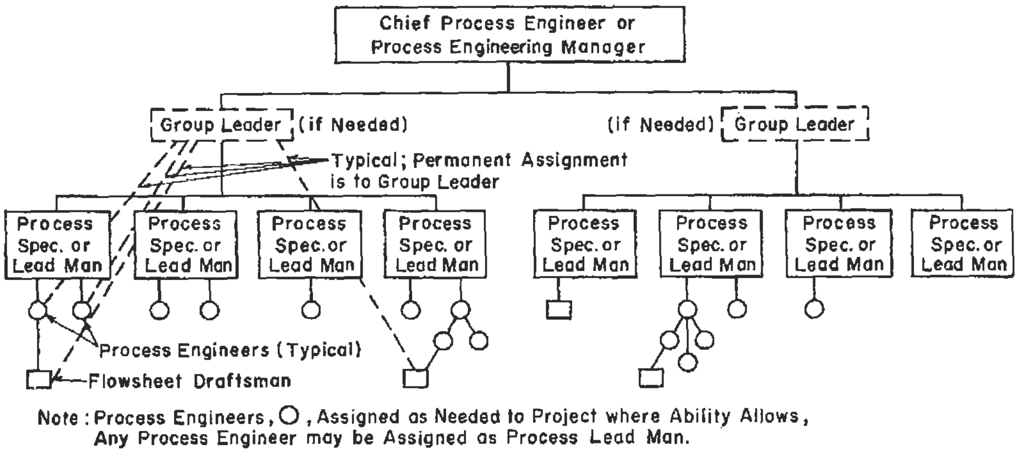


Figure 1-1. A process engineering section supervision chart. By permission, E. E. Ludwig [7].

In a consulting or engineering contractor organization, process design and/or process engineering is usually a separate group responsible for developing the process with the customer, or presenting the customer with a turnkey proposed process.

In an operating or producing chemical or petrochemical company the process engineering and design may be situated in a research, technical service, or engineering department. In most cases it is associated with an engineering department if new projects and processes are being planned for the company. If located elsewhere, the designs and planning must be closely coordinated with the engineering activity.

Most current thinking establishes a project team headed by a project engineer or manager to oversee the accomplishment of a given plant development for a process company. If the projects or jobs are small, then the scope of activity is limited and may often be consolidated in a single individual for project and process responsibility. For projects larger than $500,000, the project and process responsibility usually are best kept separate in order to expedite the specific accomplishment of the process design phase. When the process design engineer is required to interrupt calculations and specification development and to follow some electrical, structural or even expediting delivery question or problem, the design work cannot be completed at best efficiency and often the quality of process design suffers, assuming there is a fixed target date for completion of the various phases as well as the overall project.

Figure 1-2 diagrammatically suggests a team arrangement for accomplishing the planning of a process project. The arrows indicate directions of flow of communications and also the tie-in relationship of the process design function in the accomplishment of an assignment. The planning team in the box works to place the proper perspective on all phases of the engineering functions by developing a working atmosphere of understanding for accomplishing the engineering design. This is physically represented by mechanical vessels, piping, structures, electrical, instrumentation, civil and any other specialized functions. In many projects, the Lead Process Engineer and the Project Lead Engineer are the only individuals who see the details of the overall scope of the project.

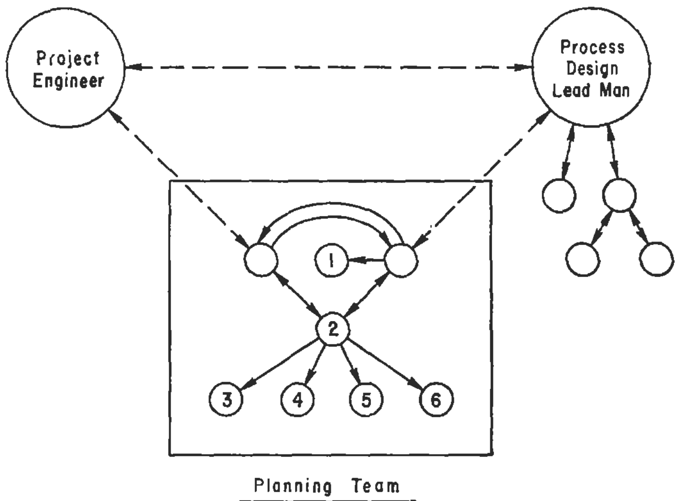


Figure 1-2. Typical organization of 'engineering planning team.' By permission, E. E. Ludwig [7].

Process Design Scope

The term *process design* is used here to include what is sometimes referred to as process engineering. Yet in some process engineering operations, all process design functions may not be carried out in detail. As discussed, process design is intended to include:

1. Process material and heat balances.
2. Process cycle development, correlation of pilot or research data, and correlation of physical data.
3. Auxiliary services material and heat balances.
4. Flowsheet development and detailed completion.
5. Chemical engineering performance design for specific items of equipment required for a flowsheet, and mechanical interpretation of this to a practical and reasonable specification. Here the process requirements are converted into hardware details to accomplish the process end results at each step in the product production process.
6. Instrumentation as related to process performance, presentation and interpretation of requirements to instrument specialists.
7. Process interpretation for proper mechanical, structural, civil, electrical, instrument, etc., handling of the respective individual phases of the project.
8. Preparation of specifications in proper form and/or detail for use by the project team as well as for the purchasing function.
9. Evaluation of bids and recommendation of qualified vendor.

Most of the functions are fairly self explanatory; therefore, emphasis will be placed only on those requiring detailed explanation.

Role of the Process Design Engineer

Although the working role of the process design engineer may include all of the technical requirements listed above, it is very important to recognize what this entails in some detail. The process design engineer, in addition to being capable of participating in evaluation of research and pilot plant data and the conversion of this data into a proposed commercial process scheme, must also:

1. Prepare heat and material balance studies for a proposed process, both "by hand" and by use of computer programs.
2. Prepare rough cost economics, including *preliminary* sizing and important details of equipment, factor to an order of magnitude capital cost estimate [34] (see also [19]), prepare a production cost estimate, and work with economic evaluation representatives to establish a payout and the financial economics of the proposed process.
3. Participate in layout planning for the proposed plant (see [46] [47]).
4. Prepare final detailed heat and material balances.
5. Prepare detailed sizing of all process equipment and possibly some utility systems. It is important that the process engineer *visualize* the flow and processing of the fluids through the system and *inside* the various items of equipment in order to adequately recognize what will take place during the process.
6. Prepare/supervise preparation of draft of process flowsheets for review by others.
7. Prepare/supervise preparation of piping or mechanical flow diagram (or P and ID), with necessary preliminary sizing of all pipe lines, distillation equipment, pumps, compressors, etc., and representation of all instrumentation for detailing by instrument engineers.
8. Prepare mechanical and process specifications for all equipment, tanks, pumps, compressors, separators, drying systems, refrigeration systems. This *must* include the selection of materials of construction and safety systems and the coordination of specifications with instrumentation and electrical requirements.
9. Determine size and specifications for all safety relief valves and/or rupture disks for process safety relief (including run-a-way reactions) and relief in case of external fire.
10. Prepare valve code specifications for incorporation on item 6 above, or select from existing company standards for the fluids and their operating conditions (see Figures 1-25 and 1-26).
11. Select from company insulation standards (or prepare, if necessary) the insulation codes to be applied to each hot or cold pipe or equipment. Note that insulation must be applied in some cases only to prevent operating personnel from contacting the base equipment. See Table 1-1 for typical insulation thickness from which code numbers can be established.
12. Establish field construction hydraulic test pressures for each process equipment. Sometimes the equipment is blanked or blocked off, and no test pressure is applied in the field, because all pressure equipment must be tested in the fabricators' or manufacturers' shop per ASME Code.
13. Prepare drafts of line schedule and/or summary sheets (Figures 1-24 A–D), and equipment summary schedules (Figures 1-27, 1-28, 1-29, 1-30), plus summary schedules for safety relief valves and rupture disks, compressors and other major equipment.
14. Prepare detailed process and mechanical specifications for developing proposals for purchase by the purchasing department.

The process design engineer actually interprets the process into appropriate hardware (equipment) to accomplish the process requirements. Therefore, the

Table 1-1
Typical Thickness Chart—Insulation for Services 70°F through 1200°F Piping, Vessels & Equipment 36" Diameter & Smaller

Pipe size	Insulation Thickness 1"	1½"	2"	2½"	3"
2½" & Smaller	700°F	1000°F	1200°F		
3"	700	900	1100	1200°F	
4"	700	900	1100	1200	
6"	600	800	1000	1200	
8"	—	800	1000	1200	
10"	—	800	1000	1200	
12"	—	800	1000	1200	
14"	—	800	1000	1100	1200°F
16"	—	800	900	1100	1200
18"	—	800	900	1100	1200
20"	—	800	900	1100	1200
24"	—	800	900	1100	1200
30"	—	800	900	1100	1200
36"	—	800	900	1000	1200

Temperatures in chart are maximum operating temperatures in degrees Fahrenheit for given thickness.
Note: All hot insulated piping shall be coded, including piping insulated for personnel protection. Thickness is a function of insulation composition.

engineer must be interested in and conversant with the layout of the plant; the relationship of equipment for maintenance; the safety relationships of equipment in the plant; the possibilities for fire and/or explosion; the possibilities for external fire on the equipment areas of the plant; the existence of hazardous conditions, including toxic materials and pollution, that could arise; and, in general, the overall picture.

The engineer's ability to recognize the interrelationships of the various engineering disciplines with the process requirements is essential to thorough design. For example, the recognition of metallurgy and certain metallurgical testing requirements as they relate to the corrosion in the process environment is absolutely necessary to obtain a reliable process design and equipment specification. An example of the importance of this is hydrogen brittlement (see latest charts [54]). Another important area is water service (see [49]). The engineer selecting the materials of construction should recognize the importance of plastics and plastic composites in the design of industrial equipment and appreciate that plastics often serve as better corrosive resistant materials than do metals.

Flowsheets—Types

The flowsheet is the "road-map" of a process, and serves to identify and focus the scope of the process for all interested and associated functions of the project. As a project progresses, the various engineering disciplines read their portions of responsibility from the flowsheet, although they may not understand the process or other details relative to some of the other phases of engineering. Here is where the process and/or project engineer serves to tie together these necessary segments of work. This often involves explanations sufficiently clear to enable these other groups to obtain a good picture of the objective and the problems associated with attaining it.

The flowsheet also describes the process to management as well as those concerned with preparing economic studies for process evaluation.

A good process flowsheet pictorially and graphically identifies the chemical process steps in proper sequence. It is done in such a manner and with sufficient detail to present to others a proper mechanical interpretation of the chemical requirements.

There are several types of flowsheets:

1. Block Diagram, Figure 1-3

Usually used to set forth a preliminary or basic processing concept without details. The blocks do not describe *how* a given step will be achieved, but rather *what* is to be done. These are often used in survey studies to management, research summaries, process proposals for "packaged" steps, and to "talk-out" a processing idea. For management presentations the diagrams of Figures 1-4, 1-5A and B and 1-6A and B are pictorial and help illustrate the basic flow cycle.

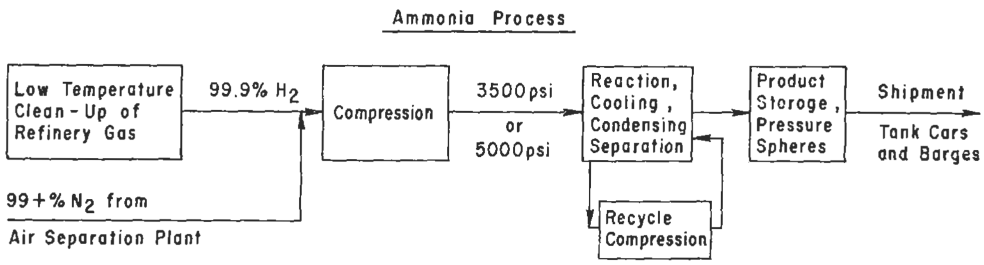


Figure 1-3. Block flow diagram.

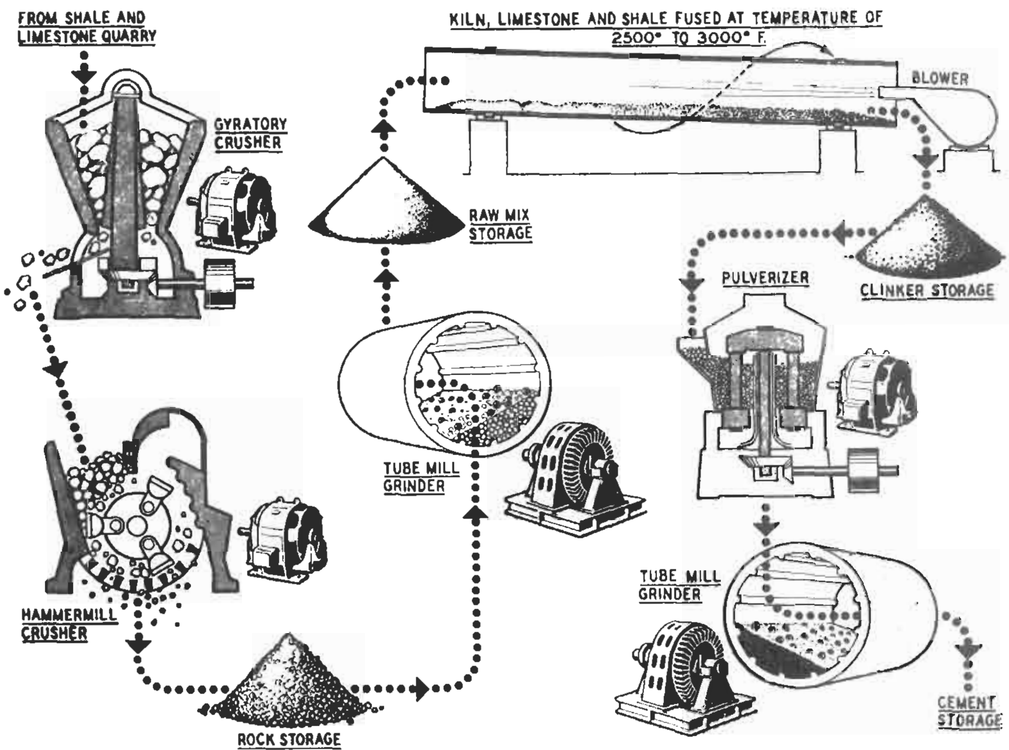


Figure 1-4. Pictorial flow diagram establishes key processing steps: Cement manufacture. By permission, E-M Synchronizer, Electric Machinery Mfg. Co.

2. Process Flowsheet or Flow Diagram, Figure 1-7

Used to present the heat and material balance of a process. This may be in broad block form with specific key points delineated, or in more detailed form identifying essentially every flow, temperature and pressure for each basic piece of process equipment or processing step. This may and usually does include auxiliary services to the process, such as steam, water, air, fuel gas, refrigeration, circulating oil, etc. This type of sheet is not necessarily distributed to the same groups as would receive and need the piping flowsheet described next, because it may contain detailed confidential process data.

3. Piping Flowsheet or Mechanical Flow Diagram, Figures 1-8, 1-9, or Piping and Instrumentation Diagram

Used to present "mechanical-type" details to piping and mechanical vessel designers, electrical engineers, instrument engineers, and other engineers not directly in need of process details. This sheet contains pipe sizes, all valves (sizes and types), temperature points, and special details needed to insure a common working basis for all persons on a project. In some engineering systems, detailed specifications cannot be completed until this flowsheet is basically complete.

4. Combined Process and Piping Flowsheet or Diagram, Figures 1-10 and 1-11

Used to serve the same purpose as both the process and the piping flow diagram combined. This necessarily results in a drawing with considerably more detail than either of types 2 and 3 just discussed. However, the advantage is in concentrating the complete data and information for a project at one point. It does require close attention in proper reading and often opens data to larger groups of persons who might misinterpret or misuse it.

Some companies do not allow the use of this sheet in their work primarily because of the confidential nature of some of the process data. Where it is used, it presents a concise summary of the complete process and key mechanical data for assembly. This type of sheet requires more time for complete preparation, but like all engineering developments preliminary issues are made as information is available. Often the sheet is not complete until the piping and other detailed drawings are finished. This then is an excellent record of the process as well as a work sheet for training operators of the plant.

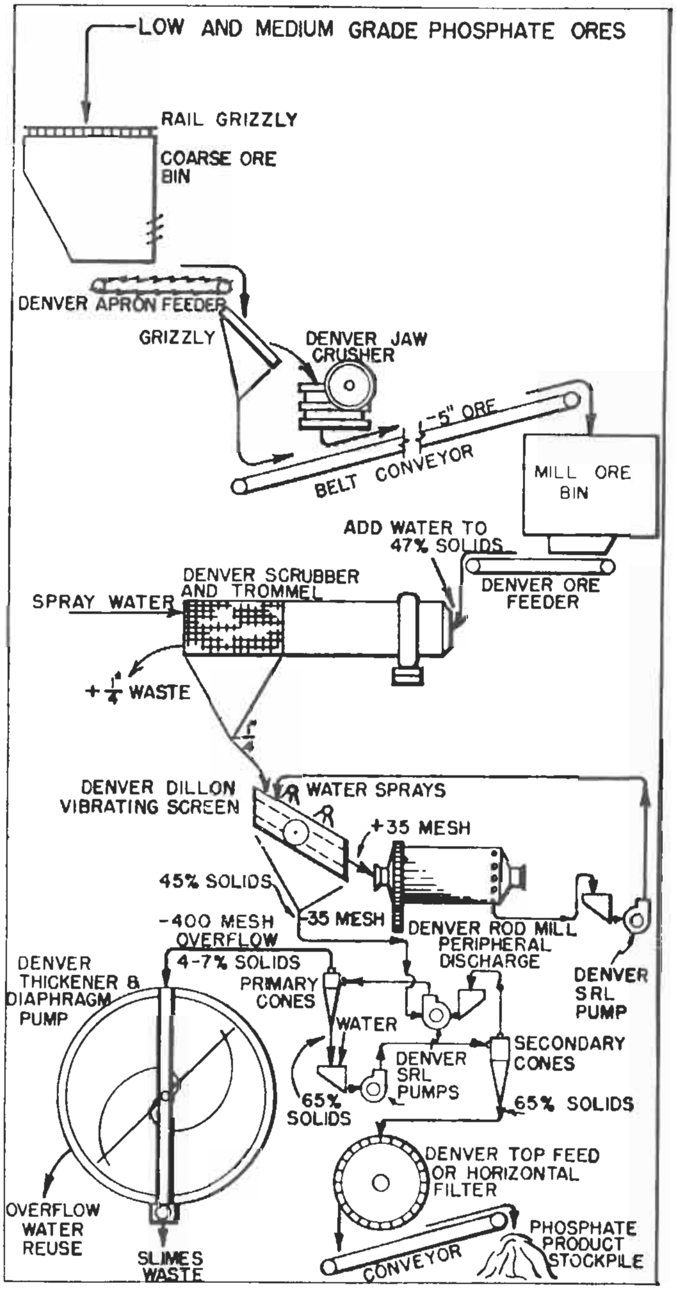


Figure 1-5A. Pictorial sections flow diagram for principal operations: phosphate recovery. By permission, Deco Trefoil, 1958, Denver Equipment Co.

5. Utility Flowsheets or Diagrams, Figures 1-12 and 1-13

Used to summarize and detail the interrelationship of utilities such as air, water (various types), steam (various types), heat transfer mediums such as Dowtherm, process vents and purges, safety relief blow-down, etc., to the basic process. The amount of detail is often too great to combine on other sheets, so separate sheets are prepared.

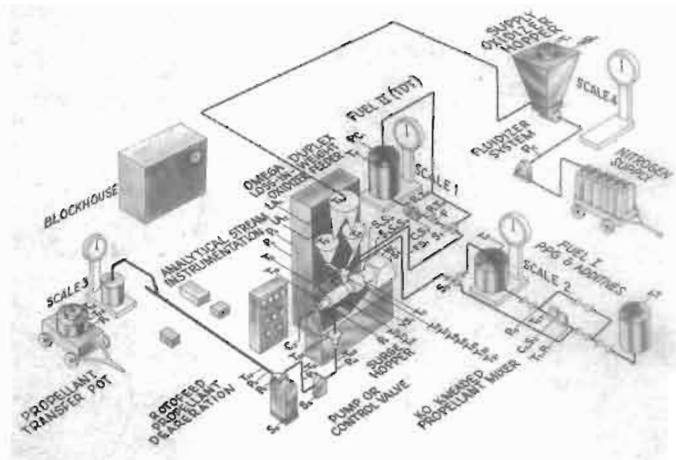


Figure 1-5B. Isometric pictorial flow diagram. By permission, J. W. Keating and R. D. Geckler, Aerojet General Corp.

These are quite valuable and time saving during the engineering of the project. They also identify the exact flow direction and sequence of tie-in relationships for the operating and maintenance personnel.

6. Special Flowsheets or Diagrams

From the basic process-containing flowsheet other engineering specialties develop their own details. For example, the instrument engineer often takes the requirements of the process and prepares a completely detailed flowsheet which defines every action of the instruments, control valves, switches, alarm horns, signal lights, etc. This is his detailed working tool.

The electrical engineer likewise takes basic process and plant layout requirements and translates them into details for the entire electrical performance of the plant. This will include the electrical requirements of the instrumentation in many cases, but if not, they must be coordinated.

O'Donnell [9] has described the engineering aspects of these special flowsheets.

7. Special or Supplemental Aids

(a) Plot Plans, Figure 1-14

Plot plans are necessary for the proper development of a final and finished process, piping or utility flowsheet. After broad or overall layout decisions are made, the detailed layout of each processing area is not only helpful but necessary in determining the first realistic estimate of the routing, lengths and sequence of piping. This is important in such specifications as pipe sizing, and pump head and compressor discharge pressures. The nature of the fluids—whether hazardous, toxic, etc.,—as well as the direction or location or availability for entrance to the

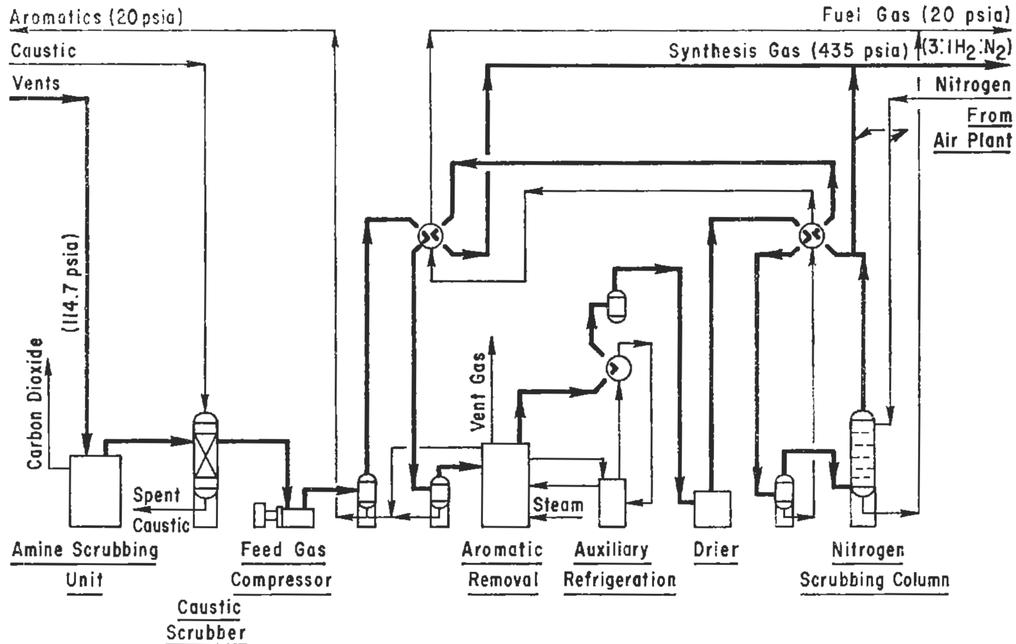


Figure 1-6A. Typical flow scheme for separation and purification of vent streams.

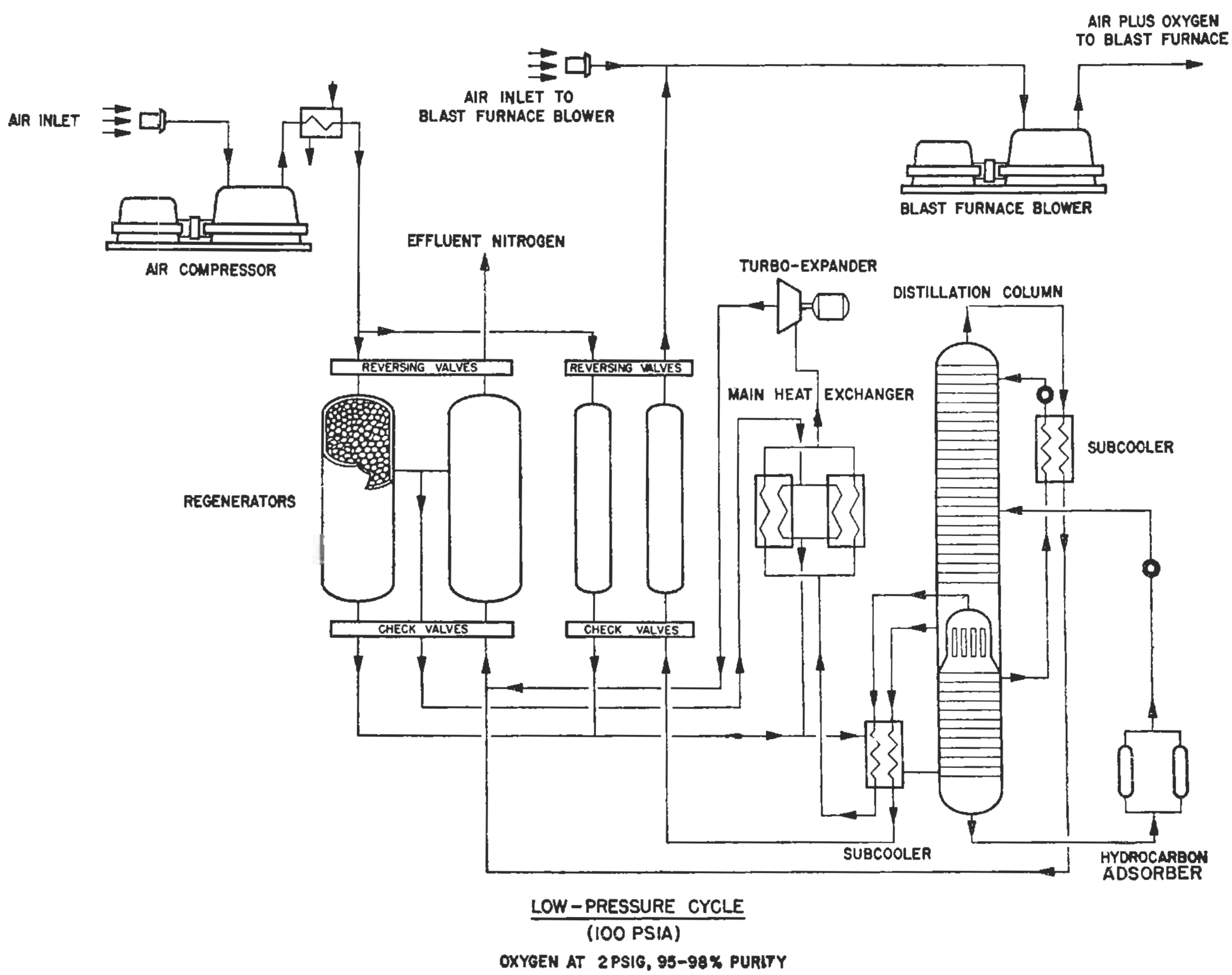


Figure 1-6B. This low pressure cycle is used for production of oxygen in steady state conditions. By permission, Air Products and Chemicals Inc.

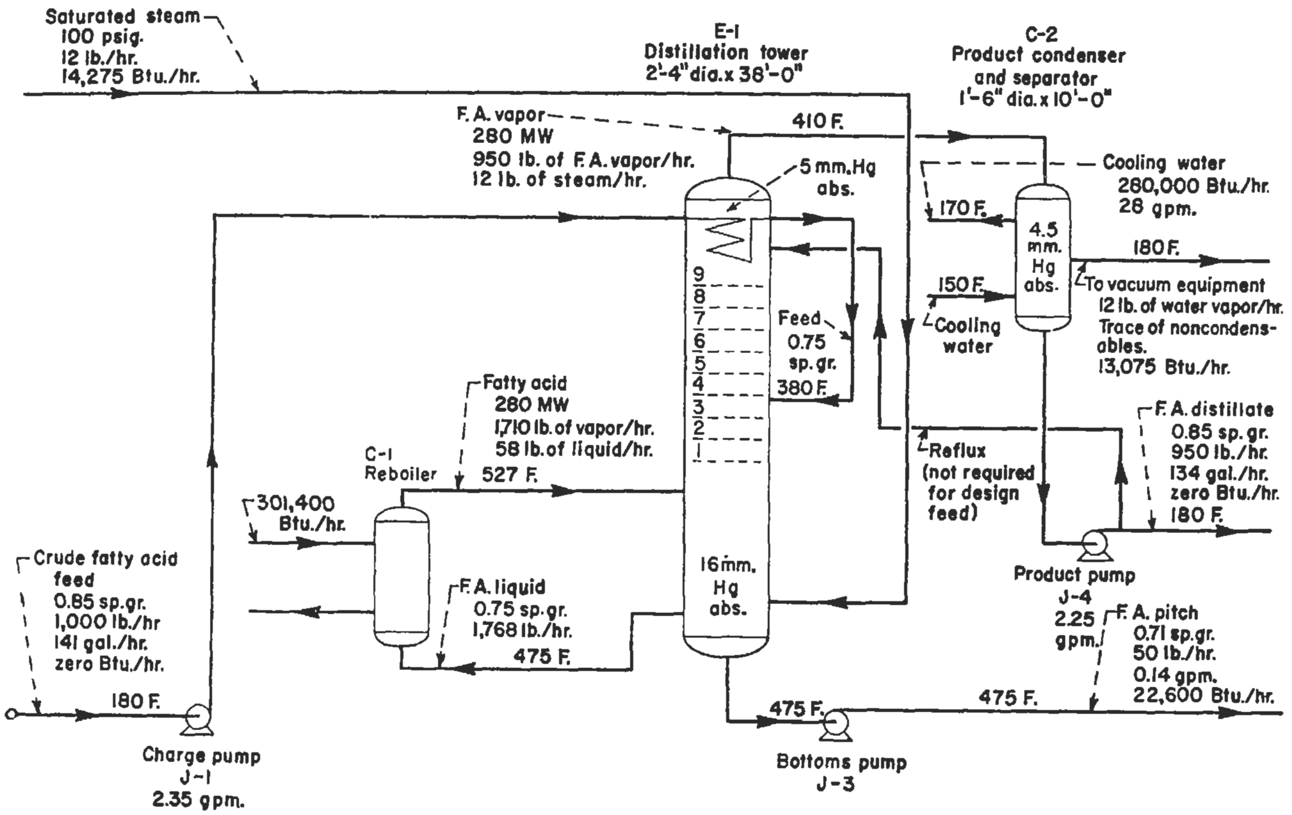


Figure 1-7. Heat and material balance establishes material and thermal requirements. By permission, J. P. O'Donnell [9].

area, definitely influences decisions regarding the equipment layout on the ground, in the structures, and in relation to buildings. Prevailing wind direction and any other unusual conditions should also be considered.

The use of pictorial isometric or oblique views of plot areas as shown in Figure 1-15 is very helpful for equipment location evaluation. With talented personnel, this type of layout study can replace model studies. These layouts are also useful for management presentations.

(b) Models, Figure 1-16A and 16B

Scale models are a real asset in the effective and efficient layout and sometimes process development of a plant. Although any reasonable scale can be used, the degree of detail varies considerably with the type of process, plant site, and overall size of the project. In some instances cardboard, wooden, or plastic blocks cut to a scale and placed on a cross-section scale board will serve the purpose. Other more elaborate units include realistic scale models of the individual items of equipment. These are an additional aid in visualizing clearances, orientation, etc.

A complete model usually includes piping, valves, ladders, floor grating, etc. This essentially completes the visualization of the condition of the layout. In fact, many engineering offices use models to varying degrees and often make direct space-clearance measurements from them. Others photograph the models, or sections, for use by the piping engineers at their desks. In some few instances, dimensioned photographs have been issued directly to construction forces in place of drawings.

The models are even more helpful to the process engineer than simple plot plans. The advantages are multiplied, as with models the process engineer can study as well as solicit the advice of other engineers in visualizing a processing condition.

Plant model costs vary depending upon the degree of detail included. Considerable decision making information can be obtained from a set-up of block layout only, and these costs would be extremely small. For a reasonably complete scale piping detail model the costs are reported[5] as 0.1 to 0.6 percent of the cost of the plant. The large plants over $20 million cost in the lower 0.1 percent range while small plant models cost in the 0.6 to 1.0

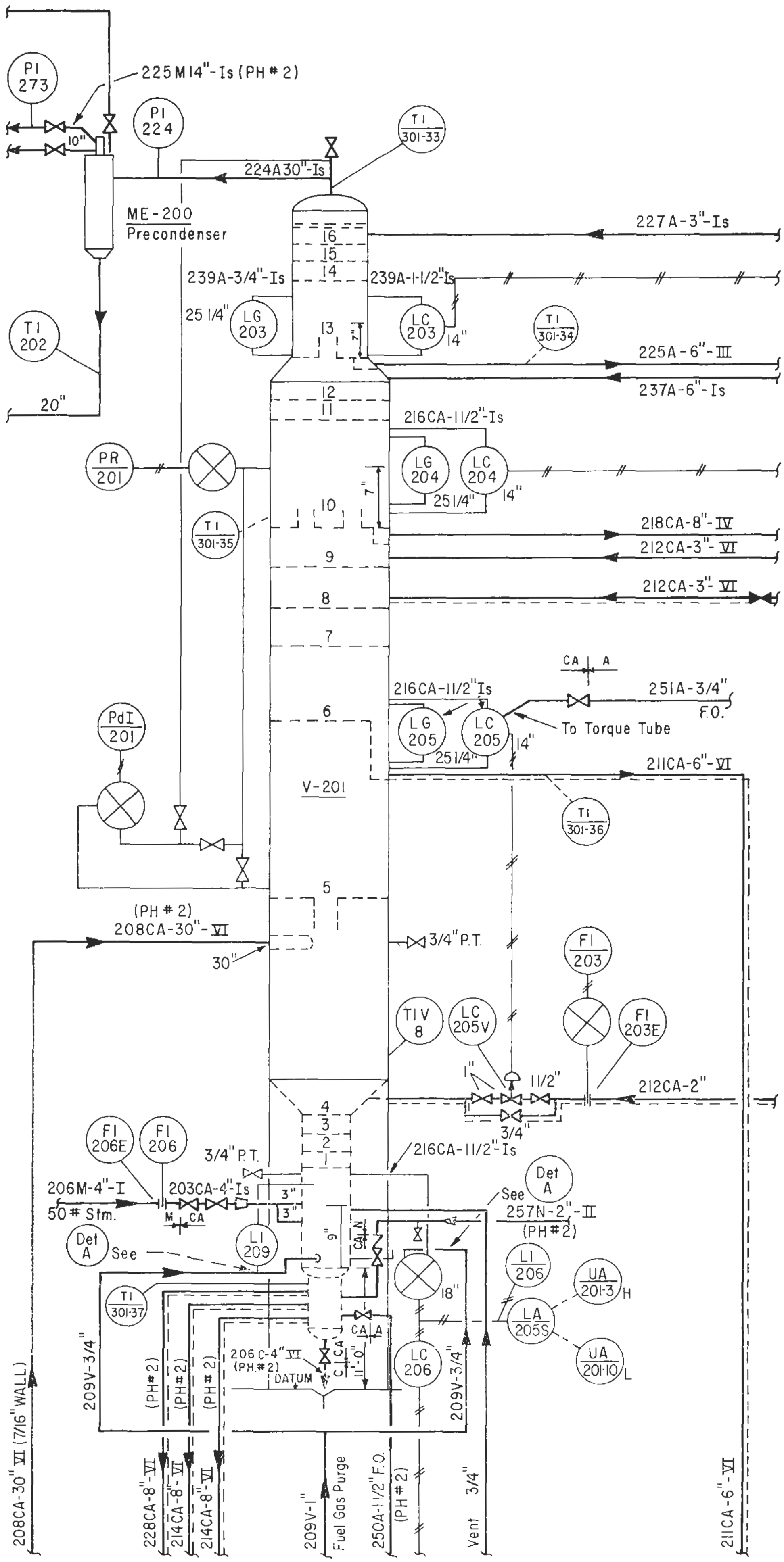


Figure 1-8. Mechanical detail flow diagram. By permission, Fluor Corp. Ltd.

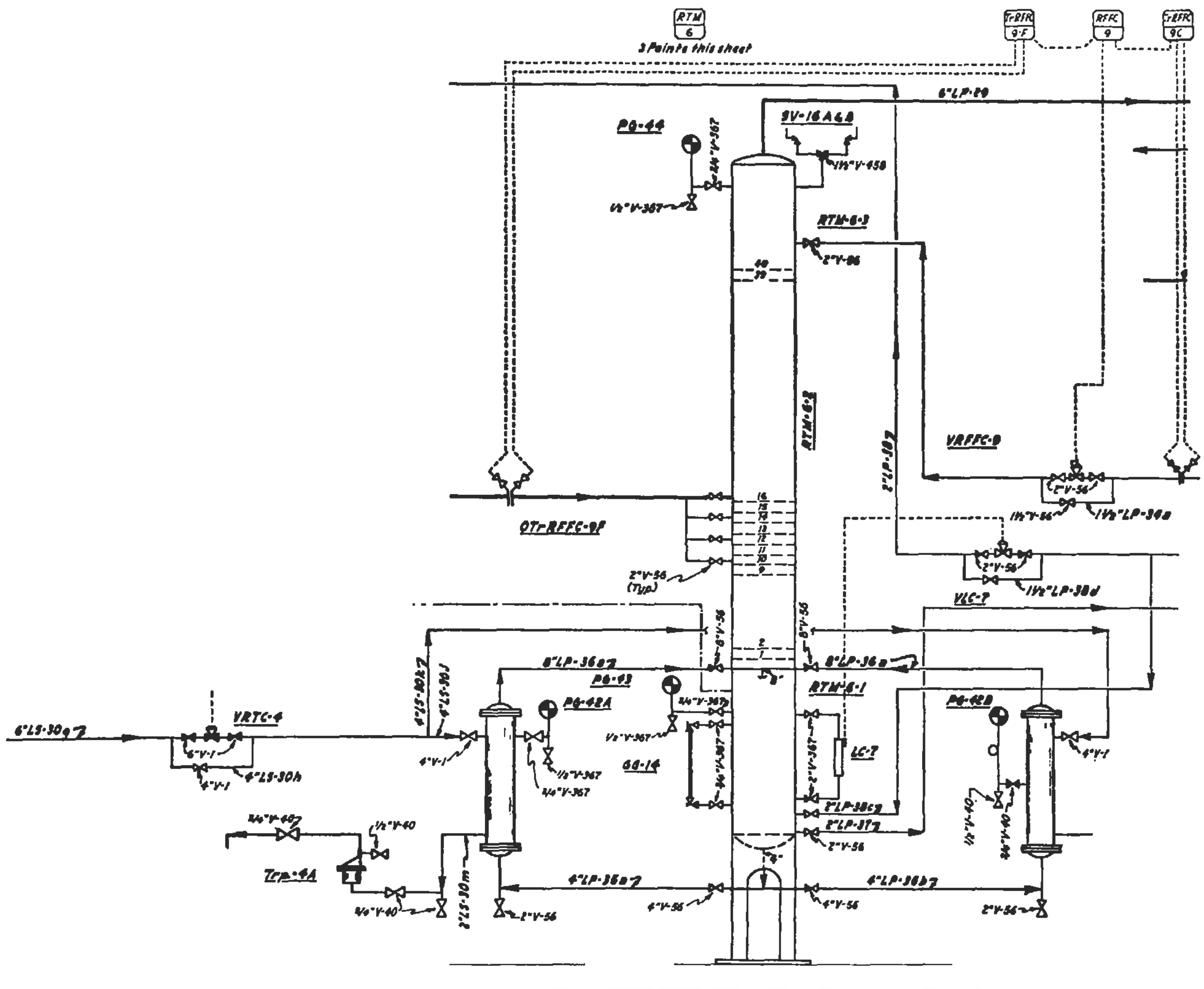


Figure 1-9. Typical process and piping flow diagram. By permission, E. E. Ludwig [56].

percent range. Even these costs can be reduced if all minute detail is avoided, and only basic decision making piping is included. The necessary model structure and rough block outline equipment for a $1 million hydrocarbon compression and processing plant costs around $1,000 to $2,000.

Paton [15] reports total model costs of 0.4 to 1.0 percent of erected plant costs for a $1 million plant. These are actual costs and do not reflect profits. Material costs are less than 10 percent of total model costs, and usually less than 5 percent. For a $30 million plant model costs run as low as 0.1 percent. These are for models which include plant layout, piping layout, and piping details. If simpler models are used the costs should be less.

Flowsheet Presentation

Experienced flowsheet layout personnel all emphasize the importance of breaking processes into systems and logical parts of systems such as reaction, compression, separating, finishing, refrigeration, storage, etc., for detailed drafting. This point cannot be overemphasized, since considerably more space is needed for final completion of all details than is usually visualized at first. The initial layout of the key equipment should be spread farther than looks good to the eye. In fact, it probably looks wasteful of drawing space.

Later as process and sometimes service lines, valves, controls and miscellaneous small accessories are added this "extra" space will be needed to maintain an easily readable sheet. As this develops, attention should be

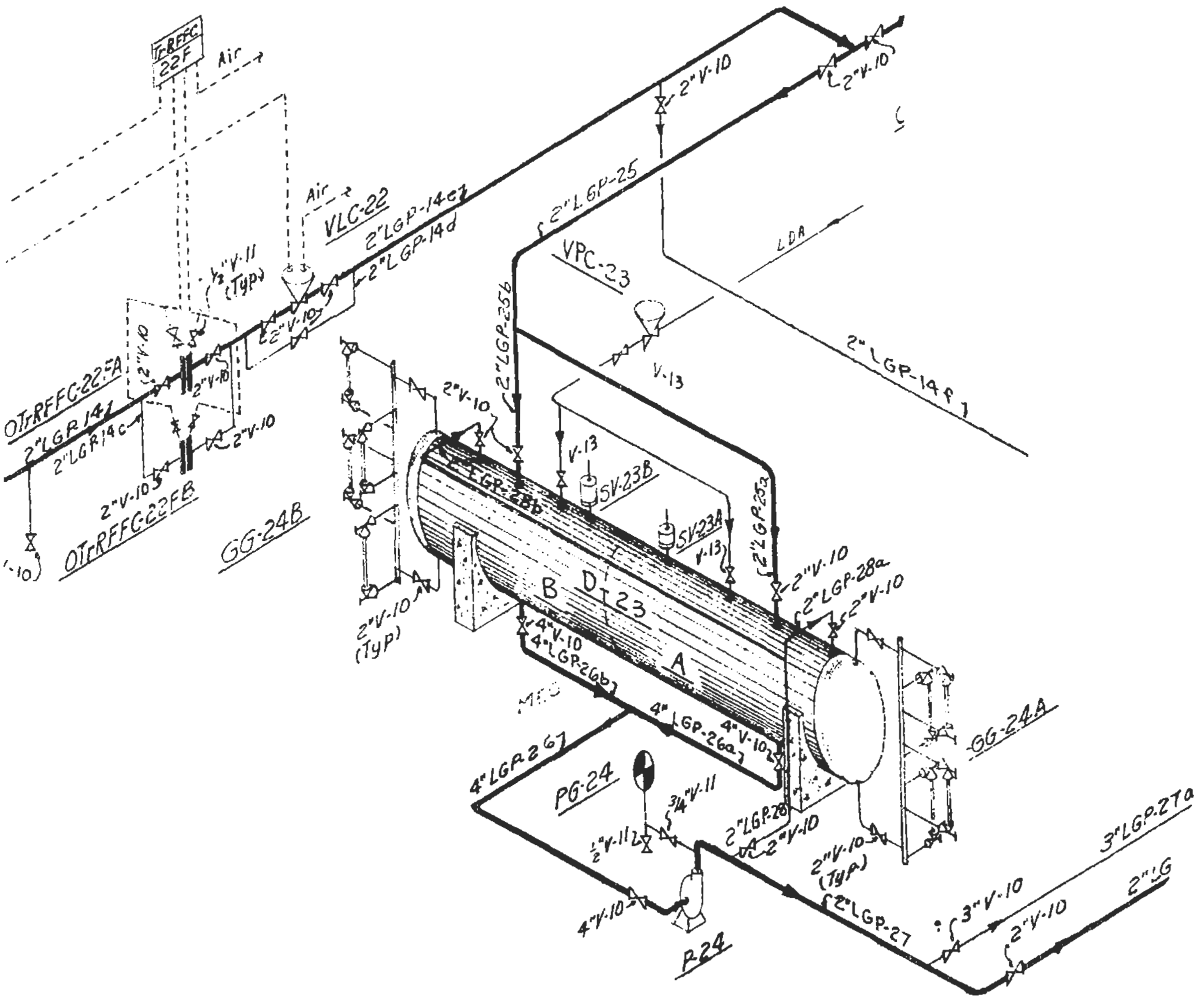


Figure 1-10. Piping detail isometric flow diagram.

given to the relative weights and styles of lines to aid in the readability of the sheets.

Figure 1-11 suggests an approach to standardization of form for general use. It can be rearranged in several ways to provide a format suitable for any one of several purposes. Of particular importance is the flexibility of adding or deleting data without changing other details. Some companies prefer to place the process data on a separate sheet, although the same basic form for the table can be retained as shown in Figure 1-11. The layout principles of Figure 1-8 are also standardized by some companies.

General Arrangements Guide

Each phase of the process is best represented on individual flowsheets. Electric power, fuel gas, drainage and the many other auxiliary system requirements are also best defined by separate individual flowsheets. These should be complete including all headers, branch take-offs, tie-ins to existing or known points, etc. Only in this way can all the decisions as well as specifications be delineated for the various parts contributing to the entire project. The master process or mechanical flowsheet must contain specific references to the other sheets for continuation of the details and complete coordination.

Flowsheet size may vary depending upon the preferences of the individuals using them. The most popular system uses one size sheet about 24 × 36 inches for *all* flowsheets. The use of miscellaneous large and small sizes to represent the entire project is often awkward when collected, and increases the possibilities of sheets becoming misplaced. Some groups use sheets from a roll and these are sized to length by systems, becoming 24 × 60 inches, 24 × 72 inches or longer. These are fine for initial study but become tedious to handle on the usual desk. These sheets can be reduce to 11 × 36 inches or 11 by 48 inches

(text continued on page 15)

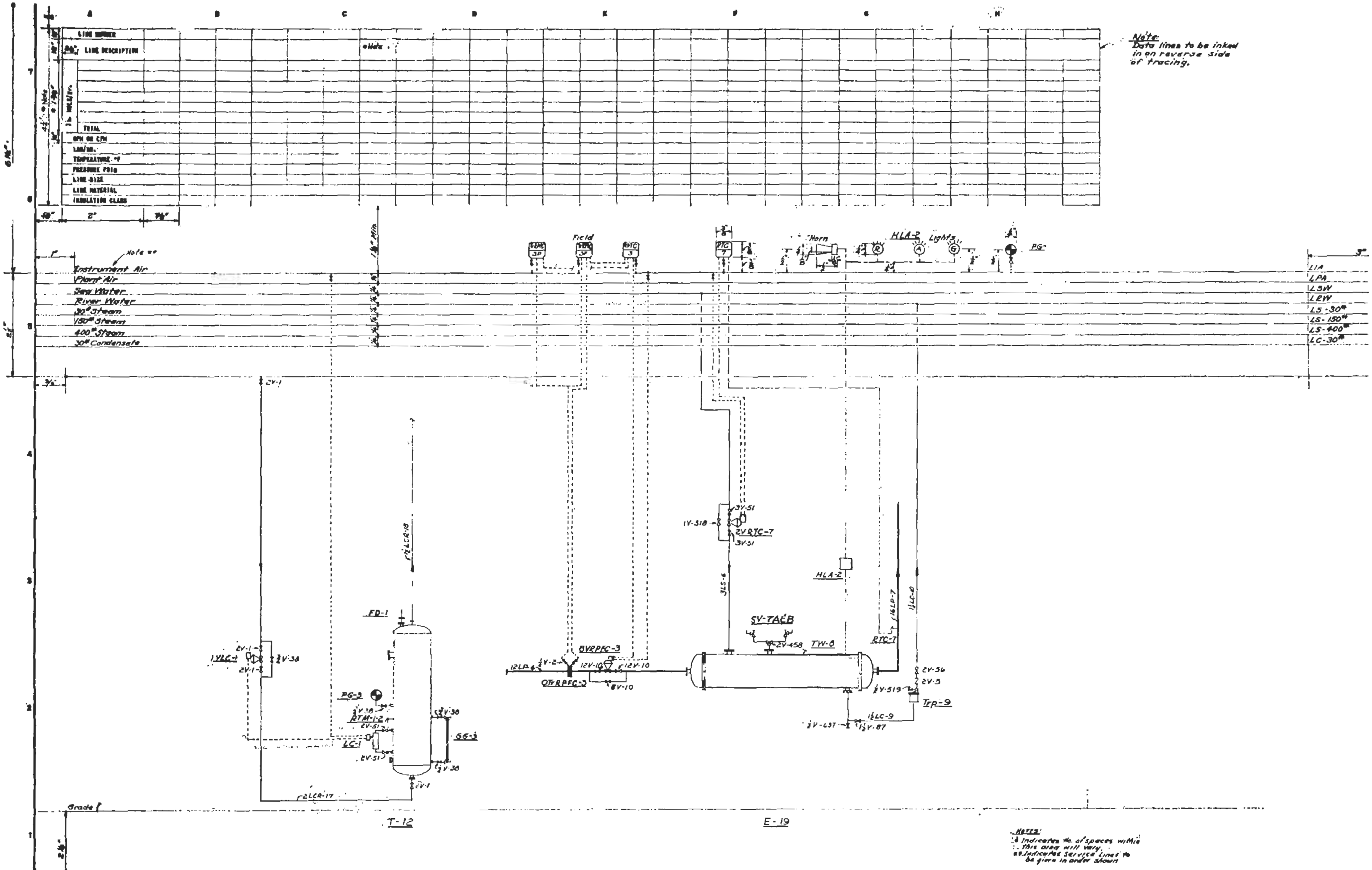


Figure 1-11. Typical material balance process flowsheet.

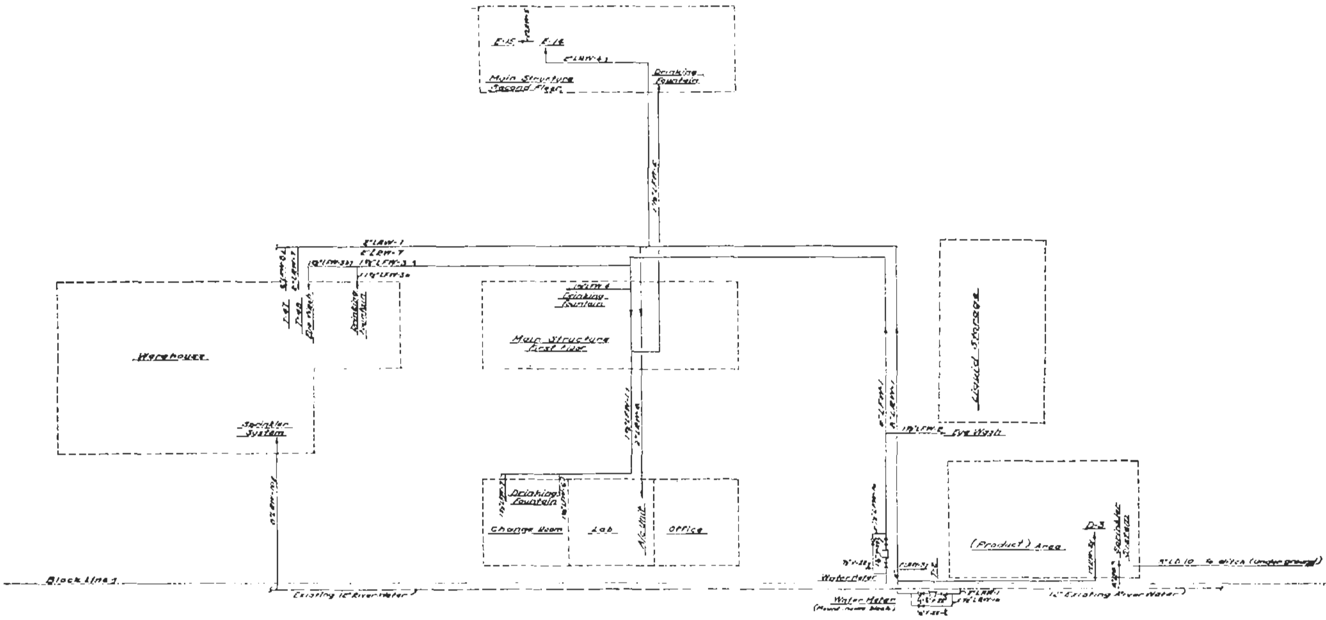


Figure 1-12. Standard type layout for service piping diagram.

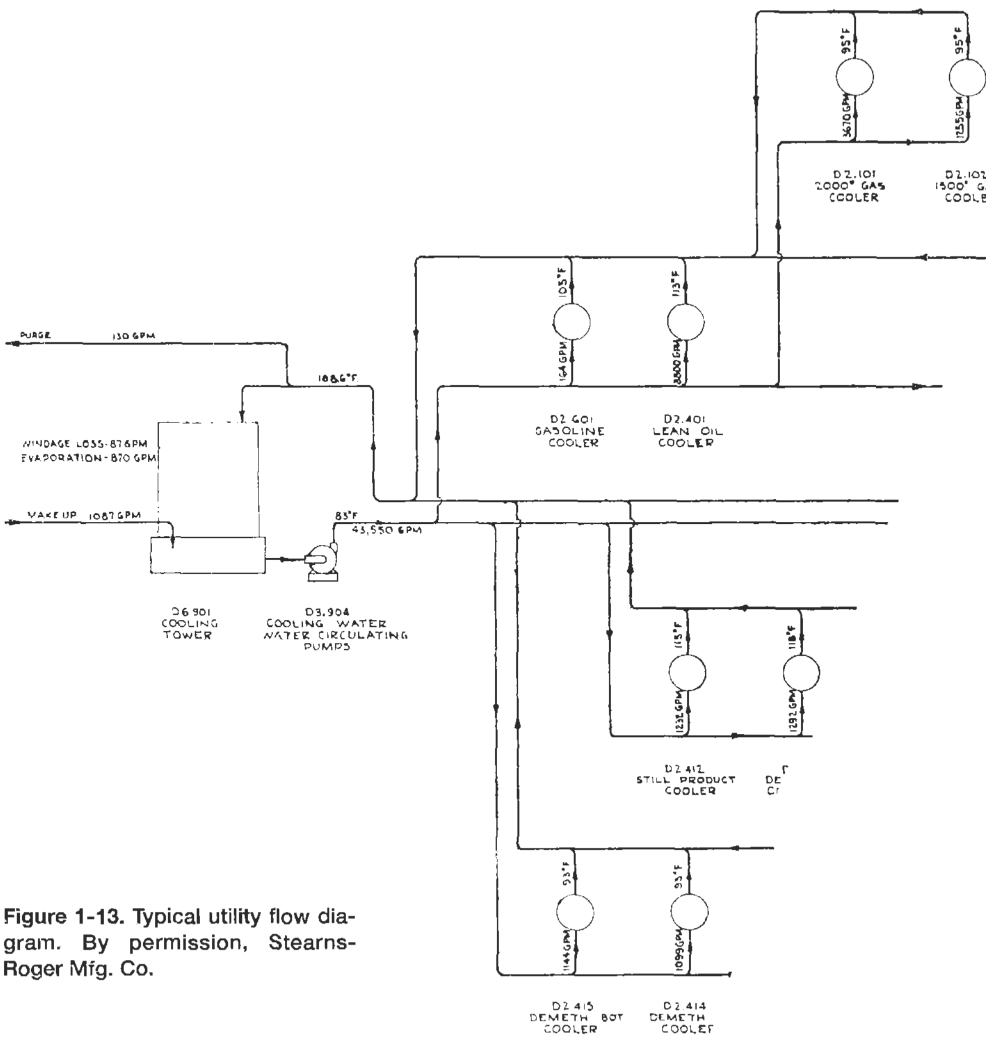


Figure 1-13. Typical utility flow diagram. By permission, Stearns-Roger Mfg. Co.

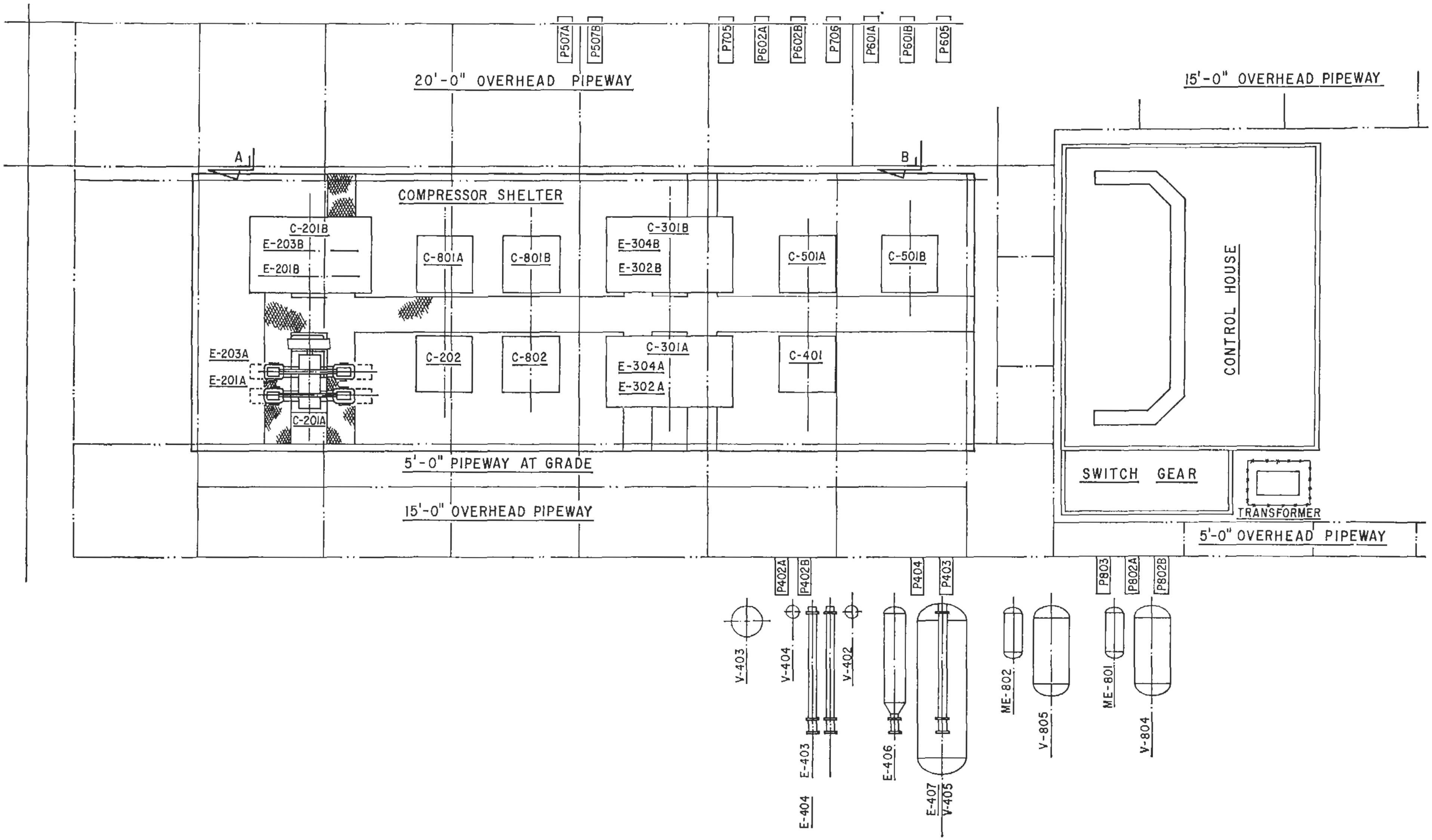


Figure 1-14. Typical process area plot plan and study elevations. By permission, Fluor Corp. Ltd.

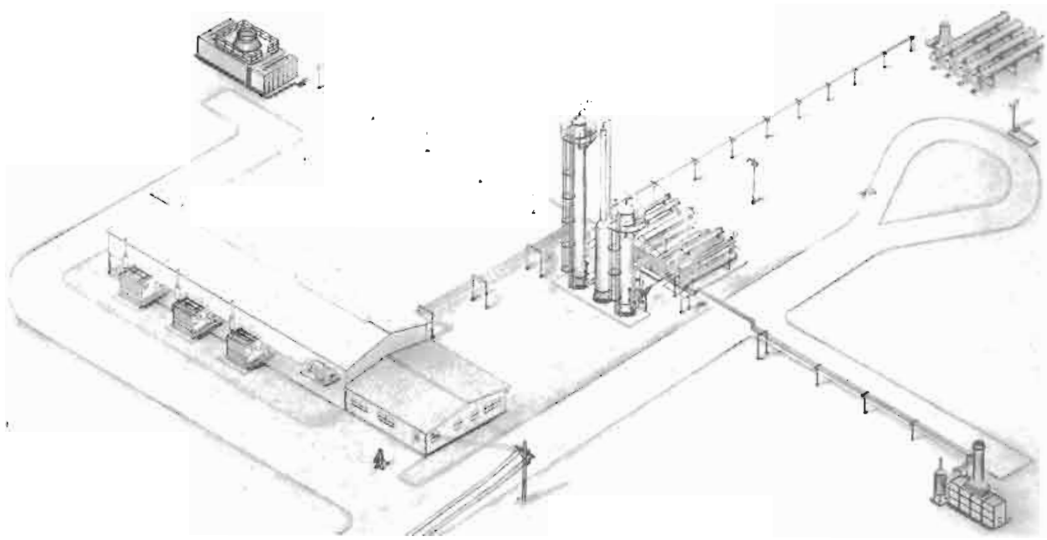


Figure 1-15. Pictorial plot plan layout. Courtesy of Prengle, Dukler and Crump, Houston, Texas.

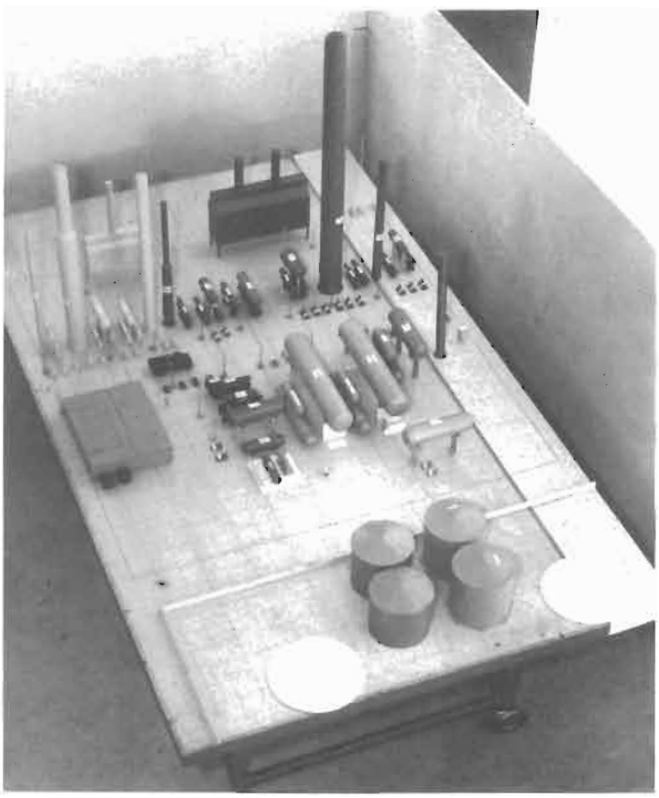


Figure 1-16A. Simple block model plant layout. Courtesy of Socony Mobil Oil Co. Inc.

(text continued from page 11)

both of which are more convenient to work with. These strip-type sheets allow large portions of the process to be grouped together, and are adaptable for folding into reports, etc.

Since the flowsheet is the primary reference for all engineers working on a project, it must contain all of the decisions, data, flow connections, vents, drains etc., which can reasonably be included without becoming confusing and difficult to read.

It is important that the various items of equipment and valves be spaced, pictorially represented and sized as to be easy to read, recognized and followed. On the surface this may sound easy, while in reality it takes an experienced flowsheet detailer to arrange the various items in an eye-pleasing and efficient arrangement. Suggestive outline figures plus shading often yields the best looking flowsheet (Figure 1-10); however, the extra time for detail costs time and money. Some compromise is often indicated. Reference to the various flowsheets illustrated here indicates that the equipment can be arranged by (1) working from a base line and keeping all heights relative and (2) by placing the various items in a straight-through flow pattern without relative heights. The first scheme is usually preferred for working flowsheets. Whenever possible, all auxiliary as well as spare equipment is shown. This facilitates the full and proper interpretation of all the details.

Figure 1-17 [2] can be used as a guide in establishing relative sizes of equipment as represented on a flowsheet. This chart is based on approximate relative proportions pictured by the mind's eye [2]. For example, the 10-foot diameter × 33-foot high tank would scale to 1.5 inches high. By using the height-developed scale factor, the diameter would be (1.5″/33′) (10′) = 0.45″ or say 0.5″ diameter on the flowsheet.

For some purposes the addition of equipment specification and performance data on the flowsheets adjacent to the item is of value. In many cases though, this additional information makes the sheets difficult to read. The

Figure 1-16B. Detailed layout and piping model for a refinery unit. Courtesy of Socony Mobil Oil Co. Inc.

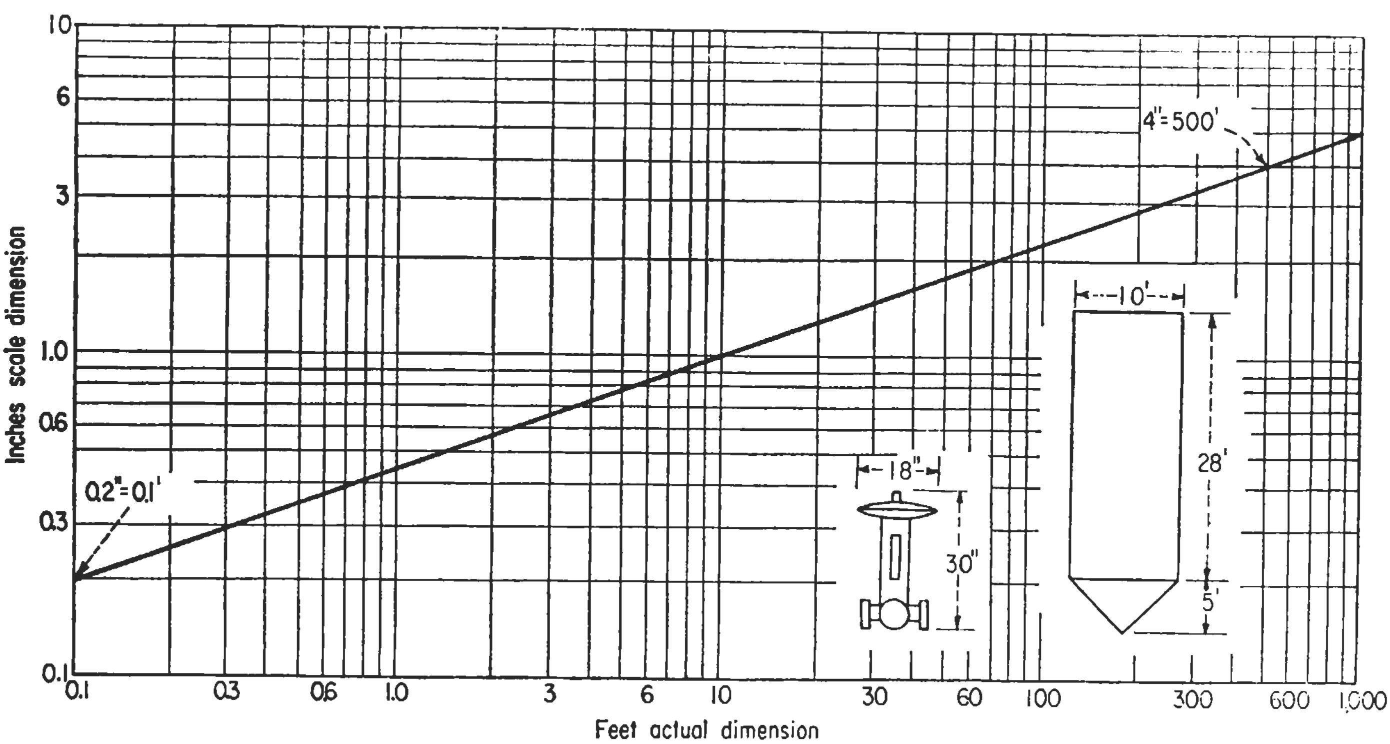


Figure 1-17. Flowsheet scale reference diagram. By permission, R. H. Berg [2].

use of equipment summary tables similar to flow and pipe data tables can avoid this objection and yet keep the information on the sheets. Some flowsheets include relief valve set pressures adjacent to the valves, volume capacities of storage tanks, etc.

Computer-Aided Flowsheet Design/Drafting

Current technology allows the use of computer programs and data bases to construct an accurate and detailed flowsheet. This may be a process type diagram or a piping and mechanical/instrument diagram, depending on the input. See Figures 1-9, 1-10, 1-18A and 1-18B.

Flowsheet Symbols

To reduce detailed written descriptions on flowsheets, it is usual practice to develop or adopt a set of symbols and codes which suit the purpose. Flowsheet symbol standardization has been developed by various professional and technical organizations for their particular fields. Most of these have also been adopted by the American National Standards Institute (ANSI). The following symbol references are related and useful for many chemical and mechanical processes:

1. American Institute of Chemical Engineers
   (a) Letter Symbols for Chemical Engineering, ANSI Y10.12
2. American Society of Mechanical Engineers
   (a) Graphic Symbols for Plumbing, ANSI or ASA Y32.4
   (b) Graphic Symbols for Railroad Maps and Profiles, ANSI or ASA Y32.7
   (c) Graphic Symbols for Fluid Power Diagrams, ANSI or ASA Y32.10
   (d) Graphic Symbols for Process Flow, ANSI or ASA Y32.11
   (e) Graphic Symbols for Mechanical and Acoustical Elements as Used in Schematic Diagrams, ANSI or ASA Y32.18
   (f) Graphic Symbols for Pipe Fittings, Valves and Piping, ANSI or ASA Z32.2.3
   (g) Graphic Symbols for Heating, Ventilating and Air Conditioning, ANSI or ASA Z32.2.4
   (h) Graphic Symbols for Heat-Power Apparatus, ANSI or ASA Z32.2.6
3. Instrument Society of America
   (a) Instrumentation Symbols and Identification, ISA-S5.1, also see Reference 27

Other symbols are established for specialized purposes.

The physical equipment symbols established in some of these standards are often not as descriptive as those the chemical, petrochemical, and petroleum industry is accustomed to using. The bare symbolic outlines given in some of the standards do not adequately illustrate the detail needed to make them useful. Accordingly, many process engineers develop additional detail to include on flowsheets, such as Figures 1-19 A–E and 1-20 A-B-C which enhance the detail in many of these standards. Various types of processing suggest unique, yet understandable, symbols, which do not fit the generalized forms.

Many symbols are pictorial which is helpful in representing process as well as control and mechanical operations. In general, experience indicates that the better the representation including *relative* locating of connections, key controls and even utility connections, and service systems, the more useful will be the flowsheets for detailed project engineering and plant design.

To aid in readability by plant management as well as engineering and operating personnel, it is important that a set of symbols be developed as somewhat standard for a particular plant or company. Of course, these can be improved and modified with time and as needed, but with the basic forms and letters established, the sheets can be quite valuable. Many companies consider their flowsheets quite confidential since they contain the majority of key processing information, even if in summary form.

Line Symbols and Designations

The two types of lines on a flowsheet are (1) those representing outlines and details of equipment, instruments, etc., and (2) those representing pipe carrying process or utility liquids, solids, or vapors and electrical or instrument connections. The latter must be distinguished among themselves as suggested by Figure 1-21.

In order to represent the basic type of solution flowing in a line, designations or codes to assign to the lines can be developed for each process. Some typical codes are:

RW — River Water
TW — Treated Water
SW — Sea Water
BW — Brackish Water
CW — Chilled Water
S — Low Pressure Steam
S150 — 150 psi Steam
S400 — 400 psi Steam
V — Vent or Vacuum
C — Condensate (pressure may be indicated)
D — Drain to sewer or pit
EX — Exhaust
M — Methane
A — Air (or PA for Plant Air)
F — Freon

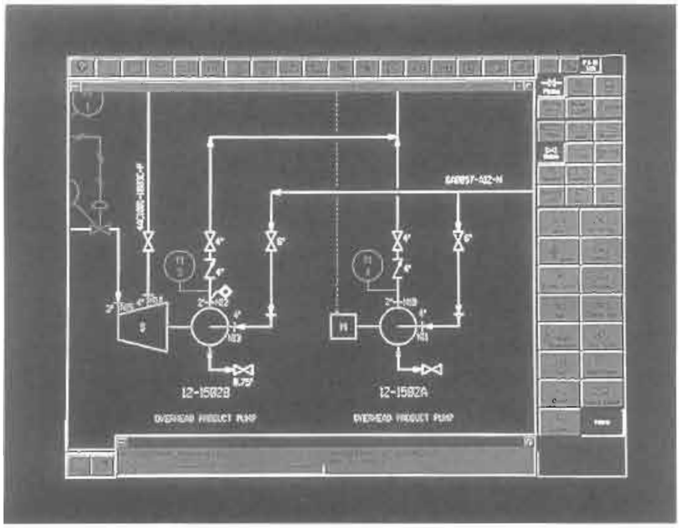


Figure 1-18A. Computer generated P. and I D. flowsheet. Courtesy of Intergraph Corp., Bul. DP016A0.

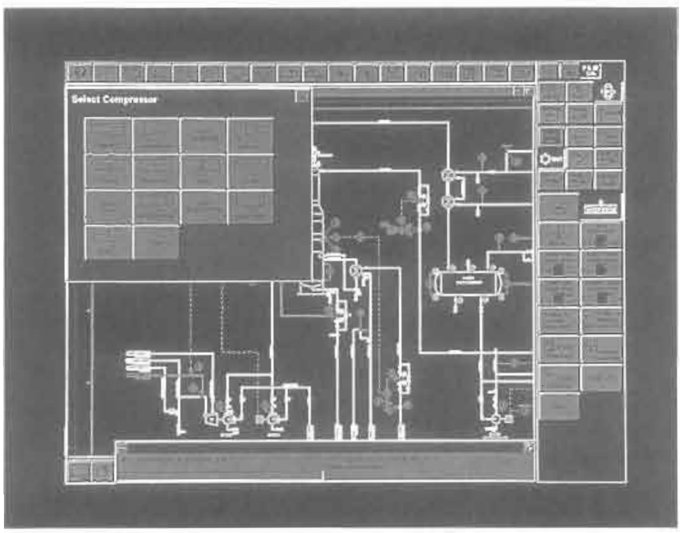


Figure 1-18B. Computer generated instrumentation detail for P. and I D. flowsheet. Courtesy of Integraph Corp., Bul. DP016A0.

G — Glycol
SA — Sulfuric Acid
B — Brine
CL — Chlorine
P — Process mixture (use for in-process lines not definitely designated by other symbols)

Sometimes it is convenient to prefix these symbols by L to indicate that the designation is for a line and not a vessel or instrument.

Materials of Construction for Lines

The process designer must also consider the corrosive nature of the fluids involved when selecting construction materials for the various process and utility service lines. Some designers attach these materials designations to the line designation on the flowsheets, while others identify them on the Line Summary Table (Figure 1-24D). Some typical pipe materials designations are:

CS40 — Carbon steel, Sch. 40
CS80 — Carbon steel, Sch. 80
SS316/10 — Stainless steel 316m Sch. 10
GL/BE — Glass bevel ends
N40 — Nickel, Sch. 40
TL/CS — Teflon-lined carbon steel
PVC/CS Polyvinyl chloride — lined CS
PP — Solid polypropylene (designate weight sch)

Test Pressure for Lines

The process designer also needs to designate the hydraulic test pressures for each line. This testing is performed after construction is essentially complete and often is conducted by testing sections of pipe systems, blanking off parts of the pipe or equipment, if necessary. Extreme care must be taken to avoid over pressuring any portion of pipe not suitable for a specific pressure, as well as extending test pressure through equipment not designed for that level. Vacuum systems must always be designed for "full vacuum," regardless of the actual internal process absolute vacuum expected. This absolute zero design basis will prevent the collapse of pipe and equipment should internal conditions vary. Some line design systems include the test pressure in the line code, but this often becomes too unwieldly for drafting purposes.

The usual complete line designation contains the following: (1) line size (nominal); (2) material code; (3) sequence number; and (4) materials of construction.

Examples: 2″—CL6—CS40
3″—CL6a—CS40
4″—RW1—CS40
16″—S150—CS40
3″—P—TL/CS

See Figures 1-23 and 1-24A through D.

Some engineers rearrange the sequence of the code although the information remains essentially the same. The line number sequence is conveniently arranged to start with one (1) or 100 for each of the fluid designations (CL, P, etc.). Since the sequence numbers are for coordi-

(text continued on page 23)

Figure 1-19A. Process vessels.

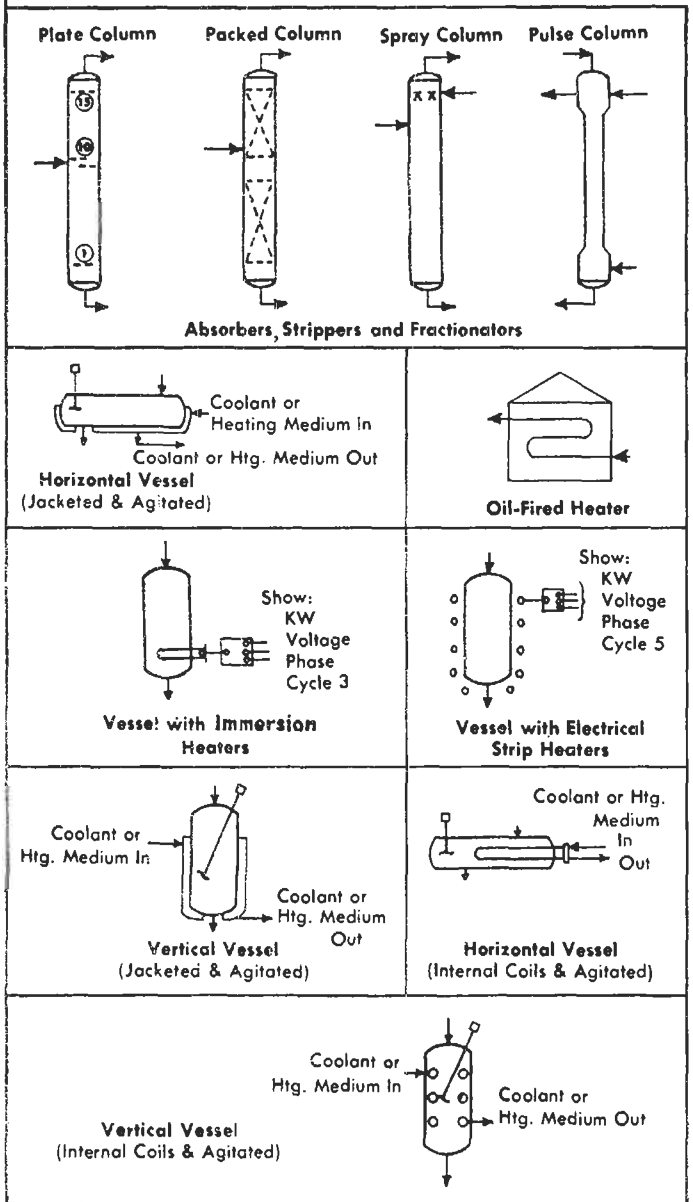


Figure 1-19B. Pumps and solids.

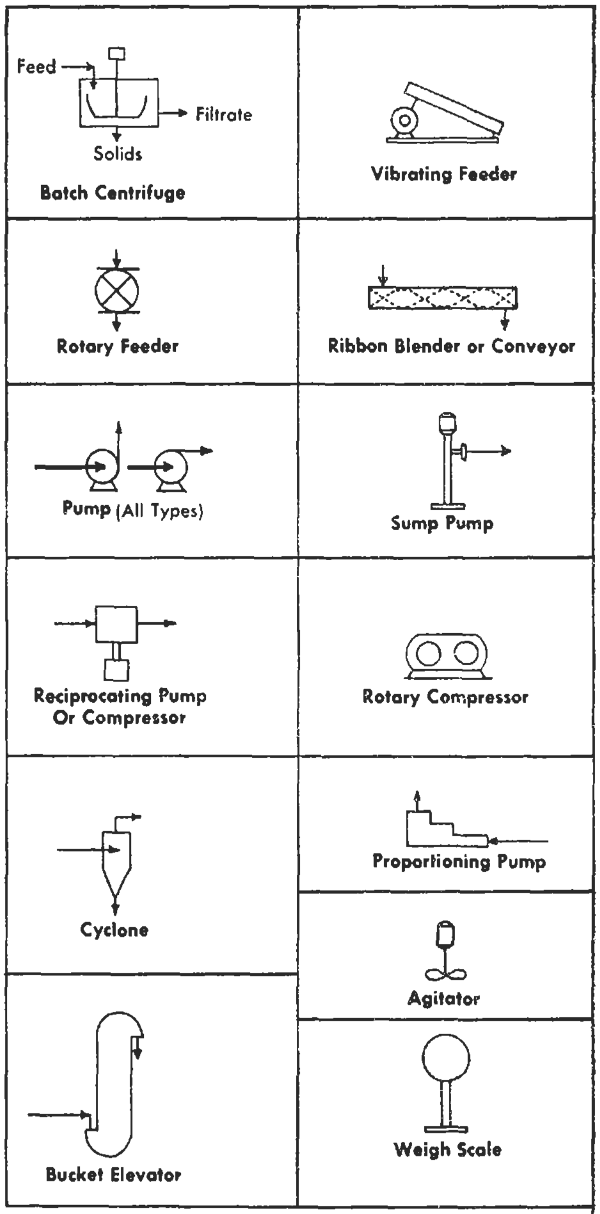
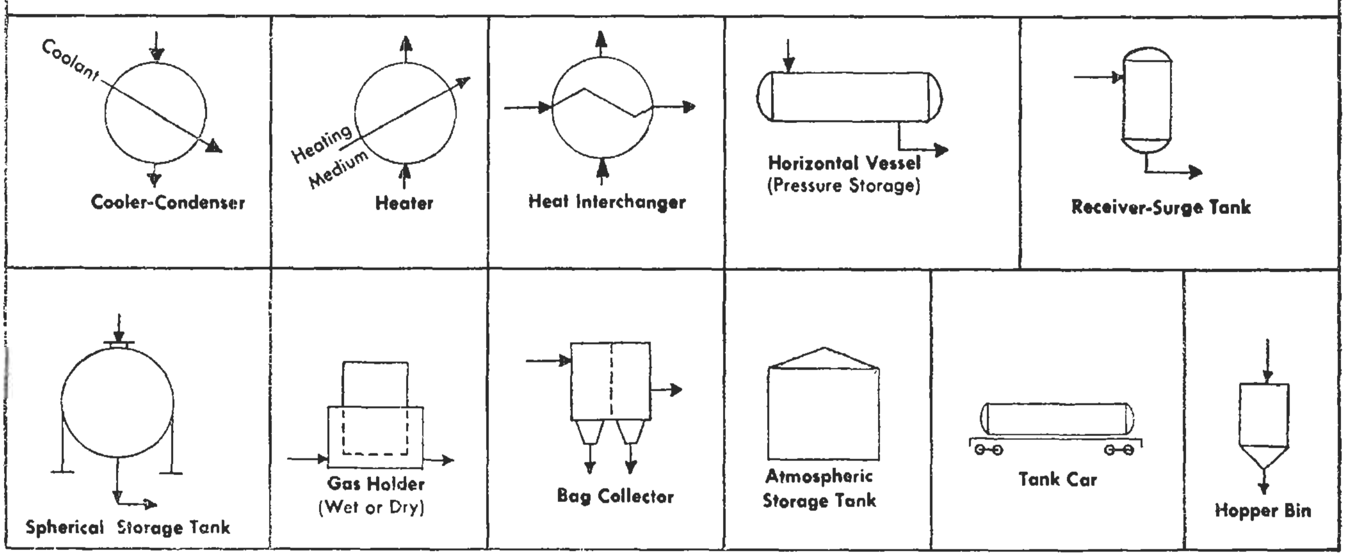


Figure 1-19C. Storage equipment.



Handling

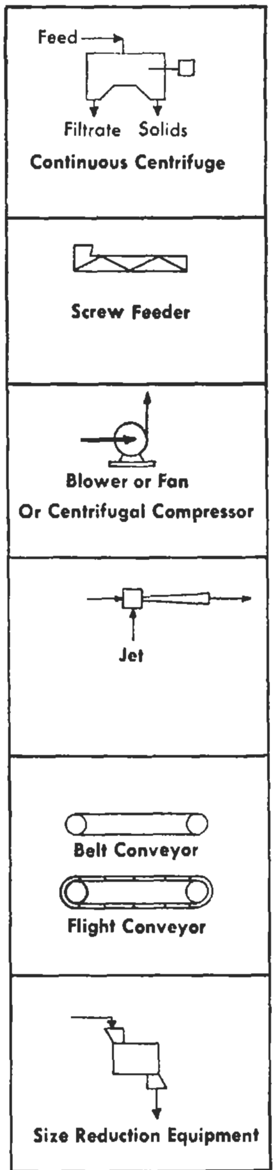
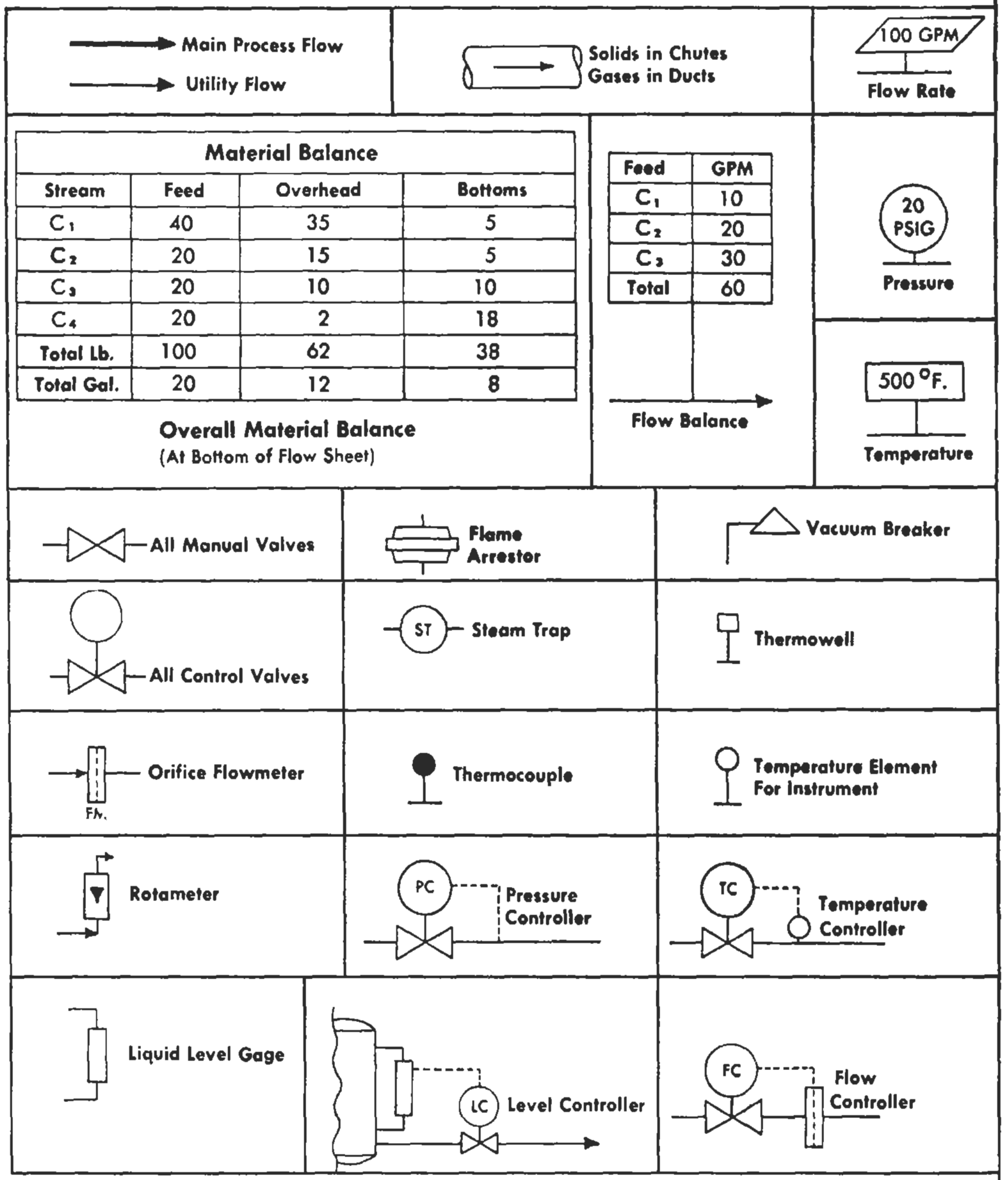


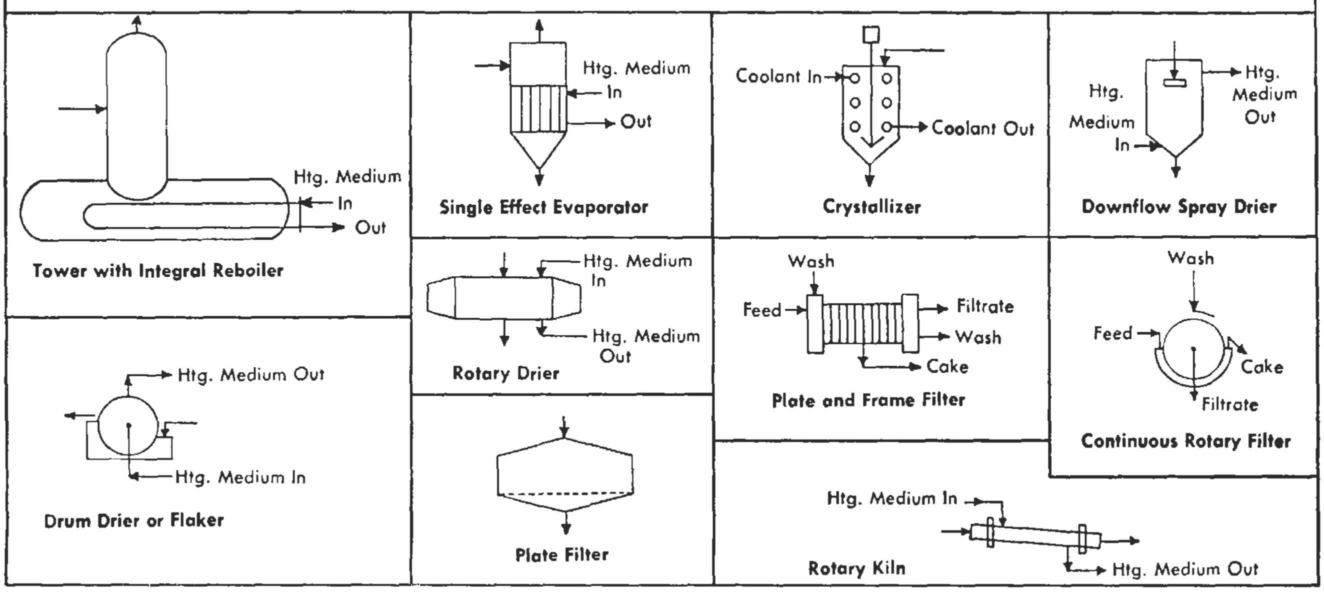
Figure 1-19D. Flow and instruments.



Material Balance			
Stream	Feed	Overhead	Bottoms
C_1	40	35	5
C_2	20	15	5
C_3	20	10	10
C_4	20	2	18
Total Lb.	100	62	38
Total Gal.	20	12	8

Feed	GPM
C_1	10
C_2	20
C_3	30
Total	60

Figure 1-19E. Filters, evaporators and driers.



By permission, D.J. Oriolo, *Oil and Gas Journal*, 56, Nov. 17, 1958, pp. 152-3.

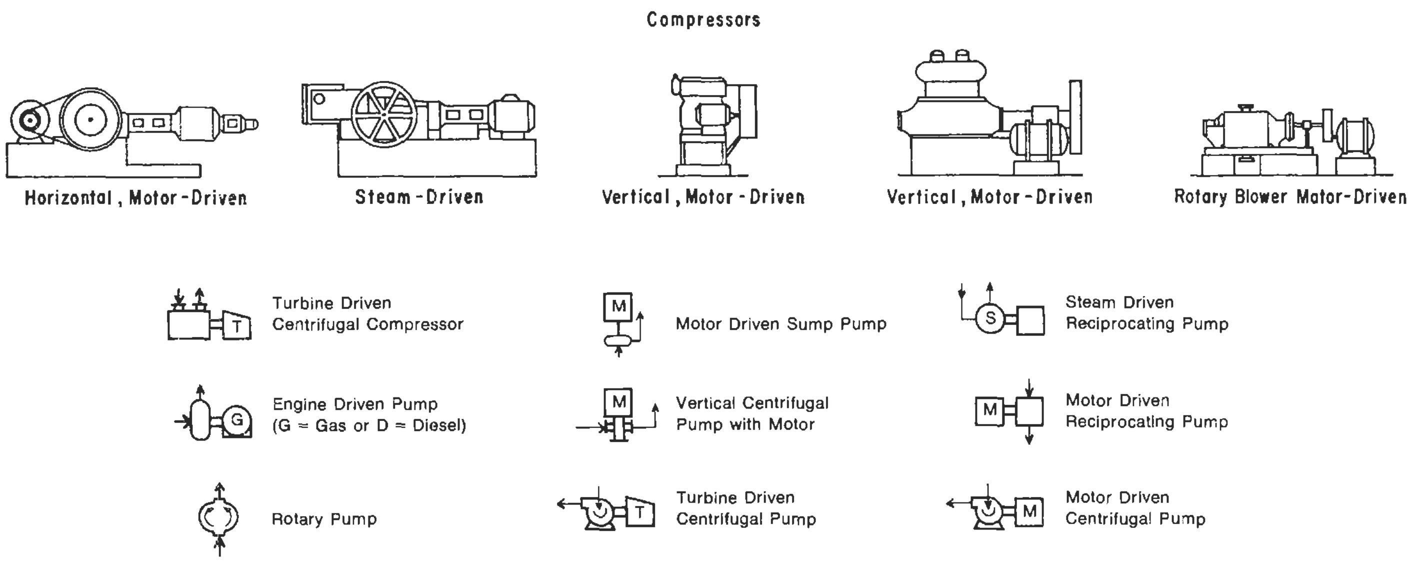


Figure 1-20A. Special types of descriptive flowsheet symbols.

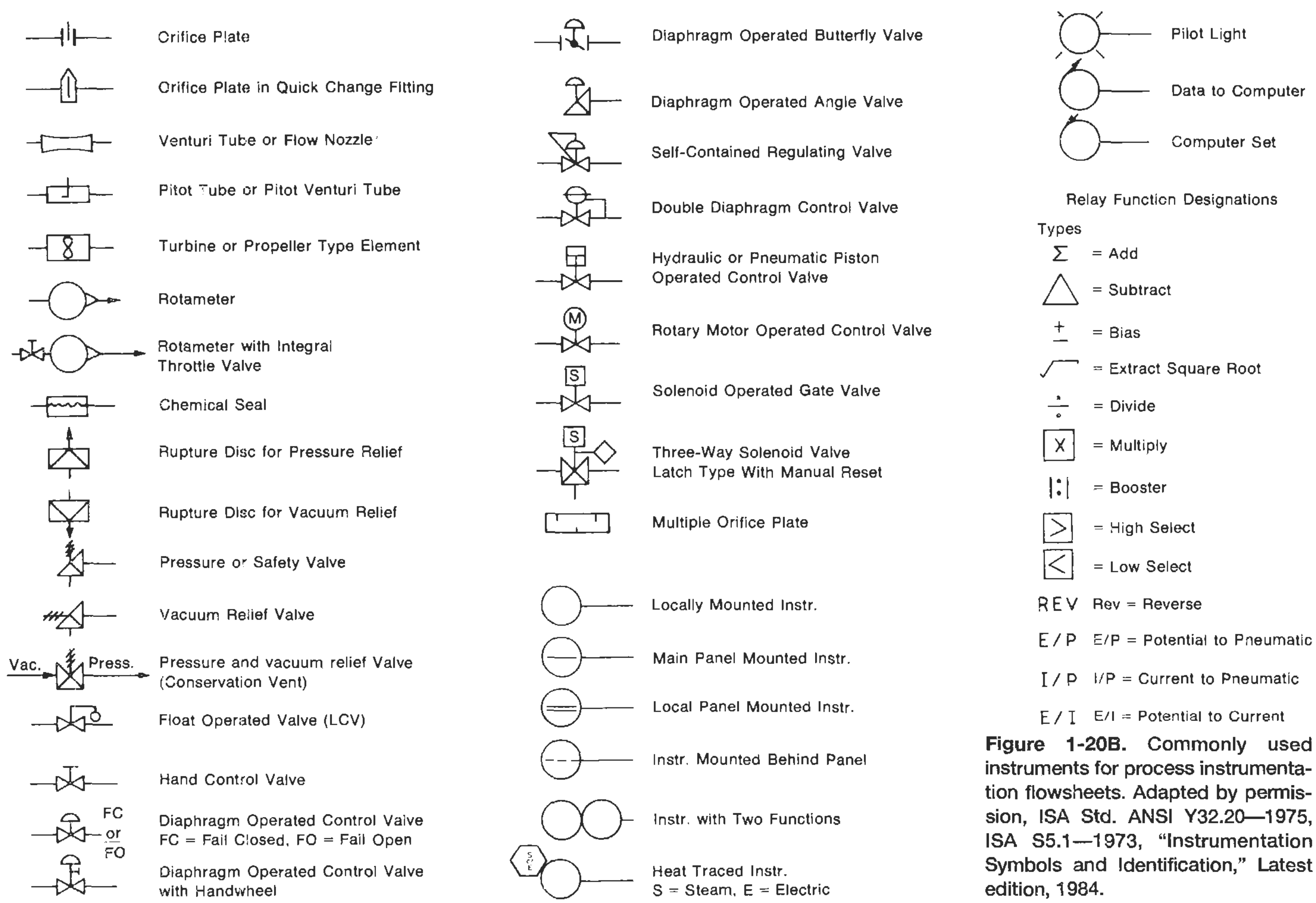


Figure 1-20B. Commonly used instruments for process instrumentation flowsheets. Adapted by permission, ISA Std. ANSI Y32.20—1975, ISA S5.1—1973, "Instrumentation Symbols and Identification," Latest edition, 1984.

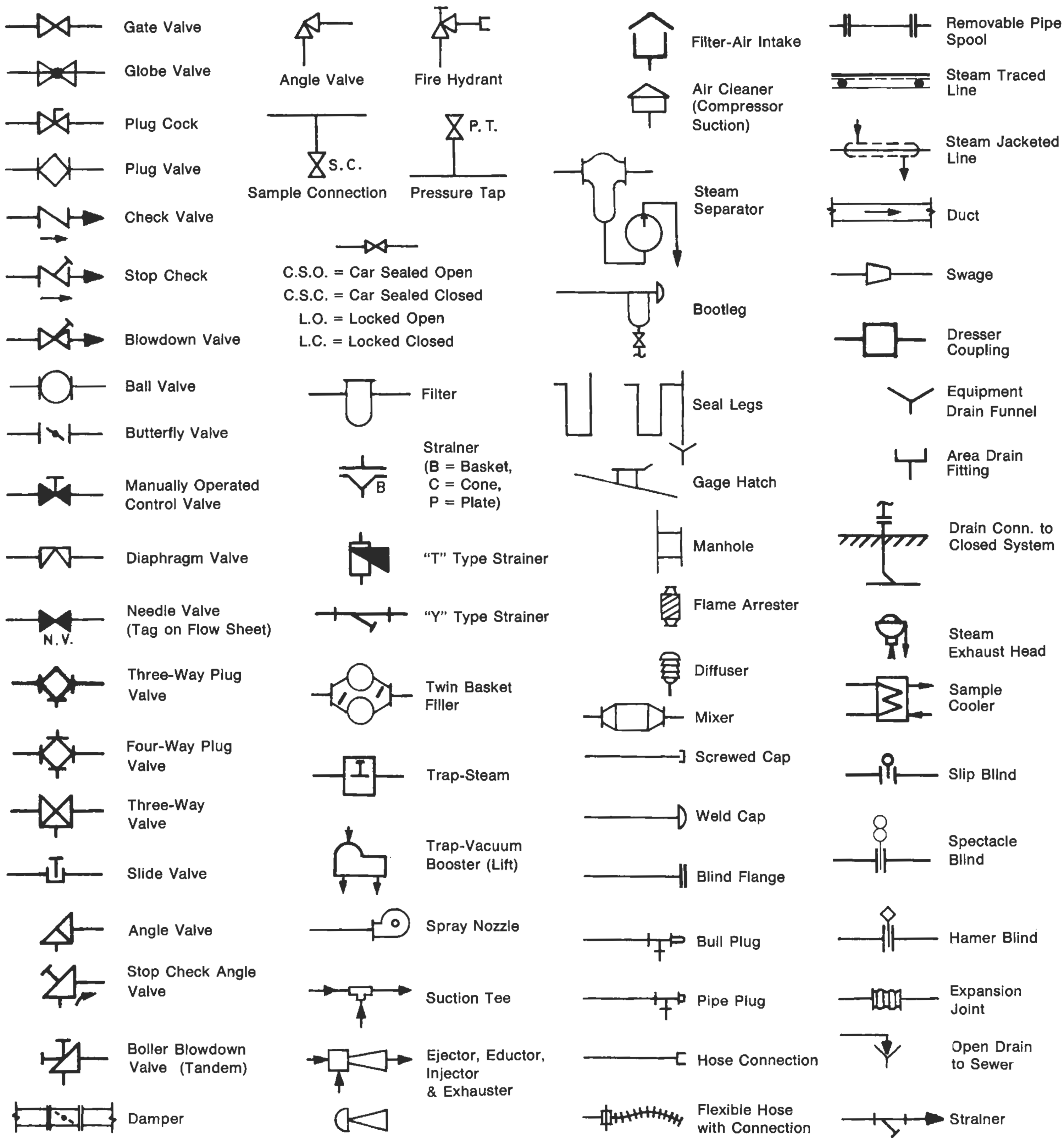


Figure 1-20C. Flow diagram symbols: valves, fittings and miscellaneous piping. (Compiled from several sources, and in particular, Fluor Corp, Ltd.)

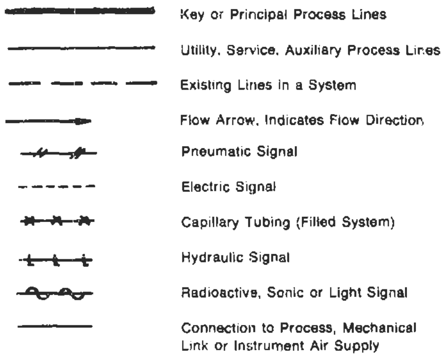


Figure 1-21. Line Symbols. By permission, ISA Std. S5.1—1973 and 1984.

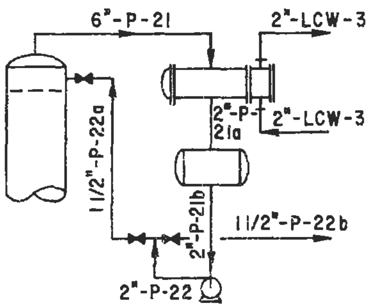


Figure 1-22. Use of alphabetical suffixes with line symbols.

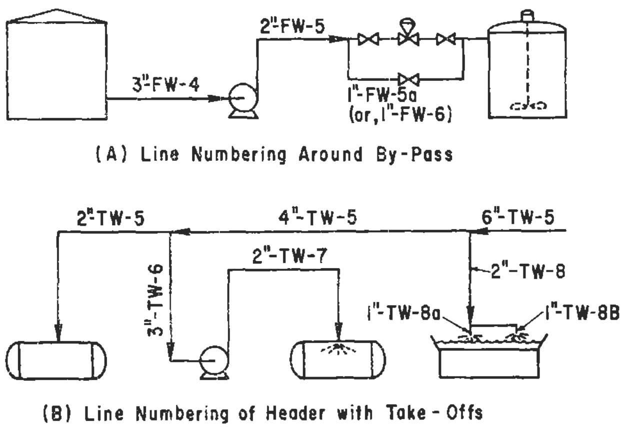


Figure 1-23. Examples of line numbering.

(text continued from page 18)

nation purposes and will appear on piping drawings, Line Schedule (Figure 1-24A through D), the number has no significance in itself. It is convenient to start numbering with the first process flow sheet and carry on sequentially to each succeeding sheet. Sometimes, however, this is not possible when several detailers are preparing different sheets, so each sheet can be given arbitrary beginning numbers such as 100, 300, 1000, etc. Although the sequential number may be changed as the line connects from equipment to equipment, it is often convenient to use the system concept and apply alphabetical suffixes to the sequence number as shown in Figures 1-22 and 1-23.

Line Schedule

Line No.	Size, In.	From	To	Line Class or Code	Insulation Code	Test Pressure, psig	Special Remarks

Figure 1-24A. Line Schedule.

This contributes materially to the readability of the flowsheets. Each line on the flowsheet must represent an actual section or run of piping in the final plant and on the piping drawings.

Suggested guides for line identification for any one principal fluid composition:

1. Main headers should keep one sequence number (Figure 1-23).
2. New sequence numbers should be assigned:
   (a) Upon entering and leaving an item of equipment
   (b) To take-off or branch lines from main headers
   (c) To structural material composition of line changes
3. Alphabetical suffixes should be used in the following situations as long as clarity of requirements is clear, otherwise add new sequence numbers.
   (a) For secondary branches from headers or header-branches
   (b) For by-pass lines around equipment, control valves, etc. Keep same sequence number as the inlet or upstream line (Figure 1-23).
   (c) For identical multiple systems, piping corresponding identical service items, and lines.

In order to coordinate the process flowsheet requirements with the mechanical piping specifications, Line Schedules are prepared as shown in Figure 1-24A through D. The complete pipe system specifications are summarized by codes on these schedules; refer to paragraph on Working Schedules.

Equipment code designations can be developed to suit the particular process, or as is customary a master coding can be established and followed for all projects. A suggested designation list (not all inclusive for all processes) for the usual process plant equipment is given in Table 1-2 and process functions in Table 1-3.

The various items are usually numbered by type and in process flow order as set forth on the flowsheets. For example:

Item Code	*Represents*
C—1a	Three compressors of identical size operating in the same process service, connected in parallel.
C—1b	
C—1c	

PIPE LINE LIST
THE FLUOR CORPORATION, LTD.

CUSTOMER ____________ SHEET NO. ____ OF ____ REV. ____
PLANT DESCR. ____________ CONTRACT NO. ____________
LOCATION ____________ BY ____________ DATE ____________

REV.	FLOW SH. NO.	LINE NO.	LINE CLASS	LINE SIZE	INSUL	COMMODITY	DESCRIPTION ORIGIN FROM	DESCRIPTION TERMINUS TO	PSIG MAX. OPER.	°F SERV. OPER.	
△											
△											
△											
△											
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Figure 1-24B. Pipe line List. By permission: Fluor Corp, Ltd.

Line Schedule Sheet

Line Number	Flow Medium	Quantity Flow, Gpm. or Scfh.	Quantity Flow, Lb./Hr.	Temp., °F.	Press., psi.	Density, Lb./Ft.^{3}	Density, Lb./Gal.	Sp. Gr. At 60 F.	Sp. Gr. At Flow Temp.	Expansion Coeff.	Chemical Formula	Mol. Wt.	Line Size	Press. Drop / 100 Ft.	Est. Line Length	Line Press. Drop	Line Origin	Line Termination

Figure 1-24C. Line schedule sheet (alternate). By permission, J. P. O'Donnell, *Chemical Engineer,* September 1957.

Line Summary Table

Title: Job No. Sheet No.

No.	Material	From	To	Line Size	Oper. Temp. °F	Oper. Press.	Test Press.	Quanity #/HR.	Quanity G.P.M. 60@	Quanity G.P.M. (Hot)	Quanity C.F.S. (Hot)	Vel. F.P.S.	Press. Drop Per 100'	Press. Drop Total	Remarks	No.

Figure 1-24D. Line summary table.

C—2	Single compressor in different service (by fluid *or* compression ratio) from C—1's above.
S—1	First separator in a process
S—2	Second separator in a process
S—3a	Two identical separators connected in
S—3b	parallel, in same process service.

Some equipment code systems number all items on first process flowsheet with 100 series, as C-101, C-102, P-106 to represent compressors number 101 and 102 in different services and pump 106 as the sixth pump on the sheet. The second sheet uses the 200 series, etc. This has some engineering convenience but is not always clear from the process view.

To keep process continuity clear, it is usually best to number all like items sequentially throughout the process, with no concern for which flowsheet they appear on. Also, another popular numbering arrangement is to identify a system such as reaction, drying, separation, purification, incineration, vent, and cooling tower waters and number all like process items within that system, for example:

Reactor System, R: Reactor is RD-1
Reactor vent cooler is RE-1
Reactor vent condenser is RE-2
Reactor recycle pump is RP-1
Level control valve is RLC-1
Relief valve is RSV-1

Then, establish the same concept for all other unit or block processing systems. This is often helpful for large projects, such as refinery or grass roots chemical processes.

Valve identification codes are usually used in preference to placing each valve specification on the flowsheet. This latter method is feasible for small systems, and is most workable when a given manufacturer (not necessarily the same manufacturer for all valves) can be selected and his valve catalog figure number used on the flowsheet. For large jobs, or where many projects are in progress at one time, it is common practice to establish valve specifications for the various process and utility services (see Figures 1-25 and 1-26) by manufacturers' catalog figure numbers. These are coded as V-11, V-12, V-13, etc., and such code numbers are used on the flowsheets wherever these valves

Table 1-2
A System of Equipment Designations

AD	Air Drier
AF	Air Filter
Ag	Agitator
B	Blower
BR	Barometric Refrigeration Unit
C	Compressor
CP	Car Puller
CT	Cooling Tower
CV	Conveyor
D	Drum or tank
DS	Desuperheater
E	Heat Exchanger, condenser, reboiler, etc.
Ej	Jet Ejector
Ex	Expansion Joint
F	Fan
FA	Flame Arrestor
Fi	Filter (line type, tank, centrifugal)
GT	Gas Turbine
MB	Motor for Blower
MC	Motor for Compressor
MF	Motor for Fan
MP	Motor for Pump
P	Pump
PH	Process Heater or Furnace
R	Reactor
S	Separator
St	Strainer
ST	Steam Turbine
Str	Steam trap
SV	Safety Valve
Tr	Trap
V	Valve
VRV	Vacuum Relief Valve

Table 1-3
Typical Identification for Flowsheet Process Functions

AS	Air Supply
BD	Blowdown
BF	Blind Flange
CBD	Continuous Blowdown
CD	Closed Drain
CH-O	Chain Operated
CSO	Car Seal Open
CSC	Car Seal Closed
DC	Drain Connection
EBD	Emerg. Blowdown Valve
ESD	Emerg. Shutdown
FC	Fail Closed
FO	Fail Open
HC	Hose Connection
IBD	Intermittent Blowdown
LO	Lock Open
ML	Manual Loading
NC	Normally Closed
NO	Normally Open
OD	Open Drain
P	Personnel Protection
QO	Quick Opening
SC	Sample Protection
SO	Steam Out
TSO	Tight Shut Off
VB	Vacuum Breaker

V-NO. MATL. SIZE 8 CONN.	CHECK VALVES DESCRIPTION		
V-11 1¼ Cr. 2½″–14″ 600# RF	PISTON LIFT, PRES. SEAL RATING: 600 psig @ 975°F BODY: C.A.S. A-217 GR WC6 STEM: SEATS: Integral Stel Alloy DISC: Body-Guided	ROCKWELL EDWARDS 690 WC6 ½″–6″	ROCKWELL EDWARDS 694 WC6 8″–14″
V-12 CS ½″–2″ 2500# SW	HORIZ. PISTON, WELD CAP RATING: 2500 psig @ 650°F BODY: C.S. A-216 Gr. WCB STEM: SEATS: Integral, Stellited DISC:	ROCKWELL EDWARDS 6674 ½″–2½″	VOGT SW-6933 ½″–2″
V-13 CS 2½″–12″ 2500# BW	HORIZ. PISTON, PRESS. SEAL RATING: 2500 @ 650°F BODY: C.S.A.-216 Gr. WCB STEM: SEATS: Integral, Stellited DISC: Piston Stellited	ROCKWELL EDWARDS 3994Y WCB 2½″–12″	POWELL 125065 WE 2½″–10″ (5) (6)
V	Add additional valves of all types as needed for project		

NOTE: 1. Vertical columns indicate valves acceptable as equivalent to the specification description.
2. V-11 is a typical valve code to use on flowsheets and piping drawings.

Figure 1-25. Typical valve codes and specifications. By permission, Borden Chemicals and Plastics Operating Limited Partnership.

are required. (Also see Figures 1-8 and 1-9.) By completely defining the valve specification in a separate specification book the various valves—gate, globe, butterfly, plug, flanged end, screwed end, welding end—can be identified for all persons involved on a project, including piping engineers and field erection contractors.

Figure 1-20C summarizes a system for representing piping components on the flow sheets.

The instrument symbols of Table 1-4 and Figures 1-23B and C are representative of the types developed by the Instrument Society of America and some companies.

Some other designation systems indicate the recording or indicating function in front of rather than behind the instrument function. For example:

RTC—1, Recording Temperature Controller No. 1

VRTC—1, Control Valve for Recording Temperature Controller No. 1

RFM—6, Recording Flow Meter No. 6

ORFM—6, Orifice flanges and plate for Recording Flow Meter No. 6

OTrRFC—1, Orifice flanges and plate used with Transmitter for Recording Flow Controller No. 1

TrRFC—1F, Flow Transmitter for Recording Flow controller No. 1

IPC—8, Indicating Pressure Controller No. 8

IFC—6, Indicating Flow Controller No. 6

IFM—2, Indicating Flow Meter No. 2

RLC— , Recording Level Controller

RLM— , Recording Level Meter

ILC— , Indicating Level Controller

LC— , Level Controller

PC— , Pressure Controller

Control valves carry the same designation as the instrument to which they are connected.

GENERAL
PIPING MATERIAL SPECIFICATIONS

GENERAL MATERIAL	:	**Carbon Steel**
MAXIMUM DESIGN PRESSURE and TEMPERATURE LIMITS	:	**275 PSIG at –20/100°F; 100 PSIG at 750°F**
LIMITED BY	:	**150# Flanges**
CORROSION ALLOWANCE	:	**See Table, This Spec.**
CONSTRUCTION	:	**1½″ and Smaller—Socket Welded** **2″ and Larger—Flanged and Butt-Welded**

TYPE	SIZE	DESCRIPTION
PIPE:	1½″ and smaller	Schedule 80, ASTM-A106 Gr. B Seamless P.E. (Plain End). Nipples: Sch. 80 ASTM-A106 Gr. B
	2″ through 10″	Schedule 40, Standard Weight, ASTM-A53 Gr. B, Seamless, B.E. (Bevel Ends)
	12″ through 24″	Standard Weight, (.375″) ASTM-A53, Gr. B, Seamless, B.E.
FITTINGS:	1½″ and smaller	3000# F.S., Socket Weld, ASTM-A105 Gr. I or II
	2″ through 10″	Schedule 40, Standard Weight, Butt-Weld ASTM-A234 Gr. WPB, Seamless
	12″ through 24″	Same Except Use Standard Weight (.375″)
BRANCHES:	Full	Use Tees
	Half header Size and larger	Straight Tee and Reducer or Reducing Tee
	Less half header Size down through 2″	Straight Tee and Reducer or Reducing Tee or Weldolets
	1½″ and smaller	Sockolets, Elbolets and Nipolets
FLANGES:	1½″ and smaller	150# ASA, 1/16″ R.F., Socket Weld ASTM-A181 Gr. I
	2″ and larger	150# ASA 1/16″ R.F. Weld Neck, ASTM-A181 Gr. I
UNIONS: (6)	1½″ and smaller	3000# F.S. Union ASTM-A105 Gr. II, Socket Weld ASA B16.11. Steel to Steel Seats, Ground Joint. No Bronze
BOLTING:	All	ASTM-A193 Gr. B7, Alloy Steel Stud Bolts, with ASTM-A194, Class 2H Heavy Series, Hex. Nuts
GASKETS:		1/16″ Thick, Compressed Asbestos Flat Ring Type. (JM 60 or Equal) 500°F and above, use Flexitallic CG.
THREAD LUBRICANT:	450°F and under	Use Teflon Tape
	Over 450°F	Use "Molycote" G Paste
GATE VALVES: (4)	1½″ and smaller	VGA-112, 800#, Socket Weld Ends, Welded Bonnet, F.S., ASTM-A105 Gr.II
	¾″ and smaller (1)	VGA-113, 800#, Screwed Ends, Welded Bonnet, F.S., ASTM-A105 Gr.II
	2″ and larger (2) (7)	VGA-101, 150#, Flanged O.S. & Y., Cast Steel Body, ASTM-A216 WCB
GLOBE VALVES: (4)	1½″ and smaller	VGL-215, 800#, Socket Weld Ends, Welded Bonnet, F.S., ASTM-105 Gr. II
	2″ through 12″ (7)	VGL-200, 150#, Flanged, O.S. & Y., Cast Steel Body, ASTM-A216 WCB
CHECK VALVES: (4)	1½″ and smaller (3)	VCH-314, 800#, Horizontal Piston Type Socket Weld Ends, F.S., ASTM-A105 Gr. II VCH-312, 800#, Combination Horizontal & Vertical Ball Type, Socket Weld Ends, F.S., ASTM-A105, Gr. II
	2″ through 16″	VCH-302, 150#, Horizontal Swing Check, Flanged, Cast Steel Body, ASTM-A216 WCB
DRAINS, VENTS and INSTRUMENTS: (4)	1″ and smaller	VGA-120, 800#, Male Socket Weld × Female Thread Ends, Welded Bonnet, F.S., ASTM-A105, Gr. II

Figure 1-26. Partial presentation of piping materials specifications for a specific process service. By permission, Borden Chemicals and Plastics, Operating Limited Partnership. *(Figure continued on next page)*

PIPING MATERIAL SPECIFICATIONS (continued)
Alternate Process Service

Press./Temp. Limits: 175 PSIG/−20 to 150°F 125 PSIG/350°F
Corrosion Allowance: 0.05 inches

ITEM	RATING & TYPE	MATERIAL OR MANUFACTURER	NOTE
SIZE INCHES			
PIPE			
2 and smaller	Sch. 40 Seamless	Carbon steel ASTM A-53, Gr. B, T&C	
3 through 6	Sch. 40 ERW	Carbon steel ASTM A-53, Gr. B, beveled	
8 through 12	Sch. 20 ERW	Carbon steel ASTM A-53, Gr. B, beveled	
14 through 20	Sch. 10 ERW	Carbon steel ASTM A-53, Gr. B, beveled	
FITTINGS			
2 and smaller	150# Screwed	Mal. iron ASTM A-197	
3 and larger	Buttweld-Sch. to match pipe	Carbon steel ASTM A-234, Gr. WPB.	
FLANGES			
2 and smaller	150# RF or FF Screwed	Carbon steel ASTM A-105	
3 and larger	150# RF or FF Slip-on or weld neck	Carbon steel ASTM A-105	
ORIFICE FLANGES			
1 and larger	300# RF Weld Neck	Carbon steel ASTM A-105	
UNIONS			
2 and smaller	300# Screwed	Mal. iron, ground joint, brass to iron seats ASTM A-197	
BRANCH CONN.			
2 and smaller	3000# Threadolet	Forged steel ASTM A-105	
3 and larger	Std. Wt. Weldolet	Forged steel ASTM A-105	
REDUCERS			
2 and smaller	150# Screwed	Mal. iron ASTM A-197	
	Sch. 80 Swage	Carbon steel ASTM A-234, Gr. WPB	
3 and larger	Buttweld-Sch. to match pipe	Carbon steel ASTM A-234, Gr. WPB	
STAINERS			
2 and smaller	150 screwed	Bronze with 30 mesh monel screen - Mueller #351 or equal.	
GASKETS			
All sizes	$\frac{1}{16}$ in. ring	Compressed	
All sizes	$\frac{1}{16}$ in. full face	Compressed	
BOLTING			
All sizes	Machine bolts w/ hex nuts	Sq. hd. ASTM A-307, Gr.B.	
VALVES (Alternate, for different process liquid/vapor service)			
2 and smaller	150# screwed gate	Bronze, ISRS, union bonnet, Powell 2714 or equal	
3 or larger	125# FF gate	IBBM, OS&Y, bolted bonnet, Powell 1793 or equal	
2 and smaller	300# screwed ball	CS body, Teflon seats & seals CS ball, Hills McCanna Fig. S302-CSTCS	
3 and larger	150# RF ball	CS body, Teflon seats & seals CS ball, Hills McCanna Fig. S151-CSTCS	
3 to 6	150# butterfly w/locking handle	Cast iron body, Buna N seat & seals, Al-Brz. disc, 316 SS stem. Keystone Fig. 100/122 or equal	

Figure 1-26 (continued). Partial presentation of piping specifications for a specific process service. By permission, Borden Chemicals and Plastics, Operating Limited partnership.

Table 1-4
Instrumentation Nomenclature—Complete General Identification*

MODIFICATION OF ISA STANDARDS

FIRST LETTER: * Process Variable or Actuation		SECOND AND THIRD LETTERS: Controlling Devices						Measuring Devices		Alarm Devices					
		Recording	Indicating	Nonindicating (Blind)	Valves	Self-Actuated Valve	Safety Valves	Recording	Indicating	Recording	Indicating	Nonindicating (Blind)	Primary Elements	Wells	Glass Devices for Visual Observation
		RC	IC	C	CV	V	SV	R	I	RA	IA	A	E	W	G
Flow	F	FRC	FIC			FV		FR	FI	FRA	FIA	FA	FE		FG
Level	L	LRC	LIC	LC	LCV	LV		LR	LI	LRA	LIA	LA	LE		LG
Pressure	P	PRC	PIC	PC	PCV	PV	PSV	PR	PI	PRA	PIA	PA	PE		
Speed	S	SRC	SIC	SC	SCV		SSV	SR	SI	SRA	SIA	SA			
Weight	W	WRC	WIC					WR	WI	WRA	WIA		WE		
Analysis	A	ARC	AIC	AC	ACV		ASV	AR	AI	ARA	AIA	AA	AE		
Hand	H		HIC	HC	HCV										
Temperature	T	TRC	TIC	TC	TCV	TV	TSV	TR	TI	TRA	TIA	TA	TE	TW	
Special	X	XRC	XIC	XC	XCV		XSV	XR	XI	XRA	XIA	XA	XE		

NOTE: Blank spaces are impossible or improbable combinations.

By permission, D. J. Oriolo, O. & G. Jour., Nov. 17, 1958; Also see ISA Stds. Latest edition.

Thermocouples carry the same designation as the recorder or indicator to which they are connected. Sequential point numbers are indicated thus (see Table 1-4):

RTM—6—4, Thermocouple connected to point No. 4 RTM instrument No. 6. Also see Figure 1-10.

Additional symbols include:

PG—6, Pressure Gage No. 6 connected in the field on some item of equipment. If panel board mounted, it becomes—6B.
LTA—1, Low Temperature Alarm No. 1
HTA—1, High Temperature Alarm No. 1
LPA—2, Low Pressure Alarm No. 2
HPA—2, High Pressure Alarm No. 2
LLA—6, Low Level Alarm No. 6
HLA—8, High Level Alarm No. 8
PG— , Push Button

Process flowsheets do not normally show companion flanges for valves unless these serve as blinds or for orifice plates. This detail is sometimes shown on the piping flowsheet, but here again the use of detail which does not contribute to the communication function of the sheets is avoided. Such detail can be time consuming when considered over the entire set of sheets for a process. Figures 1-8 and 1-9 are typical of reasonably good presentation without unnecessary detail. Such specifications as height of a seal leg, locked open valve, or other information not summarized elsewhere *must* be recorded on the flowsheets.

Working Schedules

As a direct companion of the completed flowsheet, the line schedule sheet transmits the process and mechanically necessary details for proper interpretation of the piping aspects of the flowsheet (see Figures 1-24A, B, C, D). These schedules are initiated by the process engineer to further explain the requirements of the process as shown on the flowsheets. They are often and perhaps usually cooperatively completed by other engineers, particularly the piping, mechanical and instrumentation groups.

Centrifugal Pump Summary													
Item No.	No. of Units	Service	Liquid	Oper. Temp. °F	Sp. Gr.	GPM	Avail. NPSH, Ft.	Discharge Press. PSIG	Speed RPM	BHP Pump	H. P. Driver	Driver Type	File Ref.
					Oper. Temp.	Oper. Temp.							

Figure 1-27. Centrifugal pumps summary.

Centrifugal Pump Schedule — Pump													
Item No.	No. Units	Service	Make	Model	Type	Size Connection		Weight	Reference Drawings	Purchase Order No.	Seal Type	Type Coupling
						Suction	Disch.					

Driver														
Item No.	Type	H. P.	RPM	Rotation, CW-CCW	Electrical Characteristics	NEMA Frame	Steam Conditions		Make	Model	Steam Rate	Weight	P.O. No.	
							PSIG	Temp. °F						

Figure 1-28. Centrifugal pump schedule.

A schedule similar to Figure 1-24A is used to summarize insulation process code or class, and pressure test information to the erection contractor. The process code is the complete code specification (as a separate fluid process service detailed for each fluid) tabulation for the required piping materials, fittings, valves, gaskets, thread lubricant, etc., for a specific process or utility fluid (see Figures 1-25 and 1-26.) For example, it identifies the type of gate, globe, plug, check and needle valves to be used in the fluid by specific catalog figure number of a manufacturer or its equivalent. This requires attention to materials of construction, pressure-temperature ratings, and connections (flanged, screwed, weld-end), bonnet type, packing, seat type (removable or nonremovable), stem, and any other details affecting the selection of a valve for the process fluid conditions. It also contains the specifications for pipe, fittings, flanges, unions, couplings, gaskets, thread compound, bolting and any special materials needed to properly complete the piping requirements.

Other schedules and summaries include vessels (tanks and drums), towers or columns, heat exchangers, pumps, compressors, motors, etc. These are often developed by the process engineer for organizational uses by the process designers as well as by other engineering groups. Again, these are often cooperatively and sometimes completely prepared by a particular specialty group after interpreting and designing for the needs of the process, see Figures 1-27, 1-28, 1-29, 1-30.

Two types of schedules are in use:

1. The summary sheet which summarizes process conditions and equipment selection
2. The schedule sheet which summarizes the key reference data for a particular class of equipment such as pumps, but contains no process data. The latter type is prepared for job coordination with and in the various departments, i.e., engineering, construction, purchasing, production. It primarily serves for the construction period but, of course, does have lasting cross-reference value.

From a construction viewpoint these summaries are a necessary check list to aid in keeping the construction program organized. Individuals who have no real knowledge of the scope of the job, and in particular the process, can properly tie the project together in the field by use of these schedules.

Information Checklists

The process engineer must summarize in some form the raw material and utility requirements for use by others. For example, the civil engineer is interested in waste water and sanitary sewer flows for proper layout studies. He is also in need of special requirements for site devel-

Vessel and Tank Summary

Item No.	Equipment	Type			Capacity	Thickness, ins.			Corrosion	Material		Pressure, psig			Temp., °F.		Code	Stamp	Reference Drawing No.	Notes
			O.D.	Ht. or Lnth.	Gallons	Shell	Head	Saddle Skirt Supp't.	Allow., inches	Shell & Heads	Saddle Skirt Supp't.	Oper'n.	Design	Test	Oper'n	Design				

Figure 1-29. Vessel and tank summary sheet.

Vessel and Tank Schedule

Item No.	No. Units	Service Description	Vertical or Horizontal	Diam.	Length	Thickness		Lining	Insulation Code	Test Pressure, psig	Weight	Ref. Dwg.	Purchase Order To	P.O. Number
						Head	Shell							

Figure 1-30. Vessel and tank schedule.

opment as well as railroads. The checklist of Figure 1-31 is an example of a helpful form. Others can be developed to suit the project or general plant situation.

For immediate job reference as well as for estimating requirements of a process for expansion purposes, the form shown in Figure 1-32 is convenient and can be expanded to suit the process under consideration.

Standards and Codes

The process design engineer must in effect become a good general purpose engineer who recognizes the need for integrating the various engineering disciplines into the process details as may be required. The engineer becomes what might be termed a pseudo-mechanical, corrosion, and metallurgical engineer as well as a basic chemical engineer. The design engineer must, or should soon, be knowledgeable of all types of information and specifications necessary to totally perform the process design functions in all detail and scope. A partial list of these specifications follows.

It is recommended that all pressure vessels and atmospheric vessels be designed, fabricated, tested, and code stamped according to the most applicable code as ASME or API, regardless of service application (nuclear is excluded from any discussion in these chapters):

- American Society of Mechanical Engineers (ASME) Unfired Pressure Vessel Code Section 8, Division 1
- ASME Code, Materials Specification, Part A, Ferrous Materials
- ASME Code, Materials Specification, Part B, Non-Ferrous Materials
- ASME Section V Non-Destruction Examination
- American Society for Testing Materials, Part 10, Annual Book of ASTM Standards: Metals-Physical, Mechanical and Corrosion Testing
- General Recommendations for Spacing in Refineries, Petrochemical Plants, Gasoline Plants, Terminals, Oil Pump Stations and Offshore Properties, Industrial Risk Insurance, Hartford, Conn. (See [19].)
- American Standards Association, Petroleum Refinery Piping ASA B31.3 (latest Edition)
- Standards of the Tubular Exchangers Manufacturers Association (TEMA Standards, latest edition)
- National Fire Protection Association as follows:

Standard	Code
Blower and Exhaust Systems	91
Chemical Reactions, Hazardous	491M
Chemical Data, Hazardous	49
Chimneys, Vents, Fireplaces, and Solid Fuel Burning Appliances	211
Coding, Uniform for Fire Protection	901
Dry Chemical Extinguishing Systems	17
Electrical Code, National	70
Electrical Equipment in Hazardous (Class.) Locations, Gases, Vapors, Dusts	497M
Electrical Equipment, Purged and Pressurized Enclosures for	496
Electrical Installations, Classification of Class 1	

(continued on next page)

Standards and Recommended Practices of American Petroleum Institute:

- 520 Design and Installation of Pressure-Relieving Systems in Refineries
  - Part I Design
  - Part II Installation
- 521 Guide for Pressure Relief of Depressuring Systems
- 525 Testing Procedures for Pressure Relieving Devices Discharging Against Variable Back Pressure
- 526 Flanged Steel Safety Relief Valves for Use in Petroleum Refineries
- 527 Commercial Seat Tightness of Safety Relief Valves with Metal-to-Metal Seats
- 540 Recommended Practice for Electrical Installations in Petroleum Refineries
- 550 Installation of Refinery Instruments and Control Systems
  - Part I Process Instrumentation Control
  - Part II Process Stream Analyzers
- 1101 Measurement of Petroleum Liquid Hydrocarbons by Positive Displacement Meter
- 2000 Venting Atmospheric and Low Pressure Storage tanks (Non-refrigerated and Refrigerated)
- 2545 Method of Gauging Petroleum and Petroleum Products
- 2217 Guidelines for Confined Space Work in the Petroleum Industry
- 2513 Evaporation Loss in the Petroleum Industry—Causes and Control
- 2516 Evaporation Loss from Low-Pressure Tanks
- 2517 Evaporation Loss from External Floating Roof Tanks
- 2518 Evaporation Loss from Fixed-Roof Tanks
- Chapter II Guide for Inspection of Refinery Equipment—Conditions Causing Deterioration or Failures
- Chapter IV Guide for Inspection of Refinery Equipment—Inspection Tools
- Chapter V Preparation of Equipment for Safe Entry and Work
- Chapter VI Pressure Vessels (Tower, Drums, and Reactors)
- Chapter VII Heat Exchangers, Condensers, and Cooler Boxes
- Chapter IX Fired/Heaters and Stacks
- Chapter IXX Atmospheric and Low Pressure Storage Tanks
- Chapter XIX Inspection for Accident Prevention
- Chapter XX Inspection for Fire Protection
- Std. 620 Recommended Rules for Design and Construction of Large, Welded, Low Pressure Storage Tanks
- RP-2003 Recommend Practice for Protection Against Ignitions Arising Out of Static, Lightning and Stray Currents
- 2521 Use of Pressure-Vacuum Vent Valves for Atmospheric Pressure Tanks to Reduce Evaporation Loss
- 2523 Petrochemical Evaporation Loss from Storage Tanks

Steel Structures Painting Council, Ref. SSPC-Vis 1-67

- No. 1 "Pictorial Surface Preparation Standards for Painting Steel Structures

Occupational Safety and Health Administration (OSHA) Regulations Environmental Protection Agency (EPA Regulatory section)

Metals Handbook, ASM International

- Volume 1 Properties and Selection: Irons and Steels (Latest Ed.)
- Volume 2 Properties and Selection: Nonferrous Alloys and Pure Metals (Latest Ed.)
- Volume 4 Heat Treating(1981)
- Volume 8 Mechanical Testing (1985)
- Volume 9 Metallography and Microstructures (1985)
- Volume 11 Failure Analysis and Prevention (1986)
- Volume 13 Corrosion (Latest Ed.)
- Volume 17 Nondestruction Evaluation and Quality Control (1989)

Instrument Society of America, Standards and Practices

- RP1.1-7 Thermocouples and Thermocouple Extension Wires
- RP3.1 Flowmeter Installations. Seal and Condensate Chambers
- RP3.2 Flange Mounted Sharp Edged Orifice Plates for Flow Measurement

- RP4.1 Uniform Face-to-Face Dimensions for Flanged Control Valve Bodies
- RP4.2 Standard Control Valve Manifold Designs
- S5.1 Instrumentation Flow Plan Symbols
- RP7.1 Pneumatic Control Circuit Pressure Test
- RP7.2 Color Code for Panel Tubing
- RP8.1 Instrument Enclosures for Industrial Environments
- RP12.1 Electrical Instruments in Hazardous Atmospheres
- RP12.2 Intrinsically Safe and Non-Incendive Electrical Instruments
- S12.4 Instrument Purging for Reduction of Hazardous Area Classification
- RP18.1 Specifications and Guides for the Use of General Purpose Annunciators
- RP20.1, and 20.2 Specification Forms for Instruments

Federal Safety Standards for Pipelines; Part 195-Transportation of liquids by Pipelines.

Often the process design engineer will become involved in managing a project, especially if he/she designed the specifications for fabrication and purchase of the equipment for the project. It is necessary that the process engineer participate in equipment layout/arrangement decisions for the early stages of the plant development. With all this background, the process engineer is the logical person to handle or coordinate the interrelationships of the various engineering disciplines and to review and evaluate the equipment purchase proposals from the purchasing department. The role of a project engineer often grows from the process design engineer's responsibilities (see [19]).

The process engineer should be responsible for understanding the following regulations:

1. Occupational Safety and Health Administration regulations as they relate to (a) safety of design related to injury to personnel (includes such matters as latest vessel design [53], noise level from operating equipment, etc., [20, 21, 22, 23, 24, 25, 26, 27, 28]. (b) safety of the plant layout environment which might influence the safety of the plant facilities.
2. Environmental Protection Agency regulations related to air, water, solid waste, and land contamination with toxic substances that a plant might emit/release into immediate plant area, or discharge as waste into public streams, or inject into underground aquafiers, or dump or store [29, 30, 31].

Although the U.S. chemical industry is committed to converting from American Engineering Standard units to the metric standards, or SI units, the actual progress in this conversion has been slow. This is primarily due to the fact that engineers are more familiar with the "more practical" engineering units and also few text books using SI units are available. The conversion in the industry is awkward and confusing because there is no "feel" for the practical meaning of the SI terms.

System Design Pressures

In order to coordinate the design pressures for the various vessels in a given process system, it is necessary to establish the relationship between the operating and design conditions. Figure 1-33 and Tables 1-5 and 1-6 are guides to setting the percentage for the design pressure over the operating pressure. This type of relationship can be established according to the preferences of the responsible engineer or company policy. In the range near atmospheric pressure the preferences vary, however, for

Table 1-5
Suggested System Design Pressures
(Based upon condensation at 100°F with 10°F approach)

System	Design Pressure, psig
Freon — 11 (or equivalent)	50
Freon — 12 (or equivalent)	200
Freon — 22 (or equivalent)	300
Ammonia	250
Chlorine	300

Table 1-6
Suggested Maximum Operating Pressure

System	Usual Condition of Maximum Operating Pressure
Refrigeration Systems	Refrigerant vapor pressure at temperature 10°-15° F. above condensing water.
Storage Vessels	Vapor pressure of liquid at maximum ambient temperature plus 30° F. (usually 110° to 140° F.)
Process Vessels	Depends upon operating conditions, surge conditions, insulation. toxicity, explosion hazard, etc.
A. In a compressor or pump system:	
(1) Centrifugal Type	Shut-off pressure plus 5 psi
(2) Reciprocating Type	Normal operating pressure plus 15 psi for low pressures to plus 50 psi for 200-3000 psi system.
B. Direct Injection of Steam, Air, Methane, Cooling Water, etc.	Supply line pressure plus 5 psi to 15 psi.

GENERAL SERVICES AND UTILITIES CHECK LIST

NOTE: ONE COPY OF THIS LIST TO BE GIVEN TO SERVICE AND UTILITY SECTION

JOB NO. ____________ PREPARED BY: ____________

CHG. NO. ____________ DATE: ____________

JOB TITLE:

____________ LOCATION: ____________

____

SERVICE	QUANTITY SUSTAINED - - PEAK -	UNIT	OPERATING PRESSURE & TEMPERATURE	REMARKS
ELECTRICAL 13.8 KV		*KVA		
ELECTRICAL 2300 V		*KVA		
ELECTRICAL 440 V		*KVA		
MISC. LIGHTING, ST. ETC.,		KVA		
(110 V)				
WELL WATER (CITY)		GPM		
SEA WATER		GPM		
RIVER WATER - PROCESS		GPM		
RIVER WATER-FIRE PROT.		GPM		
FUEL GAS		SCFM		
ODORIZED GAS		SCFM		
COMPRESSED AIR		SCFM		
STEAM		LB/HR		
(400 OR 475) PSIG		LB/HR		
235 PSIG		LB/HR		
150 PSIG		LB/HR		
30 PSIG		LB/HR		
CONDENSATE RETURN		LB/HR		
CONDENSATE USAGE		LB/HR		
PROCESS TRANSFER & RAW MATERIALS				
CAUSTIC (-%)		LB/HR		
BRINE (-%)		LB/HR		
LPG		LB/HR		
SITE DEV. (AREA FILL)				
RAILROADS				
BLOCK ACCESS ROADS				
SANITARY SEWERS		GPM		
WASTE WATER		GPM		

*ASSUME 1 HORSEPOWER = 1 KVA

Figure 1-31. General services and utilities checklist.

small diameter (less than 8 feet) vessels operating in a definite pressure system. Thus, the effect of a reasonable overpressure for design (as suggested by Figure 1-33) on the vessel wall thickness is usually negligible.

For the larger diameter storage vessels operating with a few ounces water to 1 psig, the selection of a design pressure must also consider the system surges in relation to the normal conditions. For example, a storage tank 20

PROCESS ENGINEERING JOB ANALYSIS SUMMARY

Job Title ____________________

Job No. __________ Charge No. __________ Date __________

Based Upon Cost Estimate Dated __________ or Actual Construction Cost

Summary Prepared By __________ Information Dated __________

Production Basis (lbs./day, tons/day, lbs./month) __________

Service Requirements:		Unit Rate	Unit Rate/ Production Basis
1. Steam (30 lbs.)	______	lbs./hr.	______
2. Steam (150 lbs.)	______	lbs./hr.	______
3. Steam (400 lbs.)	______	lbs./hr.	______
4. Steam (lbs.)	______	lbs./hr.	______
5. Treated R.W.	______	gpm	______
6. Untreated R.W.	______	gpm	______
7. Fresh Water	______	gpm	______
8. Sea Water	______	gpm	______
9. Fuel Gas (psi)	______	cfm (60°F & 1 atm.)	______
10. Air (psi)	______	cfm (60°F & 1 atm.)	______
11. Power ( )	______		______
12. Horsepower	______		______
13. Condensate	______	lbs./hr.	______
14.	______		______

Raw Materials:		Unit Rate	
1. Chlorine	______		______
2. Hydrogen (%)	______		______
3. Caustic (%)	______		______
4. Salt	______		______
5. Sat. Brine	______		______
6. Natural Gas	______		______
7. Air	______		______
8. Ethylene	______		______
9.	______		______
10.	______		______
11.	______		______

Products and By-Products:		Unit Rate	
1. Chlorine	______		______
2. HCl (%)	______		______
3. Salt (%)	______		______
4. Caustic (%)	______		______
5. Ammonia (%)	______		______
6. H_2SO_4 (%)	______		______
7. Gas ( )	______		______
8.	______		______
9.	______		______
10.	______		______
11.	______		______

Figure 1-32. Process engineering job analysis summary.

feet in diameter which will operate under 6 oz. water might be designed for 12 oz. while arbitrarily selecting a design pressure of 10 psig would be quite uneconomical. A 40-foot diameter tank for atmospheric storage would normally be designed for 2 to 3 oz. of water. (See previous listing of API codes and ASME codes.) The bottom shell of a 40′ diameter × 40′ tall vessel must withstand the greater pressure of the height of water or process liquid when the vessel is full to the vents. For a 6-foot diameter vessel operating at 3 psig, a reasonable design pressure might be 10 psig.

In some low pressure processes it is good practice to set a minimum design pressure of 10 psig or 25 psig for all vessels operating below 5 psig and no larger in diameter than 8 to 10 feet. The minimum design pressures for a vessel will be influenced by the fact that the minimum vessel wall thickness for carbon steel is usually $\frac{3}{16}$ inch to $\frac{1}{4}$ inch. Economics of the situation dictate where the cutoff pressures and/or diameters lie, as these will vary with the type of metal under consideration.

Vessels operating below atmospheric pressure are designed for *full vacuum* regardless of the actual vacuum.

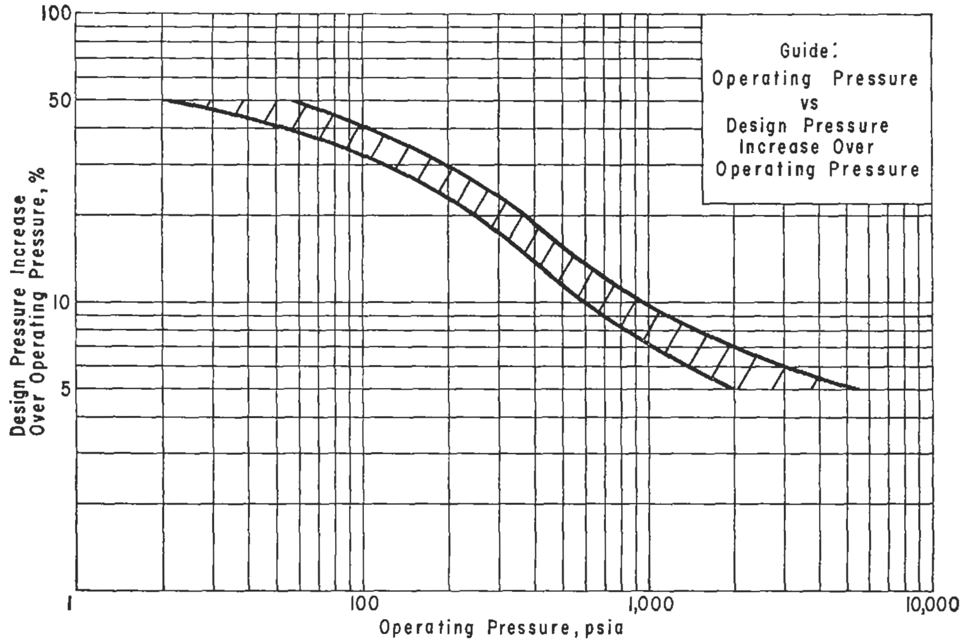

Figure 1-33. Guide: Operating pressure vs. design pressure increase over operating pressure.

If it is extremely uneconomical to design at this point, then proper vacuum control must be installed. However, this is not the usual approach to the design. If the equipment can operate alternately under vacuum or positive pressure, it must be designed for the worst or controlling condition.

Time Planning and Scheduling

Scheduling of work in process engineering or design is a near impossibility as far as pin-point accuracy is concerned. The very developmental and planning nature of the early phases, as well as the continuous follow-through and follow-up, make this difficult. It is seldom that one can foresee specific changes, delays, etc. Very few projects are clear-cut and well defined ("frozen") as to scope or design conditions except for small jobs and repeat or duplicate projects.

With new processes and/or products, the collection of physical data (either from pilot or laboratory operations, or from the literature), consideration and evaluation of alternate conditions and flow schemes with the corresponding decisions, often become a significant portion of the time required to complete the actual process calculations and preparation of design specifications. So that this early phase of work does not unnecessarily slow down the project, it is important that close guidance and supervision be given the individual designers and the use of experience, judgment and approximations be encouraged. In this way many unnecessarily detailed or time consuming calculations can be avoided, or reduced to a reasonable minimum.

On the other hand there are many situations which require the detailed work before a sound decision can be made. In addition, it is often necessary to obtain reasonably accurate prices for various items of equipment and their assembly before the final decision can be made.

For groups specializing in this type of design work it well to maintain records of the time requirements, job conditions, etc., in order to build a history upon which to base future estimating. It will be recognized that no two projects or problems are exactly alike. However, with time certain basic similarities can be recognized, with good judgment these records can be used to advantage. Thus, average information can have some value.

The size of a project does not always have a significant bearing on the schedule. Weighted judgment, taking the type of job, type of process, and type and nature of the men with the engineering and process responsibility into account is necessary to align a balanced and smooth working team.

Activity Analysis

A time study of eight graduate process engineers with a minimum of five years experience is shown in Tables 1-7 and 1-8. The time includes process calculations, preparation of specifications, discussions with vendors and handling the complete scope of small and large projects and is helpful in accounting for legitimate time which was obviously not spent in performing process calculations.

Table 1-7
Time Study

Activity of Engineers	Percent of Time: Single Study	Percent of Time: Avg. Range
Process design calculations	34.69	35—52
Conferences, consultation, unscheduled urgent assignments, information assembly	28.98	13—29
Supervision and administrative, including time schedules, discussions with salesmen, preparation of outside correspondence	4.45	4—15
Preparation of charts, forms, methods for benefit of over-all group	1.95	1—3
Marking, checking, and reviewing flow sheets (no drafting)	10.94	9—12
Group meetings, training periods, over-all department and company development	1.80	1—3
Literature review (current magazines, etc.)	1.80	0.5—2
Coffee breaks, etc.	5.55	4—6
Unaccounted, including vacation	9.84	5—10

Table 1-8
Time Study

Activity of Engineers	Percentage of Time
Consulting outside of scheduled jobs	4.4
Section supervision duties	4.7
Meetings related to scheduled jobs	13.7
Discussions with vendors	2.6
Special technical assignments	2.4
Communications within section	5.9
Process design calculations (original)	51.0
Process design calculations (checking)	3.7
Equipment schedules, line schedules, etc.	3.1
Flow sheet development, checking, revising (no drafting)	2.5
Coffee breaks, miscellaneous activity	6.0
	100

This does not include total project coordination or project engineering. (For expanded reference also see [51].)

It should be recognized that the data in these tables may not necessarily fit other situations; however, it can serve as a guide. Since it is based upon engineers associated with an engineering department located at an operating company plant site, there is a basic difference in contacts, availability of production experience, and perhaps even philosophy between this type of group and one centered at an engineering office remote from plant contacts. The interruptions and requirements for data and results although similar in many respects are certainly different in other respects. The use of this type of activity information will be combined with detailed calculation data and discussed later.

Collection and Assembly of Physical Property Data

An important but time-consuming factor in practically every design situation and in development of flowsheets is the collection and assembly of physical property data for the components of the system in question. Often it is not sufficient to obtain single data points from various tables, since many designs cover rather wide ranges of temperature and pressure and the effects of these on the properties must be taken into account.

Data may be located in many useful handbooks as well as published technical papers and company compilations. However, experience indicates that extensive literature searches may be necessary to locate specific data on a particular compound. It is surprising to find so many common compounds for which the data is incomplete and sometimes inaccurate. Empirical correlations must often be utilized, sometimes to generate a value and sometimes to check a questionable literature value.

Therefore, when developing an estimate of process engineering time required, it is important to recognize the amount of effort that may be necessary to collect physical property data before any real work can commence. This same concern exists when evaluating K values and activity data for systems.

Estimated Equipment Calculation Man-Hours

The required man-hours for a specific calculation vary with the process system, availability of physical data, and the relative familiarity of the process design engineer. Records collected over a period of years on a wide cross-section of organic and inorganic process equipment calculations are summarized in Table 1-9. It is impossible to accurately define the limits of the calculations represented, but on an average, they have been found to be helpful in establishing the *order of magnitude* of the calculation time, as well as the basis for approximating the over-all extent of the process engineering of the project.

Electronic computers, both digital and analog, can be used to great advantage in design studies and calculations. In evaluating reactor designs it is extremely helpful to develop a family of performance curves for variables involved in the system. Usually this type of calculation becomes too time consuming with the desk electronic calculator, and is a good problem for the computer.

After investing time and talent into a program for the computer, it is usually only a matter of minutes or hours before a complete series of results can be calculated.

Table 1-9
Estimated Man-Hours Required for Equipment Design [7] (updated)

Type of Equipment	**Design*	***Computer*	*□Checking*	*Total M-H*
HEAT EXCHANGERS:				
Solvent cooler	30	—	3	33
Tank heating coil	4	—	2	6
Caustic cross exchanger	32	(1)	6	38
Caustic cooler	8	(2)	2	10 (2)
Oil cross exchanger	32	(3)	5	37 (3)
Gas cooler	8	(3)	4	12 (3)
Compressor gas aftercooler	8	(2)	1	9 (2)
Slurry cooler	32	(4)	8	40 (4)
Finned tube exchanger	15	—	4	19
Gas cooler	4	(1)	1	5 (1)
CONDENSERS:				
Steam	7	(2)	—	7 (2)
Organic	6	(2)	2	8 (2)
HCl organic	10	(5)	11	21 (5)
Organic	6	(2)	2	8 (2)
Finishing	4	(1.5)	1.5	5.5 (1.5)
PARTIAL CONDENSERS:				
Organic—air	10	(3)	2	12 (3)
Organic—air	20	(3)	4	24 (3)
Organic—air	30	(4)	14	44 (4)
Inorganic—air	50	(4)	20	70 (4)
REBOILERS (THERMOSIPHON):				
Organic—steam	16	—	—	16
Organic—steam	20	(3)	5	25 (3)
Organic—steam	14	(3)	—	14 (3)
Organic—steam	10	(3)	5	15 (3)
Organic—steam	16	(3)	6	22 (3)
Organic—steam	14	(3)	5	24 (3)
Organic—steam	4	(0.5)	—	4.5 (0.5)
Organic—steam	5	(1)	—	6 (1)
Organic—steam	4	(1)	—	5 (1)
Organic—steam	5	(0.5)	1	6 (0.5)
Organic—steam	5	(0.5)	1	6 (0.5)
REBOILERS (FORCED CIRCULATION):				
Organic—steam	25	(4)	10	35 (4)
Organic—steam	19	(4)	8	27 (4)
Organic—steam	6	(1)	3	9 (1)
Organic—steam	6	(2)	3	9 (2)
DISTILLATION (PACKED):				
Carbonating tower	25	(4)	10.5	35.5 (4)
Gas cooler	20	(4)	8	28 (4)
Gas cooler	25	(6)	7	32 (6)
Cooling	16	(5)	22	38 (5)
Gas scrubber	24	(6)	4	28 (6)
Gas scrubber	12	(6)	8	20 (6)
Vent gas scrubber	5	(2)	1.5	6.5 (2)
DISTILLATION (TRAY):				
Organic—tray-by-tray	50	(12)	25	75 (12)
Organic—tray-by-tray	40	(12)	64	104 (12)
Demethanizer—tray-by-tray	31	(15)	22	53 (15)
Organic—tray-by-tray	35	(7)	4	46 (7)
Organic—tray-by-tray	10	(5)	5	15 (5)
Organic—tray-by-tray	5	(6)	3	11 (6)
Organic—tray-by-tray	2	(2)	2	4 (2)
De-ethanizer	24	(12)	15	39 (12)
Demethanizer	30	(15)	15	45 (15)
Organic—includes tray layout	24	(15)	28	52 (15)
Organic—includes tray layout	38	(20)	10	48 (20)
PUMPS				
System	8	(2)	6	14 (2)
Single	1.5	(2)	0.5	2 (2)
Single	1	(1)	1	2 (1)
Single	3	(1)	3	6 (1)
RECIPROCATING COMPRESSOR:				
BHP, temperature and	3	—	—	3
Data for vendor rating	6	—	2	8
CENTRIFUGAL COMPRESSOR: About the same as Reciprocating above.				
PROCESS LINE SIZES:				
Single	1	(0.5)	0.5	1.5 (0.5)
Single	0.5	(0.5)	0.5	1 (0.5)
System, 22 lines	20	(3)	9	29 (3)
Air header for plant	4.5	(3)	2	6.5 (3)
SAFETY VALVES:				
Single	2	—	2	4
Single	1	—	1	2
STEAM TRAPS:				
System of 4	3	—	1	4
Single	0.75	—	0.25	1
Single	1	—	1	2
MISCELLANEOUS TANKS, DRUMS, ETC.:				
Condensate level drum	0.5	—	0.5	1
Steam flash drum	6	—	3	9
Storage tank	2	—	—	2
MATERIAL BALANCES: Depends on size of system.				

Note: The man-hours listed in this table included collection of needed physical and other data and preparation of a specification or summary of the requirements.

*Using desk electronic calculators, not programmed.
**Programmed computer. Represents data input plus calculation time, sometimes multiple.
□ Checking only for "Design" calculations.
( ) Alternate calculations by programmed computer.

Computers are quite adaptable to the following calculations: distillation tray-by-tray and short-cut methods; tray hydraulics for bubble cap, sieve or perforated and "dual-flow"; absorption, heat exchange including condensation, partial condensation, cooler-condensing, reboiling; drying; compression; equilibrium flash; fluid flow including two phase and many others. It is important to remember that good results cannot be obtained from a poor or inadequate computer program. Thus, it is wise to invest the effort into the development of basically sound general purpose programs. With these many variations can be arranged to suit the special case. In order to have confidence in the results of any computer program (whether self-developed or purchased) it must be tested against extreme conditions or limits. To purchase and use a program without testing is inviting errors.

Some programs require only a few days to completely program for general purpose use, while some others require several months of continuous effort. Whenever more than one individual is expected to use the computer program, it is good practice to obtain the several views on attacking the problem, i.e., type of input data, solution approach, range of variables, fixed conditions and type and form of output or results.

Table 1-10 illustrates some reasonable time requirements for solution of problems or designs when using a medium-sized digital computer, using existing programs. A very high speed machine might reduce the pure calculation time to a matter of minutes; however, the time required for (1) data collection (specific problem conditions as well as physical data, (2) data input to the computer, and (3) evaluation of results and preparation of design specification sheets all remain essentially fixed. In some situations the complexity of the calculations requires the capabilities of the large machines, and in these cases the time advantage can be the difference between a good result and none at all. Total plant material and heat balances are a good example.

Estimated Total Process Man-Hours

After the man-hours have been estimated for all of the individual items of equipment on the project, a guide to total man-hours is:

$$\text{Total Estimated Job Process Engineering Man-Hours} = \frac{\text{Estimated Equipment man-hours (including checking)}}{0.45}$$

This applies to work where at least 50 percent of the time is by electronic desk calculator for the numerical calculations. It may not apply well to projects of less than 200 process man hours.

Table 1-10
Calculation Time Using Medium Size Digital Computer

Calculation	Hours Total Elapsed Time Preparation + Calculation
Preliminary Distillation	
Number of trays, reflux ratio	1-3
Tray-by-Tray Distillation	
To 40 trays	2-3
To 100 trays	3-5
Tray Hydraulics	
Bubble cap, sieve, perforated	1-3
Heat Exchangers	
Condensers, exchangers	0.5-1.0
Separators	0.5
Flash Vaporization	0.5
Oil Absorbers	1-3
Safety Valves	0.25

When a limited time is available to complete a project, this may be used to determine the estimates of manpower:

$$\text{Average number of men} = \frac{\text{Estimated man-hours (process)}}{\text{(Total elapsed weeks) (30 to 33)}}$$

Where: 30 to 33 represents the actual usable job-related man-hours per 40-hour week per man, allowing for average sickness, vacation, jury duty, etc.

Approximate maximum number required = (1.67) (Avg. number of engineers)

Example 1-1: Man-Hour Evaluation

From an examination of the process flowsheet the man-hours total 685* for the significant equipment. Items such as steam traps and miscellaneous small time-items can be omitted from the total. *Includes 75 man-hours for pipeline sizing.

$$\text{Total Estimated Job Man-Hours} = \frac{685}{0.45} = 1525$$

If the work must be complete, including flowsheet supervision, etc., in three weeks:

$$\text{Average no. engineers required} = \frac{1525}{(3)\ (32)} = 15.9$$

This is impractical since a job of this magnitude cannot be planned and decisions reached in this time. Therefore,

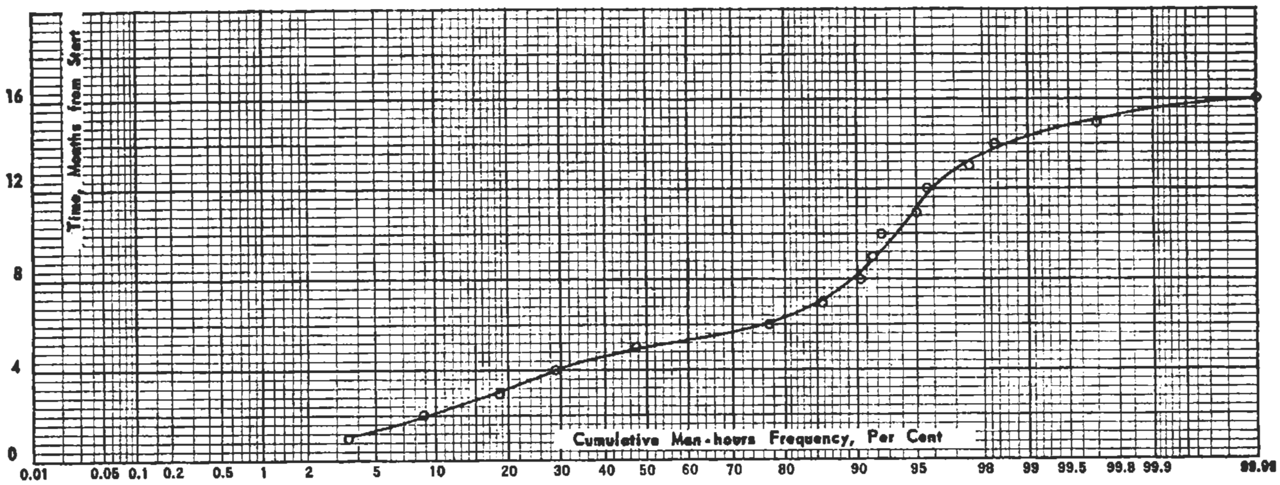

Figure 1-34A. Process engineering manhours accumulation pattern: Project A. By permission, E. E. Ludwig [7].

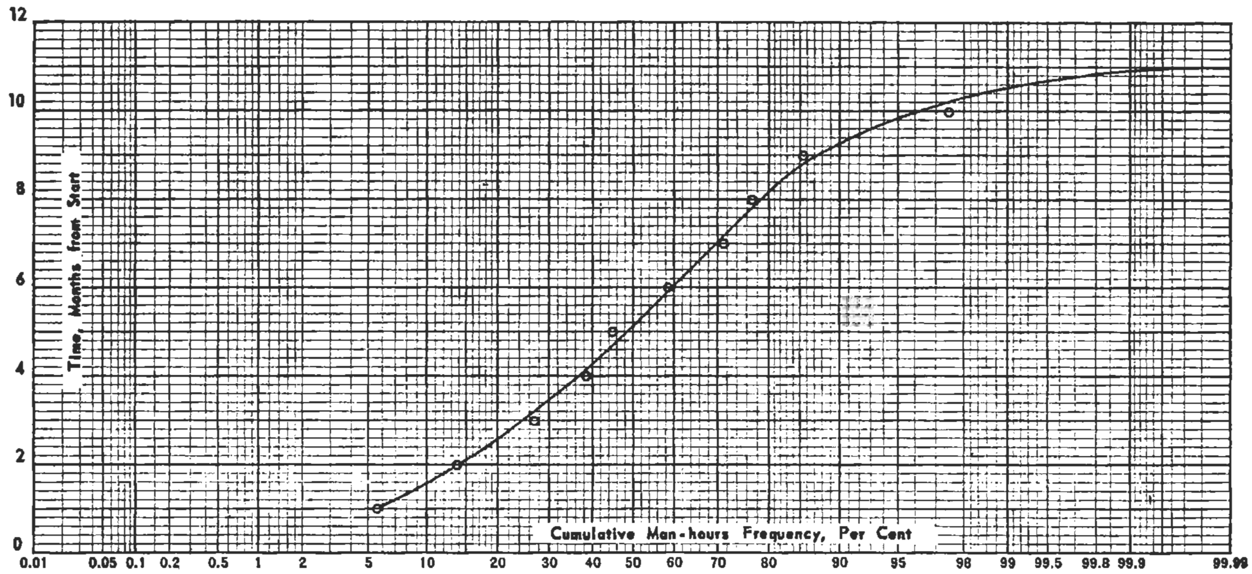

Figure 1-34B. Process engineering manhours accumulation pattern: Project B. By permission, E. E. Ludwig [7].

the men could not be kept busy. It will be necessary to spread out the time, using fewer engineers.

For a twelve weeks program:

$$\text{Average no. engineers required} = \frac{1525}{(12)\ (32)} = 3.97 \cong 4$$

$$\text{Peak man power} \cong 3.97 \times 1.67 \cong 6.6, \text{ use 7 men}$$

Near peak manpower requirements will be needed from 30 to 50 percent of the total time schedule, unless other factors influence the timing.

Typical Man-Hour Patterns

Figures 1-34, A, B, C illustrate accumulation patterns for the process engineering man-hours of a few typical projects. In general the smaller the project and the better defined the scope, the more the pattern of Project B is approached. Projects A or C represent the larger projects where there may be changes in plant capacity or location, as well as a concurrent pilot plant research program to continually obtain a better answer. The slow-down portions of the curves can be accounted for as significant changes in the process or process-related factors. In general, most large (six months or longer) process engineering projects undergo significant changes by the time 50 percent of the project has been completed. These

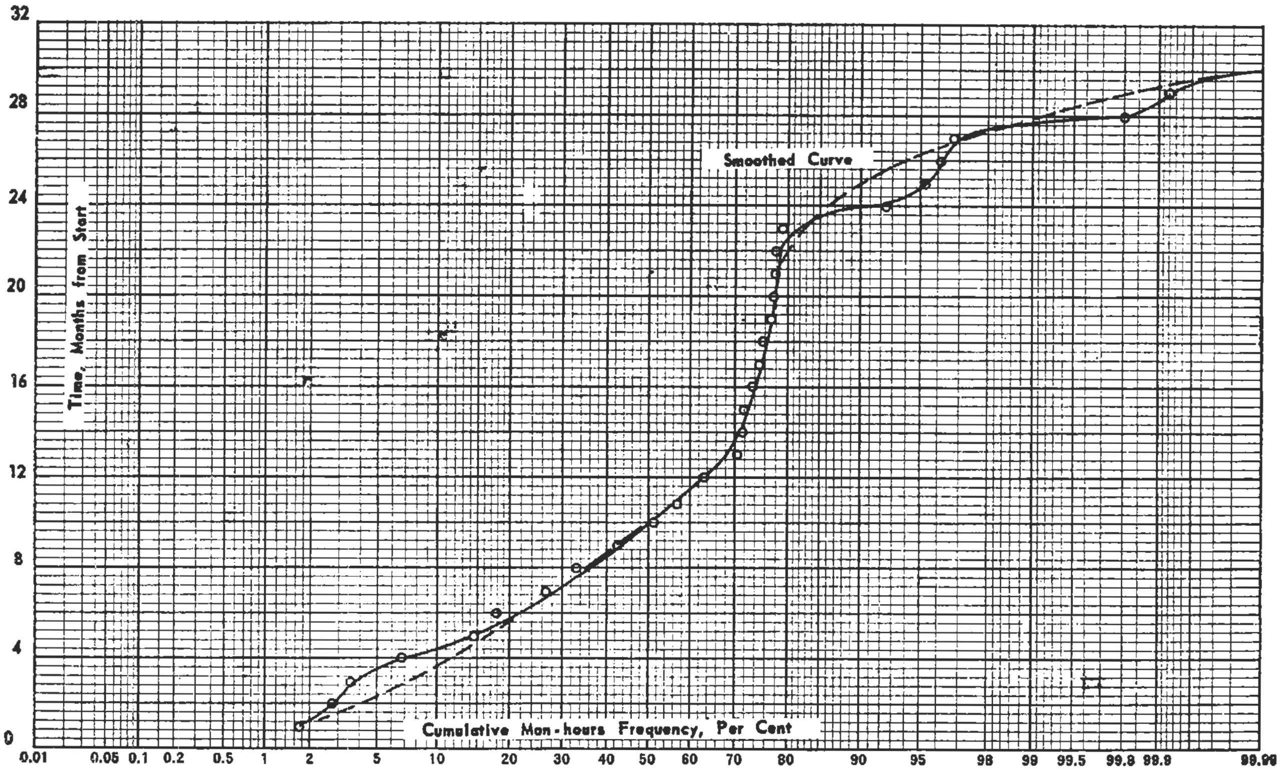

Figure 1-34C. Process engineering manhours accumulation pattern: Project C. By permission, E. E. Ludwig [7].

changes may not be setbacks, but they are reflected in the ability of the project to properly utilize the available engineering manpower in the "normal" manner.

Figure 1-35 presents some typical monthly requirements of process engineering for projects of different magnitudes. In some organizations the schedule is set by the available manpower, and does not always represent all that could be accomplished if a limitless supply of qualified manpower were available.

A summary of process engineering costs as they are related to total erected plant costs is shown in Figure 1-36. The process engineering man-hour requirements are related to total engineering for the project in Figure 1-37. These data are based on the operation of a complete process engineering section in the engineering department of a relatively large petrochemical plant complex. Since the assignment of responsibility varies with company policy and types of processes, this information is reasonably valid only for the particular plant relationship. It should establish order of magnitude information for other related operations. By studying the progress history of the individual projects, the major deviations from a so-called average straight-forward job can be recognized.

Figure 1-38 is reasonably typical of fixed-fee costs as charged by contract engineering organizations. The top curve representing the total engineering and related costs includes complete process engineering, equipment specifications, flowsheets, detailed complete plant drawings, purchasing and expediting. The lower curve represents only the process engineering including material and heat balances, equipment specifications, flowsheets, plot plan and elevations, and cost estimate. The middle curve covers the balance of all engineering detailing, purchasing and expediting.

In some cases, they may be anticipated by a knowledge of the status of the process data prior to the start of engineering activity. The larger projects are somewhat easier to group than the smaller ones. Process engineering is not always handled as completely for the small jobs. This is to say that flowsheets may be simplified, detailed equipment and line schedules may not be required, and the over-all project can be completely visualized at the outset, which is not the case with large projects.

Figure 1-39 illustrates that for average capital expenditures of \$10–30 million per year covering the very small hundred thousand dollar to very large (\$5–8 million) projects, the process engineering work leads the expenditures in a somewhat regular pattern by about three calendar quarters. This actual lead interval is a function of a company policy in scheduling its projects. The curves are believed representative for an aggressive program.

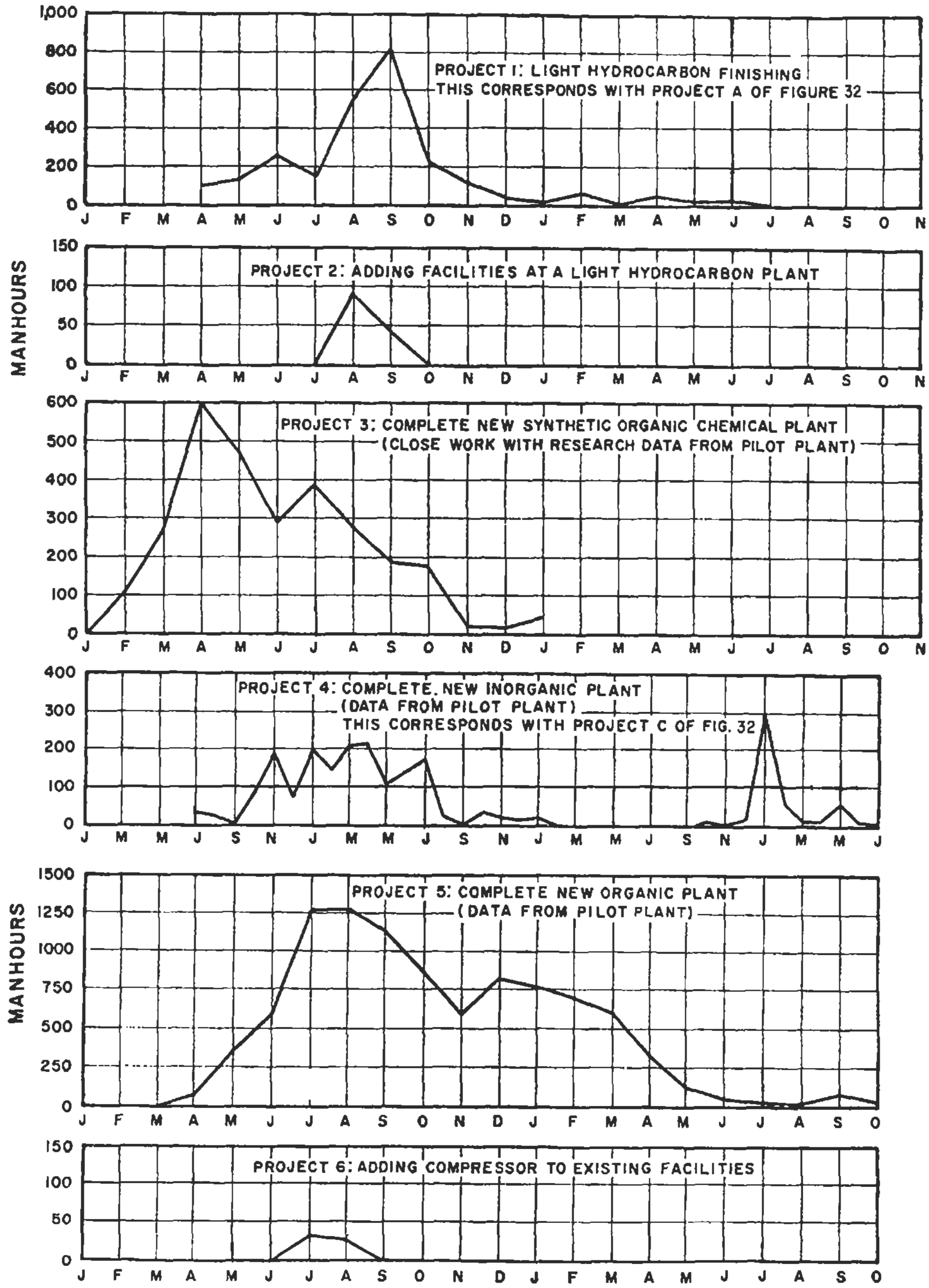

Figure 1-35. Process engineering manpower requirements by project and by months from start. By permission, E. E. Ludwig [7].

Influences

The principal factor which runs the process engineering man-hours over expected time for a "straight through" project is the comparison studies of equipment or process schemes when compared to the relatively simple and limited work after the decisions are made.

Any rushed program of process engineering development will usually be inefficient in manpower for certain parts of the work. Thus lead time for proper thinking and evaluation of significant process schemes and types of equipment will usually be reflected in efficient handling for the project when the bulk of the general engineering manpower is assigned to the detailed work. When decisions are made at the time of the need, all concerned can produce to the most benefit of the project. If the basic process can be designed and the flowsheet approved prior to initiating the detailed mechanical, structural and electrical engineering, the project usually runs well throughout the department. This situation is more likely to occur in a contractor organization than a producing company engineering department.

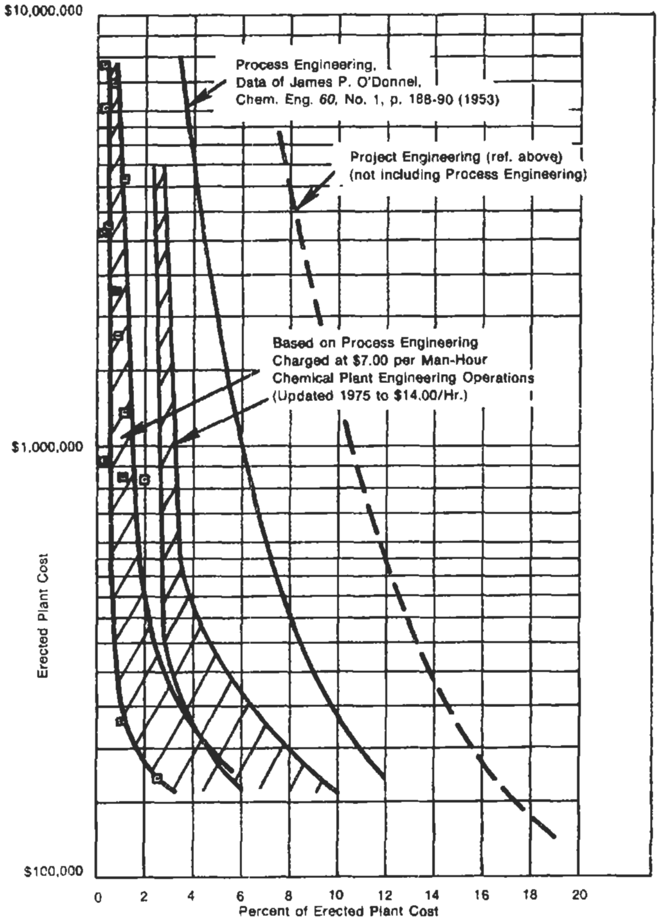

Figure 1-36. Process engineering costs (1975), based on process engineering charged at $14 per manhour. Chemical plant engineering operations, includes flowsheet development and drafting, material and heat balances, equipment designs, ratings, checking, and bid reviews and selection of equipment. By permission, E. E. Ludwig [7].

The schedule of projects must often be adjusted to reflect the influence of the key decision maker assigned to the work. If he requires complete detailed figures before reaching any decision, time will necessarily be consumed. On the other hand if he applies judgment and experience to the basic factors (less details), then the over-all direction of the project can be continually pointed in a profitable direction in the minimum of time. In reality actual "multipliers" are often applied to the time schedule of a project to reflect the type of decision-maker involved.

Assignment of Personnel

It is important to plan ahead for the proper assignment of qualified engineers to various projects as they arise. Jobs cannot be assigned on an unconsidered basis; that is, each lead or principal process engineer and others in his group on a project must be selected for their (1) basic ability to understand the process under consideration, (2) background know-how, (3) design ability for the equipment involved, and (4) compatibility with the project engineer and other key decision making representatives with whom they will be in daily contact.

There are two approaches to developing qualified personnel:

1. *The generalist approach*—each process engineer becomes competent over-all with preferential areas of specialization. With this approach, all personnel are urged to study and keep up to date in order to handle any type of project. This simplifies the assignment of the men, since there are more chances of

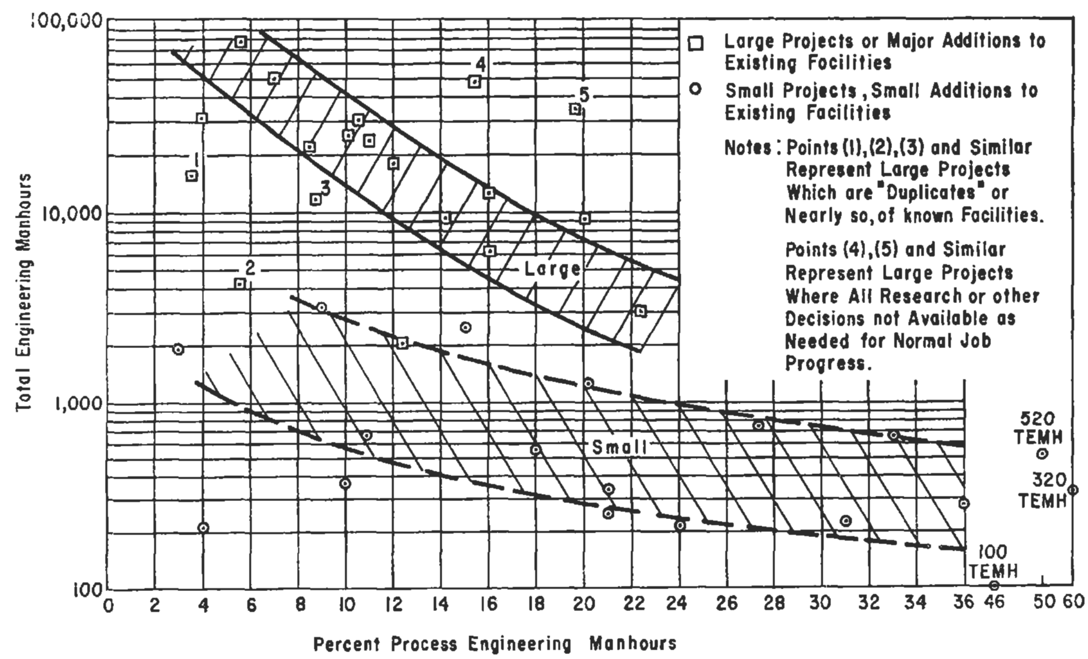

Figure 1-37. Process engineering manhours for new construction or major additions to existing facilities and small projects. By permission, E. E. Ludwig [7].

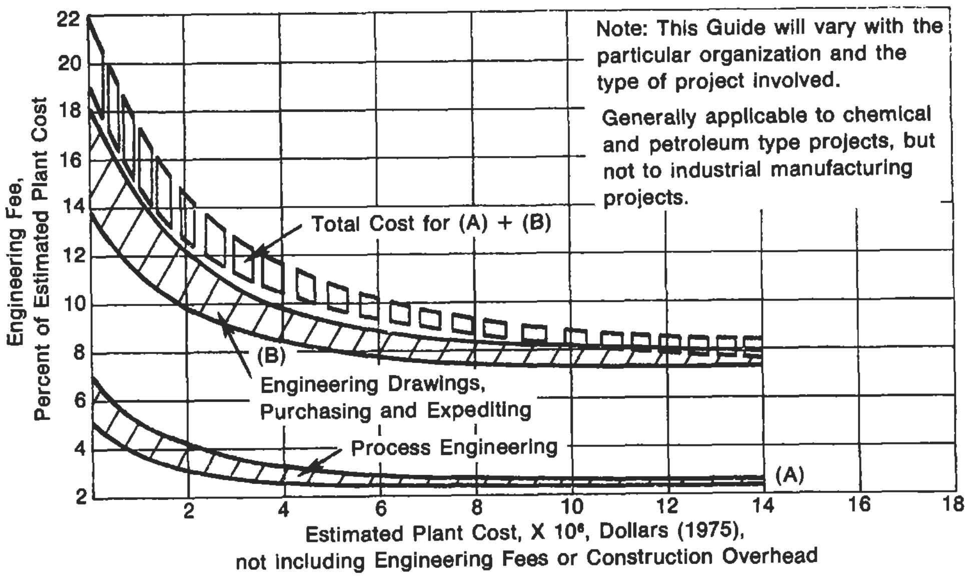

Figure 1-38. Estimating fixed engineering fees.

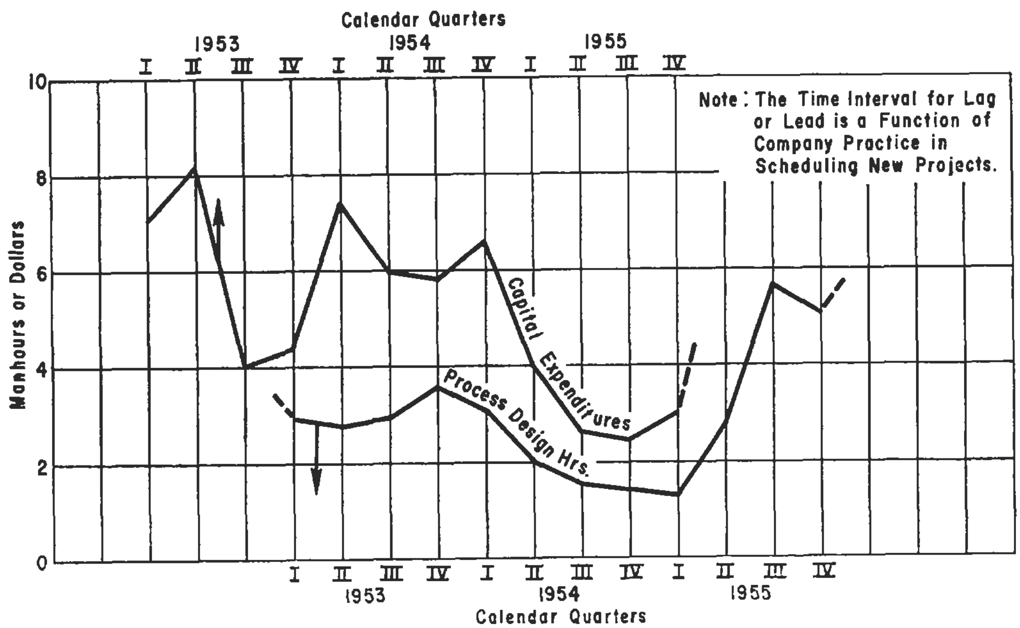

Figure 1-39. Process design manhours versus capital expenditures.

having some available who are relatively strong in the needed specialties of a particular job. This does not require that projects or specific designs be lined up waiting for the specialist. With over-all good general knowledge by each member of the group there is better appreciation for the exchange of views and understanding of specific design problems.

2. *The specialist approach*—each process engineer becomes a specialist in one or more related fields. Even in this arrangement some over-all general process engineers are needed to cover and tie together the areas handled by the specialist. Each specialist becomes an expert in a single field, or if reasonable, in a broad range of related topics. Each problem for design or study for every project of the particular type passes to the specialist for detailed handling of design, specification, and evaluation. His work passes into the project and he turns to another assignment on the same or some other project.

In general, the specialist may often be much more of an expert in a particular subject under this system than under the generalist approach, and consequently more depth into the pertinent factors of a problem may come to light for evaluation. By contrast it is easy to make a career of even a small assignment when the field of interest is narrow and the over-all project perspective is not clearly in view.

When the work load is low, it is important to have other assignments for these men. This is the time to develop standards for:

1. Design of various types and items of equipment
2. Methods of practice and general details
3. Electronic computer programs for these design standards
4. Evaluation of field data.

These should all be viewed from the long range and repetitive value to the group effort. The individuals who develop these standards in effect become specialists if they handle the assignment in good detail.

Plant Layout

The final plant layout combines the various engineering considerations for soil conditions; drainage; railroad, truck and services access; raw materials receiving; waste materials removal; climate effect on outdoor versus indoor operations and on types of structures; prevailing wind direction for vent as well as climatic moisture; corrosion; plant expansion and growth; access to public, and many other general evaluation points. From these broad considerations the details are developed to suit the particular plant process and the combined effects of the location.

The process engineer has an important responsibility in site selection as well as plant layout, since many of the decisions regarding physical location of buildings and associated equipment require a knowledge of what is taking place in the operation as well as the hazardous factors of explosion, fire, toxicity, etc. The process engineer is usually called upon to describe the process requirements and limitation to the other engineers—civil, structural, mechanical, electrical, and instrument. By progressively discussing the process each of the others can note the requirements which might affect the normal or routine design approach to each phase of the project. This review must not be limited to the design aspects of the engineering but rather must describe how the plant is to operate and how product is to be shipped, stored, etc.

After the project begins to take shape and preliminary layouts of the over-all as well as sections of the plant are partially complete, design work by the other phases of engineering will require the answering of questions as well as evaluating details of a particular phase as they are related to the process performance. Some useful considerations for selected details are given by Thompson [17] and Ludwig [19]. A general check list of factors which usually need reviewing for the proper layout considerations of chemical and petrochemical plants is given in Table 1-11.

There are many other factors which affect project planning as it is related to process engineering. However, these are usually peculiar to the process or objective of the project. On first glance some of the items listed in Table 1-11 may appear to be unrelated to the process engineering requirements, and this can be the case for some types of projects. In these situations they become more of a project engineering responsibility. However, in many cases these have a relationship either to the process engineering requirements or to the decisions which must take this into account.

Cost Estimates

Although this chapter is not intended to present the total details on preparation of capital or production/ operating/manufacturing cost estimates, it is worthwhile and helpful to provide some usable current references for the engineer who for many situations will be called on to provide total estimates or contribute to their development. As a guide to information, procedure and necessary data, references [10, 11, 12, 13, 14, 19, 21, 22, 23, 24, 25, 26, 27, 28, 29, 30, 33, 34] can be useful, but they are not all-inclusive, nor do they take the place of a thorough book on cost estimating for chemical and petrochemical plants. One of the most difficult problems is locating reliable up-to-date capital costs for equipment (see [43]). It is not "safe" to escalate or update by indexes [42] for costs that are more than six years old, and certainly not over ten years old.

The details of the preparation of cost estimates will not be covered. However, it is important to recognize that the process engineer plays a key role in estimate development. From a first draft flowsheet and a preliminary plot plan, a preliminary cost estimate can be prepared by the "factoring" or equivalent method. This basically accumulates the individual costs of each item of major equipment and then multiplies by an experience factor to produce one or all of (1) total plant cost installed with or without overhead costs (2) piping installed (3) equipment installed. For accuracy, these factors must be developed from actual plant costs, and are often peculiar to a specific type of construction or engineering approach to the project. That is, they may be a function of a "poor-boy" job, turn-key job, middle-of-the road, or "gold-plated" job. These types are peculiar to either the engineering contractor, the customer or to both. The factor of 2.5 to 6.0 usually covers most petrochemical processing plants. This factor times the costs of major equipment (pumps, compressors, tanks, columns, exchangers) but not instru-

Table 1-11
Layout and Process Development Engineering Check-List

Site (Assumes Site Selected)

1. Ground contour and its relation to general orientation of buildings and equipment.
2. Drainage and waste disposal, details to prevent erosion.
3. Set plant elevations: floor elevations of buildings and bottom of steel footings for equipment and large storage tanks.
4. Location of any existing or probable locations for new railroads, roads, power lines and power sources, telephone lines, water supply, residential and/or industrial buildings or structures.
5. Legal Requirements and Permits.
   a. Rights of way for pipe crossing of road, highway, railroad, rivers, canals, etc.
   b. Easements for pipe lines, power lines, etc.
   c. C.A.A. approval on airports, and for construction and painting of structures in certain areas in airport vicinity.
   d. Underground storage wells for chemical and hydrocarbon products.
   e. Railroad approval of road crossings, additions to existing facilities, automatic railroad gates, required state and railroad clearances.
   f. Navigable stream requirements and permits.

Climate

1. Prevailing wind; locate hazardous vents, burning flares, waste burning pits, waste settling ponds down-wind of plant proper.
2. Nature of climate. Consider seasonal and daily temperature variations, dust, fog, tornados, hurricanes, earthquakes. Define duration of conditions for design. Determine from U.S. Weather Bureau yearly statistics for above, as well as rainfall. Establish if conditions for earthquakes, hurricanes prevail. For stormy conditions, structural design for 100 miles per hour winds usually sufficient. For hurricanes, winds of 125 miles per hour may be design basis.
3. Corrosion. Plants located close (within 100 feet) to seas, oceans, bays, lakes encounter more severe corrosion than if located one-fourth mile or more away. Some highly industrial areas are more corrosive than rural or non-industrial locations. Additional details are presented by Mears.[15]
4. Pollution of Air and water. Determine allowable limits for atmospheric vent as well as liquid wastes. Consider neutralization. Determine federal, state and local regulations and effect of climatic conditions on dispersion.

Utilities and Raw Materials

1. Sources and methods of transportation and packaging.
   a. Water: potable, service, brackish, sea or ocean, cooling tower.
   b. Steam: condensate disposal, feed-water make-up
   c. Gas: (1) Process; may not be odorized
      (2) Fuel; odorized
   d. Oil: fuel, lubrication (or Liquefied Petroleum Gas)
   e. Air, (1) Utility
      (2) Instrument; must be dry below lowest equivalent dew point to prevent moisture condensation and freezing.
   f. Power
2. Warehouse receiving and storage: drums, boxes, carboys for raw processing materials as well as laboratory control and testing chemicals.

Product Shipments

1. Conditions for pipe line transfer of product to user or customer.
2. Warehouse conditions for bagging, boxing, crating, palletizing and methods of shipment (trailer truck, box car, tank car, hopper or special car). Consider in-transit and turnaround time to determine number in use.

General Layout

1. Use of models.
2. Maintenance considerations associated with each building, process area and equipment. Consider (a) access for cranes and trucks (b) work space for local repairs (c) operating conditions of adjacent parts of process to allow local repairs.
3. Initial construction sequence and problems.
4. Materials of construction for buildings.
5. Roads: paving, width.
6. Basic pattern for concrete, gravel or asphalt paving or work floors in operating and adjacent areas.
7. Fencing.
8. Plant guard or security system.

Electrical and Fire Hazards

1. Define plant areas handling hazardous and lethal materials and set rules for design considerations, such as ventilation, explosion walls, etc. Flammable storage materials may require enclosed dikes, foam systems and the like. Refer to National Board of Fire Underwriters or specific insurance company to coordinate recommended protection. Attaway[1] has details on many points to consider.
2. Define plant areas requiring explosion-proof, drip-proof and open motor and associated electrical components. Refer to National Electrical Code and National Electrical Manufacturer's Association Standards.
3. Define areas and buildings to use wet and dry sprinkler systems, foam systems, location of hand and hose fire extinguishers, fire carts, fire engines.
4. Define location of fire walls, fire hydrants.
5. Review layout for fire equipment access, and secondary and emergency exit roads from each area.
6. Review entire fire and other hazards program with insurance representatives. Industrial insurance companies have excellent facilities for evaluating the associated problems.

Safety Requirements

1. Special design problems for emergency handling of dangerous or lethal materials.
2. Safety as it is reflected in factors of safety in design of pressure vessels, pressure testing of piping and vessels, etc. Use of A.P.I., A.S.M.E. and ASA Codes; Code Stamps on equipment.
3. Areas requiring safety showers and eye wash stations.
4. Design and selection philosophy for use of safety devices for pressure relief and alarm.
5. Inside block valves on acid and caustic storage vessels.
6. Emergency power and other facilities to control safe operation or shut-down.

Future Growth

1. Define areas of future growth and associated space requirements.
2. Correlate future expansion plans to required utilities and raw materials as related to economics of required installation.
3. Consider spare equipment, present and future.

ments will give total plant costs. The plant will include usual control buildings, structure, foundations, overhead charges, construction fees, engineering costs, etc. A value of 4.0 is usually quite good.

The process designer must be aware of costs as reflected in the (1) selection of a basic process route (2) the equipment to be used in the process and (3) the details incorporated into the equipment. The designer must not arbitrarily select equipment, specify details or set pressure levels for design without recognizing the relative effect on the specific cost of an item as well as associated equipment such as relieving devices, instruments, etc.

With more and better information regarding the process and layout plans, estimating engineers can prepare detailed estimates which are often quite accurate, usually ±10 percent for the best. It is the duty of the process designer to supply the best information in order to contribute to better or improved estimates.

Estimating equipment costs is a specialty field in itself. Therefore, the estimator must have access to continuously updated basic reference costs and to graphical costs relations which are a function of capacity of this equipment. Page's [10] *Estimator's Manual of Equipment and Installation Costs* is a helpful reference. Since the equipment is only a portion of the total cost of a plant, or an addition to a chemical project, installation costs which reflect the labor portion of the total cost must also be determined. Useful and comprehensive data for such needs are presented for equipment [10], general construction [11], heating, air-conditioning, ventilating, plumbing [12], piping [13], electrical [14] and all disciplines [42] in the references indicated.

From such information even the inexperienced estimator can establish an approximation of the costs, provided he adequately visualizes the work functions and steps involved. From the same type of work reference, the experienced estimator can develop a realistic cost, usually expressed with certain contingencies to allow for unknown factors and changing conditions. The professional estimator will normally develop cost charts and tables peculiar to the nature of his responsibilities and requirements of his employer.

Six-Tenths Factor

This factor as presented by Chilton [4] has been used for scale-up of total or segments of plant cost.

$$P_b = P_a \left(\frac{C_b}{C_a}\right)^{0.6}$$

where P_b = Cost of plant or section of plant of new capacity "b."

P_a = Cost of plant or section of plant of original capacity "a."

C_b = Capacity of plant or section of new requirements.

C_a = Capacity of plant or section of original requirements.

This is applicable for any given year of installation but does not correct for the differences in cost from year to year. This is conveniently done as described in the section for year indices. Experience has indicated that this six-tenths rule is reasonably accurate for capacity scale-up of individual items of equipment. Thus, if the cost of one size of a piece of equipment is known, an estimating figure for one twice as large can be found by multiplying by $(2)^{0.6}$.

The most difficult feature of this method is that for each type of plant or plant product as well as for each type of equipment there is a break-point where the 0.6 no longer correlates the change in capacity. For small equipment or plants in reasonable pilot or semi-works size, the slope of the cost curve increases and the cost ratio is greater than 0.6, sometimes 0.75, 0.8 or 0.9. From several cost values for respective capacities a log-log plot of capacity versus cost will indicate the proper exponent by the slope of the resultant curve. Extrapolation beyond eight or ten fold is usually not too accurate.

Yearly Cost Indices

The three most used cost indices for the chemical, petrochemical, and refining industry for relating the cost level of a given year or month to a reference point are

1. *Chemical Engineering Plant Cost Index* [42]. Probably the most commonly used cost adjusting index printed/updated monthly is in *Chemical Engineering Magazine* and has established continuity over many years. Its breakdown component costs apply to plants and plant equipment/systems.
2. *Marshall and Swift Equipment Cost Index* [57]. Commonly used for process industry equipment and index numbers presented by industries in *Chemical Engineering Magazine* on a monthly basis.
3. *Nelson Index* [58]. This is generally suited to petroleum refining plants and is referenced to them. It is updated and published regularly in *The Oil and Gas Journal.*

These indices are used to up-date costs when values at some date are known. The new costs are of estimating accuracy and should be verified whenever possible, just as the results of using the 0.6 power for correlating cost and capacity.

$$EC_2 = EC_1\left(\frac{I_2}{I_1}\right)$$

where I_2 = index value for year represented by 2, (usually current)
I_1 = index value for earlier year represented by 1.
EC_2 = equipment estimated cost for year represented by 2.
EC_1 = equipment purchased cost (when available) for year represented by 1.

Return on Investment

The proper evaluation of costs as they affect the selection of processes and equipment is not included in this book. However, it is important to emphasize that every process engineer must be cognizant of the relationships. There are several methods to evaluate return on invested money, and the nomograph of Figure 1-40 represents one. It is a useful guide [6] to estimate the order of magnitude of a return on an expenditure to gain savings in labor and/or material costs. The nomograph is used to determine the investment justified by a gross annual savings, assuming a percent return, a percent annual depreciation charge, and a 50 percent Federal tax on net savings.

$$\text{Return} = \frac{(\text{Gross savings} - \text{Depreciation} \times \text{Investment})\ (1 - \text{Federal Tax})}{\text{Investment}}$$

Example 1-2: Justifiable Investment For Annual Savings [6]

Find the justifiable investment for a gross annual savings of $15,000 when a return of 10% and a depreciation rate of 15 percent are specified.

1. From Figure 1-40, connect scales A and B.
2. From the intersection with the C scale, connect a line to the D scale.
3. At the intersection of line (2) with the inclined investment scale, E, read that a $43,000 investment is justified to save $15,000 gross per year.

Accounting Coordination

All new plants as well as changes to existing facilities and plants must be coordinated with a cost accounting system. Often the building, services and utilities, and site development must be separated cost-wise from each other. Each company has reason and need for various arrangements in order to present proper information for tax purposes and depreciation. Although the project engineer is usually responsible for this phase of coordination through the engineering groups, it is often necessary that

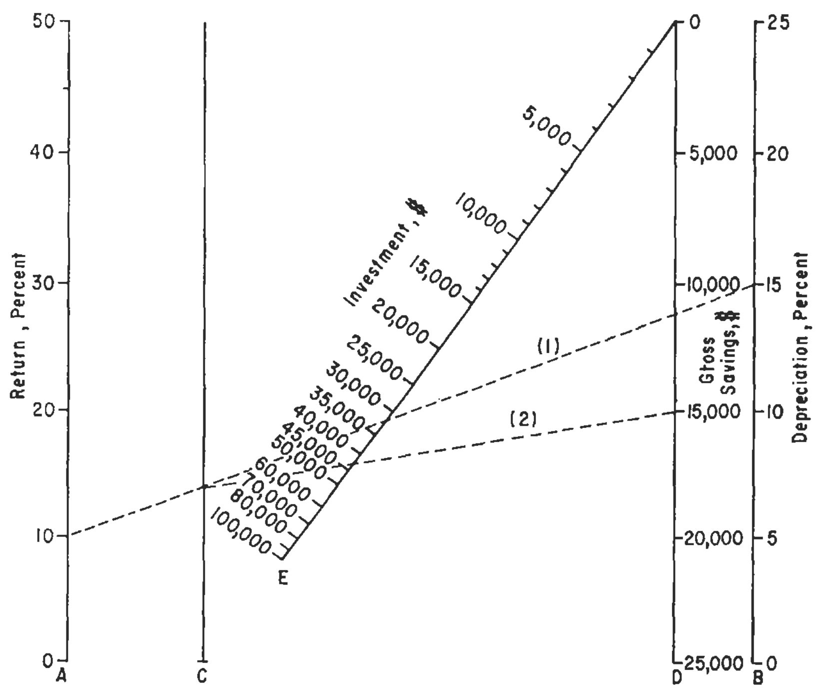

Figure 1-40. Annual saving, return, and depreciation of fixed adjustable investment. By permission, G. A. Larson [6].

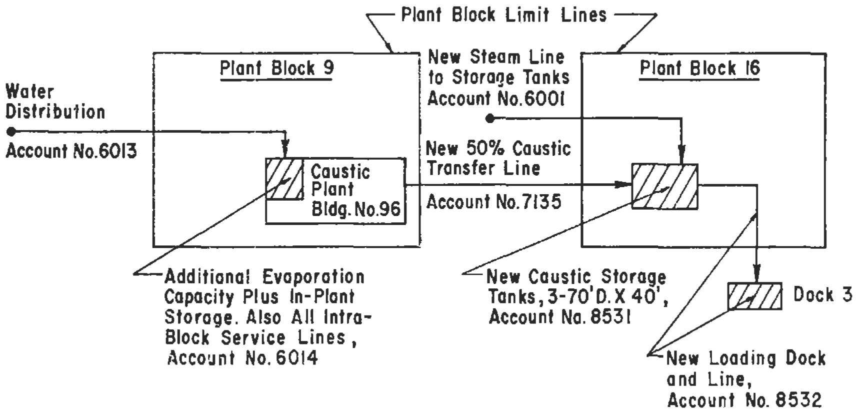

Figure 1-41. Account diagram for accumulation of project costs. Cost estimates must be made to conform to same scope basis.

the process engineer present proper breakdown details, and these then serve to coordinate the cost breakdowns. Figure 1-41 is an example of such an accounting diagram.

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Chapter

2

Fluid Flow

The flow of compressible and non-compressible liquids, gases, vapors, suspensions, slurries and many other fluid systems has received sufficient study to allow definite evaluation of conditions for a variety of process situations for Newtonian fluids. For the non-Newtonian fluids, considerable data is available. However, its correlation is not as broad in application, due to the significant influence of physical and rheological properties. This presentation is limited to Newtonian systems, except where noted.

Primary emphasis is given to flow through circular pipes or tubes since this is the usual means of movement of gases and liquids in process plants. Flow through duct systems is treated with the fan section of Compression in Volume 3.

Scope

The scope of this chapter emphasizes applied design techniques for 85%± of the usual situations occurring in the design and evaluation of chemical and petrochemical plants for pressure and vacuum systems (see Figure 2-1). Whereas computer methods have been developed to handle many of the methods described here, it is the intent of this chapter to present only design methods per se that may be applied to computer programming. First, however, a thorough understanding of design methods, their fundamental *variations* and *limitations* is critical. There is a *real danger* in losing sight of the required results of a calculation when the computer program is "hidden" from the user and the user becomes too enamored with the fact that the calculations were made on a computer. A good designer *must* know the design details built into the computer program before "blindly" using it and its "cold" results. There are many programs for process design that actually give incorrect results because the programmer was not sufficiently familiar with the design procedures and end limits/limitations of the method. Then, when such programs are purchased by others, or used in-house by others, some serious and erroneous design results can be generated. On the other hand, many design procedures that are complicated and require successive approximation (such as distillation) but are properly programmed, can be extremely valuable to the design engineers.

Except as a limited reference, computer programs are not emphasized anywhere in these volumes. Instead, important mechanical details are given to emphasize the mechanical application of the process requirements (see Figure 2-2). Many of these details are essential to the *proper functioning of the process* in the hardware. For two fundamental aspects of fluid flow, see Figures 2-1 and 2-3.

Basis

The basis for single-phase and some two-phase friction loss (pressure drop) for fluid flow follows the Darcy and Fanning concepts. The exact transition from laminar or viscous flow to the turbulent condition is variously identified as between a Reynolds number of 2000 and 4000.

For an illustration of a portion of a plant piping system (see Figure 2-2).

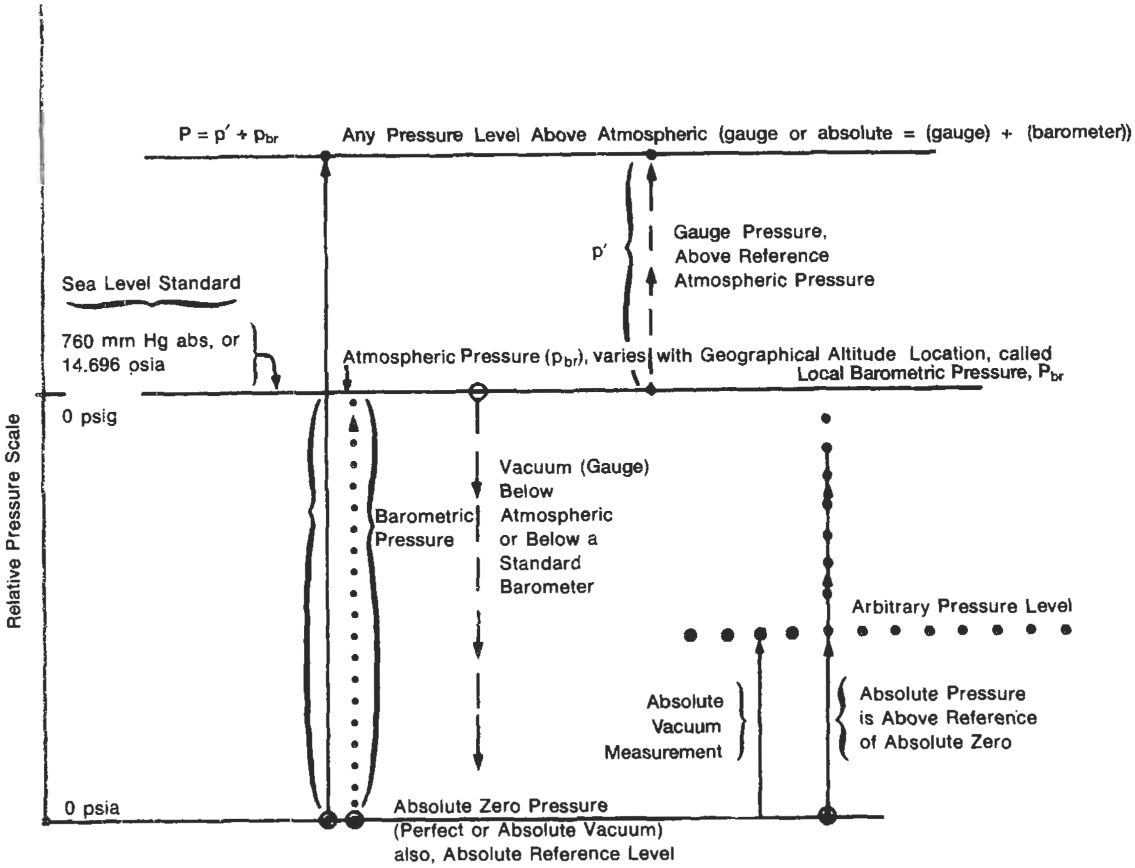
Incompressible Flow

For liquids, laminar or turbulent flow in a pipe [3]

$$\Delta P = \frac{\rho f v^2 L}{144D\,(2g)}, \text{ lbs / square in.} \tag{2-1}$$

or,

$$h_f = \frac{fLv^2}{D\,(2g)}, \text{ ft of fluid flowing} \tag{2-2}$$



Notes:

1. At sea level, barometric pressure = 14.696 pounds/sq. in. absolute, or 760 mm of mercury, referred to as "standard." This is also 0 pounds/sq. in. gauge for that location.
2. Absolute zero pressure is absolute vacuum. This is 0 psia, also known as 29.92 inches of mercury below atmospheric pressure, or 33.931 feet of water below atmospheric, all referenced at sea level.
3. Important equivalents: 1 atmospheric pressure at sea level =
   (a) 14.696 psia
   (b) 33.931 feet of water (at 60°F)
   (c) 29.921 inches mercury (at 32°F)
   (d) 760 mm H_g (at 32°F)
   (e) 1.0332 kilogram/sq. centimeter
   (f) 10,332.27 kilogram/sq. meter
4. Barometric pressure for altitudes above "standard" sea level are given in the appendix. These correct values must be used wherever the need for the local absolute barometric pressure is involved in pressure calculations.
5. Vacuum is expressed as either
   (a) Inches (or millimeters) vacuum below atmospheric or local barometric, or
   (b) Inches vacuum absolute, above absolute zero pressure or perfect vacuum.
   (c) For example, at sea level of 29.921 in Hg abs. barometer;(1) 10″ vacuum is a gauge term, indicating 10″ of mercury below local barometric pressure; (2) 10″ vacuum (gauge) is equivalent to 29.921″ Hg abs. − 10″ = 19.921″ Hg abs. vacuum.

Figure 2-1. Pressure level references. Adapted by permission from Crane Co., *Technical Paper #410,* Engineering Div., 1957.

See nomenclature for definition of symbols and units. The units presented are English engineering units, unless a conversion is required. The friction factor is the only experimental variable that must be determined by reference to the above equations and it is represented by Figure 2-3. Note that this may sometimes be referred to as the Fanning formula, and may be modified to yield a friction factor one-fourth that of the Darcy factor. Care should be observed; otherwise, the friction loss calculations for flow of liquids or gases will be too low, but not necessarily by a straight one-fourth factor. Also, it is important to note that the Figure 2-3 presented here is the friction chart recommended and consistent with the engineering data of the Hydraulic Institute [2].

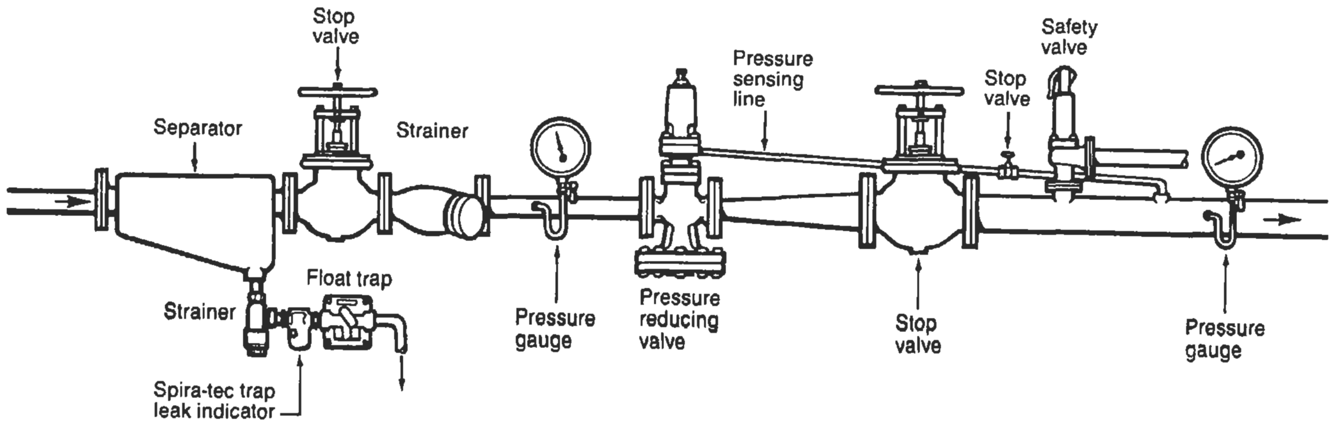


Figure 2-2. Portion of a plant piping system. By permission, Spirax-Sarco, Inc., 1991.

The many empirical correlations advanced to represent the frictional resistance to flow vary from exact results because of the specific simplifying assumptions incorporated in each. Some relations agree in one region of flow and diverge in others.

Compressible Flow: Vapors and Gases [3]

Compressible fluid flow occurs between the two extremes of isothermal and adiabatic conditions. For adiabatic flow the temperature decreases (normally) for decreases in pressure, and the condition is represented by $p'V'^{(k)}$ = constant. Adiabatic flow is often assumed in short and well-insulated pipe, supporting the assumption that no heat is transferred to or from the pipe contents, except for the small heat generated by friction during flow. Isothermal $p'Va$ = constant temperature, and is the mechanism usually (not always) assumed for most process piping design. This is in reality close to actual conditions for many process and utility service applications.

The single-phase friction loss (pressure drop) for these situations in chemical and petrochemical plants is still represented by the Darcy equation with specific limitations [3]:

1. If calculated pressure drop from inlet (upstream) to outlet (downstream) of a line system is less than about 10% of inlet pressure P_1, reasonable accuracy can be expected provided the specific volume used is based on inlet or outlet conditions.
2. If calculated pressure drop from inlet to outlet of line system (not including control or hand valves) is greater than approximately 10%, but less than about 40% of the inlet pressure P_1 (pounds per square inch gauge), the Darcy equation will yield reasonable accuracy when using a specific volume based on the average of upstream (inlet) and downstream (outlet) conditions. If these criteria do not apply, then refer to the method using the flow coefficient, K.
3. For larger pressure drops in long lines of a mile or greater in length than noted above, use methods presented with the Weymouth, Panhandle Gas formulas, or the simplified compressible flow equation.
4. For isothermal conditions [3]:

$$w_s = \sqrt{\left[\frac{144gA^2}{\overline{V}_1\left(f\frac{L}{D} + 2\log_e\frac{P_1'}{P_2'}\right)}\right]\left[\frac{(P_1')^2 - (P_2')^2}{P_1'}\right]}, \text{ lbs/sec} \quad (2\text{-}3)$$

$$w_s = 0.371\sqrt{\left[\frac{d^4}{\overline{V}_1\left(f\frac{L}{D} + 2\log_e\frac{P_1'}{P_2'}\right)}\right]\left[\frac{(P_1')^2 - (P_2')^2}{P_1'}\right]}, \text{ lbs/sec} \quad (2\text{-}4)$$

The correlations included here are believed to apply to good plant design procedures with good engineering accuracy. As a matter of good practice with the exercise of proper judgment, the designer should familiarize himself with the background of the methods presented in order to better select the conditions associated with a specific problem.

Design conditions may be:

1. Flow rate and pressure drop allowable established, determine pipe size for a fixed length

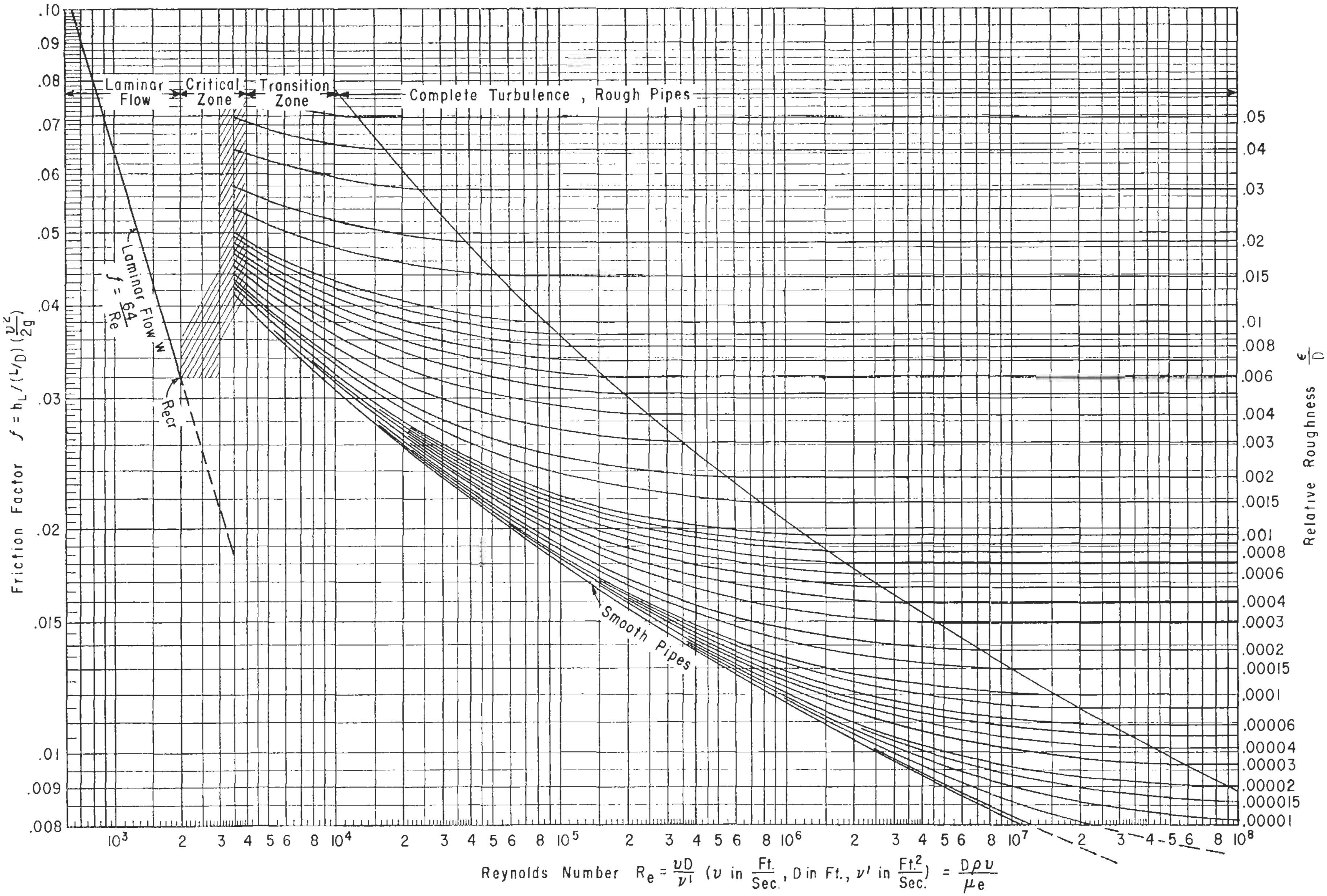


Figure 2-3. Moody or "regular" Fanning friction factors for any kind and size of pipe. Note: the friction factor read from this chart is four times the value of the f factor read from *Perry's Handbook,* 6th Ed. [5]. Reprinted by permission, *Pipe Friction Manual,* 1954 by The Hydraulic Institute. Also see *Engineering DataBook,* 1st Ed., The Hydraulic Institute, 1979 [2]. Data from L. F. Moody, "Friction Factors for Pipe Flow" by ASME [1].

2. Flow rate and length known, determine pressure drop and line size.

Usually either of these conditions requires a trial approach based upon assumed pipe sizes to meet the stated conditions. Some design problems may require determination of maximum flow for a fixed line size and length: however, this just becomes the reverse of the conditions above.

Optimum economic line size is seldom realized in the average process plant. Unknown factors such as future flow rate allowances, actual pressure drops through certain process equipment, etc., can easily over-balance any design predicated on selecting the optimum. Certain guides as to order of magnitude of costs and sizes can be established either by one of several correlations or by conventional cost estimating methods. The latter is usually more realistic for a given set of conditions, since generalized equations often do not fit a plant system.

There are many computer programs for sizing fluid flow through pipe lines. An example can be found in Reference [32]. However, before "blindly" jumping to use such programs, the designer should examine the bases and sources of such programs. Otherwise, significant errors could destroy the validity of the program for its intended purpose.

Factors of "Safety" for Design Basis

Unless noted otherwise the methods suggested here do not contain any built-in safety factors. These should be included, but only to the extent justified by the problem at hand. Although most designers place this factor on the flow rate, care must be given in analyzing the actual conditions at operating rates below this value. In some situations a large factor imposed at this point may lead to unacceptable conditions causing erroneous decisions and serious effects on the sizing of automatic control valves internal trim.

As a general guide, factors of safety of 20 percent to 30 percent on the friction factor will accommodate the change in roughness conditions for steel pipe with average service of 5 to 10 years, but will not necessarily compensate for severe corrosive conditions. Corrosion conditions should dictate the selection of the materials of construction for the system as a part of establishing design criteria. Beyond this the condition often remains static, but could deteriorate further. This still does not allow for increased pressure drop due to increased flow rates. Such factors are about 10 percent to 20 percent additional. Therefore for many applications the conservative Cameron Tables [4] give good direct-reading results for long-term service. See Table 2-22.

Important Pressure Level References

Figure 2-1 presents a diagrammatic analysis of the important relationships between absolute pressure, gauge pressures, and vacuum. These are key to the proper solution of fluid flow, fluid pumping, and compression problems. Most formulas use absolute pressures in calculations; however, there are a few isolated situations where gage pressures are used. Care must be exercised in following the proper terminology as well as in interpreting the meaning of data and results.

Pipe, Fittings, and Valves

To ensure proper understanding of terminology, a brief discussion of the "piping" components of most process systems is appropriate.

The fluids considered in this chapter consist primarily of liquids, vapors, gases, and slurries. These are transported usually under pressure through circular ducts, tubes, or pipes (except for low pressure air), and these lengths of pipe are connected by fittings (screwed or threaded, butt welded, socket welded, or flanged) and the flow is controlled (stopped, started, or throttled) by means of valves fixed in these line systems. The components of these systems will be briefly identified in this chapter, because the calculation methods presented are for flows through these components in a system. These flows always create some degree of pressure drop (or loss of pressure head), which then dictates the power required to move the fluids through the piping components (Figure 2-2).

Pipe

Process plants use round pipe of varying diameters (see pipe dimensions in Tables A-14, A-15, and A-16 in Appendix). Connections for smaller pipe below about 1½ in. to 2 in. (Figures 2-4A, 2-4B) are threaded or socket welded, while nominal pipe sizes 2 in. and larger are generally butt-welded or socket welded (Figure 2-4C) with the valves and other connections flanged into the line. Steam power plants are a notable exception. This chapter, however, does not deal with power plant design, although steam lines are included in the sizing techniques. Pipe is generally designated by nominal size, whereas calculations for flow considerations must use the actual standard inside diameter (I.D.) of the pipe. For example: (Note: O.D. refers to outside diameter of pipe.)

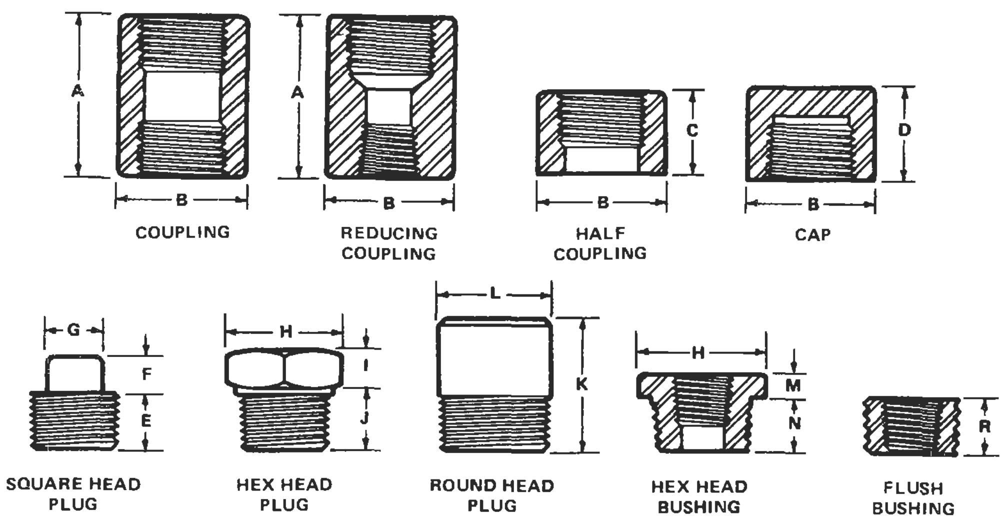


Figure 2-4A. Forged steel threaded pipe fittings, WOG (water, oil or gas service). Note: the working pressures are always well above actual plant operating levels. Pressure classes 3000 psi and 6000 psi, sizes ⅛ in. through 4 in. nominal. By permission, Ladish Co., Inc.

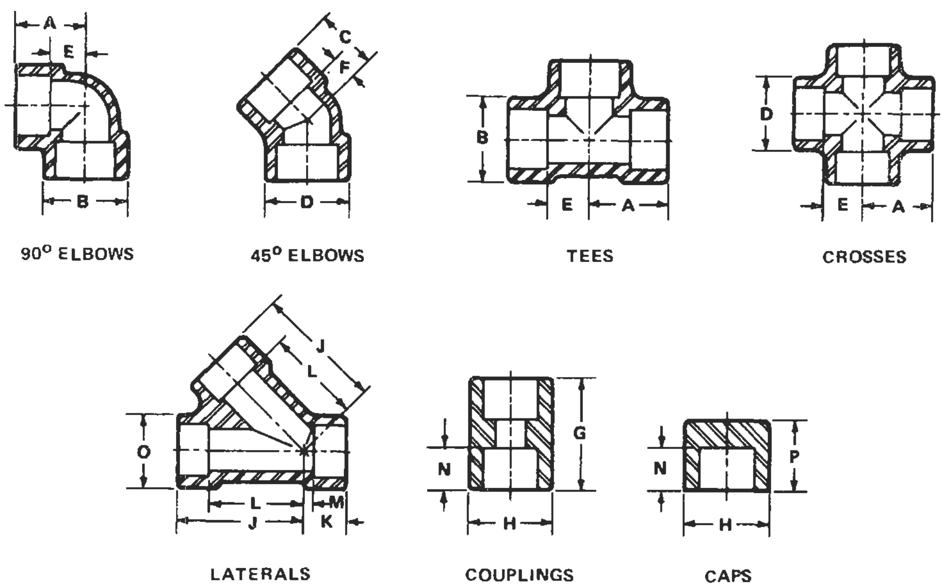


Figure 2-4B. Forged steel socket weld fittings, WOG (water, oil or gas service). Note: the working pressures are always well above actual plant operating levels and are heavy to allow for welding. Pressure classes 3000 psi and 6000 psi, sizes ⅛ in. through 4 in. nominal. Do not weld on malleable iron or cast iron fittings. (By permission, Ladish Co., Inc.)

Nominal Pipe Size	O.D. Inches		I.D. Inches	
Inches	Schedule 40	80	40	80
¾	1.050	1.050	0.824	0.742
1	1.315	1.315	1.049	0.957
1½	1.900	1.900	1.610	1.500
2	2.375	2.375	2.067	1.939
3	3.500	3.500	3.068	2.900
4	4.500	4.500	4.026	3.826

See Appendix for other sizes.

American Standards Association piping pressure classes are:

ASA Pressure Class	Schedule No. of Pipe
≤ 250 lbs./sq. in.	40
300–600	80
900	120
1500	160
2500 (½ in.–6 in.)	XX (double extra strong)
2500 (8 in. and larger)	160

Figure 2-4C. Forged steel welded-end fittings. By permission, Tube Turn Technologies, Inc.

Usual Industry Pipe Sizes and Classes Practice

Certain nominal process and utility pipe sizes are not in common use and hence their availability is limited. Those not usually used are: ⅛ in., 1¼ in., 2½ in., 3½ in., 5 in., 22 in., 26 in., 32 in., 34 in.

Some of the larger sizes, 22 in. and up, are used for special situations. Also, some of the non-standard process sizes such as 2½ in., 3½ in. and 5 in. are used by "packaged" equipment suppliers to connect components in their system for use in processes such as refrigeration, drying, or contacting.

The most common schedule in use is 40, and it is useful for a wide range of pressures defined by ANSI Std. B 36.1 (American National Standards). Lighter wall thickness pipe would be designated Schedules 10, 20, or 30; whereas, heavier wall pipe would be Schedules 60, 80, 100, 120, 140, 160 (see Appendix Table). Not all schedules are in common use, because after Schedule 40, the Schedule 80 is usually sufficient to handle most pressure situations. The process engineer must check this schedule for both pressure and corrosion to be certain there is sufficient metal wall thickness.

When using alloy pipe with greater tensile strength than carbon steel, the schedule numbers still apply, but may vary, because it is unnecessary to install thicker walled alloy pipe than is necessary for the strength and corrosion considerations. Schedules 10 and 20 are rather common for stainless steel pipe in low pressure applications.

For example, for 3-in. nominal carbon steel pipe, the Schedule 40 wall thickness is 0.216 in. If the pressure required in the system needs 0.200 in. wall and the corrosion rate over a five-year life required 0.125 in. (⅛ in.), then the 0.200 in. + 0.125 in. = 0.325 in. and the Schedule 40 pipe would not be strong enough at the end of five years. Often the corrosion is calculated for 10- or 15-years' life before replacement. Currently Schedule 80, 3-in. pipe has a 0.300 in. wall thickness, so even this is not good enough in carbon steel. Rather than use the much heavier Schedule 160, the designer should reconsider the materials of construction as well as re-examine the corrosion data to be certain there is not unreasonable conservatism. Perhaps stainless steel pipe or a "lined" pipe would give adequate strength and corrosion resistance. For a bad corrosion condition, lined pipe using linings of PVC (polyvinyl chloride), Teflon®, or Saran® typically as shown in Figure 2-5A, 2-5B, 2-5C and 2-5D can be helpful.

While threaded pipe is joined by threaded fittings (Figure 2-4A), the joints of welded pipe are connected to each other by butt welding or socket welding (Figure 2-4B) and to valves by socket welds or flanges of several types (Figure 2-6) using a gasket of composition material, rubber or metal at the joint to seal against leaks. The joint is pulled tight by bolts (see Figure 2-7).

For lower pressure systems of approximately 150 psig at 400°F or 225 psig at 100°F, and where sanitary precautions (food products or chemicals used in food products) or some corrosion resistance is necessary, tubing is used. It is joined together by butt welds (Figure 2-8) or special compression or hub-type end connectors. This style of "piping" is not too common in the

(text continued on page 62)

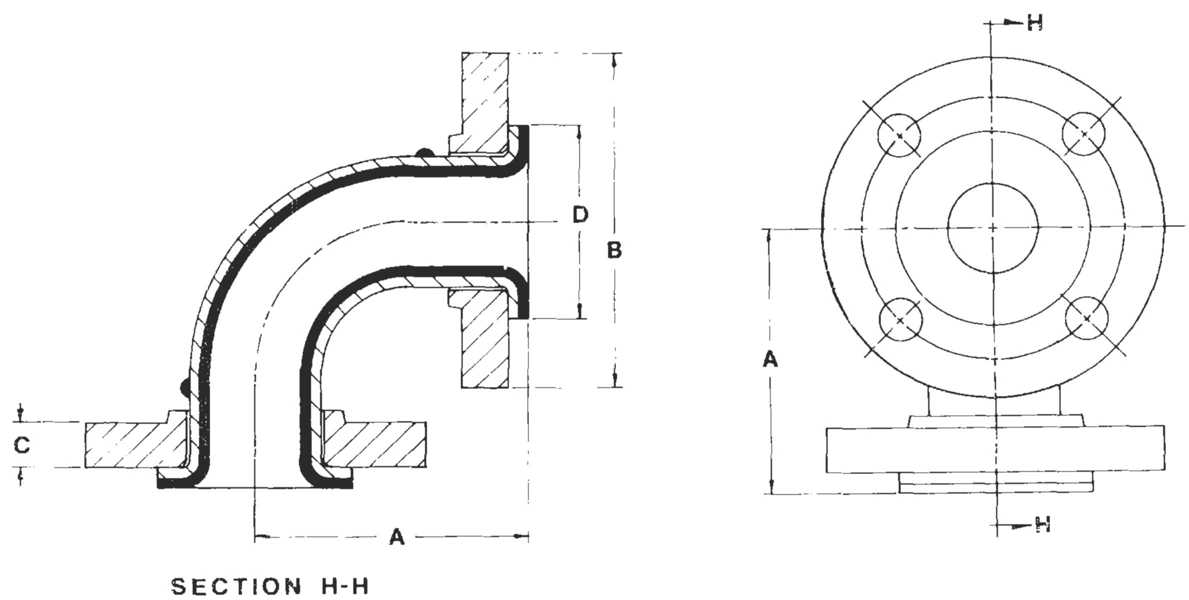


Figure 2-5A. Lined-steel pipe and fittings for corrosive service. By permission, Performance Plastics Products.

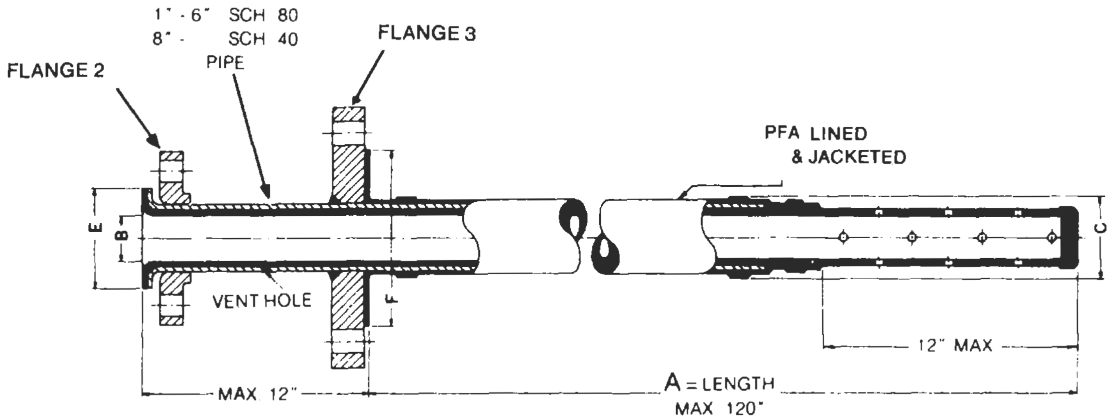
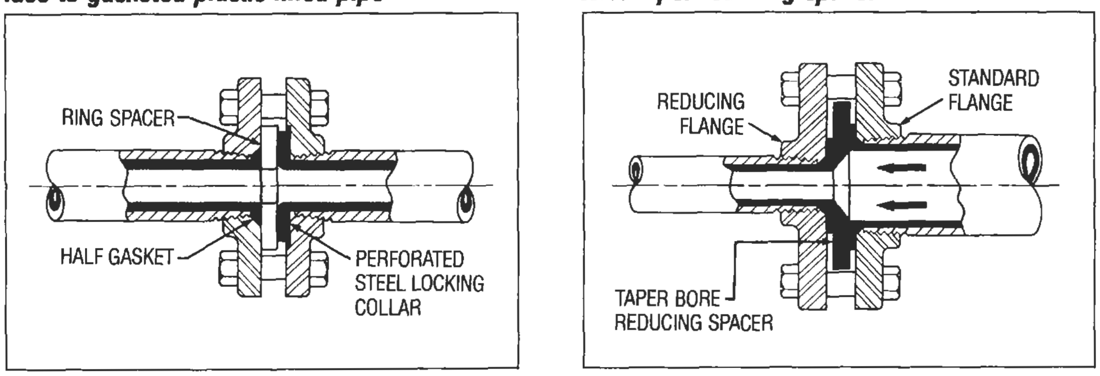


Figure 2-5B. Lined-steel pipe flanged sparger for corrosive service. By permission, Performance Plastics Products.

Connection of reinforced flared face to gasketed plastic-lined pipe

With taper reducing spacer²



[2] Only the following size reductions should be made by this technique when connecting pipe with molded raised faces: 1½×1, 2×1, 2×1½, 2½×1½, 2½×2, 3×2, 3×2½, 4×2½, 4×3, 6×4, 8×6. All other reductions require use of reducing filler flanges or concentric reducers.

Figure 2-5C. Flanged lined-steel pipe fittings for corrosive service. By permission, Dow Plastic-Lined Products, Bay City, Mich. 48707, 1-800-233-7577.

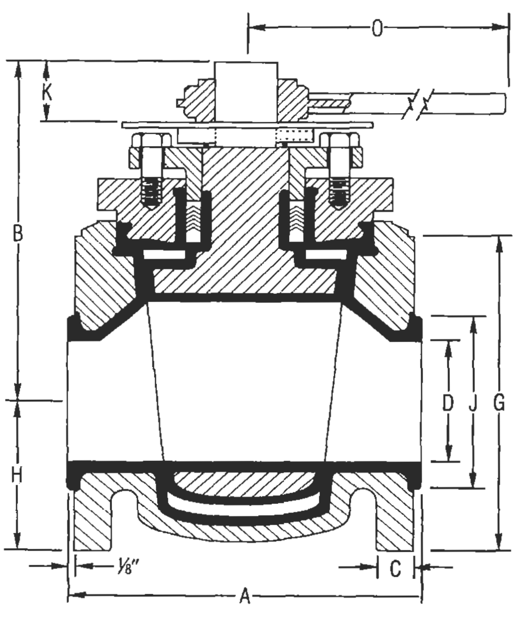
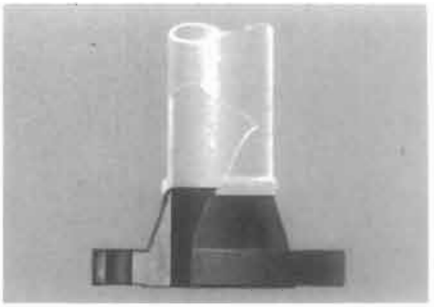
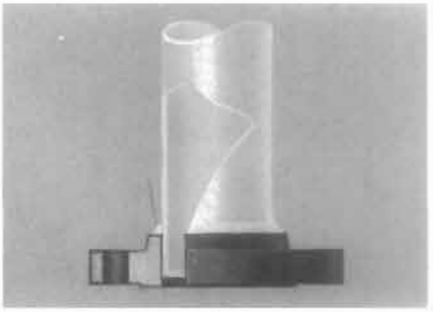


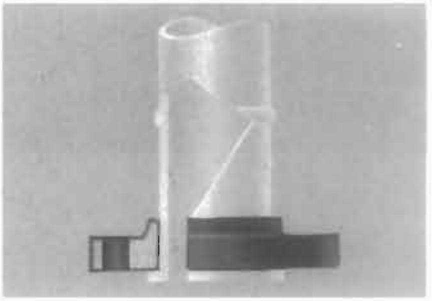
Figure 2-5D. Lined plug valve for corrosive service. By permission, Dow Plastic-Lined Products, Bay City, Mich. 48707, 1-800-233-7577.



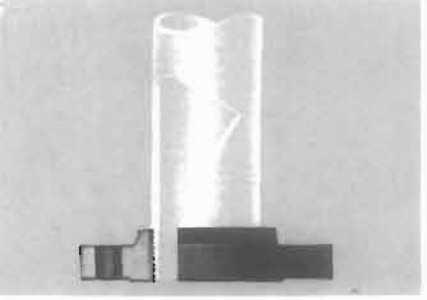
Welding neck flanges are distinguished from other types by their long tapered hub and gentle transition of thickness in the region of the butt weld joining them to the pipe. Thus this type of flange is preferred for every severe service condition, whether this results from high pressure or from sub-zero or elevated temperature, and whether loading conditions are substantially constant or fluctuate between wide limits.



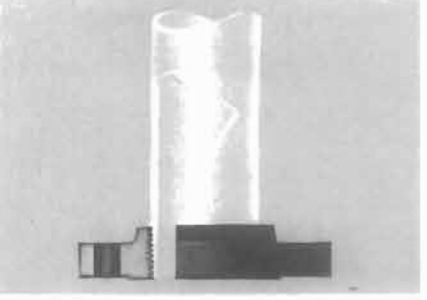
Slip-on flanges continue to be preferred to welding neck flanges by many users on account of their initially lower cost, the reduced accuracy required in cutting the pipe to length, and the somewhat greater ease of alignment of the assembly; however, their final installed cost is probably not much, if any, less than that of welding neck flanges. Their calculated strength under internal pressure is of the order of two-thirds that of welding neck flanges, and their life under fatigue is about one-third that of the latter.



Lap joint flanges are primarily employed with lap joint stubs, the combined initial cost of the two items being approximately one-third higher than that of comparable welding neck flanges. Their pressure-holding ability is little, if any, better than that of slip-on flanges and the fatigue life of the assembly is only one-tenth that of welding neck flanges. The chief use of lap joint flanges in carbon or low alloy steel piping systems is in services necessitating frequent dismantling for inspection and cleaning and where the ability to swivel flanges and to align bolt holes materially simplifies the erection of large diameter or unusually stiff piping. Their use at points where severe bending stress occurs should be avoided.

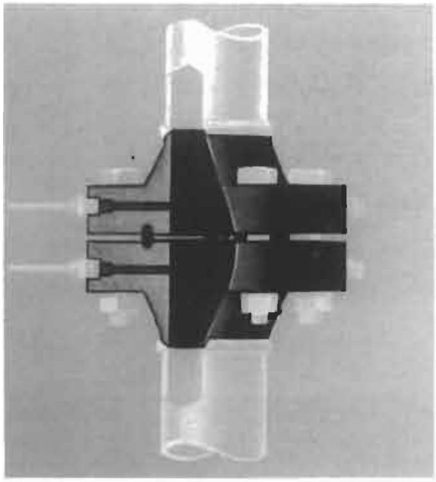


Threaded flanges made of steel, are confined to special applications. Their chief merit lies in the fact that they can be assembled without welding; this explains their use in extremely high pressure services, particularly at or near atmospheric temperature, where alloy steel is essential for strength and where the necessary post-weld heat treatment is impractical. Threaded flanges are unsuited for conditions involving temperature or bending stresses of any magnitude, particularly under cyclic conditions, where leakage through the threads may occur in relatively few cycles of heating or stress; seal welding is sometimes employed to overcome this, but cannot be considered as entirely satisfactory.

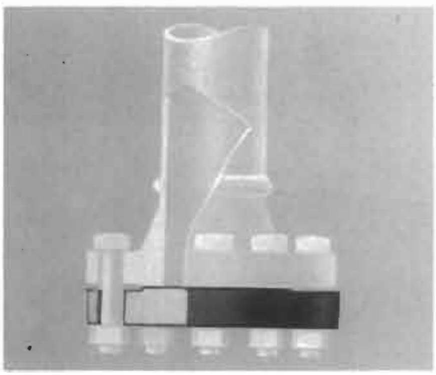


Socket welding flanges were initially developed for use on small-size high pressure piping. Their initial cost is about 10% greater than that of slip-on flanges; when provided with an internal weld as illustrated, their static strength is equal to, but their fatigue strength 50% greater than double-welded slip-on flanges. Smooth, pocketless bore conditions can readily be attained (by grinding the internal weld) without having to bevel the flange face and, after welding, to reface the flange as would be required with slip-on flanges.

Figure 2-6. **(continued on next page)**



Orifice flanges are widely used in conjunction with orifice meters for measuring the rate of flow of liquids and gases. They are basically the same as standard welding neck, slip-on and screwed flanges except for the provision of radial, tapped holes in the flange ring for meter connections and additional bolts to act as jack screws to facilitate separating the flanges for inspection or replacement of the orifice plate.



Blind flanges are used to blank off the ends of piping, valves and pressure vessel openings. From the standpoint of internal pressure and bolt loading, blind flanges, particularly in the larger sizes, are the most highly stressed of all American Standard flange types; however, since the maximum stresses in a blind flange are bending stresses at the center, they can safely be permitted to be higher than in other types of flanges.

1.) In Tube Turns tests of all types of flanged assemblies, fatigue failure invariably occurred in the pipe or in an unusually weak weld, never in the flange proper. The type of flange, however, and particularly the method of attachment, greatly influence the number of cycles required to cause fracture.
2.) ANSI B16.5-1961—Steel Pipe Flanges and Flanged Fittings.
3.) ASME Boiler and Pressure Vessel Code 1966, Section I, Par. P-300.

Figure 2-6. Continued. Forged steel companion flanges to attach to steel pipe by the methods indicated. By permission, Tube Turn Technologies, Inc.

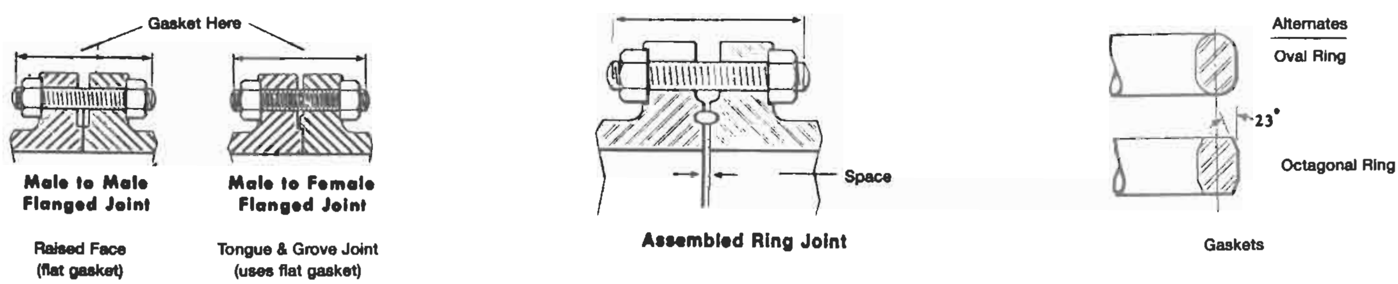
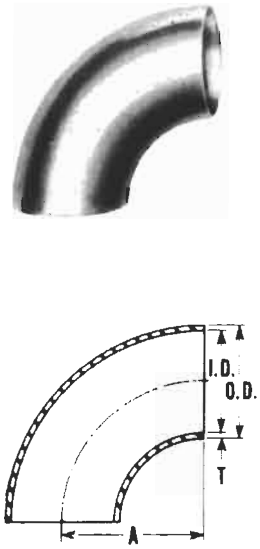


Figure 2-7. Most common flange connection joints. Cross section of a pair of flanges with bolts to draw joint tight.

(text continued from page 59)

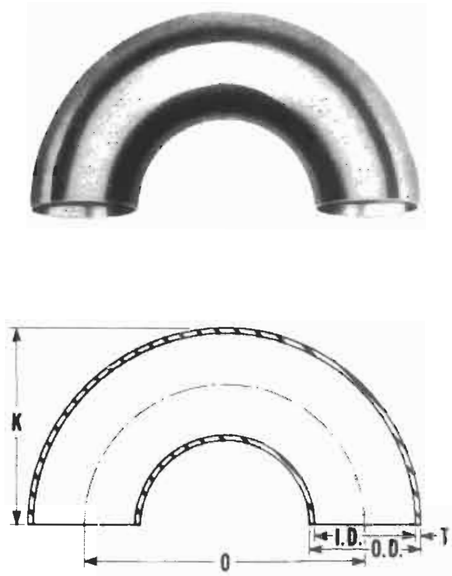
chemical/petrochemical industries, except for instrument lines (sensing, signal transmission) or high pressures above 2,000 psig.

Figure 2-9 compares the measurement differences for tubes (outside diameter) and iron or steel pipe size (IPS), nominal inside diameter. One example of dimensional comparison for IPS pipe for Schedules 5 and 10 are compared to one standard scale of tubing in Table 2-1. The tubing conforms to ANSI/ASTM A-403-78 Class CR (stainless) or MSS Manufacturers Standard Society SP-43, Sch 5S.



90° ELBOW
long radius
2WCL

45° ELBOW
long radius
2WK

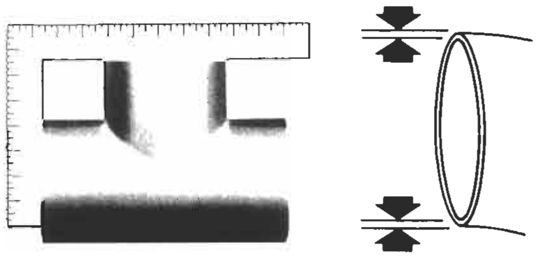


180° RETURN
long radius
2WUL
(304 S/S only)

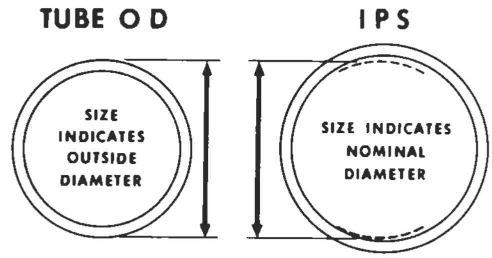
TUBE O.D. SIZE	OUTSIDE DIAMETER	INSIDE DIAMETER	WALL THICKNESS	GAUGE NUMBER	90°ELBOW	45°ELBOW	180°RETURN	
					LONG RADIUS	LONG RADIUS	LONG RADIUS	
	O.D.	I.D.	T		A	B	K	O
¾	.750	.625	.065	18	1⅛	⅝	1½	2¼
1	1.000	.870	.065	16	1½	⅝	2	3
1½	1.500	1.370	.065	16	2¼	15/16	3	4½
2	2.000	1.870	.065	16	3	1¼	4	6
2½	2.500	2.370	.065	16	3¾	1 9/16	5	7½
3	3.000	2.870	.065	16	4½	1⅞	6	9
4	4.000	3.834	.083	14	6	2½	8	12
6	6.000	5.782	.109	12	9	3¾	12 5/16	18
8	8.000	7.782	.109	12	12	5	16 5/16	24
10	10.000	9.732	.134	10	15	6¼	20⅜	30
12	12.000	11.732	.134	10	18	7½	24⅜	36

For Weights — see page 8.

Figure 2-8. Light weight stainless steel butt-weld fittings/tubing for low pressure applications. By permission, Tri-Clover, Inc.



Dependable performance; fast, easy installation
Uniformity of wall thickness and geometric accuracy of ends permit precise alignment of joints.



How Tube OD Differs from IPS
In Tube OD the size specified indicates its outside diameter . . . whereas in Iron Pipe Size (I P S), the size has reference to a nominal diameter. See Table 2-1.

Figure 2-9. Dimension comparison of tubing and IPS (iron pipe size) steel piping. By permission, Tri-Clover, Inc.

Table 2-1
Comparison of dimensions and flow area for Tubing and Iron Pipe Size (IPS) Steel Pipe.

O D TUBING				IPS PIPE					
						SCHEDULE 5S		SCHEDULE 10S	
O D TUBING SIZE	OUTSIDE DIAMETER	INSIDE DIAMETER*	FLOW AREA SQ. IN.	IPS PIPE SIZE	OUTSIDE DIAMETER	INSIDE DIAMETER	FLOW AREA SQ. IN.	INSIDE DIAMETER	FLOW AREA SQ. IN.
¾	.750	.625	.307	¾	1.050	.920	.665	.884	.614
1	1.000	.870	.595	1	1.315	1.185	1.10	1.097	.945
1½	1.500	1.370	1.47	1½	1.900	1.770	2.46	1.682	2.22
2	2.000	1.870	2.75	2	2.375	2.245	3.96	2.157	3.65
2½	2.500	2.370	4.41	2½	2.875	2.709	5.76	2.635	5.45
3	3.000	2.834	6.31	3	3.500	3.334	8.73	3.260	8.35
3½	. . .	. . .	. . .	3½	4.000	3.834	11.55	3.760	11.10
4	4.000	3.834	11.55	4	4.500	4.334	14.75	4.260	14.25
6	6.000	5.782	26.26	6	6.625	6.407	32.24	6.357	31.75
8	8.000	7.782	47.56	8	8.625	8.407	55.5	8.329	54.5
10	10.000	9.732	74.4	10	10.750	10.482	86.3	10.420	85.3
12	12.000	11.732	108.	12	12.750	12.438	121.	12.390	120.

*Based on wall thickness listed on following pages.
**Indicates greater latitude in selecting line size with capacity closest to flow requirement.

By permission Tri-Clover, Inc.

Total Line Pressure Drop

The total piping system pressure drop for a particular pipe installation is the sum of the friction drop in pipe valves and fittings, plus other pressure losses (drops) through control valves, plus drop through equipment in the system, plus static drop due to elevation or pressure level. For example, see Figure 2-2.

This total pressure loss is not necessarily required in determining the *frictional* losses in the system. It is necessary when establishing gravity flow or the pumping head requirements for a complete system.

Design practice breaks the overall problem into small component parts which allow for simple analysis and solution. This is the recommended approach for selection and sizing of process piping.

Background Information (Also see Chapter 3)

Gas or vapor density following perfect gas law:

$$\rho = 144\, P' (T)(1544/MW), \text{ lbs/cu ft} \quad (2\text{-}5)$$

Gas or vapor specific gravity referred to air:

$$S_g = \text{MW of gas/MW of air} = \text{MW of gas}/29 \quad (2\text{-}6)$$

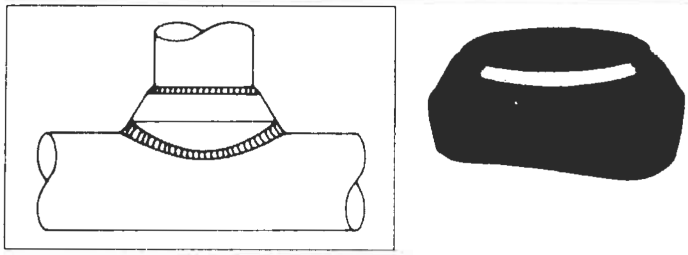
Conversion between fluid head loss in feet and pressure drop in psi, any fluid:

$$\text{Pressure drop, pounds/sq in., } \Delta P = h_L \rho/144 \quad (2\text{-}7)$$

$$\text{For water, } \Delta P = h_L/2.31, \text{ psi} \quad (2\text{-}8)$$

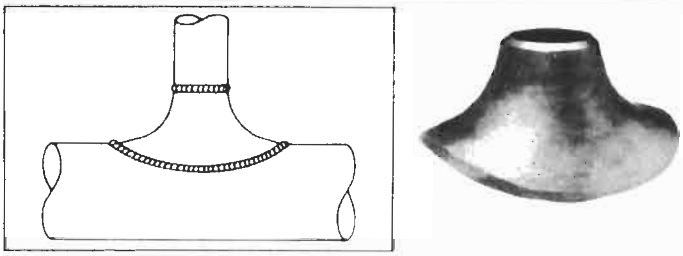
Equivalent diameter and hydraulic radius for non-circular flow ducts or pipes

Weldolet



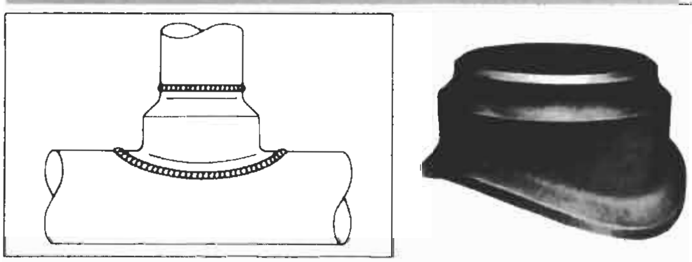
Weldolet®, an economical butt-weld branch connection, is designed to minimize stress concentrations and provide integral reinforcement.

Sweepolet



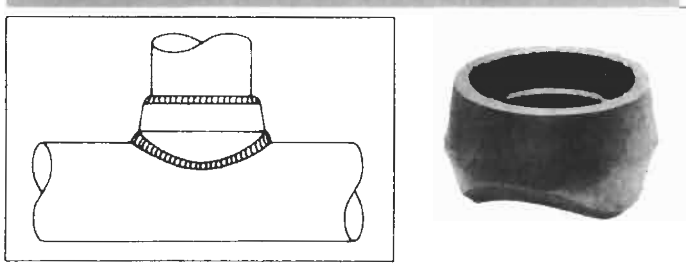
Sweepolet® is a contoured, integrally reinforced, butt-weld branch connection with a low stress intensification factor for low stresses and long fatigue life. The attachment weld is easily examined by radiography, ultrasound and other standard non-destructive techniques.

Insert Weldolet



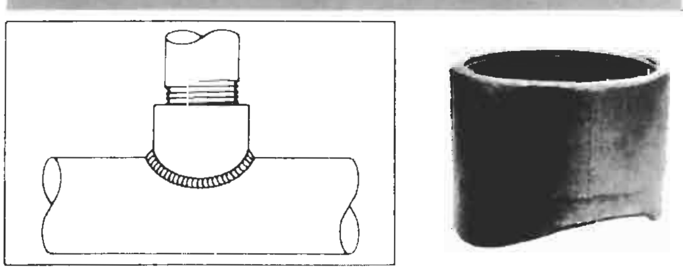
Insert Weldolet® is another contoured butt-weld branch connection used in less critical applications. Like the Sweepolet, the attachment welds are easily examined by radiography, ultrasound and other standard non-destructive techniques.

Sockolet



Sockolet® utilizes the basic Weldolet design configuration and incorporates a socket-weld outlet.

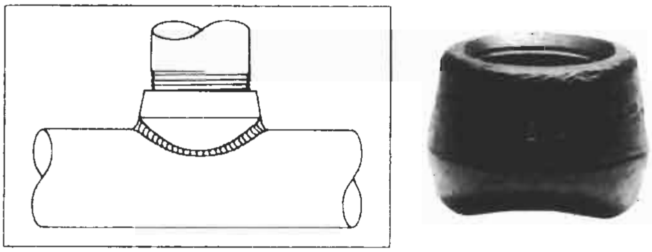
Coupolet



Coupolet® fittings are designed for use in fire protection sprinkler systems and other low pressure piping applications.

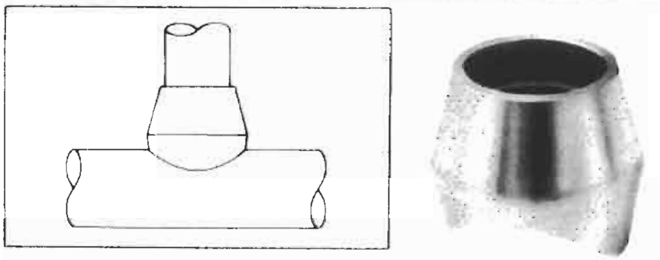
Figure 2-10. Branch connections for welding openings into steel pipe. See Figure 2-4C for alternate welding fittings. By permission, Bonney Forge Corp., Allentown, PA.

Thredolet



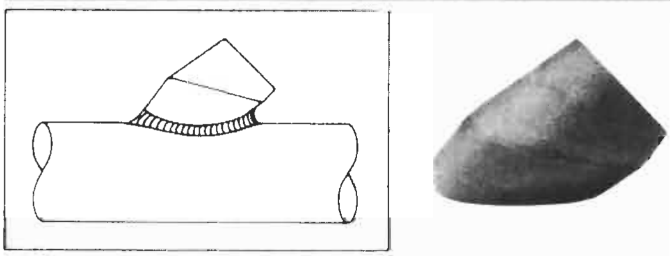
Thredolet® utilizes the basic Weldolet configuration, provides a threaded outlet branch connection.

Brazolet



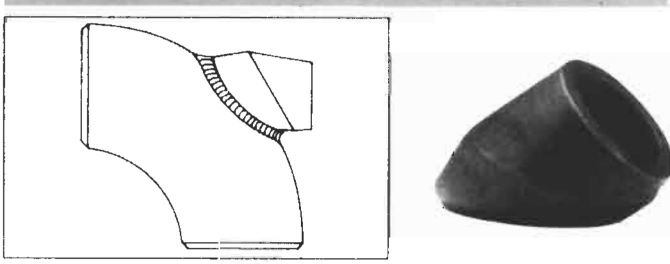
Brazolet® is designed for use with KLM and IPS brass or copper piping or copper tubing.

Latrolet



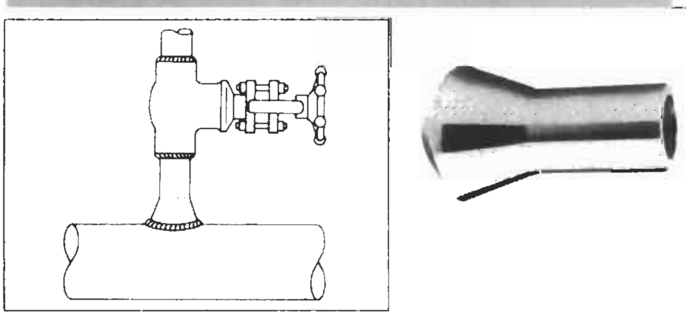
Latrolet® used for 45° lateral connections, is available butt-weld to meet your specific reinforcement requirements, and 3000# or 6000# classes for socket weld and threaded applications.

Elbolet



Elbolet® is used on 90° Long Radius Elbows (can be manufactured for Short Radius Elbows) for thermowell and instrumentation connections. Available butt-weld to meet your specific reinforcement requirements, and 3000# and 6000# classes for socket weld and threaded applications.

Nipolet



Nipolet® is a one piece fitting for valve take-offs, drains and vents. Available with male socket-weld or male threaded outlets.

Figure 2-10. Continued.

R_H = hydraulic radius, ft

$$R_H = \frac{\text{cross-section for fluid flow, sq ft}}{\text{wetted perimeter for fluid flow, ft}} \quad (2\text{-}9)$$

D_H = hydraulic diameter, (equivalent diameter), ft

$$D_H = 4\,R_H,\ \text{ft} \quad (2\text{-}10)$$

d_H = hydraulic diameter, (equivalent diameter), in.

$$d_H = 48\,R,\ \text{in.} \quad (2\text{-}11)$$

$$d_H = \frac{4\ (\text{cross-section area for flow}),\ \text{sq in.}}{(\text{wetted perimeter for fluid flow}),\ \text{in.}} \quad (2\text{-}12)$$

For the narrow shapes with width small relative to length, the hydraulic radius is approximately [3]:

$$R_H \approx 1/2\ (\text{width of passage}) \quad (2\text{-}13)$$

For those non-standard or full circular configurations of flow, use d equivalent to actual flow area diameter, and D equivalent to $4R_H$.

$$d = 4\left[\frac{\text{cross-section available for fluid flow, of duct}}{\text{wetted perimeter of duct}}\right]$$

This also applies to circular pipes or ducts and oval and rectangular ducts not flowing full. The equivalent diameter is used in determining the Reynolds number for these cases, but does not apply to very narrow or slotted or annular flow cross-sections.

Minimum size of pipe is sometimes dictated by structural considerations, i.e., 1½-inch Schedule 40 steel pipe is considered the smallest size to span a 15′ to 20′ pipe rack without intermediate support.

Gravity flow lines are often set at 1¼ inch to 2 inch minimum, disregarding any smaller calculated size as a potential source of trouble.

Overflow pump suction lines are designed for about a one foot/second velocity, unless a higher velocity is necessary to keep small solids or precipitates in suspension. Suction line sizes should be larger than discharge sizes.

Flooded suction lines to pumps must be designed so that pressure drop in the pipe is safely less than head available.

As a *general guide,* for pipe sizes use:

threaded pipe—up to and including 1½ in. or 2 in. nominal

welded pipe—2 in. and larger

Situations may dictate deviations.

Never use cast iron fittings or pipe in process situations unless there is only gravity pressure head (or not over 10 psig) or the fluid is nonhazardous. One exception is in some concentrated sulfuric acid applications; however, extreme caution must be used in the design of the safety of the system area. Never use in pulsing or shock service.

Never use malleable iron fittings or pipe unless the fluid is nonhazardous and the pressure not greater than 25 psig. Always use a pressure rating at least four times that of the maximum system pressure. Also, never use cast iron or malleable iron fittings or valves in pressure pulsating systems or systems subject to physical shock.

Use forged steel fittings for process applications as long as the fluid does not create a serious corrosion problem. These fittings are attached to *steel* pipe and/or each other by threading, socket welding, or direct welding to steel pipe. For couplings attached by welding to pipe, Figure 2-4B, use either 2,000 psi or 6,000 psi rating to give adequate area for welding without distortion, even though the process system may be significantly lower (even atmospheric). Branch connections are often attached to steel pipe using forged Weldolets® or Threadolets® (Figure 2-10).

® = Bonney Forge, Allentown, Pa.

Mean pressure in a gas line [57].

$$P\ (\text{mean or average}) = \frac{2}{3}\left[(P_1 + P_2) - \frac{P_1 P_2}{P_1 + P_2}\right] \quad (2\text{-}14)$$

This applies particularly to long flow lines.

The usual economic range for pressure loss due to liquid flow; (a) Suction piping—½ to 1¼ psi per 100 equivalent feet of pipe.

(b) Discharge piping—1 to 5 psi per 100 equivalent feet of pipe.

The Appendix presents useful carbon steel and stainless steel pipe data.

Reynolds Number, R_e (Sometimes used N_{RE})

This is the basis for establishing the condition or type of fluid flow in a pipe. Reynolds numbers below 2000 to 2100 are usually considered to define laminar or viscous flow; numbers from 2000 to 3000–4000 to define a transition region of peculiar flow, and numbers above 4000 to define a state of turbulent flow. Reference to Figure 2-3 and Figure 2-11 will identify these regions, and the friction factors associated with them [2].

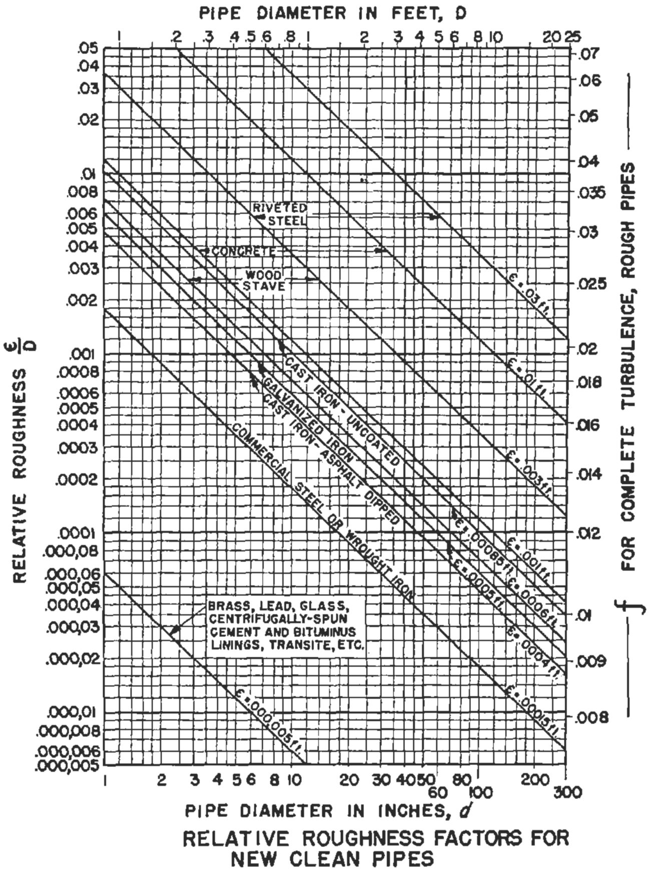


Figure 2-11. Relative roughness factors for new clean pipe. Reprinted by permission from *Pipe Friction Manual,* 1954, The Hydraulic Institute. Also see *Engineering Data Book,* 1st Ed., 1979, The Hydraulic Institute. Data from L. F. Moody, see note Figure 2-3.

$$R_e = \frac{Dv\rho}{\mu_e} = \frac{123.9\, dv\rho}{\mu} = \frac{6.31\, W}{d\mu} \qquad (2\text{-}15)$$

$$R_e = \frac{22,700\, q\, \rho}{d\mu} = \frac{50.6\, Q\rho}{d\mu} = \frac{0.482\, q_h'\, S_g}{d\mu} \qquad (2\text{-}16)$$

Friction Factor, f

For laminar or viscous flow:

$$f = 64/R_e \qquad (2\text{-}17)$$

For transition and turbulent flow, use Figure 2-11 with Figure 2-3, and Figure 2-12A and 2-12B as appropriate.

Friction factor in long steel pipes handling wet (saturated with water vapor) gases such as hydrogen, carbon monoxide, carbon dioxide, nitrogen, oxygen and similar materials should be considered carefully, and often increased by a factor of 1.2 to 2.0 to account for corrosion.

Important Note: The Moody [1] friction factors reproduced in this text (Figure 2-3) are consistent with the published values of references [1, 2, 3], and cannot be used with the values presented in *Perry's Handbook* [5], as Perry's values for, f, are one-fourth times the values cited in this chapter. It is essential to use f values with the corresponding formulas offered in the appropriate text.

The Colebrook equation [6, 58] is considered a reliable approach to determining the friction factor, f (Moody factor)

$$\frac{1}{\sqrt{f}} = -\log_{10}\left(\frac{\varepsilon}{3.7D} + \frac{2.51}{R_e\sqrt{f}}\right) \qquad (2\text{-}18)$$

$$R_e = \frac{vD}{v'}$$

Note that the term ε/D is the relative roughness from Figure 2-11. The solution of the above equation is trial and error. Colebrook [6] also proposed a direct solution equation that is reported [7] to have

$$f = 1.8 \log_{10} (R_e/7)^{-2} \qquad (2\text{-}19)$$

The equation proposed by Churchill [8] is also a direct solution with good accuracy [7].

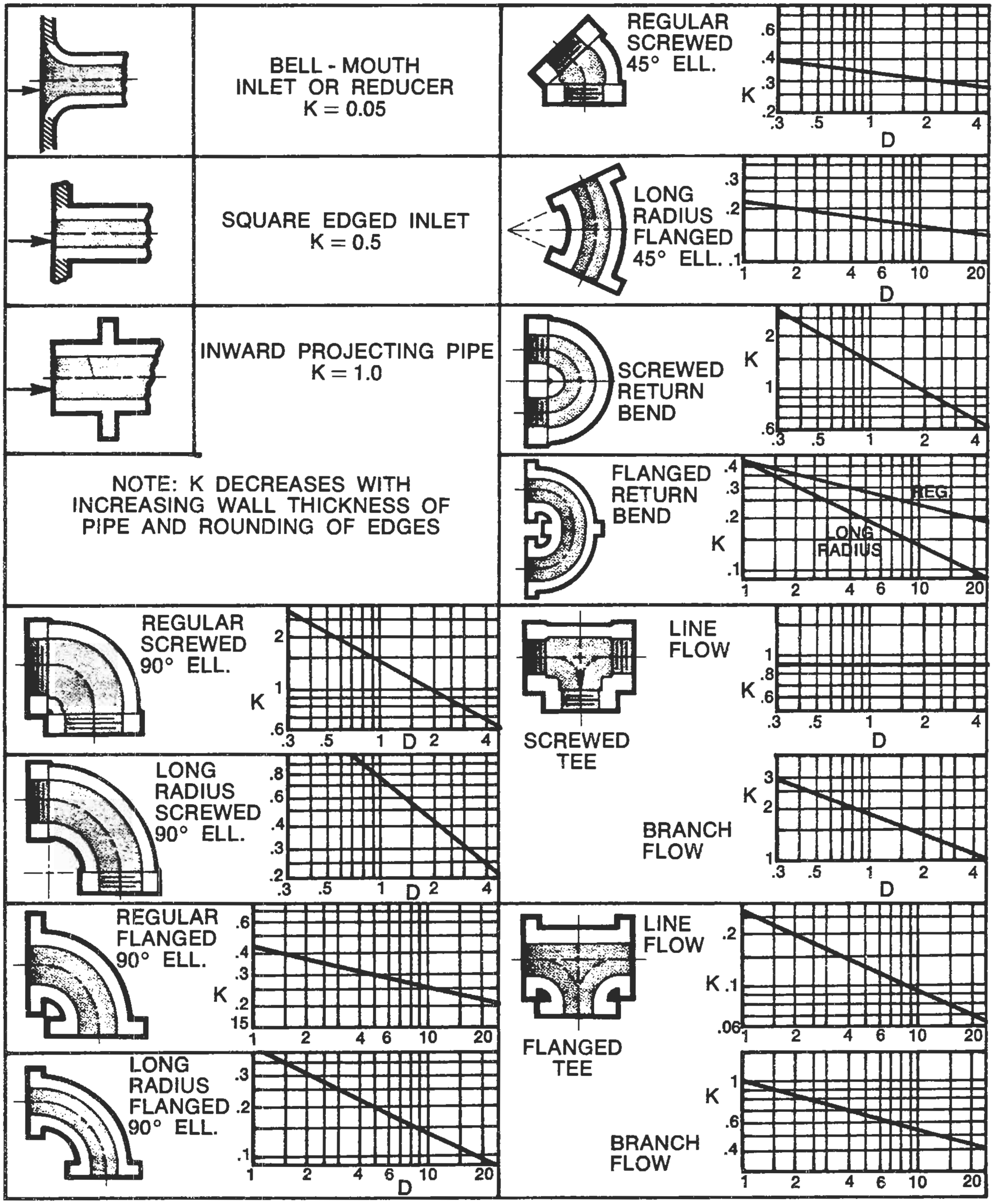
Friction Head Loss (Resistance) in Pipe, Fittings, and Connections

Friction head loss develops as fluids flow through the various pipes, elbows, tees, vessel connections, valves, etc. These losses are expressed as loss of fluid static head in feet of fluid flowing.

Pipe—Relative Roughness

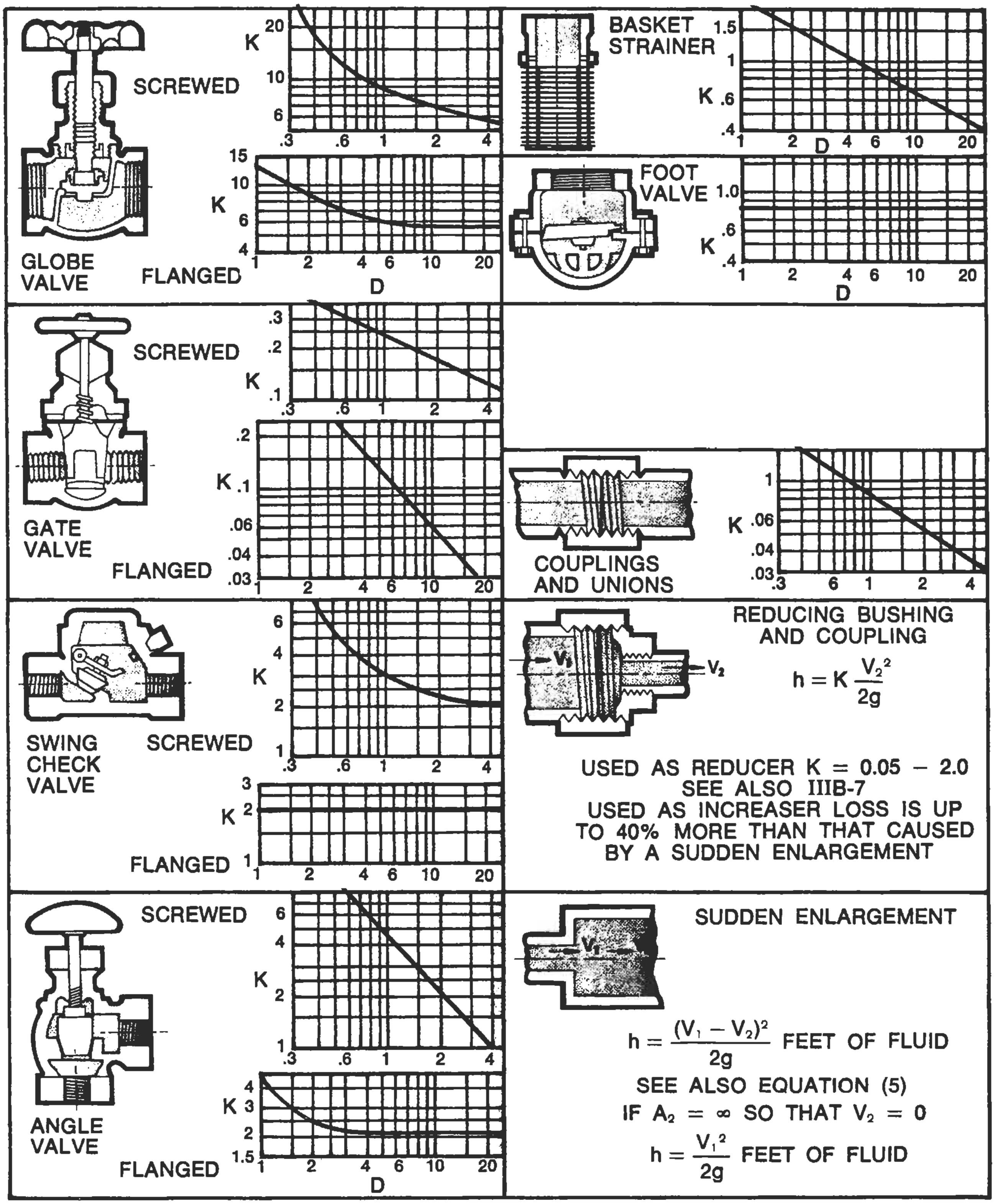
Pipe internal roughness reflects the results of pipe manufacture or process corrosion, or both. In designing a flow system, recognition must be given to (a) the initial internal pipe condition as well as (b) the expected condition after some reasonable life period, such as 10, 15, or 20 years in service. Usually a 10- to 15-year life period is a reasonable expectation. It is not wise to expect smooth internal conditions over an extended life, even for water, air, or oil flow because some actual changes can occur in the internal surface condition. Some fluids are much worse in this regard than others. New, clean steel pipe can be adjusted from the initial clean condition to some situation allowing for the additional roughness. The design roughened condition can be interpolated from Figure 2-11 to achieve a somewhat more roughened condition, with the corresponding *relative roughness* ε/D value.

ε = epsilon, absolute roughness factor, ft
D = pipe inside diameter, ft



$$h = K\frac{V^2}{2g} \text{ FEET OF FLUID}$$

Figure 2-12A. Resistance coefficients for fittings. Reprinted by permission, Hydraulic Institute, *Engineering Data Book*, 1st Ed., 1979, Cleveland, Ohio.



$$h = K \frac{V^2}{2g} \text{ FEET OF FLUID}$$

Figure 2-12B. Resistance coefficients for valves and fittings. Reprinted by permission, Hydraulic Institute, *Engineering Data Book*, 1st Ed., 1979, Cleveland, Ohio.

Note that the ε/D factor from Figure 2-11 is used directly in Figure 2-3. As an example that is only applicable in the range of the charts used, a 10% increase in ε/D to account for increased roughness, yields from Figure 2-3, an f of only 1.2% greater than a commercial condition pipe. Generally the accuracy of reading the charts does not account for large fluctuations in f values. Of course, f, can be calculated as discussed earlier, and a more precise number can be achieved, but this may not mean a significantly greater accuracy of the calculated pressure drop. Generally, for industrial process design, experience should be used where available in adjusting the roughness and effects on the friction factor. Some designers increase the friction factor by 10% to 15% over standard commercial pipe values.

Pressure Drop in Straight Pipe: Incompressible Fluid

The *frictional* resistance or pressure drop due to the flow of the fluid, h_f, is expressed by the Darcy equation:

$$h_f = \frac{fLv^2}{D(2g)}, \text{ ft of fluid, resistance} \qquad (2\text{-}2)$$

$$\text{or, } \Delta P = \frac{\rho f v^2 L}{144D\,(2g)}, \text{ resistance loss, lbs/sq in.} \qquad (2\text{-}1)$$

Note: these values for h_f and ΔP are *differentials* from point (1) upstream to point (2) downstream, separated by a length, L. These are *not absolute pressures,* and cannot be meaningfully converted to such units. Feet of fluid, h_f, can be converted to pounds per square inch by:

$$h_f = \frac{\Delta P(144)}{\rho} = \text{ft, for any fluid} \qquad (2\text{-}20)$$

Referenced to water, convert psi to feet of water:

$$h_f\ (\text{ft}) = \frac{[(1\ \text{lb/sq in.})]\ (144)}{62.3\ \text{lb/cu ft}} = 2.31\ \text{ft} \qquad (2\text{-}21)$$

For conversion, 1 psi h 2.31 ft of water head

This represents a column of water at 60°F, 2.31 feet high. The bottom pressure is one pound per square inch (psi) on a gauge. The pressure at the bottom as psi will vary with the density of the fluid. For fluids other than water, the relationship is:

$$1\ \text{psi} = 2.31/(\text{Sp Gr rel. to water}),\ \text{ft fluid} \qquad (2\text{-}22)$$

With extreme velocities of liquid in a pipe, the downstream pressure may fall to the vapor pressure of the liquid and cavitation with erosion will occur. Then the calculated flow rates or pressure or pressure drops are not accurate or reliable.

Pressure Drop in Fittings, Valves, Connections: Incompressible Fluid

The resistance to flow through the various "piping" components that make up the system (except vessels, tanks, pumps—items which do not necessarily provide frictional resistance to flow) such as valves, fittings, and connections into or out of equipment (not the loss through the equipment) are established by test and presented in the published literature, but do vary depending on the investigator.

Resistance to fluid flow through pipe and piping components is brought about by (1) pipe component internal surface roughness along with the density and viscosity of the flowing fluid, (2) directional changes in the system through the piping components, (3) obstructions in the path to flow, and (4) changes in system component cross-section and shape, whether gradual or sudden.

$$h_f = K\,(v^2/2g),\ \text{ft of the fluid flowing} \qquad (2\text{-}23)$$

Velocity and Velocity Head

The average or mean velocity is determined by the flow rate divided by the cross section area for flow in feet per second, v. The velocity in a pipe is related to the decrease in static head *due to the velocity only* by:

$$h_L = h_f = v^2/2g,\ \text{termed velocity head, ft} \qquad (2\text{-}24)$$

Note the static reduction (loss) due to fluid flowing through a system component (valve, fitting, etc.) is expressed in terms of velocity head, using the resistance coefficient, K, in the equation above. This K represents the number of velocity heads lost due to flow through the respective system component. It is always associated with diameter for flow, hence, velocity through the component. Actually, for most system components, the static losses due to pipe friction due to internal roughness and the actual length of flow path are minor when compared to one or more of the other losses listed in the previous paragraph [3]. The resistance coefficient, K, is considered independent of friction factor or Reynolds number and is treated as a constant for any component obstruction (valve or fitting) in a piping system under all conditions of flow, including laminar.

From the Darcy equation [3]:

$$K = (f\,L/D) \tag{2-25}$$

Head loss through a pipe, $h_L = (f)\ (L/D)\ (v^2/2g)$ (2-26)

Head loss through a valve (for instance),

$$h_L = K(v^2/2g) \tag{2-27}$$

where L/D is the equivalent length in pipe diameters of straight pipe that will cause or develop the same pressure drop as the fitting, component, or other obstruction under the same flow conditions. K is a constant for all flow conditions through a given system component; thus, the value of L/D for the specific component must vary inversely with the change in friction factor for varying flow conditions [3].

For various components' K values, see Figures 2-12A, 2-12B, 2-13A, 2-13B through 2-16 and Tables 2-2 and 2-3.

Common Denominator for Use of "K" Factors in a System of Varying Sizes of Internal Dimensions

K value can be adjusted to a common reference pipe size:

$$K_2 = K_1\ (d_2/d_1)^4 \tag{2-28}$$

where subscript 1 is the known resistance from standard K factor tables or charts (these are based on standard ANSI pipe/fitting dimensions), and subscript 2 is the corrected resistance coefficient used to identify the inside diameter of the actual *pipe* into which the valve or fitting is connected or installed.

The K values determined for various valves, fittings, etc., are specific to the system, particularly valves. For example, most reliable data[8] have been developed with valves and fittings installed in pipe of specific dimensions, then, if a larger or smaller inside diameter valve or fitting is to be installed in a pipe of different inside diameter, a correction of the K value should be made.

Pressure drop through line systems containing more than one pipe size can be determined by (a) calculating the pressure drop separately for each section at assumed flows, or (b) determining the K *totals* for each pipe size separately, and then converting to one selected size and using that for pressure drop calculations. For example, using

$$K_2 = K_1\ (d_2/d_1)^4 \tag{2-28}$$

and thereby converting to a common base K, they are then additive, *when all referenced to the same size pipe.* Flow then can be determined for a fixed head system by

$$\text{GPM liquid} = 19.65\ d^2\ (h_L/K)^{1/2} \tag{2-29}$$

Of course, by selecting the proper equation, flows for vapors and gases can be determined in the same way, as the K value is for the fitting or valve and *not* for the fluid.

The head loss has been correlated as a function of the velocity head equation using K as the resistance coefficient in the equation.

$$h_L = K\,v^2/2g = K\,v^2/64.4,\ \text{ft of fluid} \tag{2-27}$$

For a system of multiple components of valves, pipe, and fittings, Equation 2-25 can be used to establish a component size to which each separate resistance can be expressed as a "common denominator," or common pipe size. Under these conditions, the "corrected" K values are additive and can be used as one number in Equation 2-27. These types of corrections should be made to improve and more accurately represent the pressure drop calculations.

An example procedure connecting 3-in. and 6-in. pipe and fittings, using 6-in. as the final reference, is as follows:

1. From Table 2-2, read for 3 in. Sch. 40 pipe, $f_T = 0.018$.
2. Calculate R_e for each pipe size.
3. Read friction factor, f, from Figure 2-3, using Figure 2-11.
4. Calculate K for 6-in. pipe:

   $K = 0.018\ (L/d)\ (12)$, $L_{6''}$ = ft 6-in. pipe.
5. Calculate K for 3-in. pipe, using $L_{3''}$ = ft of 3-in. pipe.
6. Then, referencing to the 6-in. pipe throughout the system:

   $K_2 = (K_{3''})\ (d_{3''}/d_{6''})^4$, representing entire pipe part of system.
7. Add K values for individual fittings and valves from Figures 2-12A through 2-16 and Tables 2-2 and 2-3.
8. Using sum of K values for 6-in. pipe, 3-in. pipe equivalent calculated above in step 6, and all items in step 7 above [3]:

$$h_L = (0.00259\ K\ Q^2)/(d_{6''})^4 \tag{2-30}$$

(text continued on page 77)

Table 2-2
"K" Factor Table: Representative Resistance coefficients (K) for Valves and Fittings

Pipe Friction Data for Clean Commercial Steel Pipe with Flow in Zone of Complete Turbulence

Nominal Size	½"	¾"	1"	1¼"	1½"	2"	2½, 3"	4"	5"	6"	8-10"	12-16"	18-24"
Friction Factor (f_T)	.027	.025	.023	.022	.021	.019	.018	.017	.016	.015	.014	.013	.012

Formulas for Calculating "K" Factors for Valves and Fittings with Reduced Port

- **Formula 1**

$$K_2 = \frac{0.8 \sin\frac{\theta}{2}(1 - \beta^2)}{\beta^4}$$

- **Formula 2**

$$K_2 = \frac{0.5\,(1 - \beta^2)\sqrt{\sin\frac{\theta}{2}}}{\beta^4}$$

- **Formula 3**

$$K_2 = \frac{2.6 \sin\frac{\theta}{2}(1 - \beta^2)^2}{\beta^4}$$

- **Formula 4**

$$K_2 = \frac{(1 - \beta^2)^2}{\beta^4}$$

- **Formula 5**

$$K_2 = \frac{K_1}{\beta^4} + \text{Formula 1} + \text{Formula 3}$$

$$K_2 = \frac{K_1 + \sin\frac{\theta}{2}[0.8\,(1 - \beta^2) + 2.6\,(1 - \beta^2)^2]}{\beta^4}$$

- **Formula 6**

$$K_2 = \frac{K_1}{\beta^4} + \text{Formula 2} + \text{Formula 4}$$

$$K_2 = \frac{K_1 + 0.5\sqrt{\sin\frac{\theta}{2}}\,(1 - \beta^2) + (1 - \beta^2)^2}{\beta^4}$$

- **Formula 7**

$$K_2 = \frac{K_1}{\beta^4} + \beta\,(\text{Formula 2} + \text{Formula 4}) \text{ when } \theta = 180^\circ$$

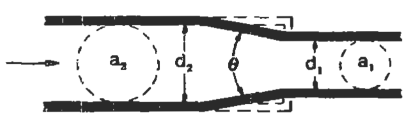
$$K_2 = \frac{K_1 + \beta\left[0.5\,(1 - \beta^2) + (1 - \beta^2)^2\right]}{\beta^4}$$

$$\beta = \frac{d_1}{d_2}$$

$$\beta^2 = \left(\frac{d_1}{d_2}\right)^2 = \frac{a_1}{a_2}$$

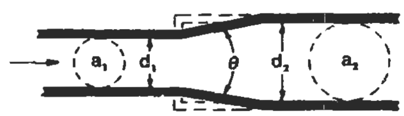
Subscript 1 defines dimensions and coefficients with reference to the smaller diameter.
Subscript 2 refers to the larger diameter.

SUDDEN AND GRADUAL CONTRACTION



If: $\theta \leq 45^\circ$ .... K_2 = Formula 1
$\theta > 45^\circ \leq 180^\circ$ ... K_2 = Formula 2

SUDDEN AND GRADUAL ENLARGEMENT

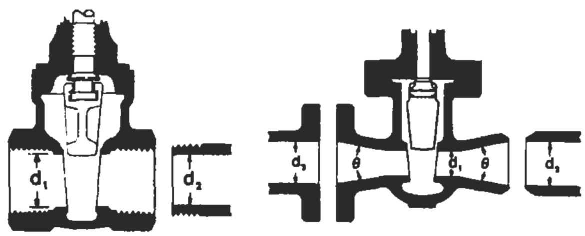


If: $\theta \leq 45^\circ$ .... K_2 = Formula 3
$\theta > 45^\circ \leq 180^\circ$ ... K_2 = Formula 4

(continued on next page)

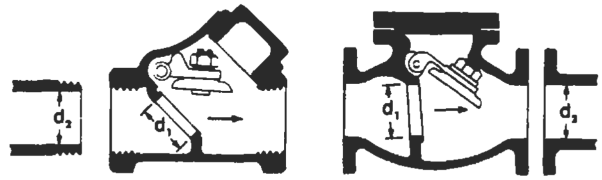
(table 2-2 continued from previous page)

GATE VALVES
Wedge Disc, Double Disc, or Plug Type



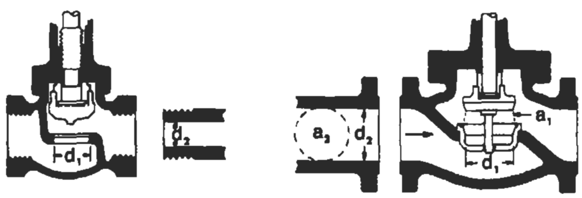
If: $\beta = 1, \theta = 0 \ldots\ldots\ldots\ldots K_1 = 8 f_T$
$\beta < 1$ and $\theta \leqq 45° \ldots\ldots\ldots K_2 =$ Formula 5
$\beta < 1$ and $\theta > 45° \leqq 180° \ldots K_2 =$ Formula 6

SWING CHECK VALVES

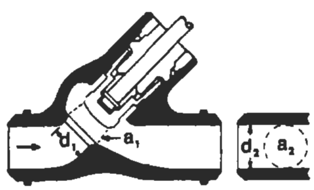


$K = 100 f_T$	$K = 50 f_T$
Minimum pipe velocity (fps) for full disc lift $= 35 \sqrt{\bar{V}}$	Minimum pipe velocity (fps) for full disc lift $= 48 \sqrt{\bar{V}}$

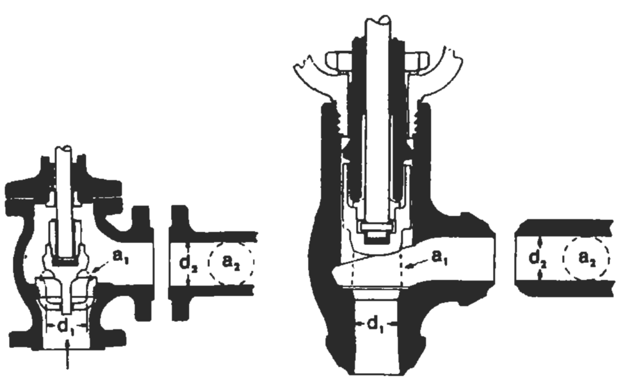
GLOBE AND ANGLE VALVES



If: $\beta = 1 \ldots K_1 = 340 f_T$



If: $\beta = 1 \ldots K_1 = 55 f_T$

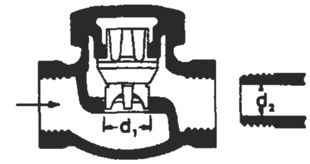


If: $\beta = 1 \ldots K_1 = 150 f_T$ If: $\beta = 1 \ldots K_1 = 55 f_T$

All globe and angle valves,
whether reduced seat or throttled,

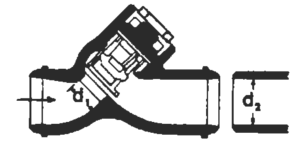
If: $\beta < 1 \ldots K_2 =$ Formula 7

LIFT CHECK VALVES



If: $\beta = 1 \ldots K_1 = 600 f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

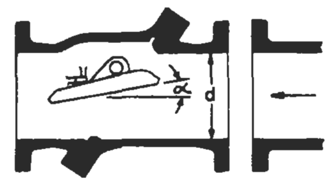
Minimum pipe velocity (fps) for full disc lift
$= 40\ \beta^2 \sqrt{\bar{V}}$



If: $\beta = 1 \ldots K_1 = 55 f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

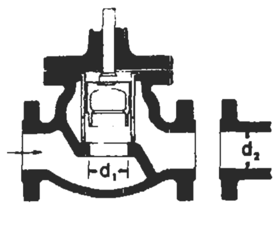
Minimum pipe velocity (fps) for full disc lift
$= 140\ \beta^2 \sqrt{\bar{V}}$

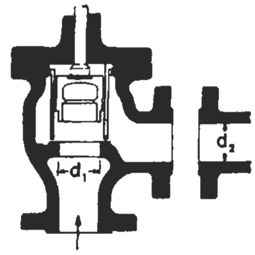
TILTING DISC CHECK VALVES



	$\alpha = 5°$	$\alpha = 15°$
Sizes 2 to 8″ . . . $K =$	$40 f_T$	$120 f_T$
Sizes 10 to 14″ . . . $K =$	$30 f_T$	$90 f_T$
Sizes 16 to 48″ . . . $K =$	$20 f_T$	$60 f_T$
Minimum pipe velocity (fps) for full disc lift $=$	$80 \sqrt{\bar{V}}$	$30 \sqrt{\bar{V}}$

STOP-CHECK VALVES
(Globe and Angle Types)



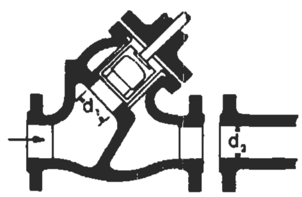


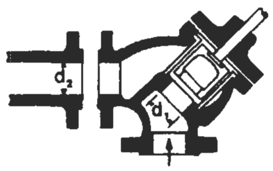
If:
$\beta = 1 \ldots K_1 = 400\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity for full disc lift
$= 55\, \beta^2 \sqrt{\bar{V}}$

If:
$\beta = 1 \ldots K_1 = 200\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity for full disc lift
$= 75\, \beta^2 \sqrt{\bar{V}}$

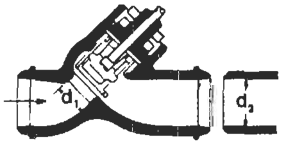


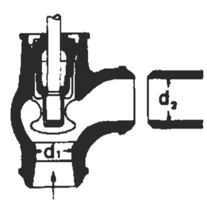


If:
$\beta = 1 \ldots K_1 = 350\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

If:
$\beta = 1 \ldots K_1 = 300\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity (fps) for full disc lift
$= 60\, \beta^2 \sqrt{\bar{V}}$





If:
$\beta = 1 \ldots K_1 = 55\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

If:
$\beta = 1 \ldots K_1 = 55\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity (fps) for full disc lift
$= 140\, \beta^2 \sqrt{\bar{V}}$

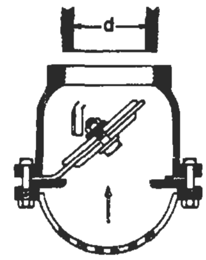
FOOT VALVES WITH STRAINER

Poppet Disc

$K = 420 f_T$

Minimum pipe velocity (fps) for full disc lift
$= 15 \sqrt{\bar{V}}$

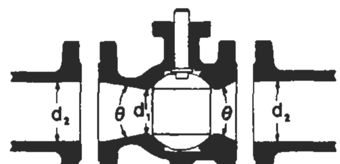
Hinged Disc



$K = 75 f_T$

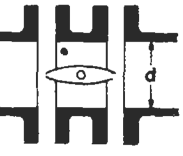
Minimum pipe velocity (fps) for full disc lift
$= 35 \sqrt{\bar{V}}$

BALL VALVES



If: $\beta = 1, \theta = 0 \ldots\ldots\ldots\ldots\ldots K_1 = 3\, f_T$
$\beta < 1$ and $\theta \leq 45° \ldots\ldots\ldots K_2 =$ Formula 5
$\beta < 1$ and $\theta > 45° \leq 180° \ldots K_2 =$ Formula 6

BUTTERFLY VALVES



Sizes 2 to 8"$\ldots K = 45\, f_T$
Sizes 10 to 14"$\ldots K = 35\, f_T$
Sizes 16 to 24"$\ldots K = 25\, f_T$

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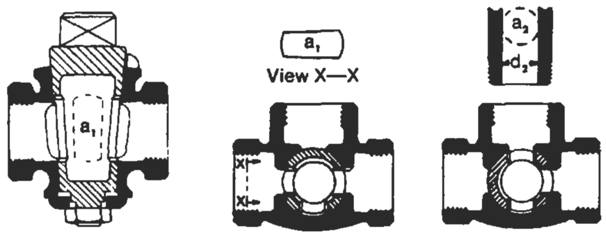
(table continued from previous page)

PLUG VALVES AND COCKS

Straight-Way **3-Way**

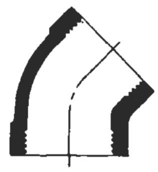


If: $\beta = 1$, $K_1 = 18 f_T$ If: $\beta = 1$, $K_1 = 30 f_T$ If: $\beta = 1$, $K_1 = 90 f_T$

If: $\beta < 1 \ldots K_2$ = Formula 6

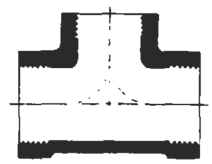
STANDARD ELBOWS

90° 45°

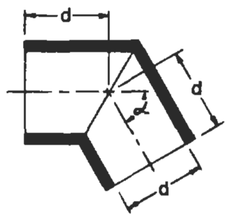
$K = 30 f_T$ $K = 16 f_T$

STANDARD TEES



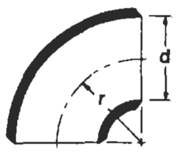
Flow thru run. . . . . . . $K = 20 f_T$
Flow thru branch. . . . $K = 60 f_T$

MITRE BENDS



α	K
0°	2 f_T
15°	4 f_T
30°	8 f_T
45°	15 f_T
60°	25 f_T
75°	40 f_T
90°	60 f_T

90° PIPE BENDS AND FLANGED OR BUTT-WELDING 90° ELBOWS



r/d	K	r/d	K
1	20 f_T	10	30 f_T
2	12 f_T	12	34 f_T
3	12 f_T	14	38 f_T
4	14 f_T	16	42 f_T
6	17 f_T	18	46 f_T
8	24 f_T	20	50 f_T

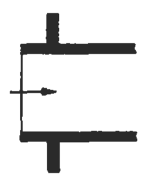
The resistance coefficient, K_B, for pipe bends other than 90° may be determined as follows:

$$K_B = (n - 1)\left(0.25\ \pi f_T \frac{r}{d} + 0.5\ K\right) + K$$

n = number of 90° bends
K = resistance coefficient for one 90° bend (per table)

PIPE ENTRANCE

Inward Projecting

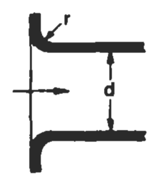


$K = 0.78$

r/d	K
0.00*	0.5
0.02	0.28
0.04	0.24
0.06	0.15
0.10	0.09
0.15 & up	0.04

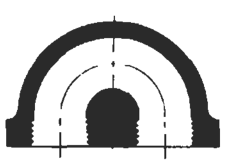
*Sharp-edged

Flush



For K, see table

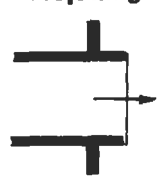
CLOSE PATTERN RETURN BENDS



$K = 50 f_T$

PIPE EXIT

Projecting **Sharp-Edged** **Rounded**



$K = 1.0$ $K = 1.0$ $K = 1.0$

By permission, Crane Co. *Technical Paper #410,* Engineering Div., 1976.

Table 2-3
Resistance Coefficients for Valves and Fittings

Approximate Range of Variation for K		
Fitting		**Range of Variation**
90 Deg. Elbow	Regular Screwed	±20 per cent above 2 inch size
	Regular Screwed	±40 per cent below 2 inch size
	Long Radius, Screwed	±25 per cent
	Regular Flanged	±35 per cent
	Long Radius, Flanged	±30 per cent
45 Deg. Elbow	Regular Screwed	±10 per cent
	Long Radius, Flanged	±10 per cent
180 Deg. Bend	Regular screwed	±25 per cent
	Regular Flanged	±35 per cent
	Long Radius, Flanged	±30 per cent
Tee	Screwed, Line or Branch Flow	±25 per cent
	Flanged, Line or Branch Flow	±35 per cent
Globe Valve	Screwed	±25 per cent
	Flanged	±25 per cent
Gate Valve	Screwed	±25 per cent
	Flanged	±50 per cent
Check Valve	Screwed	±30 per cent
	Flanged	+200 per cent −80 per cent
Sleeve Check Valve		Multiply flanged values by .2 to .5
Tilting Check Valve		Multiply flanged values by .13 to .19
Drainage Gate Check		Multiply flanged values by .03 to .07
Angle Valve	Screwed	±20 per cent
	Flanged	±50 per cent
Basket Strainer		±50 per cent
Foot Valve		±50 per cent
Couplings		±50 per cent
Unions		±50 per cent
Reducers		±50 per cent

Notes on the use of Figures 2-12 A and B, and Table 2-3
1. The value of D given in the charts is nominal IPS (Iron Pipe Size).
2. For velocities below 15 feet per second, check valves and foot valves will be only partially open and will exhibit higher values of K than that shown in the charts.
3. Reprinted by permission Hydraulic Institute, *Engineering Data Handbook*, 1st Ed., 1979, Cleveland, Ohio.

(text continued from page 72)

Validity of K Values

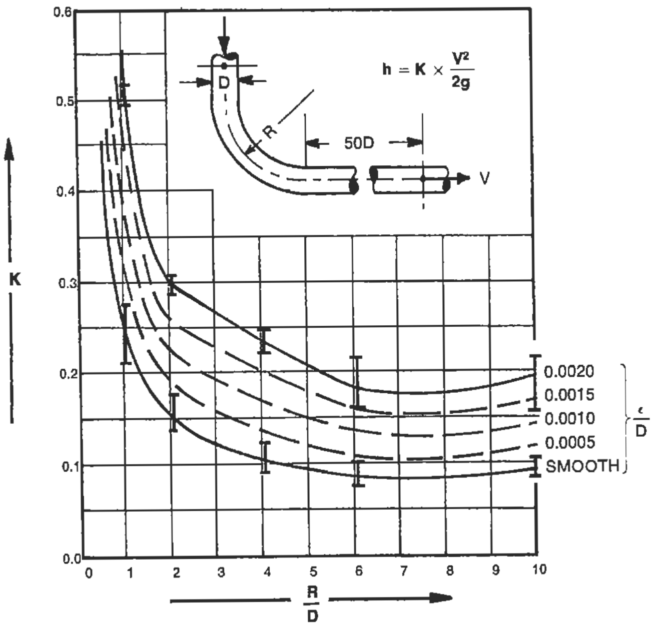
Equation 2-25 is valid for calculating the head loss due to valves and fittings for all conditions of flows: laminar, transition, and turbulent [3]. The K values are a related function of the pipe system component internal diameter and the velocity of flow for $v^2/2g$. The values in the standard tables are developed using standard ANSI pipe, valves, and fittings dimensions for each schedule or class [3]. The K value is for the size/type of pipe, fitting, or valve and not for the fluid, regardless of whether it is liquid or gas/vapor.

Laminar Flow

When the Reynolds number is below a value of 2000, the flow region is considered laminar. The pipe friction factor is defined as:

$$f = 64/R_e \tag{2-17}$$

Between R_e of 2000 and 4000, the flow is considered unsteady or unstable or transitional where laminar motion and turbulent mixing flows may alternate randomly [3]. K values can still be calculated from the



D

Note: 1.) Use 0.00085 ft for ε/D for uncoated cast iron and cast steel elbows.
2.) Not reliable when R/D < 1.0.
3.) R = radius of elbow, ft

Figure 2-13A. Resistance Coefficients for 90° bends of uniform diameter for water. Reprinted by permission, Hydraulic Institute, *Engineering Data Book,* 1st Ed., 1979, Cleveland, Ohio.

Reynolds number and the friction factor for all conditions of flow using the appropriate f and K values.

$$K = f\,(L/D) \qquad (2\text{-}25)$$

$$h_f = K(v_1 - v_2)^2/2g \qquad (2\text{-}31)$$

and:

$$h_f = (f\,L/D)\,(v^2/2g),\ \text{ft fluid for pipe} \qquad (2\text{-}26)$$

$$h_f = (K)\,(v^2/2g),\ \text{ft fluid for valves and fittings} \qquad (2\text{-}27)$$

$$\Delta P/100\ \text{eq. ft}^* = 0.0668\,(\mu v/d^2) = 0.0273\,\mu Q/d^4,\ \text{psi}/100\ \text{eq. ft} \qquad (2\text{-}32)$$

$$\Delta P = (\Delta P/100)\,(L_{eq}),\ \text{psi} \qquad (2\text{-}33)$$

*Equivalent feet of straight pipe; i.e., straight pipe plus *equivalents* for valves, fittings, other system components (except vessels, etc.). Therefore,

ΔP/100 eq. ft = pressure drop (friction) per 100 equivalent feet of straight pipe

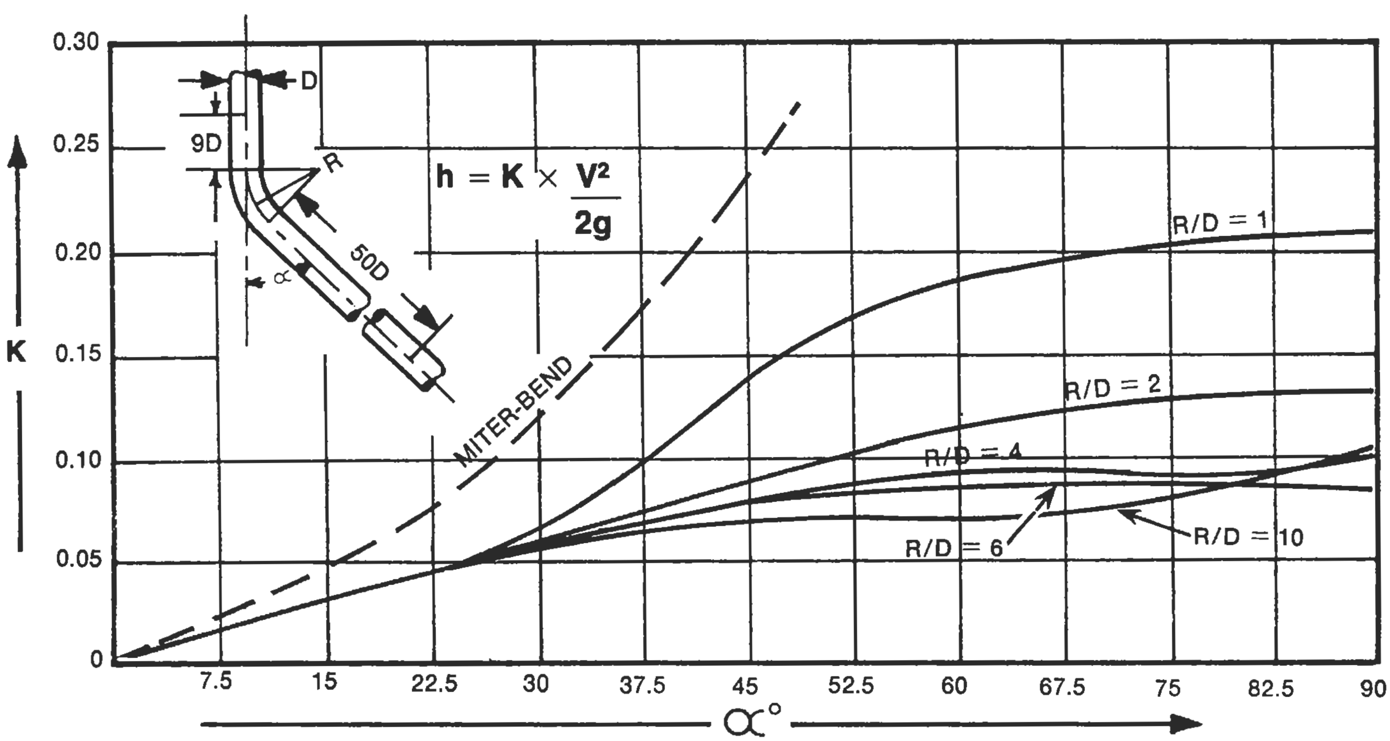
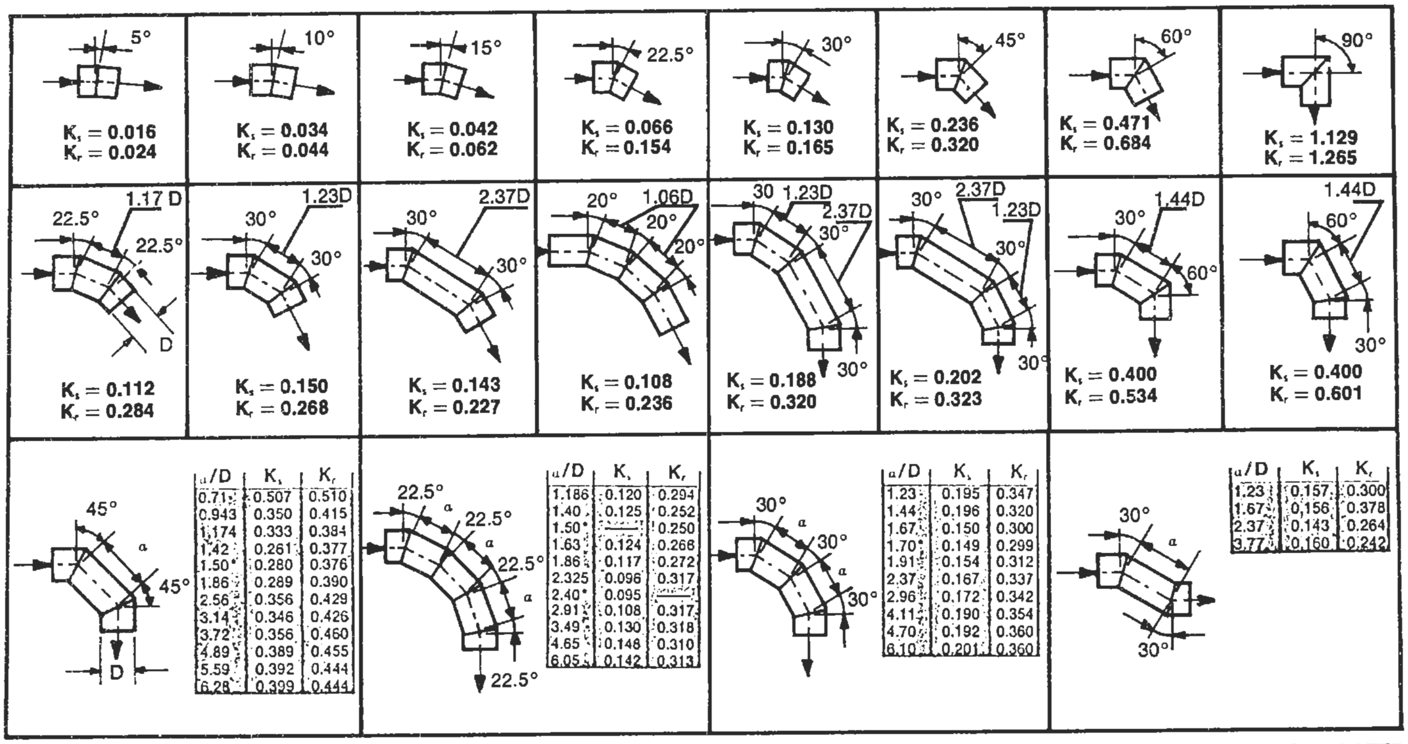


Figure 2-13B. Resistance coefficients for bends of uniform diameter and smooth surface at Reynolds number = 2.25×10^5. Reprinted by permission, Hydraulic Institute, *Engineering Data Book,* 1st Ed., 1979, Cleveland, Ohio.



α/D	K_s	K_r
0.71	0.507	0.510
0.943	0.350	0.415
1.174	0.333	0.384
1.42	0.261	0.377
1.50*	0.280	0.376
1.86	0.289	0.390
2.56	0.356	0.429
3.14	0.346	0.426
3.72	0.356	0.460
4.89	0.389	0.455
5.59	0.392	0.444
6.28	0.399	0.444

α/D	K_s	K_r
1.186	0.120	0.294
1.40	0.125	0.252
1.50*	—	0.250
1.63	0.124	0.266
1.86	0.117	0.272
2.325	0.096	0.317
2.40*	0.095	—
2.91	0.108	0.317
3.49	0.130	0.318
4.65	0.148	0.310
6.05	0.142	0.313

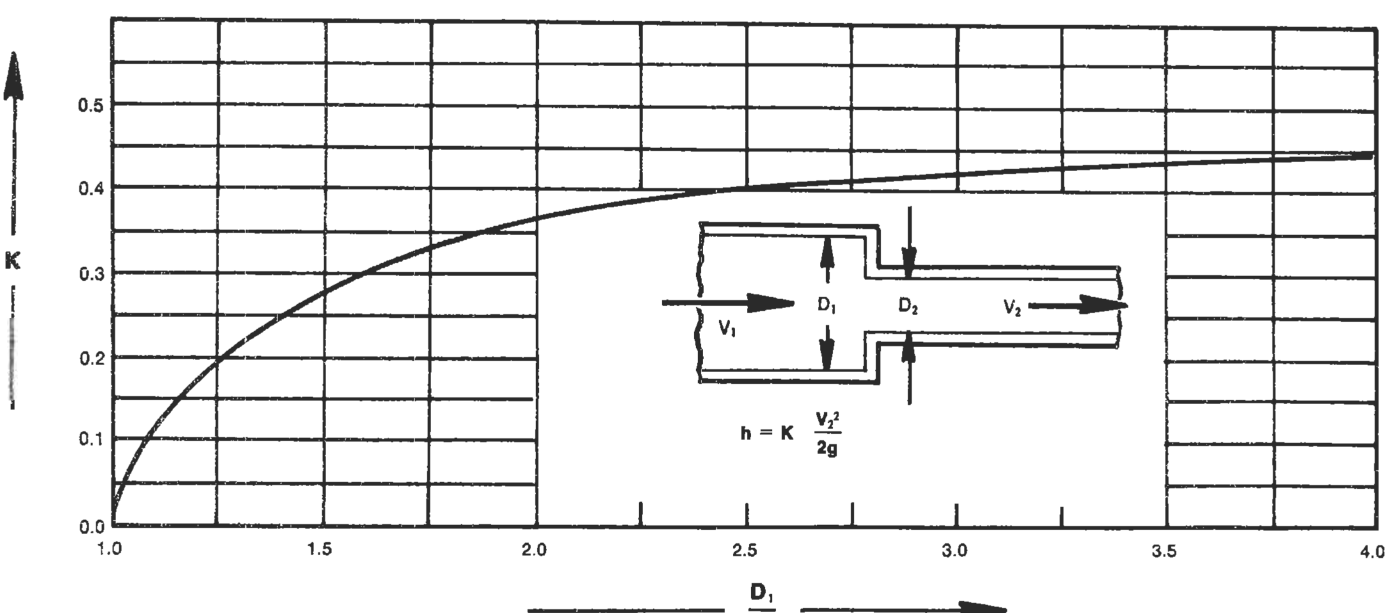
α/D	K_s	K_r
1.23	0.195	0.347
1.44	0.196	0.320
1.67	0.150	0.300
1.70*	0.149	0.299
1.91	0.154	0.312
2.37	0.167	0.337
2.96	0.172	0.342
4.11	0.190	0.354
4.70	0.192	0.360
6.10	0.201	0.360

α/D	K_s	K_r
1.23	0.157	0.300
1.67	0.156	0.378
2.37	0.143	0.264
3.77	0.160	0.242

K_s = RESISTANCE COEFFICIENT FOR SMOOTH SURFACE
K_r = RESISTANCE COEFFICIENT FOR ROUGH SURFACE, $\frac{\epsilon}{D} \cong 0.0022$

*OPTIMUM VALUE OF α INTERPOLATED

Figure 2-14. Resistance coefficients for miter bends at Reynolds number = 2.25×10^5 for water. Reprinted by permission, Hydraulic Institute *Engineering Data Book,* 1st Ed., 1979, Cleveland, Ohio.



RESISTANCE COEFFICIENTS FOR REDUCERS

Figure 2-15. Resistance coefficients for reducers for water. Reprinted by permission, Hydraulic Institute, *Engineering Data Book,* 1st Ed., 1979, Cleveland, Ohio.

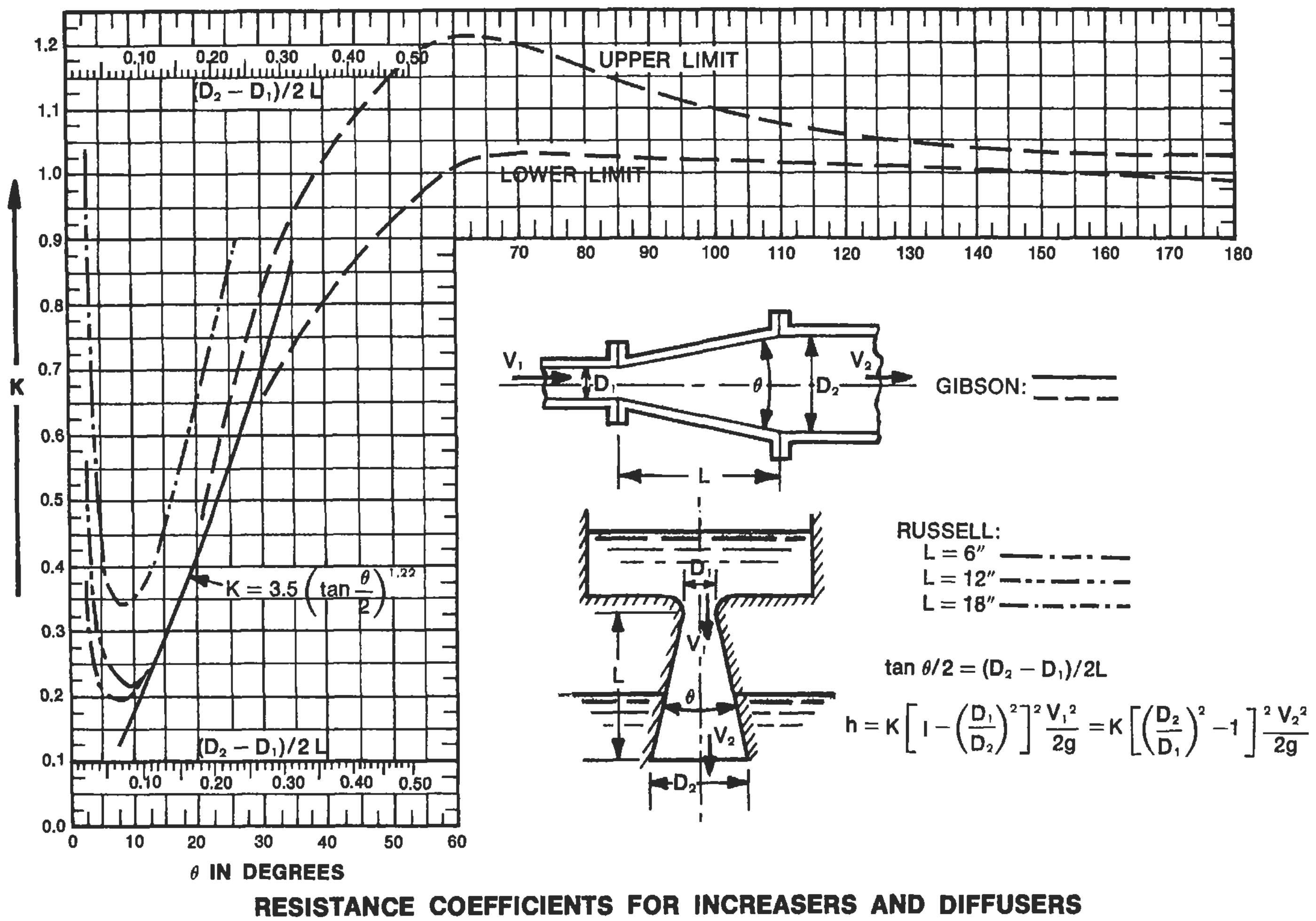


Figure 2-16. Resistance coefficients for increasers and diffusers for water. Reprinted by permission, Hydraulic Institute, *Engineering Data Book,* 1st Ed., 1979, Cleveland, Ohio.

Sudden Enlargement or Contraction [2]

For sudden enlargements in a pipe system when there is an abrupt change from a smaller pipe flowing into a larger pipe, the resistance coefficient, K, is given by:

For sudden enlargement:

$$K_1 = (1 - D_1^2/D_2^2)^2 = (1 - \beta^2)^2 \quad (2\text{-}28)$$

where subscripts 1 and 2 refer to the smaller (upstream) and larger pipes respectively [3],
or,

$$h_f = K\,(v^2/2g), \text{ ft of fluid, friction} \quad (2\text{-}27)$$

$$h_f = K_1\,[1 - (d_1^2/d_2^2)^2]^2\,(v^2{}_1/2g), \text{ ft of fluid} \quad (2\text{-}34)$$

$$K_1 = (1 - d_1^2/d_2^2)^2 \quad (2\text{-}35)$$

For sudden pipe system contractions as represented in Figure 2-12A through 2-16, the values of the resistance coefficient, K, can be read from the charts. For more details for various angles of enlargements and contractions, see References [3] and [2].

For sudden contractions:

$$K_1\ 0.5\,(1 - d_1^2/d_2^2) = 0.5\,(1 - \beta^2) \quad (2\text{-}36)$$

Note: Subscripts 1 and 2 indicate small and large pipes respectively.

Then, $h_f = K_1\,(v_1^2/2g)$, ft (2-27)

Piping Systems

The K coefficient values for each of the items of pipe, bends, valves, fittings, contractions, enlargements, entrance/exits into/from vessels are additive as long as they are on the same size basis (see Table 2-2 and Figures 2-12A through 2-16). Thus the resistance equation is applicable to calculate the head or pressure loss through the specific system when the combined K value is used.

$$h_f = K\,(v^2/2g) \quad (2\text{-}27)$$

$$\text{or, } h_f = f\,(L/D)\,(v^2/2g) \quad (2\text{-}26)$$

where K = summation of all K values in a specific system, when all are on the same size (internal flow) basis. See discussion in "Common Denominator" section.

Resistance of Valves

Figure 2-12B and Table 2-2 present several typical valves and connections, screwed and flanged, for a variety of sizes or internal diameters. These do not apply for mixtures of suspended solids in liquids; rather specific data for this situation is required (see [2]). Reference [3] presents data for specific valves.

Valves such as globes and angles generally are designed with changes in flow direction internally, and thereby, exhibit relatively high flow resistances. These same types of valves exhibit even greater resistances when they are throttled down from the "wide open" position for control of flow to a smaller internal flow path. For design purposes, it is usually best to assume a ½ or ¼ open position, rather than wide open. Estimated K values can be determined [3] by reference to Figures 2-12A through 2-16 and Tables 2-2 and 2-3.

where K_1 = refers to coefficient for smaller diameter
K_2 = refers to coefficient for larger diameter
β = ratio of diameters of smaller to larger pipe size
θ = angles of convergence or divergence in enlargements or contractions in pipe systems, degrees.

From Reference [3], K values for straight-through valves, such as gate and ball (wide open), can also be calculated. These types of valves are not normally used to throttle flow, but are either open or closed.

For sudden and gradual (Note: Sub 1 = smaller pipe; Sub 2 = larger pipe)

$$K_2 = K_1/\beta^4, \quad (2\text{-}37)$$

for $\theta \gtreqless 45°$, as enlargements:

$$K_2 = 2.6\,[(\sin\theta/2)\,(1-\beta^2)^2]/\beta^4 \quad (2\text{-}38)$$

for $\theta \gtreqless 45°$, as contractions

$$K_2 = [0.8\,(\sin\theta/2)\,(1-\beta^2)]/\beta^4 \quad (2\text{-}39)$$

For higher resistance valves, such as globes and angles, the losses are less than sudden enlargements or contractions situations. For these reduced seat valves the resistance coefficient K, can be calculated as [3]:

At $\theta \gtreqless 180$, for sudden and gradual enlargements:

$$K_2 = [(1-\beta^2)^2]/\beta^4 \quad (2\text{-}40)$$

At $\gtreqless 180$, for gradual contraction:

$$K_2 = [\{0.5\,(\sin\theta/2)^{1/2}\}\,(1-\beta^2)]/\beta^4 \quad (2\text{-}41)$$

The use of these equations requires some assumptions or judgment regarding the degree of opening for fluid flow. Even so, this is better than assuming a wide open or full flow condition, which would result in too low a resistance to flow for the design situation.

Flow Coefficients for Valves, C_v

Flow coefficients (not resistance) for valves are generally available from the manufacturer. The C_v coefficient of a valve is defined as the flow of water at 60°F, in gallons per minute, at a pressure drop of one pound per square inch across the valve [3], regardless of whether the valve ultimately will be flowing liquid or gases/vapors in the plant process. It is expressed:

$$C_v = 29.9\,d^2/(K)^{1/2} \quad (2\text{-}42)$$

$$C_v = Q\,[\rho/(\Delta P_c)\,(62.4)]^{1/2} \quad (2\text{-}43)$$

$$Q = C_v\,[\Delta P_c\,(62.4/\rho)]^{1/2} \quad (2\text{-}44)$$

$$= 7.90\,C_v\,[\Delta P_c/\rho]^{1/2} \quad (2\text{-}44A)$$

$$\Delta P = [Q/C_v]^2\,[\rho/62.4] \quad (2\text{-}45)$$

Nozzles and Orifices [3]

These piping items shown in Figures 2-17 and 2-18 are important pressure drop or head loss items in a system and must be accounted for to obtain the total system pressure loss. For liquids:

$$q = C'A\sqrt{2g(144)(\Delta P)/\rho} = C'A[2gh_L]^{1/2} \qquad (2\text{-}46)$$

where q = cubic ft/sec of fluid at *flowing conditions*
C' = flow coefficient for nozzles and orifices

$$C' = C_d/\sqrt{1-\beta^4}\text{, corrected for velocity of approach} \qquad (2\text{-}47)$$

Note: C' = C for Figures 2-17 and 2-18, corrected for velocity of approach.
C_d = discharge coefficient for nozzles and orifices
h_L = differential static head or pressure loss across flange taps when C or C′ values come from Figures 2-17 and 2-18, ft of fluid. Taps are located one diameter upstream and 0.5 diameter down from the device.
A = cross section area of orifice, nozzle or pipe, sq ft
h = static head loss, ft of fluid flowing
ΔP = differential static loss, lbs/sq in. of fluid flowing, under conditions of h_L above
β = ratio of small to large diameter orifices and nozzles and contractions or enlargements in pipes

For discharging incompressible fluids to atmosphere, take C values from Figures 2-17 or 2-18 if h_L or ΔP is taken as upstream head or gauge pressure.

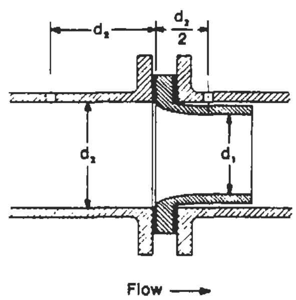
For flow of compressible fluids use the net expansion factor Y (see later discussion) [3]:

$$q = YC'A\,[2g\,(144)\,(\Delta P)/\rho]^{1/2} \qquad (2\text{-}48)$$

where Y = net expansion factor for compressible flow through orifices, nozzles, and pipe.

C' = flow coefficient from Figures 2-17 or 2-18. When discharging to atmosphere, P = inlet gauge pressure. (Also see critical flow discussion.)

For estimating purposes in usual piping systems, the values of pressure drop across an orifice or nozzle will range from 2 to 5 psi. For more exact system pressure drop calculations, the loss across these devices should be calculated using some size assumptions.



$$C = \frac{C_d}{\sqrt{1-\beta^4}}$$

Example: The flow coefficient C for a diameter ratio β of 0.60 at a Reynolds number of 20,000 (2×10^4) equals 1.03.

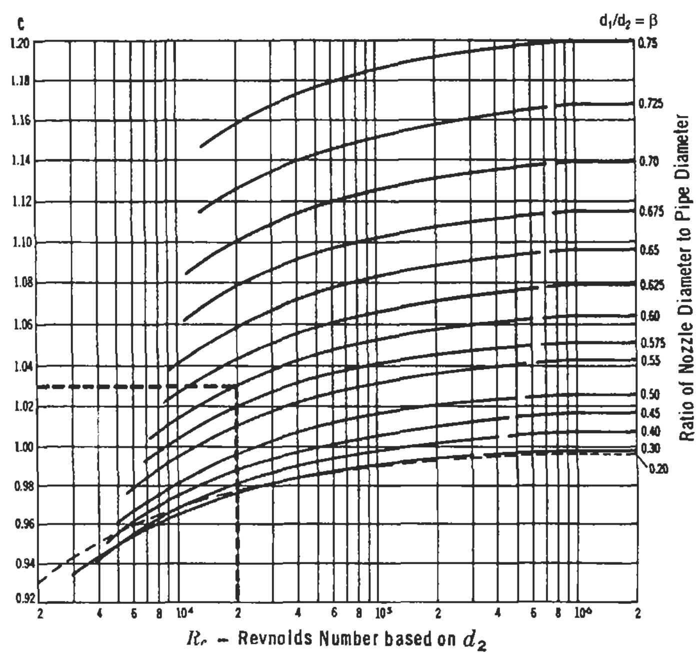
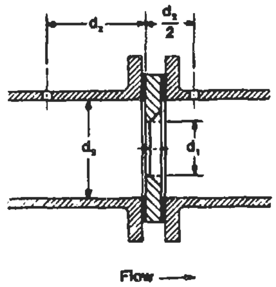


Figure 2-17. Flow coefficient "C" for nozzles. C based on the internal diameter of the upstream pipe. By permission, Crane Co. [3]. Crane reference [9] is to *Fluid Meters*, American Society of Mechanical Engineers, Part 1–6th Ed., 1971. Data used to construct charts. Chart not copied from A.S.M.E. reference.



$$C = \frac{C_d}{\sqrt{1-\beta^4}}$$

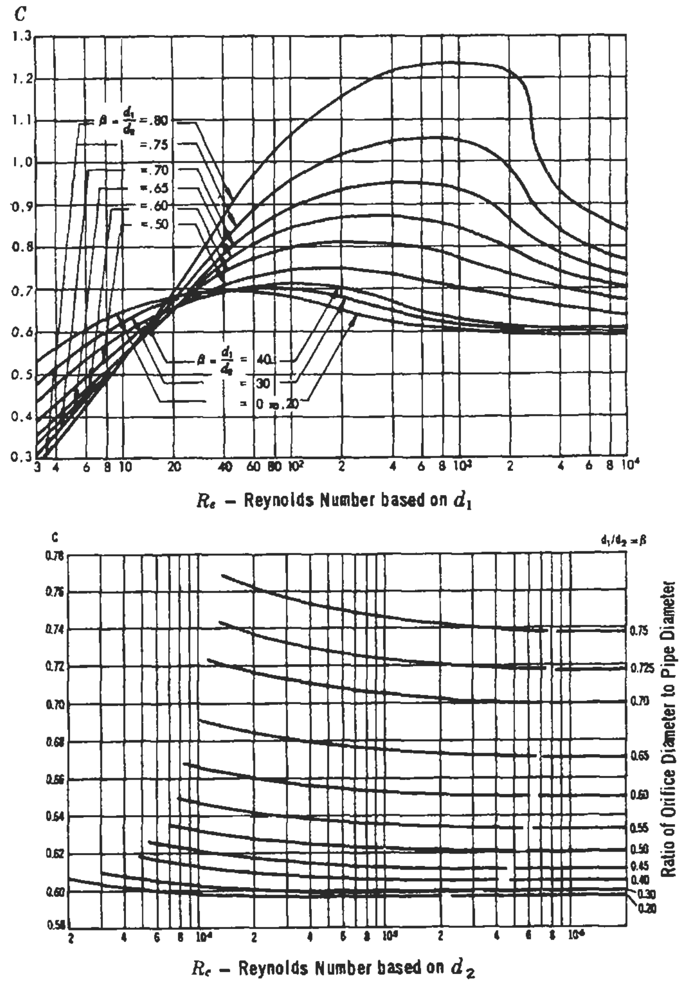


Figure 2-18. Flow coefficient "C" for square edged orifices. By permission, Crane Co. [3], *Technical Paper 410* Engineering Div. (1976) and *Fluid Meters, Their Theory and Application Part 1,* 6th Ed., 1971, American Society of Mechanical Engineers and, Tuve, G. L. and Sprenkle, R. E., "Orifice Discharge Coefficients for Viscous Liquids," *Instruments* Nov. 1933, p. 201.

Example 2-1: Pipe Sizing Using Resistance Coefficients, K

A plant decides to add a nitrogen blanket (at 5 psig) to a storage tank holding up to 25,000 gallons of a hydrocarbon mixture having kerosene-like properties and pumps this material into a process reactor operating at 30 psig. (See Figure 2-19)

The flow rate needs to be 20 gpm. Connections of pipe and valve are flanged, with the 6°–90° elbows added in the line.

Pump suction velocity = 2 ft/sec (Selected low in accordance with good pump suction practice, from Table 2-4 or Table 2-7).

Estimated flow velocity for assumed 2 in. Sch. 40 pipe (See Appendix A-16)

$$= \frac{(20\text{ gpm})(8.33\text{ lb/gal})(0.81\text{ SpGr})}{(62.3\times 0.81)(3.355\text{ in}^2)(60\text{ sec/min})/144}$$

$= 1.91$ ft/sec

$$\text{Velocity head } \frac{v^2}{2g} = \frac{(1.91)^2}{2(32.2)} = 0.05664 \text{ ft of fluid}$$

$$\text{Reynolds number } = \frac{50.6\, Q\rho}{d\mu} \qquad (2\text{-}49)$$

$$= \frac{50.6\,(20)(0.81\times 62.3)}{(2.06)(1.125\text{ cp})}$$

$R_e = 22{,}036$ (turbulent)

$\varepsilon/D = 0.00088$, Figure 2-11

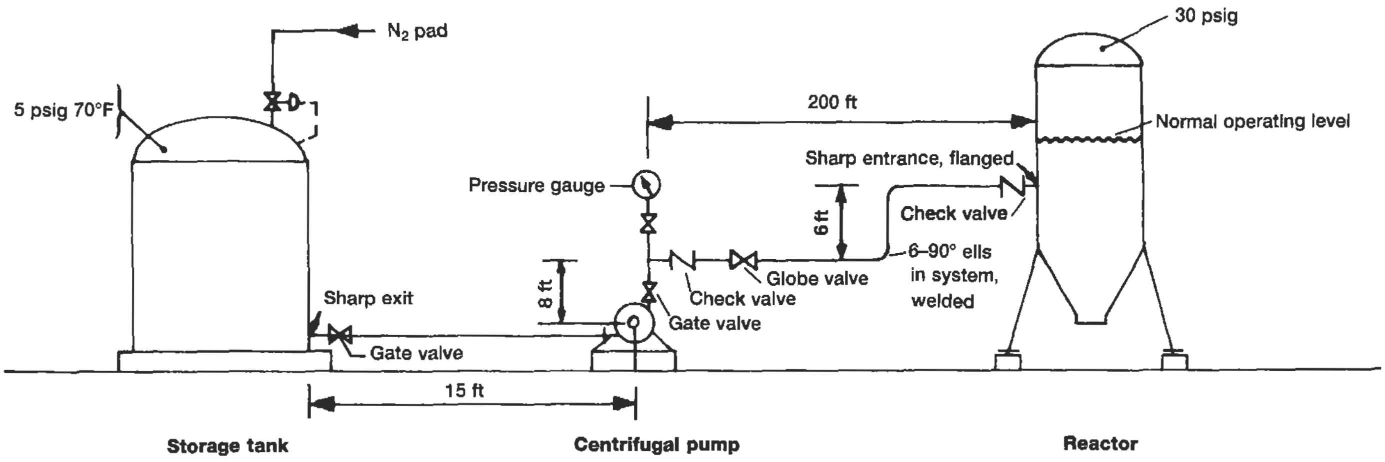


Figure 2-19. Pipe sizing using resistance coefficients, K. Illustration for Example 2-1.

From Figure 2-3 (friction factor), ft = 0.0205

$$h_f = \frac{fLv^2}{D(2g)} \qquad (2\text{-}2)$$

$$= \frac{0.0205\,(15)(1.91)^2}{\left(\dfrac{2.067 \text{ in}}{12}\right) 2(32.2)}$$

$h_f = 0.101$ ft of kerosene fluid, pipe friction, for 15 ft

Loss through pump suction fittings:

a. Square edged inlet (tank to pipe), K = 0.5, Figure 2-12A

b. Gate valve flanged, open, in suction line, from Table 2-2, with $\beta = 1$, $K = 8\ f_T$

$K = 8\ (0.0205) = 0.164$

$h_f = Kv^2/2g = (0.5 + 0.164)\ (1.91)^2/2\ (32.2) = 0.0853$ ft fluid

Total suction pipe side friction *loss:*

$\Sigma h_f = 0.101 + 0.0853 = 0.1863$ ft kerosene

Note: when used for pump system balance, this Σh_f must be used as a negative number (-0.1863) because it is a pressure loss associated with the fluid flowing. For pipe *line sizing,* the pressure head on the tank of 5 psig and any elevation difference between tank outlet nozzle and pump suction centerline do not enter into the calculations.

Pump Discharge Line Sizing (only)

The pump discharge can flow at a higher velocity than the suction line, due in part to NPSH conditions on the suction side of any pump (which are not considered directly in these pipe sizing calculations).

From Table 2-4, select 6 ft/sec as design velocity for estimating pipe size.

For 20 gpm, cross-section area for flow required:

$$A = \frac{20}{7.48 \text{ gal/cu ft } (60 \text{ sec/min}) (6 \text{ ft/sec})}$$

$= 0.00742$ sq ft
$= (0.007427)(144) = 1.009$ sq in.

From Appendix A-16, Standard Schedule 40 pipe
For 1-in. pipe, A = 0.8640 sq in. (too small)
1¼-in. pipe, A = 1.495 sq in. (too large)
Try 1¼-in. pipe, ID = 1.38 in.

(Note: Usually do not select this size. Could go to 1½-in. Velocity would be even slower.)

Actual velocity would be: (1¼-in. pipe)

$v = 20\ (144)/[(60)\ (7.48)\ (1.495)] = 4.29$ ft/sec

This is acceptable. Practical usage range is 3 ft/sec to 9 ft/sec, although 1¼-in. pipe is not the best size for some plants.

Reynolds number, $R_e = 50.6Q\rho/d\mu$ (2–49)
$= 50.6\ (20)(0.81 \times 62.3)/1.38\ (1.125)$
$= 32,894$ (turbulent)
For 1¼″ $\varepsilon/D = 0.0014$, Figure 2-11. (2-49A)

Table 2-4
Suggested Fluid Velocities in Pipe and Tubing: Liquids, Gases, and Vapors at low/moderate pressure to 50 psig and 50° to 100°F

The velocities are suggestive only and are to be used to approximate line size as a starting point for pressure drop calculations. The final line size should be such as to give an economical balance between pressure drop and reasonable velocity

Fluid	Suggested Trial Velocity	Pipe Material
Acetylene (Observe pressure limitations)	4000 fpm	Steel
Air, 0 to 30 psig	4000 fpm	Steel
Ammonia		
Liquid	6 fps	Steel
Gas	6000 fpm	Steel
Benzene	6 fps	Steel
Bromine		
Liquid	4 fps	Glass
Gas	2000 fpm	Glass
Calcium Chloride	4 fps	Steel
Carbon Tetrachloride	6 fps	Steel
Chlorine (Dry)		
Liquid	5 fps	Steel, Sch. 80
Gas	2000—5000 fpm	Steel, Sch. 80
Chloroform		
Liquid	6 fps	Copper & Steel
Gas	2000 fpm	Copper & Steel
Ethylene Gas	6000 fpm	Steel
Ethylene Dibromide	4 fps	Glass
Ethylene Dichloride	6 fps	Steel
Ethylene Glycol	6 fps	Steel
Hydrogen	4000 fpm	Steel
Hydrochloric Acid		
Liquid	5 fps	Rubber Lined
Gas	4000 fpm	R. L., Saran, Haveg
Methyl Chloride		
Liquid	6 fps	Steel
Gas	4000 fpm	Steel
Natural Gas	6000 fpm	Steel
Oils, lubricating	6 fps	Steel
Oxygen	1800 fpm Max.	Steel (300 psig Max.)
(ambient temp.)	4000 fpm	Type 304 SS
(Low temp.)		
Propylene Glycol	5 fps	Steel

Fluid	Suggested Trial Velocity	Pipe Material
Sodium Hydroxide		
0—30 Percent	6 fps	Steel
30—50 Percent	5 fps	and
50—73 Percent	4	Nickel
Sodium Chloride Sol'n.		
No Solids	5 fps	Steel
With Solids	(6 Min.— 15 Max.) 7.5 fps	Monel or nickel
Perchlorethylene	6 fps	Steel
Steam		
0-30 psi Saturated*	4000—6000 fpm	Steel
30-150 psi Saturated or superheated*	6000-10000 fpm	
150 psi up superheated	6500-15000 fpm	
*Short lines	15,000 fpm (max.)	
Sulfuric Acid		
88—93 Percent	4 fps	S. S.—316, Lead
93—100 Percent	4 fps	Cast Iron & Steel, Sch. 80
Sulfur Dioxide	4000 fpm	Steel
Styrene	6 fps	Steel
Trichlorethylene	6 fps	Steel
Vinyl Chloride	6 fps	Steel
Vinylidene Chloride	6 fps	Steel
Water		
Average service	3-8 (avg. 6) fps	Steel
Boiler feed	4-12 fps	Steel
Pump suction lines	1-5 fps	Steel
Maximum economical (usual)	7-10 fps	Steel
Sea and brackish water, lined pipe	5-8 fps } 3 (Min.)	R. L., concrete, asphalt-line, saran-lined, transite
Concrete	5-12 fps	

Note: R. L. = Rubber-lined steel.

From Figure 2-3, read, $f = 0.0219 = f_T$
then, pipe only friction loss:

$$h_f = (f\,L/D)\ (v^2/2g) \tag{2-2}$$

D = pipe, I.D., in ft = 1.38/12 = 0.1150 ft

$$h_f = 0.0219\ \frac{(8 + 6 + 200)}{0.1150}\ \frac{(4.29)^2}{(2)(32.2)}$$

$h_f = 11.64$ ft of kerosene flowing (pipe only)

Loss through discharge fittings, valves, connections, using K factors using Table 2-2:

2 check valves, swing, threaded, $100\ f_T = 100\ (0.0219) = 2.19 = 4.38$ (for 2)

1 globe valve (open), $\beta = 1$: $K = 340\ f_T = 340\ (0.0219) = 7.446$

6 90° elbows, r/d = 1.88/1.38 = 1.36

$$K = 30\ f_T = 30\ (0.0219) = 0.657$$

For 6: $6 \times 0.657 = 3.94$

1 sharp edged entrance (sudden enlargement) = 1.0

1 gate valve, open, $\beta = 1.0$, $K = 8\ f_T$; $K = 8\ (0.0219) = 0.175$

Summation:

$K = [4.38 + 7.45^* + 3.94 + 0.175^* + 1.0] = 16.941$

*Threaded, from Table 2-2.

For fittings:

$$\text{then, } h = Kv^2/2g = \frac{16.941\ (4.29^2)}{2(32.2)} = 4.84 \text{ ft kerosene}$$

Total friction loss for discharge side pump *due to friction:*

$h = 11.64 + 4.84 = 16.48$ ft fluid kerosene

$h_f = \Delta p = 16.48/[(2.31)/(0.81)] = 5.77$ psi

Example 2-2: Laminar Flow Through Piping System

A heavy weight oil, No. 5 fuel oil, is to be pumped through 350 ft of existing 4-in. Schedule 40 pipe at 350 gpm. Oil data:

Temperature = 100°F
Viscosity = 150 cp
Sp Gr = 0.78 = 48.6 lb/cu ft
Pipe I.D. = 4.026 in. = 0.3355 ft

$$\begin{aligned}\text{Reynolds number} &= 50.6\ Q\rho/(d\mu)\\ &= 50.6\ [(350)(48.6)/(4.026)(150)]\\ &= 1425 \qquad (2\text{-}50)\end{aligned}$$

Flow is <2000, therefore, flow of viscous or laminar system consists of friction factor, f_T, for 4-in. pipe = 0.017 (Table 2-2).

1 gate valve = $K = 8\ f_T = 8\ (0.017) = 0.136$, (Table 2-2)

3 90° = $K = 20\ f_T = 20\ (0.017) = 0.345$

1 90° = $R/D = 5$; $5/D = 0.00045$ (Figure 2-11); $K = 0.1$ (Figure 2-13A)

1 pipe entrance to tank projecting inward, $K = 0.78$ (Table 2-2)

For 350 ft pipe, $K = f\ (L/D) = 0.0449\ (350/0.3355) = 51.12$

[For f, see calculations below]

$f = 64/R_e$

$f = 64/1425 = 0.0449 \qquad (2\text{-}17)$

Total K values $= 51.12 + 0.78 + 0.1 + 0.136 + 0.345 = 52.48$

$$\begin{aligned}\text{Velocity } v_s &= 0.408\ Q/d^2 = 0.408\ (350)/(4.026)^2\\ &= 8.8 \text{ ft/sec} \qquad (2\text{-}51)\end{aligned}$$

$$\begin{aligned}\text{Pressure drop} = \Delta P &= 0.00001799\ K\rho\ Q^2/d^4 \qquad (2\text{-}52)\\ &= 0.00001799\ (52.48)\ (48.6)\ (350)^2/(4.026)^4\\ \Delta P &= 21.2 \text{ psi friction pressure loss only (no elevation change)}\end{aligned}$$

Alternate Calculation Basis for Piping System Friction Head Loss: Liquids

Pressure loss in a piping system (not including the tanks, heat exchangers, distillation columns, etc.) is usually expressed in units of *feet of flowing fluid,* or the *equivalent converted to pounds per square inch.* Some published pressure loss data is expressed as per *100 equivalent feet* of the size pipe being used or estimated.

Equivalent Feet Concept for Valves, Fittings, Etc.

With pipe of any specified size as the basis, the total footage of *straight* pipe in a system is just the measured length (totaled).

For fittings, valves, etc., in the same system, these can be expressed as *equivalent straight pipe,* then added to the straight pipe described above, to arrive at a total equivalent straight length of pipe of the specific size in question.

Figure 2-20 presents equivalent lengths of straight pipe (feet) for various pipe system components. For example, a standard threaded 6-inch 90° elbow is equivalent to adding 17 feet of straight pipe to the system. This 17 feet is additive to the lengths of nominal 6-inch straight pipe in the system (dotted line). However, there is an important consideration in the use of this chart, i.e., use only for threaded or screwed pipe/fittings, and only for sizes under 2-inch nominal size. It is not practical in current industry practice to thread a process or utility system much greater in nominal diameter than 2 inches. For special situations, the larger sizes can be used, but from a handling standpoint, sizes greater than 3 inches or 4 inches are not practical.

For pipe sizes greater than 2 inches nominal, industry practice is to weld the pipe and fittings into one continuous system, and then use flanged or special bolted connections for attaching the valves, orifices, and connections to vessels or other equipment. For special lethal, high pressure, and steam power plant high temperature/high pressure utility systems, even the valves and connections to vessels are welded into the system (See ASME and ANSI Codes). For these situations of about 1½-inch to 2-inch nominal pipe size and larger, use Figure 2-21 to determine the equivalent pipe lengths for these fittings, valves, etc. For example, a 45° welding elbow, or an open 6-inch gate valve (see line on chart) have an equivalent length of 6-inch pipe of four feet (straight), which is an addition to the actual straight pipe in the system. In

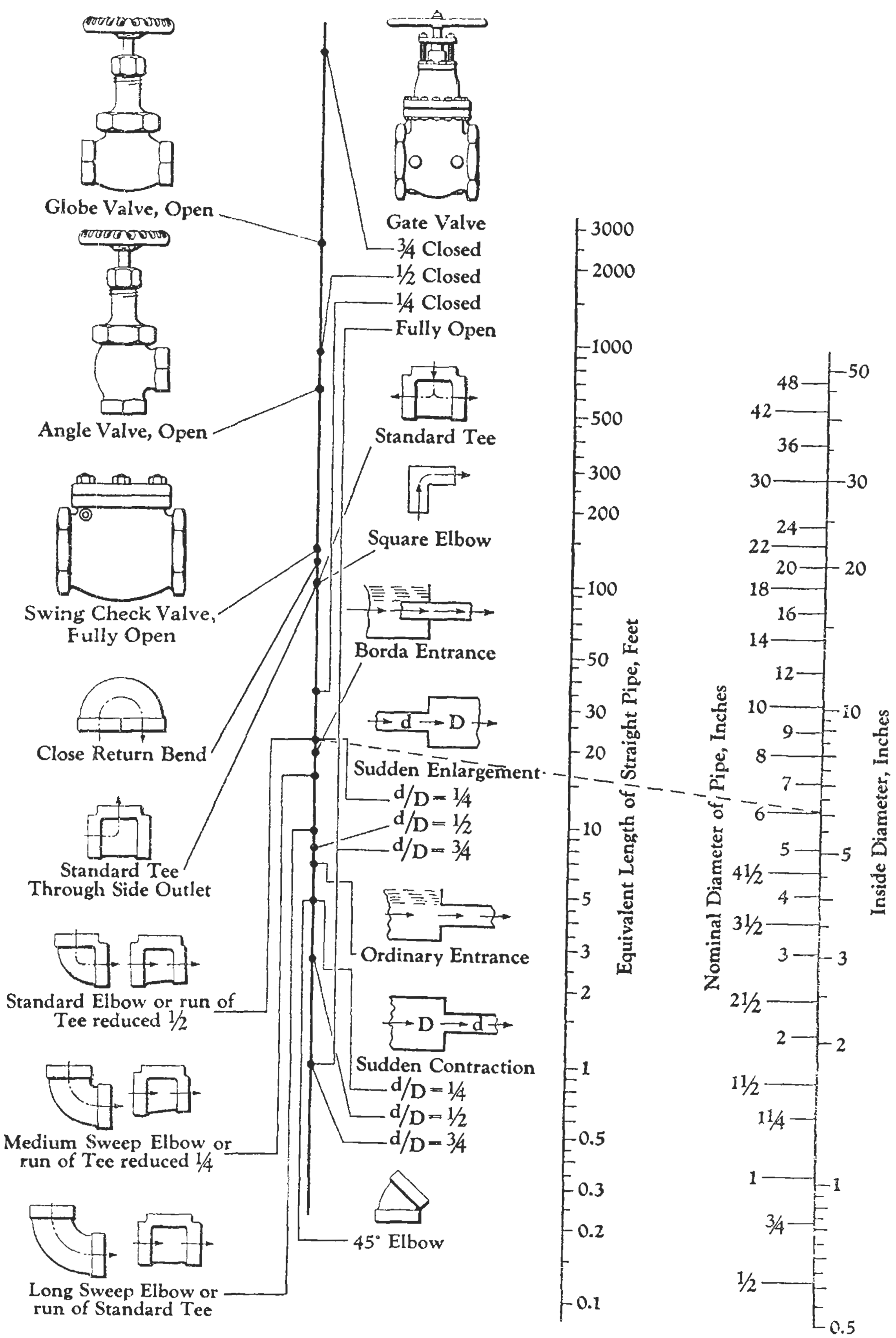


Figure 2-20. Equivalent length resistance of valves and fittings to flow of fluids. Note: apply to 2 in. and smaller threaded pipe for process applications (this author). By permission, Crane Co., *Technical Paper #409,* Engineering Div., 1942, also see [3].

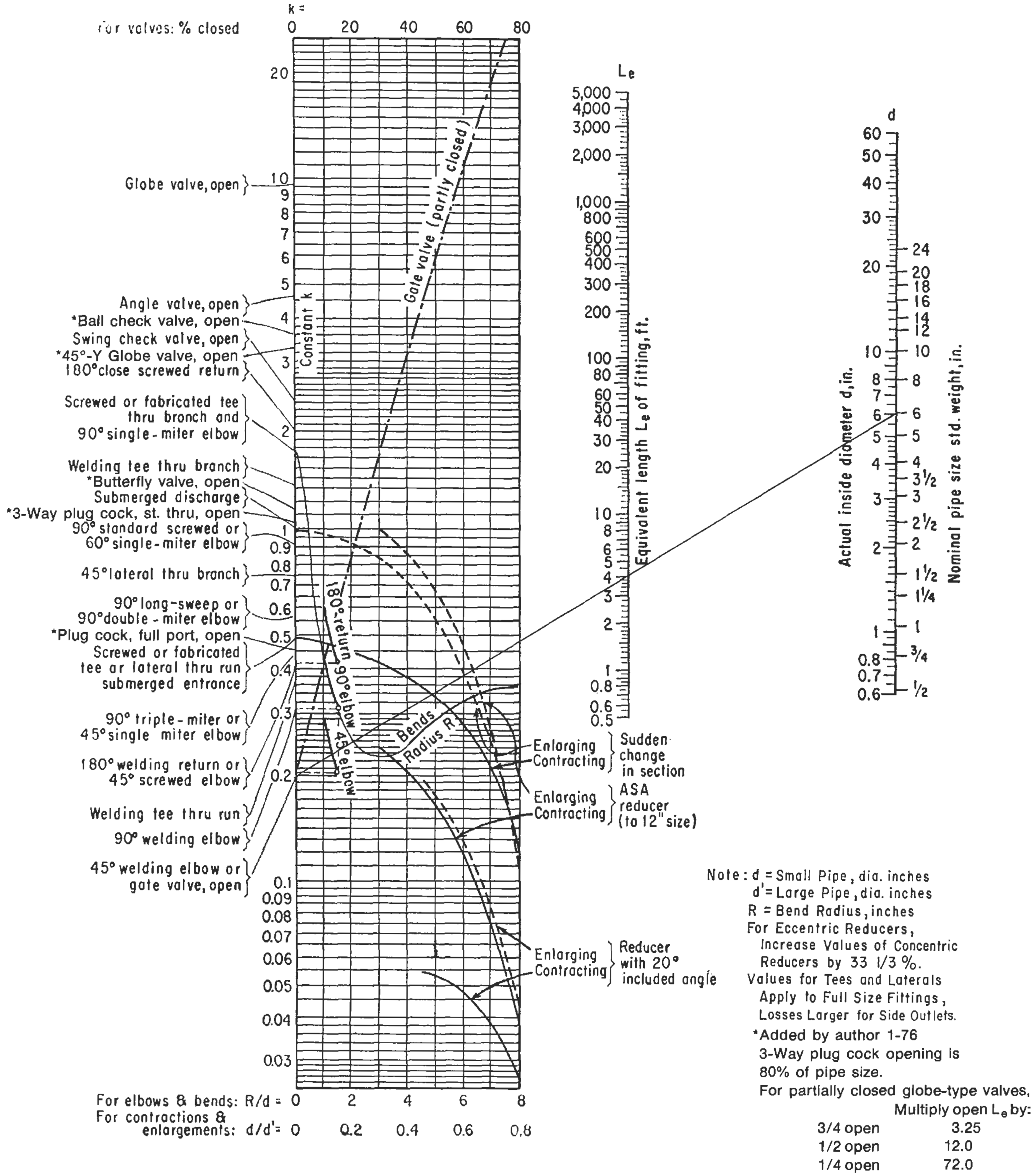


Figure 2-21. Equivalent length of fittings for pipe systems. Note: preferred use for 1½ in. and larger pipe butt-welded or socket-welded connections (this author). By permission, Tube Turns Div., Chemitron Corp. Bull. TT 725, 1952, reference now to Tube Turns Technologies, Inc.

summation, these equivalent lengths for all the components determine the total pipe length to use in the pressure loss (pressure drop) equations to be described later.

Friction Pressure Drop for Non-Viscous Liquids

The only significance in differentiating between water and liquids of different densities and viscosities is the convenience in having a separate simplified table for water.

1. Using known flow rate in gallons per minute and a suggested velocity from Tables 2-4 to 2-8 or Figure 2-22, estimate first pipe size. Mean velocity of any liquid flowing in a pipe [3] is given by Figure 2-22 and Equation 2-51.

$$v = 0.408\ Q/d^2 = 0.0509\ W/(d^2)\ (\rho),\ \text{ft/sec} \tag{2-51}$$

$$d = (0.408\ Q/v)^{1/2} = (0.0509\ W/v\rho)^{1/2},\ \text{in.} \tag{2-53}$$

$$v = q/A = w_s/A\rho = 183.3\ (q/d^2),\ \text{ft/sec} \tag{2-54}$$

2. Estimate or otherwise determine the linear feet of straight pipe in the system, L.

3. Estimate (or use actual tabulation) number of fittings, valves, etc. in system. Convert these to equivalent straight pipe using Figures 2-20 or 2-21, L_{eg}, or head by Figures 2-12 through 2-16 and Table 2-2.

   Note preferred pipe size type for charts.

4. Determine expansion or contraction losses, if any, including tank or vessel entrance or exit losses from Figures 2-12A, 2-15, or 2-16. Convert units to psi, head loss in feet times 0.4331 = psi (for water), or adjust for Sp Gr of other liquids.

5. Estimate pressure drop through orifices, control valves, and other items in the system, but not equipment. For control valves, estimate ΔP from paragraph to follow.

6. Determine pressure drop per unit of length.
   a. Calculate Reynolds number [3]

$$R_e = 50.6\ Q\rho/(d\mu) = 6.31\ W/(d\mu) \tag{2-16}$$

   b. From Reynolds Number-Friction Factor Chart, Figure 2-3, read friction factor, f, at ε/d value taken from Figure 2-11.

   c. Calculate pressure drop per 100 feet of (straight and/or equivalent) pipe [3] as psi/100 ft. Establish piping system friction pressure drop (loss), liquids (Figure 2-23):

$$\text{For turbulent flow: } \Delta P/100\ \text{ft} = 0.0216\ f\ \rho Q^2/d^5 \tag{2-55}$$

$$= 0.000336\ fW^2/(d^5)\ (\rho) \tag{2-55A}$$

$$\text{For laminar flow: } \Delta P/100\ \text{ft} = 0.0668\ (\mu)\ v/d^2 \tag{2-56}$$

$$= 0.0273\ (\mu)\ Q/d^4 \tag{2-56A}$$

Table 2-5
Typical Design Vapor Velocities* (ft/sec)

Fluid	Line Sizes ≤6″	8″-12″	≥ 14″
Saturated Vapor			
0 to 50 psig	30-115	50-125	60-145
Gas or Superheated Vapor			
0 to 10 psig	50-140	90-190	110-250
11 to 100 psig	40-115	75-165	95-225
101 to 900 psig	30- 85	60-150	85-165

* Values listed are guides, and final line sizes and flow velocities must be determined by appropriate calculations to suit circumstances. Vacuum lines are not included in the table, but usually tolerate higher velocities. High vacuum conditions require careful pressure drop evaluation.

Table 2-6
Usual Allowable Velocities for Duct and Piping Systems*

Service/Application	Velocity, ft./min.
Forced draft ducts	2,500 - 3,500
Induced-draft flues and breeching	2,000 - 3,000
Chimneys and stacks	2,000
Water lines (max.)	600
High pressure steam lines	10,000
Low pressure steam lines	12,000 - 15,000
Vacuum steam lines	25,000
Compressed air lines	2,000
Refrigerant vapor lines	
High pressure	1,000 - 3,000
Low pressure	2,000 - 5,000
Refrigerant liquid	200
Brine lines	400
Ventilating ducts	1,200 - 3,000
Register grilles	500

*By permission, *Chemical Engineer's Handbook,* 3rd Ed., McGraw-Hill Book Co., New York, N.Y., p. 1642.

Table 2-7
Typical Design* Velocities for Process System Applications

Service	Velocity, ft./sec.
Average liquid process	4 - 6.5
Pump suction (except boiling)	1 - 5
Pump suction, boiling	0.5 - 3
Boiler feed water (disch., pressure)	4 - 8
Drain lines	1.5 - 4
Liquid to reboiler (no pump)	2 - 7
Vapor-liquid mixture out reboiler	15 - 30
Vapor to condenser	15 - 80
Gravity separator flows	0.5 - 1.5

* To be used as guide, pressure drop and system environment govern final selection of pipe size.
For heavy and viscous fluids, velocities should be reduced to about ½ values shown.
Fluids not to contain suspended solid particles.

Table 2-8
Suggested Steam Pipe Velocities in Pipe Connecting to Steam Turbines

Service—Steam	Typical range, ft./sec.
Inlet to turbine	100 - 150
Exhaust, non-condensing	175 - 200
Exhaust, condensing	500 - 400

7. Total pressure drop for system:

$$\Delta P, \text{psi} = (L + \Sigma L_{eq})\ (\Delta P/100 \text{ ft from 6 c above}) + 4 \text{ above} + 5 \text{ above} \tag{2-57}$$

Note: L_{eq} is from 3 above.

If this pressure drop is too large or too small, recheck the steps using larger or smaller pipes as may be indicated. The tables in Cameron [57], Table 2-22, or Figure 2-24 are very convenient to use, although they give much more conservative results (about twice unit head loss) than the method outlined above. When using Figure 2-24, the results agree acceptably well with tests on 15 to 20 year old steel pipe. Also see Table 2-22.

For brine, Table 2-9 gives multipliers to use with the water unit losses of Figure 2-24. Figure 2-25 gives direct-reading values with Dowtherm® liquid.

It is important to note that comparison of results from these charts does not yield exact checks on any particular fitting. Calculations should never be represented as being more accurate than the basic information. Therefore, rounded values to no more than one decimal place are limits for such head loss calculations.

The head losses calculated using K coefficients by these figures can be added directly to the total friction head loss for the straight pipe portions of a system. When equivalent lengths are determined, they must be added to the straight pipe before determining the total head loss, as shown in the example calculations for a water system.

Friction loss in rubber-lined pipe is usually considered equivalent to that in new steel pipe of one-half to one nominal size smaller, with little or no change due to aging, unless known conditions can be interpolated. For a given inside diameter, the friction loss is the same (or slightly less) than clean steel pipe.

In the turbulent flow range, friction loss in glass pipe is 70 to 85 percent of clean steel.

For 2-inch (nominal) and larger vinyl, saran, or hard rubber pipe, the friction loss does not exceed clean steel. With saran and rubber-lined pipe the loss is about equal to clean steel at the 2.5-inch size, increasing to 2 to 4 times the loss at the 1-inch size.

Estimation of Pressure Loss across Control Valves: Liquids, Vapors, and Gases

Despite the need for good control in many process systems, most engineers do not allow the proper pressure drop for the control valves into their calculations. Many literature sources ignore the problem, and many plant operators and engineers wonder why the actual plant has control problems.

Rather than assuming a pressure drop across the control as 25%, 33%, or 40% of the other friction losses in the system, a logical approach [9] is summarized here. The control valve pressure drop has nothing to do with the valve size, but is determined by the pressure balance (See Equation 2-59 [9]).

Control valve pressure drop:

$$P_S = P_e + F_D + P_c \tag{2-58}$$

$$\text{Available } \Delta P_c = (P_S - P_e) - F_D, \text{ psi} \tag{2-59}$$

where P_S = total pressure at beginning (higher pressure) of system, psig, including any static heads to reach final pressure, P_e.
P_e = pressure at lower end of system, psig
F_D = friction loss at design basis, total, for the system, psi, including equipment and piping, at Q_D rate
Q_M = maximum anticipated flow rate for system, gpm, or ACFM
F_M = friction pressure drop at maximum flow rate Q_M, psi
Q_D = design flow rate, gpm, or ACFM

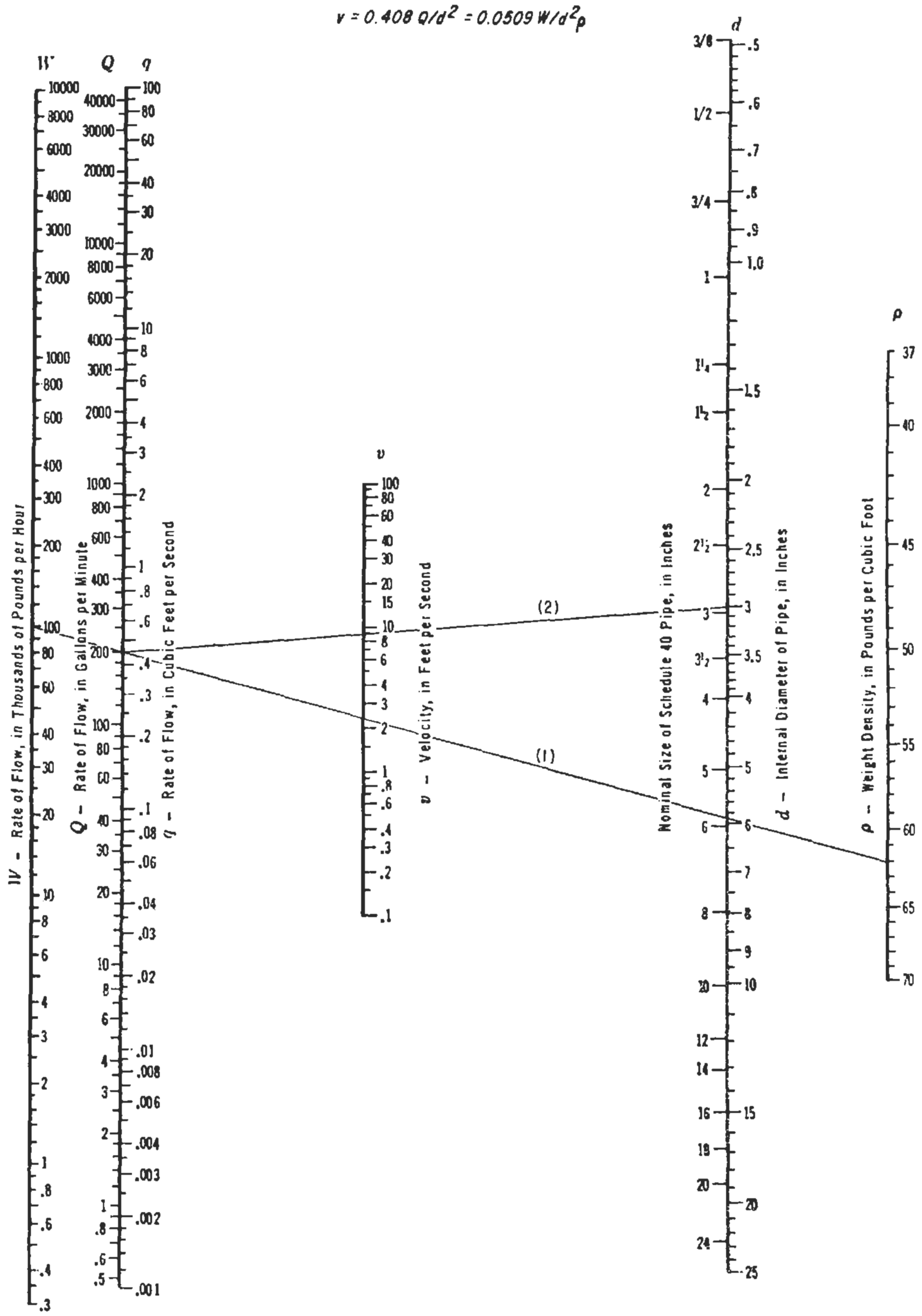


Figure 2-22. Velocity of liquid in pipe. By permission, Crane Co., *Technical Paper #410,* Engineering Div., 1957. Also see 1976 edition.

ΔP_c = pressure drop across control valve

F_M = friction pressure drop at maximum flow rate, psi

Friction loss or drop at higher flow rates than design:

$$\text{Increased pressure drop} = [F_D (Q_M / Q_D)^2 - F_D],$$

$$\text{or} \left[\left(\frac{Q_M}{Q_D}\right)^2 - 1\right](F) \qquad (2\text{-}60)$$

Allowing 10% factor of safety, expected maximum increase in friction pressure drop allowance:

$$= 1.1\ [(Q_M/Q_D)^2 - 1](F_D) \qquad (2\text{-}61)$$

At maximum flow rate, Q_M, the *friction drop* will become:

$$F_M = F_D\ (Q_M/Q_D)^2 \qquad (2\text{-}62)$$

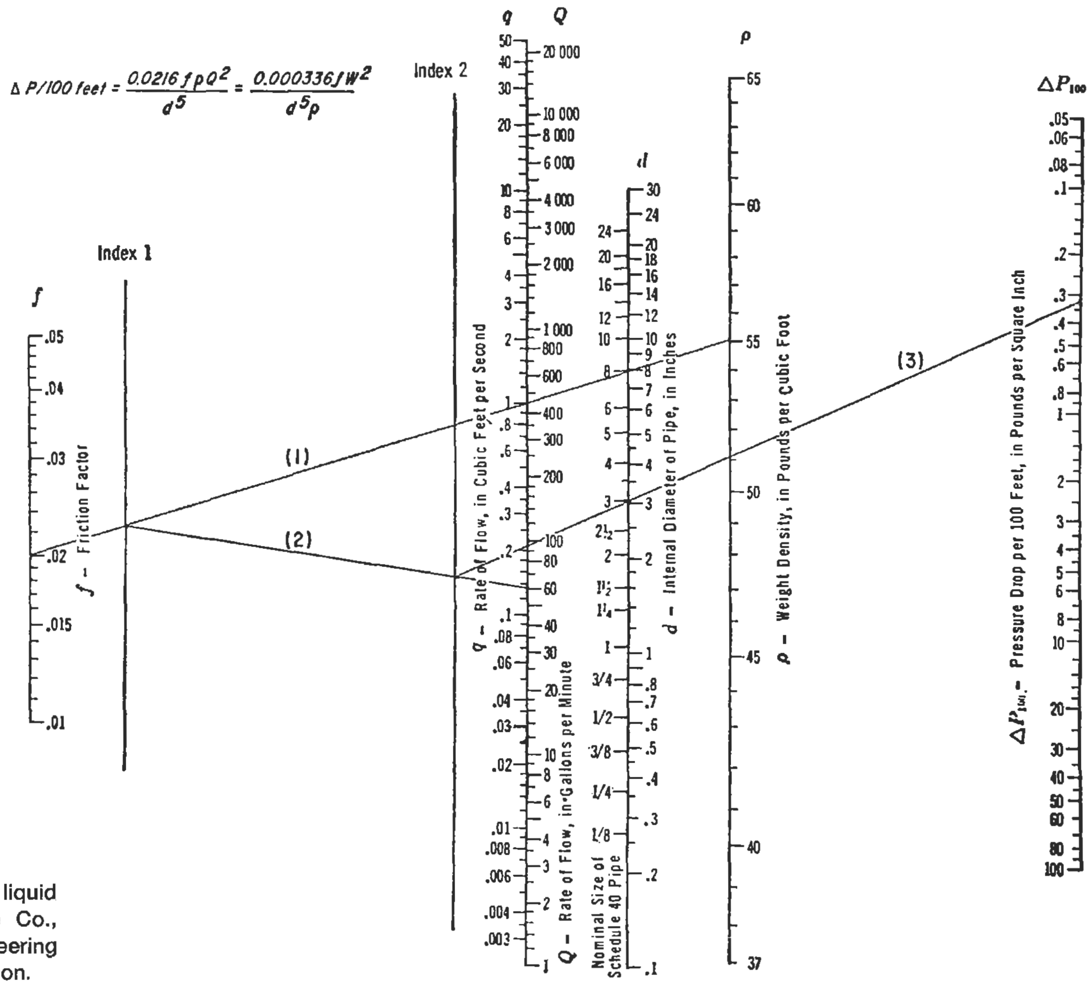


Figure 2-23. Pressure drop in liquid lines. By permission Crane Co., *Technical Paper #410*, Engineering Div., 1957. Also see 1976 edition.

The friction loss or pressure drop, F_D, is determined at the design flow rate, Q_D, for the piping, valves, and friction producing equipment (such as tubular heat exchangers, tubular furnaces/heaters), orifice or other meters, and control valves. Because the system friction pressure loss changes with flow rate through the system, recognition must be given to the changes in flow rate (increase or decrease) as it affects the pressure loss through the control valves. For any design, the beginning and end points of the system should be relatively constant for good process operations.

For good control by the valve, the pressure drop across (or through) the valve must always be greater than the friction losses of the system by perhaps 10% to 20% (see [9]).

Example 2-3: Establishing Control Valve Estimated Pressure Drop, using Connell's Method [9].

Refer to Figure 2-26 for an example to determine the pressure loss (drop) through the control valve.

where P_c = system end pressure = 22 + 15 = 37 psig (not friction)

Piping system pipe friction @ Q_D flow rate = 6 psi
Heater, friction = 65 psi
Separator, friction = 1 psi
Preheaters, 10 + 12 (friction) = 22 psi
Orifice, allow, friction = 2 psi
Total friction, excluding control valve, F_D = 96 psi

Assume pressure loss through control valve = 35 psi

Then $\Delta P_c = (P_S - P_c) - F_D$, psi (2-59)

$35 = (P_S - 37) - 96$

P_S = 168 psi, at pump discharge, using assumed control valve pressure drop of 35 psi

Note that P_c = 22 psig + 15 psi static Hd. = 37 psig

Assume that allowances must be made for a 10% increase in process flow rate, above design, Q_D. Pressure drop varies as the square of the flow rate.

Correction Factors									
Value of C	60	70	80	90	100	110	120	130	140
Multiplier to Correct Chart	2.57	1.93	1.50	1.22	1.0	.84	.71	.62	.54

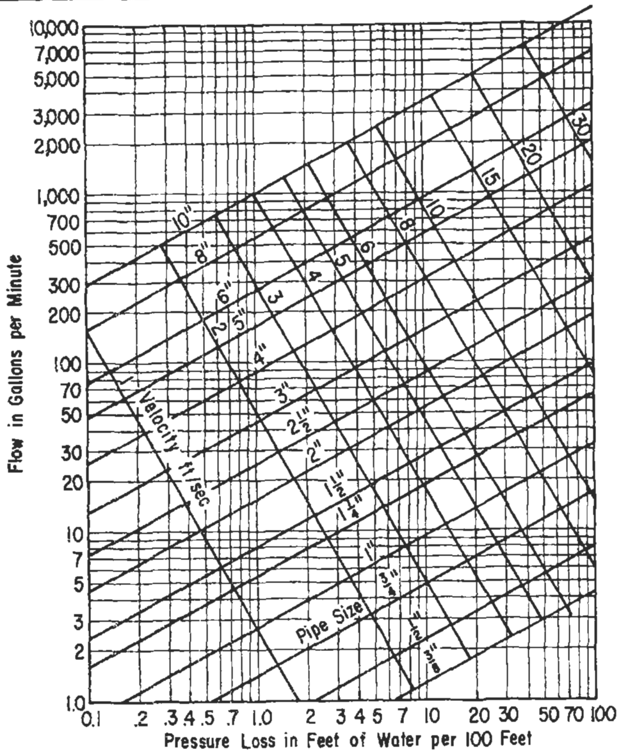


Figure 2-24. Friction loss for flow of water in steel pipes. Note C = pipe roughness factor. See Tables 2-9 and 2-22. Courtesy of Carrier Corp.

New flow rate = 110% (Q_D)

Friction pressure drop will increase to 121% of F_D;

$1.21\ (96) = 116$ psi $= F_M$

Friction increase = 116 − 96 = 20 psi added for relatively constant P_S and P_e

Available $\Delta P_c = (168 - 37) - 116$
ΔP_c = 15 psig through the control valve, which means that the valve has to open more and reduce its sensitivity of response, from its design ΔP_c of 35 psig

For design purposes, the assumed 35 psi for the control valve could be used; however, decreasing the pipe friction of 6 psi to perhaps ½ or ⅓ by increasing the line size will help the control of the valve. It would be better to have the *available* valve pressure drop equal to or greater than the assumed.

Factors to consider in evaluating the control valve pressure drop are:

A. Allowance for increase in friction drop

Establish the ratio of the *maximum anticipated* flow rate for system, Q_M, to the design basis rate, Q_D or Q_M/Q_D. When Q_M is not known, nor can it be anticipated, use: Q_M/Q_D of 1.1 for flow control and 1.25 for level pressure and temperature control valves to anticipate the flow rate transients as the control loop recovers from a disturbance [9].

At the maximum flow rate Q_M, the friction drop will become:

$$F_M = F_D\ (Q_M/Q)^2 \tag{2-62}$$

The *increase* in pressure drop will be:

$$\Delta F_M = F_D\ (Q_M/Q_D)^2 - F_D \tag{2-63}$$

or, $$\Delta F_M = F_D\ [(Q_M/Q_D)^2 - 1] \tag{2-64}$$

F_D may not necessarily be very accurate at the design stage where final drawing dimensions for the system are being estimated. For this reason a 10% increase allowance is suggested to ΔF_M.

B. Allowance for possible falloff in: overall system pressure drop, $P_S - P_e$

If there is an increase in system flow rate

Overall system pressure drop = PF (all) = 0.05 PS (2-65)

C. Allowance for control valve (base pressure drop at full-open position [9]

This varies with the type and design of valve and can be obtained from the manufacturer. It is identified here as base pressure drop B for the valve itself. Using average line velocities and assuming that the control valve will be one pipe size smaller than the pipe line it is connected to, using average B values over a range of sizes, the B values for estimating purposes are [9]:

Control Valve Type	B, psi
Single Plug	11
Double Plug	7
Cage (unbalanced)	4
Cage (balanced)	4
Butterfly	0.2
V-Ball	1

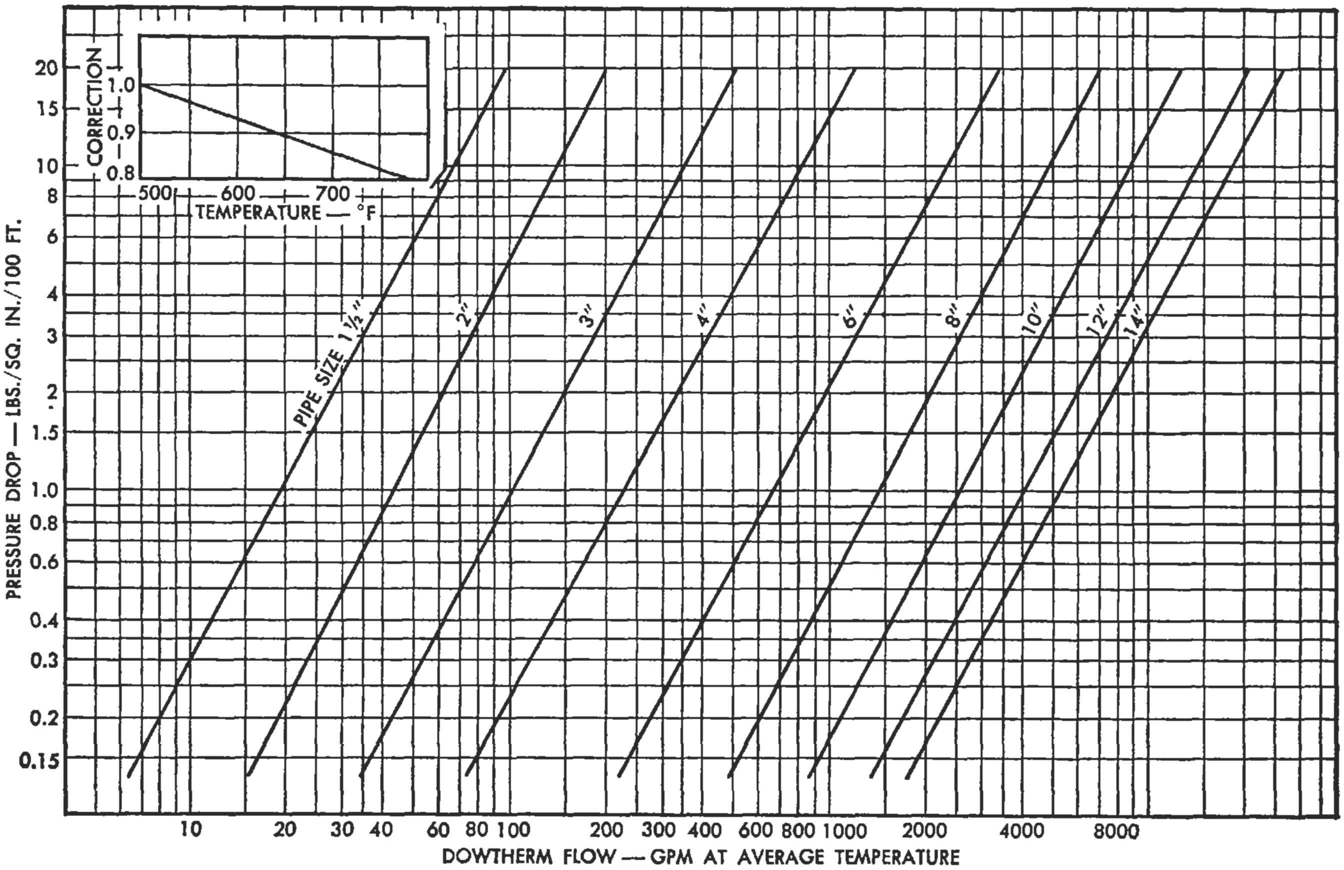


Figure 2-25. Pressure drop for Dowtherm® liquid in schedule 40 pipe. By permission, Struthers Wells Corp. Bull. 4-45, 1956.

Then, incorporating the requirements of A, B, and C above, the estimated overall control valve drop is:

$$\text{Required } \Delta P_c = 0.05\ P_S + 1.1\ [(Q_M/Q_D)^2 - 1]\ F_D + B, \text{ psi} \quad (2\text{-}66)$$

B = base pressure drop for control valve with valve in wide-open position, psi. (see list above).

Example 2-4: Using Figure 2-26, Determine Control Valve Pressure Drop and System Start Pressure (See Example 2-3)

To determine P_S, the value of ΔP through the control valve must be known.

$P_e = 37$ psi
$F_D = 96$ psi (all except control valve), psi

Assume, $Q_M = 120\%$ of Q

$Q_M/Q_D = 1.2$

Use cage type valve, B = 4

From Equation 2-59,

$$\text{Available } \Delta P_c = (P_S - P_c) - F_D = (P_S - 37) - 96 = P_S - 133$$

From Equation 2-66,

$$\begin{aligned}\text{Required } \Delta P &= 0.05P_S + 1.1\ [Q_M/Q_D)^2 - 1]\ F_D + B \\ &= 0.05\ (P_G) + 1.1\ [(1.2)^2 - 1]\ (96) + 4 \\ &= 0.05\ P_S + 46.5 + 4 = 0.05P_S + 50\end{aligned}$$

Substituting:

$$\begin{aligned}P_S - 133 &= 0.05P_S + 50 \\ 0.95P_S &= 183 \\ P_S &= 192 \text{ psi, start pressure at the pump}\end{aligned}$$

Table 2-9
Brine Pipe Friction Multiples
For Use With Water Friction Data, Figure 2-24

BRINE	Specific Gravity	BRINE TEMPERATURE, °F 0	10	20	30	40	50	60	70
Sodium Chloride	1.10	. . .	1.23	1.20	1.18	1.16	1.14	1.13	1.12
	1.15	1.43	1.33	1.29	1.26	1.24	1.22	1.21	1.20
	1.20	1.53	1.44	1.38	1.35	1.32	1.30	1.28	1.27
Calcium Chloride	1.05	. . .	. . .	1.15	1.12	1.10	1.08	1.07	1.06
	1.10	. . .	1.28	1.23	1.20	1.18	1.16	1.14	1.12
	1.15	1.41	1.35	1.31	1.28	1.25	1.22	1.21	1.20
	1.20	1.49	1.43	1.39	1.36	1.33	1.30	1.28	1.27
	1.25	1.56	1.53	1.49	1.45	1.42	1.40	1.38	1.37
	1.30	1.65	1.61	1.58	1.55	1.52	1.50	1.49	1.48

NOTE: To find brine friction loss, multiply loss from Fig. 2-10 by multiplier from above Table.
By permission, Crocker, S., *Piping Handbook*, McGraw-Hill Book Co.

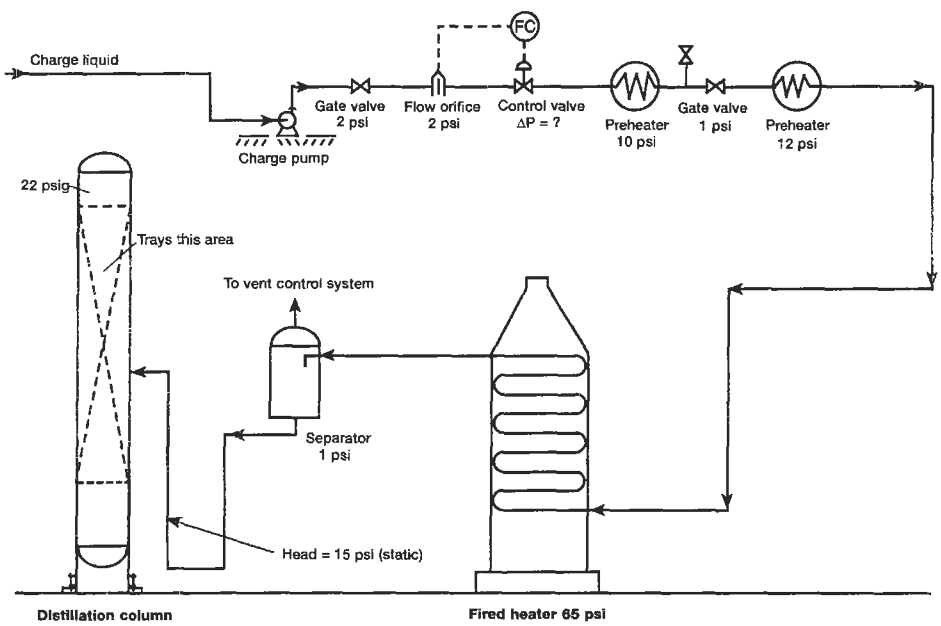


Figure 2-26. Establishing control valve estimated pressure drop.

Control valve pressure drop:

$\Delta P_c = 0.05\ P_S + 50 = 0.05\ (192) + 50 = 59.6$ psi

Use this as estimated control valve pressure drop for the system design.

The Direct Design of a Control Valve

This does not require the system balance as outlined in A through C above; however, without first preparing a pressure balance, the designer cannot be confident of properly estimating the valve pressure drop. From Shinsky [10],

$$GPM = a'(C_v')\sqrt{\Delta P_c / SpGr} \qquad (2\text{-}66)$$

where
a' = fractional opening of control valve, generally assume 60% = 0.60
C'_v = standard valve coefficient from manufacturer's catalog
ΔP_c = pressure drop across valve, psi
SpGr = specific gravity of fluid, relative to water at same temperature

or, from [11], for gases or vapors:

Flow, SCFH *†

$$q'_h = \frac{42.2\, C'_v \sqrt{(P_1 - P_2)(P_1 + P_2)}}{\sqrt{S_G}} \qquad (2\text{-}67)$$

Flow, SCFH (temperature corrected)†

$$q'_h = \frac{963\, C'_v \sqrt{(P_1 - P_2)(P_1 + P_2)}}{\sqrt{S_G T}} \qquad (2\text{-}67A)$$

where S_g = specific gravity relative to air = 1.0
P_1 = inlet pressure (14.7 + psig)
P_2 = outlet pressure (14.7 + psig)
q_h = flow rate, standard cu ft./hr (SCFH)
T = flowing temperature, °R abs, (°F + 460)
C'_v = valve coefficient of flow, full open (from manufacturer's tables)

*The effect of flowing temperatures on gas flow can be disregarded for temperatures between 30°F and 150°F. Corrections should apply to other temperatures above or below [11].

†When outlet pressure P_2 is less than ½ inlet pressure P_1 the square root term becomes 0.87 P_1 [11].

Friction Loss For Water Flow

Table 2-10 is quite convenient for reading friction loss in standard schedule 40 pipe. It is based upon Darcy's rational analysis (equivalent to Fanning).

Suggested procedure:

1. Using known flow rate in gallons/minute, and a suggested velocity from Tables 2-4, 2-5, 2-6, 2-7 and 2-8 select an approximate line size.
2. Estimate (or use actual drawing or measured tabulation) total linear feet of pipe, L.
3. Estimate (or use actual tabulation) number of elbows, tees, crosses, globe valves, gate valves and other fittings in system. Convert these to equivalent straight pipe using Figure 2-20 or 2-21, L_{eq}, or to head loss using Figures 2-12 through 2-16. Note preferred pipe size/type for charts.
4. Determine expansion and contraction losses (if any) from Figures 2-12, 2-15, and 2-16. Convert units: head loss in feet times 0.4331 = psi. (This term can usually be neglected for most liquids at reasonable velocities < 10′/sec.)
5. Estimate pressure drop through orifices, control valves and other items that may be in system, per prior discussion.
6. Total pressure drop.

$$\Delta P = (L + \Sigma L_{eq})\ (\Delta P/100' \text{ from Table 2-10}) + \text{Item (4)} + \text{Item (5)} \qquad (2\text{-}57)$$

If this pressure drop is too large (or too small), recheck the steps using larger or smaller pipe as may be indicated. Table 2-22 [53] or Figure 2-24 are convenient to use, although they give much more conservative results (about twice unit head loss) than the method and figures just referenced. When using Figure 2-24 the results agree acceptably well with tests on 15–20-year-old steel pipe.

Example 2-5: Water Flow in Pipe System

The system of Figure 2-27 consists of 125 feet of unknown size schedule 40 steel pipe on the discharge side of a centrifugal pump. The flow rate is 500 gallons per minute at 75°F. Although the tank is located above the pump, note that this elevation difference does *not* enter into the pipe size-friction drop calculations. However it will become a part of selection of the pump for the service (see Chapter 3). For quick estimate follow these steps:

1. From Table 2-4, select 6 fps as a reasonable and usually economical water rate.

   From Table 2-10, a 6-inch pipe has a velocity of 5.55 fps at 500 gpm and a head loss of 0.720 psi/100 ft. The 5-inch pipe has a velocity of 8.02 fps and might be considered; however 5-inch pipe is not commonly stocked in many plants, and the velocity is above usual economical pumping velocities. Use the 6-inch pipe (rough estimate).

2. Linear feet of straight pipe, L = 125 feet.
3. From Figure 2-20, the equivalent length of fitting is: 6 inch-90° ell ≅ 14 feet straight pipe (using medium sweep elbow to represent a welding ell). Note that this is given as 6.5 feet from Figure 2-21. This illustrates the area of difference in attempting to obtain close or exact values.

   3 90° ells = 3 (14) = L_{eq} = 42 ft (conservative)
   1 tee = 1 (12) = L_{eq} = 12 (Run of std. tee)
   1 6″ open Gate Valve = (1) (3.5) = L_{eq} = 3.5
   1 sudden enlargement in tank @ d/d′ = 0; = 10′, Figure 2-21
   Total L_{eq} = 67.5 feet

4. Neglect expansion loss at entrance to tank, since it will be so small.
5. No orifices or control valves in system.
6. From Table 2-10, at 500 gpm, loss = 0.72 psi/100 eq ft.

Table 2-10
Flow of Water Through Schedule 40 Steel Pipe*
Note: This Table closely matches results of Hydraulic Institute

Discharge		Pressure Drop per 100 feet and Velocity in Schedule 40 Pipe for Water at 60 F.															
		Veloc-ity	Press. Drop	Veloc-ity	Press. Drop	Veloc-ity	Press. Drop	Veloc-ity	Press. Drop	Veloc-ity	Press. Drop	Veloc-ity	Press. Drop	Veloc-ity	Press. Drop	Veloc-ity	Press. Drop
Gallons per Minute	Cubic Ft. per Second	Feet per Second	Lbs. per Sq. In.	Feet per Second	Lbs. per Sq. In.	Feet per Second	Lbs. per Sq. In.	Feet per Second	Lbs. per Sq. In.	Feet per Second	Lbs. per Sq. In.	Feet per Second	Lbs. per Sq. In.	Feet per Second	Lbs. per Sq. In.	Feet per Second	Lbs. per Sq. In.
		1/8"		1/4"		3/8"		1/2"									
.2	0.000446	1.13	1.86	0.616	0.359												
.3	0.000668	1.69	4.22	0.924	0.903	0.504	0.159	0.317	0.061	3/4"							
.4	0.000891	2.26	6.98	1.23	1.61	0.672	0.345	0.422	0.086								
.5	0.00111	2.82	10.5	1.54	2.39	0.840	0.539	0.528	0.167	0.301	0.033						
.6	0.00134	3.39	14.7	1.85	3.29	1.01	0.751	0.633	0.240	0.361	0.041						
.8	0.00178	4.52	25.0	2.46	5.44	1.34	1.25	0.844	0.408	0.481	0.102	1"		1 1/4"			
1	0.00223	5.65	37.2	3.08	8.28	1.68	1.85	1.06	0.600	0.602	0.155	0.371	0.048			1 1/2"	
2	0.00446	11.29	134.4	6.16	30.1	3.36	6.58	2.11	2.10	1.20	0.526	0.743	0.164	0.429	0.044		
3	0.00668			9.25	64.1	5.04	13.9	3.17	4.33	1.81	1.09	1.114	0.336	0.644	0.090	0.473	0.043
4	0.00891			12.33	111.2	6.72	23.9	4.22	7.42	2.41	1.83	1.49	0.565	0.858	0.150	0.630	0.071
5	0.01114	2"				8.40	36.7	5.28	11.2	3.01	2.75	1.86	0.835	1.073	0.223	0.788	0.104
6	0.01337	0.574	0.044	2 1/2"		10.08	51.9	6.33	15.8	3.61	3.84	2.23	1.17	1.29	0.309	0.946	0.145
8	0.01782	0.765	0.073			13.44	91.1	8.45	27.7	4.81	6.60	2.97	1.99	1.72	0.518	1.26	0.241
10	0.02228	0.956	0.108	0.670	0.046			10.56	42.4	6.02	9.99	3.71	2.99	2.15	0.774	1.58	0.361
15	0.03342	1.43	0.224	1.01	0.094	3"				9.03	21.6	5.57	6.36	3.22	1.63	2.37	0.755
20	0.04456	1.91	0.375	1.34	0.158	0.868	0.056	3 1/2"		12.03	37.8	7.43	10.9	4.29	2.78	3.16	1.28
25	0.05570	2.39	0.561	1.68	0.234	1.09	0.083	0.812	0.041	4"		9.28	16.7	5.37	4.22	3.94	1.93
30	0.06684	2.87	0.786	2.01	0.327	1.30	0.114	0.974	0.056			11.14	23.8	6.44	5.92	4.73	2.72
35	0.07798	3.35	1.05	2.35	0.436	1.52	0.151	1.14	0.704	0.882	0.041	12.99	32.2	7.51	7.90	5.52	3.64
40	0.08912	3.83	1.35	2.68	0.556	1.74	0.192	1.30	0.095	1.01	0.052	14.85	41.5	8.59	10.24	6.30	4.65
45	0.1003	4.30	1.67	3.02	0.668	1.95	0.239	1.46	0.117	1.13	0.064			9.67	12.80	7.09	5.85
50	0.1114	4.78	2.03	3.35	0.839	2.17	0.288	1.62	0.142	1.26	0.076	5"		10.74	15.66	7.88	7.15
60	0.1337	5.74	2.87	4.02	1.18	2.60	0.406	1.95	0.204	1.51	0.107			12.89	22.2	9.47	10.21
70	0.1560	6.70	3.84	4.69	1.59	3.04	0.540	2.27	0.261	1.76	0.143	1.12	0.047			11.05	13.71
80	0.1782	7.65	4.97	5.36	2.03	3.47	0.687	2.60	0.334	2.02	0.180	1.28	0.060			12.62	17.59
90	0.2005	8.60	6.20	6.03	2.53	3.91	0.861	2.92	0.416	2.27	0.224	1.44	0.074	6"		14.20	22.0
100	0.2228	9.56	7.59	6.70	3.09	4.34	1.05	3.25	0.509	2.52	0.272	1.60	0.090	1.11	0.036	15.78	26.9
125	0.2785	11.97	11.76	8.38	4.71	5.43	1.61	4.06	0.769	3.15	0.415	2.01	0.135	1.39	0.055	19.72	41.4
150	0.3342	14.36	16.70	10.05	6.69	6.51	2.24	4.87	1.08	3.78	0.580	2.41	0.190	1.67	0.077		
175	0.3899	16.75	22.3	11.73	8.97	7.60	3.00	5.68	1.44	4.41	0.774	2.81	0.253	1.94	0.102		
200	0.4456	19.14	28.8	13.42	11.68	8.68	3.87	6.49	1.85	5.04	0.985	3.21	0.323	2.22	0.130	8"	
225	0.5013	...	...	15.09	14.63	9.77	4.83	7.30	2.32	5.67	1.23	3.61	0.401	2.50	0.162	1.44	0.043
250	0.557	...	...	...	...	10.85	5.93	8.12	2.84	6.30	1.46	4.01	0.495	2.78	0.195	1.60	0.051
275	0.6127	...	...	...	...	11.94	7.14	8.93	3.40	6.93	1.79	4.41	0.583	3.05	0.234	1.76	0.061
300	0.6684	...	...	...	...	13.00	8.36	9.74	4.02	7.56	2.11	4.81	0.683	3.33	0.275	1.92	0.072
325	0.7241	...	...	...	...	14.12	9.89	10.53	4.09	8.19	2.47	5.21	0.797	3.61	0.320	2.08	0.083
350	0.7798			...	...	...	...	11.36	5.41	8.82	2.84	5.62	0.919	3.89	0.367	2.24	0.095
375	0.8355			...	...	...	...	12.17	6.18	9.45	3.25	6.02	1.05	4.16	0.416	2.40	0.108
400	0.8912			...	...	...	...	12.98	7.03	10.08	3.68	6.42	1.19	4.44	0.471	2.56	0.121
425	0.9469			...	...	...	...	13.80	7.89	10.71	4.12	6.82	1.33	4.72	0.529	2.73	0.136
450	1.003	10"		...	...	...	...	14.61	8.80	11.34	4.60	7.22	1.48	5.00	0.590	2.89	0.151
475	1.059	1.93	0.054			...	...	...	...	11.97	5.12	7.62	1.64	5.27	0.653	3.04	0.166
500	1.114	2.03	0.059			...	...	...	...	12.60	5.65	8.02	1.81	5.55	0.720	3.21	0.182
550	1.225	2.24	0.071			...	...	...	...	13.85	6.79	8.82	2.17	6.11	0.861	3.53	0.219
600	1.337	2.44	0.083			...	...	...	...	15.12	8.04	9.63	2.55	6.66	1.02	3.85	0.258
650	1.448	2.64	0.097	12"		...	...	...	...	...	...	10.43	2.98	7.22	1.18	4.17	0.301
700	1.560	2.85	0.112	2.01	0.047			...	...	...	...	11.23	3.43	7.78	1.35	4.49	0.343
750	1.671	3.05	0.127	2.15	0.054	14"		...	...	...	...	12.03	3.92	8.33	1.55	4.81	0.392
800	1.782	3.25	0.143	2.29	0.061			...	...	...	...	12.83	4.43	8.88	1.75	5.13	0.443
850	1.894	3.46	0.160	2.44	0.068	2.02	0.042	...	...	...	...	13.64	5.00	9.44	1.96	5.45	0.497
900	2.005	3.66	0.179	2.58	0.075	2.13	0.047	...	...	...	...	14.44	5.58	9.99	2.18	5.77	0.554
950	2.117	3.86	0.198	2.72	0.083	2.25	0.052			...	...	15.24	6.21	10.55	2.42	6.09	0.613
1 000	2.228	4.07	0.218	2.87	0.091	2.37	0.057	16"		...	...	16.04	6.84	11.10	2.68	6.41	0.675
1 100	2.451	4.48	0.260	3.15	0.110	2.61	0.068			...	...	17.65	8.23	12.22	3.22	7.05	0.807
1 200	2.674	4.88	0.306	3.44	0.128	2.85	0.080	2.18	0.042	...	...	...	...	13.33	3.81	7.70	0.948
1 300	2.896	5.29	0.355	3.73	0.150	3.08	0.093	2.36	0.048	...	...	...	...	14.43	4.45	8.33	1.11
1 400	3.119	5.70	0.409	4.01	0.171	3.32	0.107	2.54	0.055					15.55	5.13	8.98	1.28
1 500	3.342	6.10	0.466	4.30	0.195	3.56	0.122	2.72	0.063	18"				16.66	5.85	9.62	1.46
1 600	3.565	6.51	0.527	4.59	0.219	3.79	0.138	2.90	0.071					17.77	6.61	10.26	1.65
1 800	4.010	7.32	0.663	5.16	0.276	4.27	0.172	3.27	0.088	2.58	0.050			19.99	8.37	11.54	2.08
2 000	4.456	8.14	0.808	5.73	0.339	4.74	0.209	3.63	0.107	2.87	0.060	20"		22.21	10.3	12.82	2.55
2 500	5.570	10.17	1.24	7.17	0.515	5.93	0.321	4.54	0.163	3.59	0.091					16.03	3.94
3 000	6.684	12.20	1.76	8.60	0.731	7.11	0.451	5.45	0.232	4.30	0.129	3.46	0.075	24"		19.24	5.59
3 500	7.798	14.24	2.38	10.03	0.982	8.30	0.607	6.35	0.312	5.02	0.173	4.04	0.101			22.44	7.56
4 000	8.912	16.27	3.08	11.47	1.27	9.48	0.787	7.26	0.401	5.74	0.222	4.62	0.129	3.19	0.052	25.65	9.80
4 500	10.03	18.31	3.87	12.90	1.60	10.67	0.990	8.17	0.503	6.46	0.280	5.20	0.162	3.59	0.065	28.87	12.2
5 000	11.14	20.35	4.71	14.33	1.95	11.85	1.21	9.08	0.617	7.17	0.340	5.77	0.199	3.99	0.079	...	...
6 000	13.37	24.41	6.74	17.20	2.77	14.23	1.71	10.89	0.877	8.61	0.483	6.93	0.280	4.79	0.111	...	...
7 000	15.60	28.49	9.11	20.07	3.74	16.60	2.31	12.71	1.18	10.04	0.652	8.08	0.376	5.59	0.150	...	...
8 000	17.82	...	...	22.93	4.84	18.96	2.99	14.52	1.51	11.47	0.839	9.23	0.488	6.38	0.192	...	...
9 000	20.05	...	...	25.79	6.09	21.34	3.76	16.34	1.90	12.91	1.05	10.39	0.608	7.18	0.242	...	...
10 000	22.28	...	...	28.66	7.46	23.71	4.61	18.15	2.34	14.34	1.28	11.54	0.739	7.98	0.294	...	...
12 000	26.74	...	...	34.40	10.7	28.45	6.59	21.79	3.33	17.21	1.83	13.85	1.06	9.58	0.416	...	...
14 000	31.19	...	...	...	...	33.19	8.89	25.42	4.49	20.08	2.45	16.16	1.43	11.17	0.562	...	...
16 000	35.65	...	...	...	...	...	...	29.05	5.83	22.95	3.18	18.47	1.85	12.77	0.723	...	...
18 000	40.10	...	...	...	...	...	...	32.68	7.31	25.82	4.03	20.77	2.32	14.36	0.907	...	...
20 000	44.56	...	...	...	...	...	...	36.31	9.03	28.69	4.93	23.08	2.86	15.96	1.12	...	...

For pipe lengths other than 100 feet, the pressure drop is proportional to the length. Thus, for 50 feet of pipe, the pressure drop is approximately one-half the value given in the table . . . for 300 feet, three times the given value, etc.

Velocity is a function of the cross sectional flow area; thus, it is constant for a given flow rate and is independent of pipe length.

For pipe other than Schedule 40: $v = v_{40}\ (d_{40}/d)^2$ and $\Delta P = \Delta P_{40}\ (d_{40}/d)^5$ where subscript 40 refers to the conditions for schedule 40 pipe.

*By permission, "Technical Paper No. 410," Crane Co., Engineering Div., Chicago (1957)

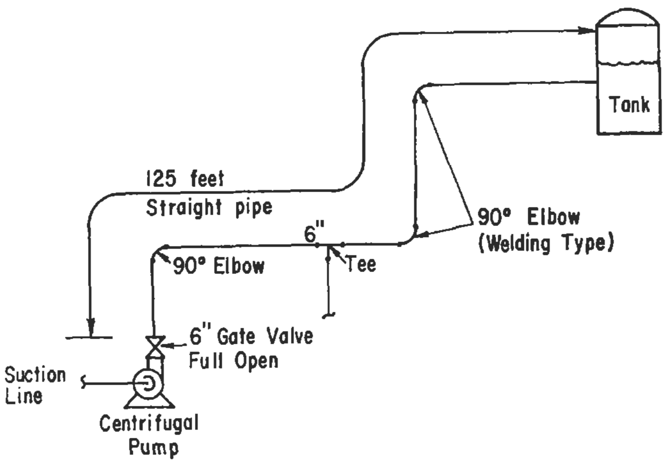


Figure 2-27. Example 2-5, pipe system for pipe sizing calculations.

Total pressure drop from face of discharge flange on pump to nozzle connection on tank:

$$\Delta P = (125 + 67.5)\ [(0.720)/100] + 0$$
$$\Delta P = 1.386 \text{ psi}$$
$$\Delta P = 1.386 \text{ psi } (2.31 \text{ feet/psi}) = 3.20 \text{ feet water}$$

Note that a somewhat more accurate result may be obtained by following the detailed loss coefficients given in Figures 2-12 through 2-16. However, most preliminary engineering design calculations for *this* type of water system do not warrant the extra detail.

Flow of Water from Open-End Horizontal Pipe

The equation of Brooke [36] is useful in estimating water or similar fluids flow from the end of open pipes:

$$GPM = 1.04\ a\ (l) \tag{2-68}$$

where GPM = flow rate, gallons per minute
a = internal cross-sectional area for flow in pipe, sq in.
l = horizontal distance from pipe opening to point where flow stream has fallen one ft, in.

Water Hammer [19]

Water hammer is an important problem that occurs in some liquid control systems. It is defined as hydraulic shock that occurs when a non-viscous liquid flowing in a pipe experiences a sudden change in velocity, such as the fast closing of a valve. The kinetic energy of the moving mass of liquid upon sudden stoppage or abrupt change of direction is transformed into pressure energy, thereby causing an abrupt pressure rise in the system, often resulting in severe mechanical damage [53].

The pressure that can develop from the shock wave can be destructive to the containing system hardware, particularly in long pipe. Examples of conditions that can develop water hammer are:

1. start, stop, or an abrupt change in a pump's speed
2. power failure
3. rapid closing of a valve (usually a control valve, which can slam shut in one or two seconds)

The magnitude of this shock wave can be expressed [19, 20]:

$$h_{wh} = a_w (v_w)/g = \frac{4660\ (v_w)}{g\sqrt{1 + K_{hs} B_r}} \tag{2-69}$$

For water:

$$a_w = 4660/(1 + K_{hs}\ B_r)^{1/2}, \text{ ft/sec} \tag{2-70}$$

where h_{wh} = maximum pressure developed by hydraulic shock, ft of water
v_w = reduction in velocity, ft/sec (actual flowing velocity, ft/sec)
g = gravitational constant, 32.2 ft/sec
K_{hs} = ratio of elastic modulus of water to that of the pipe material (See list below)
B_r = ratio of pipe diameter (I.D.) to wall thickness
a_w = velocity of propagation of elastic vibration in the discharge pipe, ft/sec

Some typical K_{hs} values for water/metal are [19]:

Metal	K_{hs}
Copper	0.017
Steel	0.010
Brass	0.017
Wrought iron	0.012
Malleable cast iron	0.012
Aluminum	0.030

The time interval t_s, required for the pressure wave to travel back and forth in the pipe is:

$$t_s = 2\ L/a_w, \text{ sec} \tag{2-71}$$

L = length of pipe, ft (not equivalent ft)

When the actual abrupt closing of a device to stop the flow has a time shorter than t_s, then the maximum pressure, h_{wh}, will be exerted on the closed device and line. Note that the value of, h_{wh}, is added to the existing static pressure in the system.

Example 2-6: Water Hammer Pressure Development

An 8-inch process pipe for transferring 2000 GPM of methanol of Sp Gr = 0.75 from the manufacturing plant site to a user plant location is 2,000 feet long, and the liquid is flowing at 10.8 ft/sec.

Maximum pressure developed (preliminary solution) when an emergency control valve suddenly closes:

$$h_{wh} = a_w (v_w)/g \qquad (2\text{-}69)$$

Since methanol has many properties similar to water:

$$a_w = 4660/(1 + K_{hs} B_r)^{1/2}$$
$$= 4660/[1 + 0.01 (24.7^*)]^{1/2} = 4175 \text{ ft/sec}$$

*For 8-inch std pipe, $B_r = 7.981/0.322 = 24.78$

Time interval for pressure wave travel:

$$t_s = 2L/a_w = 2 (2000)/4175 = 0.95 \text{ sec} \qquad (2\text{-}71)$$

If the shutoff time for the valve (or a pump) is less than 0.95 seconds, the water hammer pressure will be:

$$h_{wh} = 4175 (10.8)/32.2 = 1400 \text{ ft of methanol}$$
$$= (1400)/[(2.31)/0.75)] = 454 \text{ psi hydraulic shock}$$

Then total pressure on the pipe system

= 454 + (existing pressure from process/or pump)

This pressure level would most likely rupture an 8-inch Sch. 40 pipe. For a more exact solution, refer to specialty articles on the subject.

Example 2-7: Pipe Flow System With Liquid of Specific Gravity Other Than Water

This is illustrated by line size sheet, Figure 2-28.

Figure 2-29 represents a liquid reactor system discharging crude product similar to glycol through a flow control valve and orifice into a storage tank. The reactor is at 350 psig and 280°F with the liquid of 0.93 specific gravity and 0.91 centipoise viscosity. There is essentially no flashing of liquid across the control valve.

Flow rate: 11,000 lbs/hr

GPM actual = 11,000/(60) (8.33) (0.93) = 23.7

Design rate = 23.7 (1.05) = 25 gpm

1. From Table 2-4, selected velocity = 6 fps.

Estimated pipe diameter, $d = (0.408\ Q/v)^{1/2}$
$= [(0.408)\ 25/6]^{1/2} = 1.3$ inch

Try 1½-inch (i.d. = 1.61), since 1¼-inch (i.d. = 1.38) is not stocked in every plant. If it is an acceptable plant pipe size, then it should be considered and checked, as it would probably be as good pressure drop-wise as the 1½-inch. The support of 1¼-inch pipe may require shorter support spans than the 1½-inch. Most plants prefer a minimum of 1½-inch valves on pressure vessels, tanks, etc. The valves at the vessels should be 1½ inch even though the pipe might be 1¼ inch The control valve system of gate and globe valves could very well be 1¼ inch. For this example, use 1½-inch pipe, Schedule 40:

2. Linear length of straight pipe, L = 254 ft.
3. Equivalent lengths of fittings, valves, etc.

Estimated Fittings	Type	Eq. Feet (from Figure 2-20)
10	1½"-90° Elbows	4′ (10) = 40
8	1½"-Tees	3′ (8) = 24
4	1½"-Gate Valves	1′ (4) = 4
		68 ft. Use 75 ft.

4. No expansion or contraction losses (except control valve).
5. Pressure drop allowance assumed for orifice plate = 5 psig.

Control valve loss will be by difference, trying to maintain minimum 60% of pipe friction loss as minimum drop through valve, but usually not less than 10 psi.

6. Reynolds number, $R_e = 50.6\ Q\rho/d\mu$ (2-49)
$= 50.6\ (25)\ [0.93\ (62.3)]/(1.61)\ (0.91)$
$= 50{,}025$ (turbulent)

7. From Figure 2-11, $\varepsilon/d = 0.0012$ for 1½-inch steel pipe.

From Figure 2-3, at $R_e = 50{,}025$, read f = 0.021

8. Pressure drop per 100 feet of pipe:

$$\Delta P/100' = 0.0216\ f\rho\ Q^2/d^5 \qquad (2\text{-}72)$$
$$= 0.0216\ (0.021)\ (62.3)\ (0.93)\ (25)^2/(1.61)^5$$
$$= 1.52 \text{ psi/100 ft equivalent}$$

SHEET NO. ______

LUDWIG CONSULTING ENGINEERS

By ABC

LINE SIZE SHEET

Job No. ______

Date ______

Charge No. ______

Line No. LP - 51

Flow Sheet Drawing No. ______

Line Description Reactor Discharge

Fluid in line Crude Product

Temperature 280 °F

GPM (Calc.) 23.7

GPM (des.) 25

Pressure 350 psig

CFM (Calc.) ______

CFM (des.) ______

Sp. Gr. 0.93

Lbs./hr.(Calc.) ______

Lbs./hr.(des.) ______

Sp. Vol. ______ cu.ft./lb.

Recommended Velocity 6.0 fps

Viscosity 0.91 cp

Straight pipe, fittings, valves expansion, contraction, etc.				
Item	No.	Unit Eq. Ft.	Total Eq. Ft.	
Pipe	254	1	254	
90°Elbow	10	4	40	*
Tee	8	3	24	*
Gate Va.	4	1	4	*
		Total	329	*

* Rounded total to 75 feet
** By difference

Inlet = 350 psig; Outlet = 0 psig
Friction Loss = 10 (includes orifice loss)
Balance for Valve = 340 psi

Item	Pressure Drop in psi
Pipe & Equivalent	5)
Orifice	5) =10
Motor Valve (control)	340
Miscellaneous	
Total	350

Estimated line size $1\frac{1}{2}$" (verified)

Actual Velocity 3.9 fps

Unit Loss per 100 ft. 1.52 psi (see below)

Total head loss in feet of liquid 869

Total pressure drop in psi 350

Selected pipe size $1\frac{1}{2}$"

Material & Weight Schedule 40 steel

Calculations: $Re = 50.6\ Q\rho/d\mu = 50.6(25)(0.93 \times 62.3)/(1.61)(0.91) = 50{,}025$

$\epsilon/d = 0.0012$; $f = 0.021$ (Figures 2-3 and 2-11)

$\Delta P/100' = 0.0216\ f\rho Q^2 / d^5 = 0.0216(0.021)(62.3)(0.93)(25)^2/(1.65)^5$

$= 1.52$ psi/100 ft.

Total Pipe System Friction $\Delta P = ((329)(1.52/100)) + 5^* = 10$ psi for friction; *Orifice

Total Loss, Feet Liquid = 350(2.31ft./psi)(1/0.93) = 869 feet of Liquid

Checked by: ______ Date: ______

Figure 2-28. Line sizing sheet for example problem, Example 2-7.

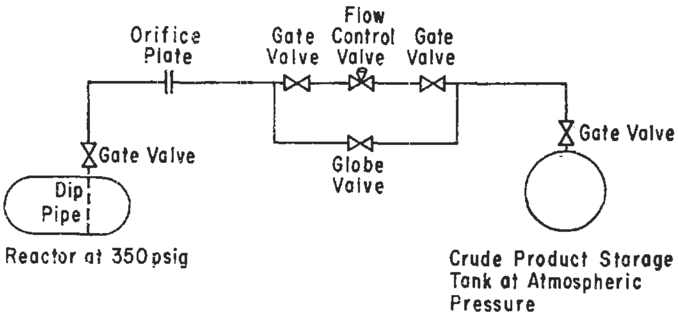


Figure 2-29. Liquid flow system, Example 2-7.

9. Total Pressure Drop

The control valve must be sized to take the residual pressure drop, as long as it is an acceptable minimum. Pressure drop accounted for:

Total psi drop = (245 + 75) (1.52/100) + 5 = 10 psi

Drop required across control valve:

Reactor	= 350 psig
Storage	= 0 psig
Differential	= 350 psi
ΔP	= 10 psi (sys. friction)
Control Valve ΔP	= 340 psi

Note that this control valve loss exceeds 60 percent of this system loss, since the valve must take the difference. For other systems where this is not the situation, the system loss must be so adjusted as to assign a value (see earlier section on control valves) of approximately 10 to 20 psi or 25 to 60 percent of the system other than friction losses through the valve. For very low pressure systems, this minimum value of control valve drop may be lowered at the sacrifice of sensitive control.

Friction Pressure Drop For Compressible Fluid Flow

Vapors and Gases

The flow of compressible fluids such as gas, vapor, steam, etc., is considered in general the same as for liquids or non-compressible fluids. Specific semi-empirical formulas have been developed which fit particular systems and have been shown to be acceptable within engineering accuracy.

Because of the importance of the relationship between pressure and volume for gases and vapors as they flow in a piping or process system, there may be (1) adiabatic flow where for practical purposes there is no exchange of heat into or from the pipe. This is expressed by:

$$P' V_a^k = \text{constant (adiabatic)} \tag{2-73}$$

or, (2) isothermal flow, which is flow at constant temperature (often close to practical experience) and:

$$P' V_a = \text{constant (isothermal)} \tag{2-74}$$

Often for a large variety of process gases, some relationship in between expresses the pressure-volume relationship by:

$$P' V_a^n = \text{constant (polytropic)} \tag{2-75}$$

For gases/vapors flowing in a pipe system from point 1 with pressure P_1 and point 2 with pressure P_2, the $P_1 - P_2$ is the pressure drop, ΔP, between the points [3].

Velocity of Compressible Fluids in Pipe

$$v_m = \frac{3.06\, W\overline{V}}{d^2} = \frac{3.06W}{d^2\rho} \tag{2-76}$$

where v_m = mean velocity in pipe, at conditions stated for $\overline{V}$, ft/min.

W = flow rate, lbs/hr

$\overline{V}$ = fluid specific volume, cu ft/lb, at T and P

d = inside pipe diameter, in.

ρ = fluid density, lbs/cu ft, at T and P

P' = pressure, pounds per sq foot absolute

k = ratio of specific heats, c_p/c_v

Note that determining the velocity at the inlet conditions to a pipe may create significant error when results are concerned with the outlet conditions, particularly if the pressure drop is high. Even the average of inlet and outlet conditions is not sufficiently accurate for some systems; therefore conditions influenced by pressure drop can produce more accurate results when calculations are prepared for successive sections of the pipe system (long or high pressure).

Friction Drop for Flow of Vapors, Gases, and Steam Figure 2-30

A. The Darcy rational relation for compressible flow [3] is:

$$\Delta P / 100 \text{ ft} = \frac{0.000336 f W^2 \overline{V}}{d^5} \qquad (2\text{-}77)$$

$$\text{or, } \Delta P / 100 \text{ ft} = \frac{0.000001959 f (q'_h)^2 S_g{}^2}{d^5 \rho} \qquad (2\text{-}78)$$

The general procedures outlined previously for handling fluids involving the friction factor, f, and the R_e chart are used with the above relations. This is applicable to compressible flow systems under the following conditions [3].

where S_g = specific gravity of gas relative to air = the ratio of molecular weight of the gas to that of air.

q'_h = rate of flow, cu ft/hr at standard conditions (14.7 psia and 60°F), SCFH.

1. When calculated ΔP total < 10 percent inlet pressure, use ρ or $\overline{V}$ based on inlet or outlet conditions.
2. When calculated ΔP total > 10 percent inlet pressure, but < 40 percent, use *average* ρ or $\overline{V}$ based on inlet and outlet conditions.
3. When calculated ΔP total, P_1 to P_2 is > 40% of inlet pressure, primarily for long lines, use the following choices, or break the line into segments and calculate ΔP for each as above.

Also use Babcock formula given in another paragraph for steam flow.

$$q'_h = 24{,}700\ [Yd^2/S_g]\ (\Delta P\ \rho_1/K)^{1/2}, \text{ CFH @ 14.7 psia and 60°F} \qquad (2\text{-}79)$$

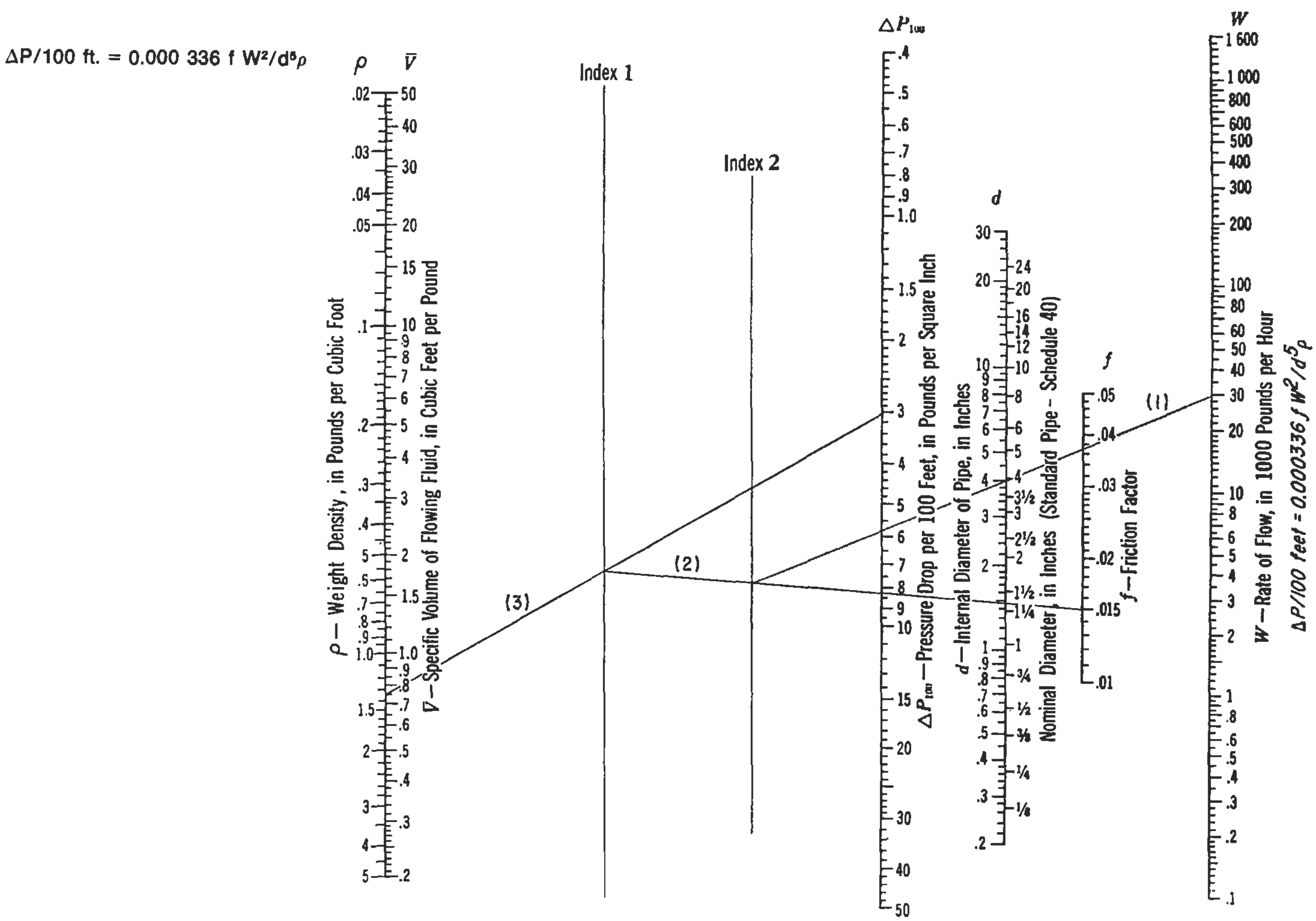


Figure 2-30. Pressure drop in compressible flow lines. By permission, Crane Co., *Technical Paper #410,* Engineering Div. 1957. Also see 1976 edition.

$$q'_h = 40{,}700\, Y d^2 \left[(\Delta P)\,(P'_1)/(K\, T_1\, S_g)\right]^{1/2} \quad (2\text{-}80)$$

same units as Equation 2-79 above

where Y = net expansion factor for compressible flow through orifices, nozzles, or pipe
K = resistence coefficient, ft
P′ = pressure, lbs/sq in. absolute
w_s = flow rate, lbs/sec.

Isothermal conditions, usually long pipe lines [3]:

$$w_s = \sqrt{\left[\frac{144\, gA^2}{\bar{V}_1\left(\frac{fL}{D} + 2\log_e \frac{P'_1}{P'_2}\right)}\right]\left[\frac{(P'_1)^2 - (P'_2)^2}{P'_1}\right]},$$

lbs/sec (2-3)

plus the conditions listed. The equation is based on steady flow, perfect gas laws, average velocity at a cross section, constant friction factor, and the pipe is straight and horizontal between end points.

D = pipe ID, ft
L = pipe length, ft
A = cross-sectional area for flow for pipe, sq ft

B. Alternate Vapor/Gas Flow Methods

Note that all specialized or alternate methods for solving are convenient simplifications or empirical procedures of the fundamental techniques presented earlier. They are not presented as better approaches to solving the specific problem.

Figure 2-31 is useful in solving the usual steam or any vapor flow problem for turbulent flow based on the modified Darcy relation with fixed friction factors. At low vapor velocities the results may be low; then use Figure 2-30. For steel pipe the limitations listed in (A) above apply.

1. Determine C_1 and C_2 from Figure 2-31 and Table 2-11 for the steam flow rate and assumed pipe size respectively. Use Table 2-4 or Table 2-8 to select steam velocity for line size estimate.
2. Read the specific volume of steam at conditions, from steam tables.
3. Calculate pressure drop (Figure 2-31) per 100 feet of pipe from

$$\Delta P/100 \text{ feet} = C_1 C_2 \bar{V} \quad (2\text{-}81)$$

4. From Figure 2-20 or 2-21 determine the equivalent lengths of all fittings, valves, etc.
5. Determine expansion and contraction losses, fittings and at vessel connections.
6. Determine pressure drops through orifices and control valves.
7. Total system pressure drop

$$\Delta P \text{ Total} = (L + L_{eq})\,(\Delta P/100) + \text{Item 5} + \text{Item 6} \quad (2\text{-}57)$$

8. If pressure drop is too large, re-estimate line size and repeat calculations (see paragraph (A) above) and also examine pressure drop assumptions for orifices and control valves.

C. Air

For quick estimates for air line pressure drop, see Tables 2-12A and 2-12B.

D. Babcock Empirical Formula for Steam

Comparison of results between the various empirical steam flow formulas suggests the Babcock equation as a good average for most design purposes at pressure 500 psia and below. For lines smaller than 4 inches, this relation may be 0–40 percent high [56].

$$p_1 - p_2 = \Delta P = 0.000131\,(1 + 3.6/d)\,\frac{w^2 L}{\rho\, d^5} \quad (2\text{-}82)$$

$$\Delta P/100 \text{ feet} = w^2 F/\rho \quad (2\text{-}83)$$

Figure 2-32 is a convenient chart for handling most in-plant steam line problems. For long transmission lines over 200 feet, the line should be calculated in sections in order to re-establish the steam specific density. Normally an estimated average ρ should be selected for each line increment to obtain good results.

Table 2-13 for "F" is convenient to use in conjunction with the equations.

Darcy Rational Relation for Compressible Vapors and Gases

1. Determine first estimate of line size by using suggested velocity from Table 2-4.
2. Calculate Reynolds number R_e and determine friction factor, f, using Figure 2-3 or Figure 2-33 (for steel pipe).
3. Determine total straight pipe length, L.
4. Determine equivalent pipe length for fittings, valves, L_{eq}.
5. Determine or assume losses through orifice plates, control valves, equipment, contraction and expansion, etc.

Pressure Drop per 100 feet Pipe:

$$\Delta P_{100} = C_1 C_2 \overline{V} = \frac{C_1\ C_2}{P}$$

$$C_1 = \frac{\Delta P_{100}}{C_2 \overline{V}} = \frac{\Delta P_{100} P}{C_2} \qquad C_2 = \frac{\Delta P_{100}}{C_1 \overline{V}} = \frac{\Delta P_{100} P}{C_1}$$

C_1 = Discharge Factor from Chart
C_2 = Size Factor, from Table 2-11

For ΔP>40% P_1, do not use this method.
For ΔP between 10% and 40% of P_1, use average for $\overline{V}$.
For ΔP<10% P_1, use $\overline{V}$ at P_1 or P_2.
ΔP_{100} = Psi, pressure drop per 100′ pipe.
$\overline{V}$ = Specific Volume, cu ft/lb.

Note: For quick estimates; not as accurate as friction loss calculations

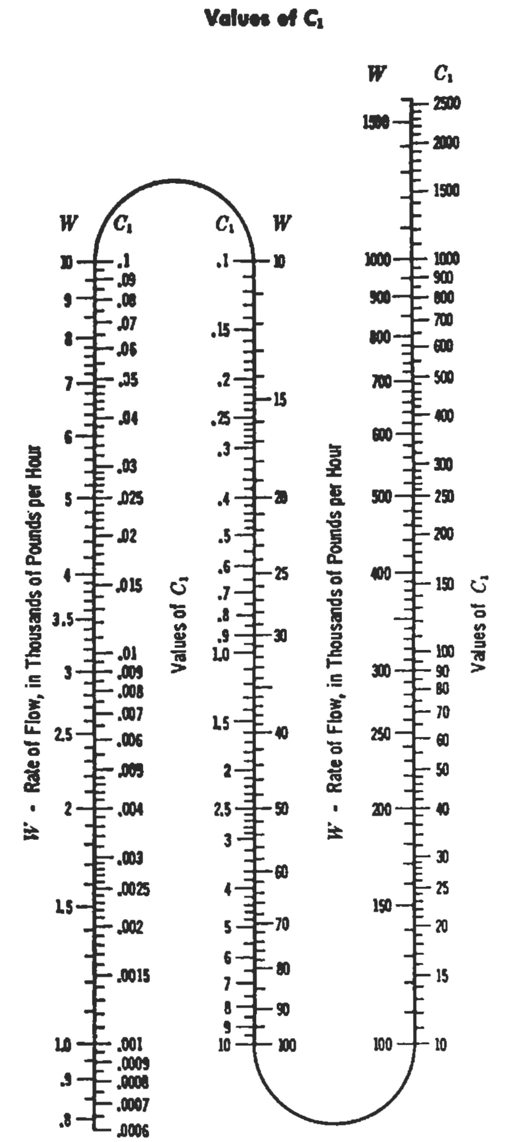


Figure 2-31. Simplified flow formula for compressible fluids. By permission, Crane Co., *Technical Paper #410*, 1957. Also see 1976 edition.

6. Calculate pressure drop, ΔP/100 ft (or use Figure 2-34).

$$\Delta P/100 \text{ feet} = \frac{0.000336\, f W^2}{\rho d^5} \qquad (2-77)$$

$$= \frac{0.000000726\, f T S_g\, (q_h')^2}{P' d^5} \qquad (2\text{-}77A)$$

7. Total pressure drop, ΔP total

$$= (L + L_{eq})\,(\Delta P/100) + \text{Item 5} \qquad (2\text{-}57)$$

8. If total line or system drop is excessive, examine the portion of drop due to pipe friction and that due to other factors in the system. If the line drop is a small portion of the total, little will be gained by increasing pipe size. Consider reducing losses through items in step 5 above. Recheck other pipe sizes as may be indicated.

Example 2-8 Pressure Drop for Vapor System

The calculations are presented in Figure 2-35, Line Size Specification Sheet.

Figure 2-36 is convenient when using Dowtherm vapor.

Alternate Solution to Compressible Flow Problems

There are several good approaches to recognizing the effects of changing conditions on compressible flow [44, 47].

Table 2-11
Simplified Flow Formula For Compressible Fluids Pressure Drop, Rate of Flow and Pipe Sizes*
(Use With Figure 2-31)

Values of C_2

Nominal Pipe Size Inches	Schedule Number	Value of C_2
1/8	40 s	7 920 000.
	80 x	26 200 000.
1/4	40 s	1 590 000.
	80 x	4 290 000.
3/8	40 s	319 000.
	80 x	718 000.
1/2	40 s	93 500.
	80 x	186 100.
	160	4 300 000.
	. . . xx	11 180 000.
3/4	40 s	21 200.
	80 x	36 900.
	160	100 100.
	. . . xx	627 000.
1	40 s	5 950.
	80 x	9 640.
	160	22 500.
	. . . xx	114 100.
1 1/4	40 s	1 408.
	80 x	2 110.
	160	3 490.
	. . . xx	13 640.
1 1/2	40 s	627.
	80 x	904.
	160	1 656.
	. . . xx	4 630.
2	40 s	169.
	80 x	236.
	160	488.
	. . . xx	899.
2 1/2	40 s	66.7
	80 x	91.8
	160	146.3
	. . . xx	380.0
3	40 s	21.4
	80 x	28.7
	160	48.3
	. . . xx	96.6
3 1/2	40 s	10.0
	80 x	37.7
4	40 s	5.17
	80 x	6.75
	120	8.94
	160	11.80
	. . . xx	18.59

Nominal Pipe Size Inches	Schedule Number	Value of C_2
5	40 s	1.59
	80 x	2.04
	120	2.69
	160	3.59
	. . . xx	4.93
6	40 s	0.610
	80 x	0.798
	120	1.015
	160	1.376
	. . . xx	1.861
8	20	0.133
	30	0.135
	40 s	0.146
	60	0.163
	80 x	0.185
	100	0.211
	120	0.252
	140	0.289
	. . . xx	0.317
	160	0.333
10	20	0.039 7
	30	0.042 1
	40 s	0.044 7
	60 x	0.051 4
	80	0.056 9
	100	0.066 1
	120	0.075 3
	140	0.090 5
	160	0.105 2
12	20	0.015 7
	30	0.016 8
	. . . s	0.017 5
	40	0.018 0
	. . . x	0.019 5
	60	0.020 6
	80	0.023 1
	100	0.026 7
	120	0.031 0
	140	0.035 0
	160	0.042 3
14	10	0.009 49
	20	0.009 96
	30 s	0.010 46
	40	0.010 99
	. . . x	0.011 55
	60	0.012 44
	80	0.014 16
	100	0.016 57
	120	0.018 98
	140	0.021 8
	160	0.025 2

Nominal Pipe Size Inches	Schedule Number	Value of C_2
16	10	0.004 63
	20	0.004 21
	30 s	0.005 04
	40 x	0.005 49
	60	0.006 12
	80	0.007 00
	100	0.008 04
	120	0.009 26
	140	0.010 99
	160	0.012 44
18	10	0.002 47
	20	0.002 56
	. . s	0.002 66
	30	0.002 76
	. . x	0.002 87
	40	0.002 98
	60	0.003 35
	80	0.003 76
	100	0.004 35
	120	0.005 04
	140	0.005 73
	160	0.006 69
20	10	0.001 41
	20 s	0.001 50
	30 x	0.001 61
	40	0.001 69
	60	0.001 91
	80	0.002 17
	100	0.002 51
	120	0.002 87
	140	0.003 35
	160	0.003 85
24	10	0.000 534
	20 s	0.000 565
	. . x	0.000 597
	30	0.000 614
	40	0.000 651
	60	0.000 741
	80	0.000 835
	100	0.000 972
	120	0.001 119
	140	0.001 274
	160	0.001 478

Note
The letters **s**, **x**, and **xx** in the columns of Schedule Numbers indicate Standard, Extra Strong, and Double Extra Strong pipe respectively.

By permission, Crane Co., *Technical Paper #410,* Engineering Div., 1957. See author's note at Figure 2-31.

Table 2-12A
Flow of Air Through Schedule 40 Pipe*
(Use for estimating; for detailed calculations use friction factors, f)

For lengths of pipe other than 100 feet, the pressure drop is proportional to the length. Thus, for 50 feet of pipe, the pressure drop is approximately one-half the value given in the table . . . for 300 feet, three times the given value, etc.

The pressure drop is also inversely proportional to the absolute pressure and directly proportional to the absolute temperature.

Therefore, to determine the pressure drop for inlet or average pressures other than 100 psi and at temperatures other than 60 F, multiply the values given in the table by the ratio:

$$\left(\frac{100+14.7}{P+14.7}\right)\left(\frac{460+t}{520}\right)$$

where:

"P" is the inlet or average gauge pressure in pounds per square inch, and,

"t" is the temperature in degrees Fahrenheit under consideration.

The cubic feet per minute of compressed air at any pressure is inversely proportional to the absolute pressure and directly proportional to the absolute temperature.

To determine the cubic feet per minute of compressed air at any temperature and pressure other than standard conditions, multiply the value of cubic feet per minute of free air by the ratio:

$$\left(\frac{14.7}{14.7+P}\right)\left(\frac{460+t}{520}\right)$$

Calculations for Pipe Other than Schedule 40

To determine the velocity of water, or the pressure drop of water or air, through pipe other than Schedule 40, use the following formulas:

$$v_a = v_{40}\left(\frac{d_{40}}{d_a}\right)^2$$

$$\Delta P_a = \Delta P_{40}\left(\frac{d_{40}}{d_a}\right)^5$$

Subscript "a" refers to the Schedule of pipe through which velocity or pressure drop is desired.

Subscript "40" refers to the velocity or pressure drop through Schedule 40 pipe, as given in the tables on these facing pages.

Free Air q'_m Cubic Feet Per Minute at 60 F and 14.7 psia	Compressed Air Cubic Feet Per Minute at 60 F and 100 psig	Pressure Drop of Air In Pounds per Square Inch Per 100 Feet of Schedule 40 Pipe. For Air at 100 Pounds per Square Inch Gauge Pressure and 60 F Temperature								
		1/8"	1/4"	3/8"	1/2"					
1	0.128	0.361	0.083	0.018						
2	0.256	1.31	0.285	0.064	0.020					
						3/4"				
3	0.384	3.06	0.605	0.133	0.042					
4	0.513	4.83	1.04	0.226	0.071					
5	0.641	7.45	1.58	0.343	0.106	0.027				
							1"			
6	0.769	10.6	2.23	0.408	0.148	0.037				
8	1.025	18.6	3.89	0.848	0.255	0.062	0.019			
								1 1/4"	1 1/2"	
10	1.282	28.7	5.96	1.26	0.356	0.094	0.029			
15	1.922	...	13.0	2.73	0.834	0.201	0.062			
20	2.563	...	22.8	4.76	1.43	0.345	0.102	0.026		
25	3.204	...	35.6	7.34	2.21	0.526	0.156	0.039	0.019	
30	3.845	...	...	10.5	3.15	0.748	0.219	0.055	0.026	
35	4.486	...	...	14.2	4.24	1.00	0.293	0.073	0.035	
40	5.126	...	...	18.4	5.49	1.30	0.379	0.095	0.044	
										2"
45	5.767	...	...	23.1	6.90	1.62	0.474	0.116	0.055	
50	6.408			28.5	8.49	1.99	0.578	0.149	0.067	0.019
		2 1/2"								
60	7.690			40.7	12.2	2.85	0.819	0.200	0.094	0.027
70	8.971			...	16.5	3.83	1.10	0.270	0.126	0.036
80	10.25	0.019		...	21.4	4.96	1.43	0.350	0.162	0.046
90	11.53	0.023		...	27.0	6.25	1.80	0.437	0.203	0.058
			3"							
100	12.82	0.029			33.2	7.69	2.21	0.534	0.247	0.070
125	16.02	0.044			...	11.9	3.39	0.825	0.380	0.107
150	19.22	0.062	0.021		...	17.0	4.87	1.17	0.537	0.151
175	22.43	0.083	0.028		...	23.1	6.60	1.58	0.727	0.205
				3 1/2"						
200	25.63	0.107	0.036		...	30.0	8.54	2.05	0.937	0.264
225	28.84	0.134	0.045	0.022		37.9	10.8	2.59	1.19	0.331
250	32.04	0.164	0.055	0.027		...	13.3	3.18	1.45	0.404
275	35.24	0.191	0.066	0.032		...	16.0	3.83	1.75	0.484
300	38.45	0.232	0.078	0.037		...	19.0	4.56	2.07	0.573
325	41.65	0.270	0.090	0.043		...	22.3	5.32	2.42	0.673
					4"					
350	44.87	0.313	0.104	0.050		...	25.8	6.17	2.80	0.776
375	48.06	0.356	0.119	0.057	0.030	...	29.6	7.05	3.20	0.887
400	51.26	0.402	0.134	0.064	0.034	...	33.6	8.02	3.64	1.00
425	54.47	0.452	0.151	0.072	0.038	...	37.9	9.01	4.09	1.13
450	57.67	0.507	0.168	0.081	0.042	...	...	10.2	4.59	1.26
475	60.88	0.562	0.187	0.089	0.047		...	11.3	5.09	1.40
500	64.08	0.623	0.206	0.099	0.052		...	12.5	5.61	1.55
550	70.49	0.749	0.248	0.118	0.062		...	15.1	6.79	1.87
600	76.90	0.887	0.293	0.139	0.073		...	18.0	8.04	2.21
						5"				
650	83.30	1.04	0.342	0.163	0.086		...	21.1	9.43	2.60
700	89.71	1.19	0.395	0.188	0.099	0.032		24.3	10.9	3.00
750	96.12	1.36	0.451	0.214	0.113	0.036		27.9	12.6	3.44
800	102.5	1.55	0.513	0.244	0.127	0.041		31.8	14.2	3.90
850	108.9	1.74	0.576	0.274	0.144	0.046		35.9	16.0	4.40
							6"			
900	115.3	1.95	0.642	0.305	0.160	0.051		40.2	18.0	4.91
950	121.8	2.18	0.715	0.340	0.178	0.057	0.023	...	20.0	5.47
1 000	128.2	2.40	0.788	0.375	0.197	0.063	0.025	...	22.1	6.06
1 100	141.0	2.89	0.948	0.451	0.236	0.075	0.030	...	26.7	7.29
1 200	153.8	3.44	1.13	0.533	0.279	0.089	0.035	...	31.8	8.63
1 300	166.6	4.01	1.32	0.626	0.327	0.103	0.041	...	37.3	10.1
1 400	179.4	4.65	1.52	0.718	0.377	0.119	0.047			11.8
1 500	192.2	5.31	1.74	0.824	0.431	0.136	0.054			13.5
								8"		
1 600	205.1	6.04	1.97	0.932	0.490	0.154	0.061			15.3
1 800	230.7	7.65	2.50	1.18	0.616	0.193	0.075			19.3
2 000	256.3	9.44	3.06	1.45	0.757	0.237	0.094	0.023		23.9
									10"	
2 500	320.4	14.7	4.76	2.25	1.17	0.366	0.143	0.035		37.3
3 000	384.5	21.1	6.82	3.20	1.67	0.524	0.204	0.051	0.016	
3 500	448.6	28.8	9.23	4.33	2.26	0.709	0.276	0.068	0.022	
4 000	512.6	37.6	12.1	5.66	2.94	0.919	0.358	0.088	0.028	
										12"
4 500	576.7	47.6	15.3	7.16	3.69	1.16	0.450	0.111	0.035	
5 000	640.8	...	18.8	8.85	4.56	1.42	0.552	0.136	0.043	0.018
6 000	769.0	...	27.1	12.7	6.57	2.03	0.794	0.195	0.061	0.025
7 000	897.1	...	36.9	17.2	8.94	2.76	1.07	0.262	0.082	0.034
8 000	1025	...	...	22.5	11.7	3.59	1.39	0.339	0.107	0.044
9 000	1153	...	...	28.5	14.9	4.54	1.76	0.427	0.134	0.055
10 000	1282	...	...	35.2	18.4	5.60	2.16	0.526	0.164	0.067
11 000	1410	...	...	...	22.2	6.78	2.62	0.633	0.197	0.081
12 000	1538	...	...	...	26.4	8.07	3.09	0.753	0.234	0.096
13 000	1666	...	...	...	31.0	9.47	3.63	0.884	0.273	0.112
14 000	1794	...	...	...	36.0	11.0	4.21	1.02	0.316	0.129
15 000	1922	...	...	...	...	12.6	4.84	1.17	0.364	0.148
16 000	2051	...	...	...	...	14.3	5.50	1.33	0.411	0.167
18 000	2307	...	...	...	...	18.2	6.96	1.68	0.520	0.213
20 000	2563	...	...	...	...	22.4	8.60	2.01	0.642	0.260
22 000	2820	...	...	...	...	27.1	10.4	2.50	0.771	0.314
24 000	3076	...	...	...	...	32.3	12.4	2.97	0.918	0.371
26 000	3332	...	...	...	...	37.9	14.5	3.49	1.12	0.435
28 000	3588	...	...	...	...	...	16.9	4.04	1.25	0.505
30 000	3845	...	...	...	...	...	19.3	4.64	1.42	0.520

*By permission *Technical Paper No. 410*, Crane Co., Engineering Div., Chicago, 1957.

Table 2-12B
Discharge of Air Through an Orifice*

In cubic feet of free air per minute at standard atmospheric pressure of 14.7 lb. per sq. in. absolute and 70° F.

Gauge Pressure before Orifice in Pounds per sq. in.	DIAMETER OF ORIFICE										
	$\frac{1}{64}$"	$\frac{1}{32}$"	$\frac{1}{16}$"	⅛"	¼"	⅜"	½"	⅝"	¾"	⅞"	1"
	Discharge in Cubic feet of free air per minute										
1............	.028	.112	.450	1.80	7.18	16.2	28.7	45.0	64.7	88.1	115
2............	.040	.158	.633	2.53	10.1	22.8	40.5	63.3	91.2	124	162
3............	.048	.194	.775	3.10	12.4	27.8	49.5	77.5	111	152	198
4............	.056	.223	.892	3.56	14.3	32.1	57.0	89.2	128	175	228
5............	.062	.248	.993	3.97	15.9	35.7	63.5	99.3	143	195	254
6	.068	.272	1.09	4.34	17.4	39.1	69.5	109	156	213	278
7	.073	.293	1.17	4.68	18.7	42.2	75.0	117	168	230	300
9	.083	.331	1.32	5.30	21.2	47.7	84.7	132	191	260	339
12	.095	.379	1.52	6.07	24.3	54.6	97.0	152	218	297	388
15	.105	.420	1.68	6.72	26.9	60.5	108	168	242	329	430
20	.123	.491	1.96	7.86	31.4	70.7	126	196	283	385	503
25	.140	.562	2.25	8.98	35.9	80.9	144	225	323	440	575
30	.158	.633	2.53	10.1	40.5	91.1	162	253	365	496	648
35	.176	.703	2.81	11.3	45.0	101	180	281	405	551	720
40	.194	.774	3.10	12.4	49.6	112	198	310	446	607	793
45	.211	.845	3.38	13.5	54.1	122	216	338	487	662	865
50	.229	.916	3.66	14.7	58.6	132	235	366	528	718	938
60	.264	1.06	4.23	16.9	67.6	152	271	423	609	828	1082
70	.300	1.20	4.79	19.2	76.7	173	307	479	690	939	1227
80	.335	1.34	5.36	21.4	85.7	193	343	536	771	1050	1371
90	.370	1.48	5.92	23.7	94.8	213	379	592	853	1161	1516
100	.406	1.62	6.49	26.0	104	234	415	649	934	1272	1661
110	.441	1.76	7.05	28.2	113	254	452	705	1016	1383	1806
120	.476	1.91	7.62	30.5	122	274	488	762	1097	1494	1951
125	.494	1.98	7.90	31.6	126	284	506	790	1138	1549	2023

Table is based on 100% coefficient of flow. For well rounded entrance multiply values by 0.97. For sharp edged orifices a multiplier of 0.65 may be used for approximate results.

Values for pressures from 1 to 15 lbs. gauge calculated by standard adiabatic formula.

Values for pressures above 15 lb. gauge calculated by approximate formula proposed by S. A. Moss.

$$W_s = .5303 \frac{aCP_1}{\sqrt{T_1}}$$

Where:
W_s = discharge in lbs. per sec.
a = area of orifice in sq. in.
C = Coefficient of flow
P_1 = Upstream total pressure in lbs. per sq. in. absolute
T_1 = Upstream temperature in °F. abs.

Values used in calculating above table were; C = 1.0, P_1 = gauge pressure + 14.7 lbs./sq. in. T_1 = 530° F. abs.

Weights (W) were converted to volumes using density factor of 0.07494 lbs./cu. ft. This is correct for dry air at 14.7 lbs. per sq. in. absolute pressure and 70° F.

Formula cannot be used where P_1 is less than two times the barometric pressure.

*By permission "Compressed Air Data," F. W. O'Neil, Editor, *Compressed Air Magazine*, 5th Edition, New York, 1939 [49].

Friction Drop for Air

Table 2-12A is convenient for most air problems, noting that both free air (60°F and 14.7 psia) and compressed air at 100 psig and 60°F are indicated. The corrections for other temperatures and pressures are also indicated. Figure 2-37 is useful for quick checking. However, its values are slightly higher (about 10 percent) than the rational values of Table 2-11, above about 1000 cfm of free air. Use for estimating only.

Example 2-9: Steam Flow Using Babcock Formula

Determine the pressure loss in 138 feet of 8-inch Schedule 40 steel pipe, flowing 86,000 pounds per hour of 150 psig steam (saturated).

Use Figure 2-32, w = 86,000/60 = 1432 lbs/min

Reading from top at 150 psig, no superheat, down vertically to intersect the horizontal steam flow of 1432 lbs/min, follow diagonal line to the horizontal pipe size

Table 2-13
Factor "F" For Babcock Steam Formula*

Nominal Pipe Size Inches	*Standard Weight Pipe	#Extra Strong Pipe
½	955.1 x 10^{-3}	2.051 x
¾	184.7 x 10^{-3}	340.8 x 10^{-3}
1	45.7 x 10^{-3}	77.71 x 10^{-3}
1¼	9.432 x 10^{-3}	14.67 x 10^{-3}
1½	3.914 x 10^{-3}	5.865 x 10^{-3}
2	951.9 x 10^{-6}	1.365 x 10^{-3}
2½	351.0 x 10^{-6}	493.8 x 10^{-6}
3	104.7 x 10^{-6}	143.2 x 10^{-6}
3½	46.94 x 10^{-6}	62.95 x 10^{-6}
4	23.46 x 10^{-6}	31.01 x 10^{-6}
5	6.854 x 10^{-6}	8.866 x 10^{-6}
6	2.544 x 10^{-6}	3.354 x 10^{-6}
8	587.1 x 10^{-9}	748.2 x 10^{-9}
10	176.3 x 10^{-9}	225.3 x 10^{-9}
12	70.32 x 10^{-9}	90.52 x 10^{-9}
14 O.D.	42.84 x 10^{-9}	55.29 x 10^{-9}
16 O.D.	21.39 x 10^{-9}	27.28 x 10^{-9}
18 O.D.	11.61 x 10^{-9}	14.69 x 10^{-9}
20 O.D.	6.621 x 10^{-9}	8.469 x 10^{-9}
24 O.D.	2.561 x 10^{-9}	3.278 x 10^{-9}

*Factors are based upon I.D. listed as Schedule 40.
#Factors are based upon I.D. listed as Schedule 80.
†By permission The Walworth Co.

of 8 inches, and then vertically down to the pressure drop loss of 3.5 psi/100 feet.

For 138 feet (no fittings or valves), total ΔP is 138 (3.5/100) = 4.82 psi.

For comparison, solve by equation, using value of F = 587.1×10^{-9} from Table 2-13.

$$\Delta P/100\text{ ft} = (1432)^2\,(587.1 \times 10^{-9})/0.364$$
$$= 3.32\text{ psi}/100\text{ ft}$$
$$\Delta P\text{ total} = (3.32/100)\,(138) = 4.75\text{ psi}$$

These values are within graphical accuracy.

Sonic Conditions Limiting Flow of Gases and Vapors

The sonic or critical velocity (speed of sound in the fluid) is the maximum velocity which a *compressible* fluid can attain in a pipe [3].

$$v_s = [(c_p/c_v)\,(32.2)\,(1544/MW)\,(460 + t)]^{1/2} \tag{2-84}$$
$$= 68.1\,[(c_p/c_v)\,P'/\rho]^{1/2},\text{ ft/sec}$$

where the properties are evaluated at the condition of sonic flow.

This applies regardless of the downstream pressure for a fixed upstream pressure. This limitation must be evaluated separately from pressure drop relations, as it is not included as a built in limitation.

Sonic velocity will be established at a restricted point in the pipe, or at the outlet, if the pressure drop is great enough to establish the required velocity. Once the sonic velocity has been reached, the pressure drop in the system will not increase, as the velocity will remain at this value even though the fluid may be discharging into a vessel at a lower pressure than that existing at the point where sonic velocity is established.

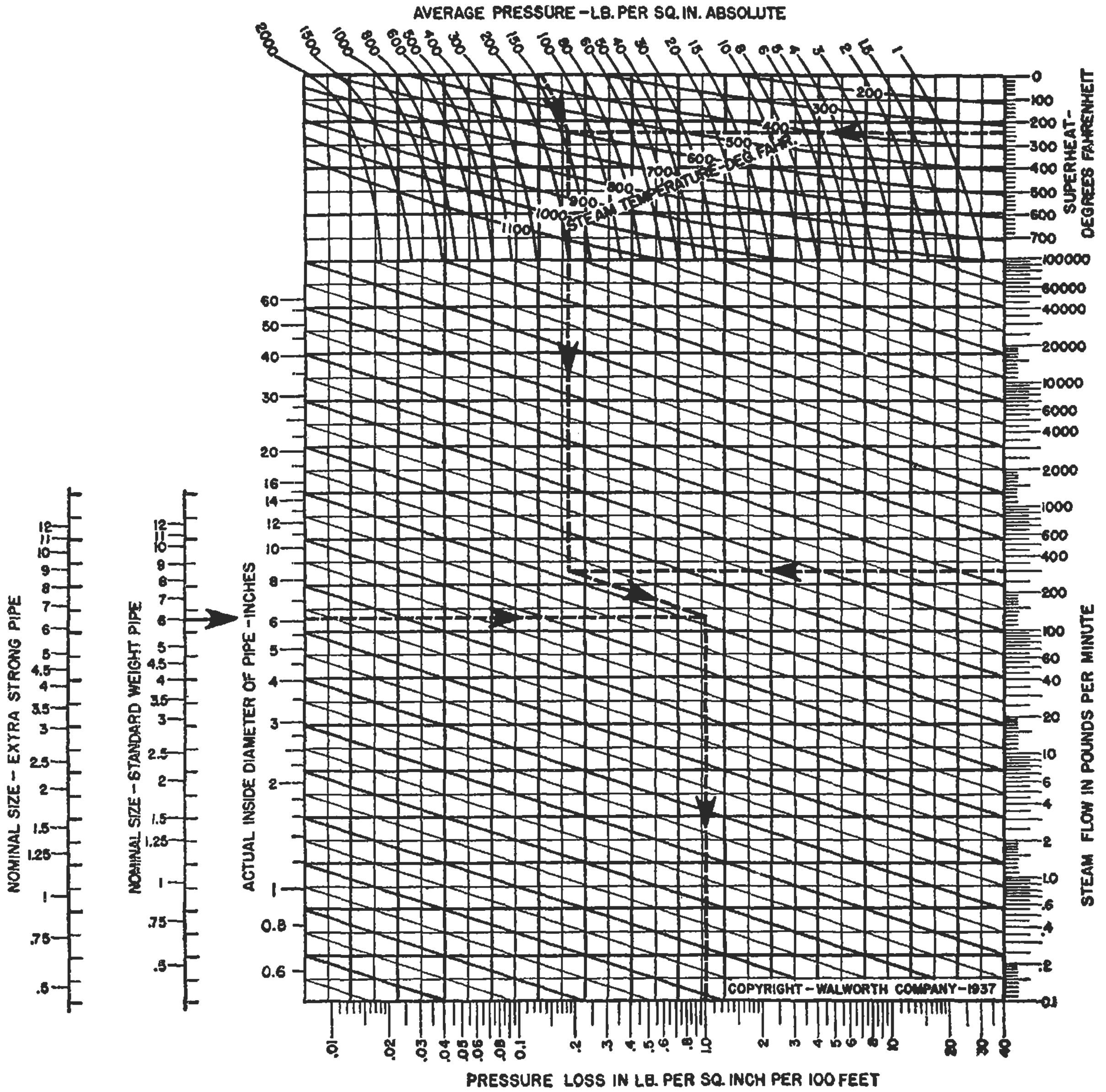
In general, the sonic or critical velocity is attained for an outlet or downstream pressure equal to or less than one half the upstream or inlet absolute pressure condition of a system. The discharge through an orifice or nozzle is usually a limiting condition for the flow through the end of a pipe. The usual pressure drop equations do not hold at the sonic velocity, as in an orifice. Conditions or systems exhausting to atmosphere (or vacuum) from medium to high pressures should be examined for critical flow, otherwise the calculated pressure drop may be in error.

All flowing gases and vapors (compressible fluids) including steam (which is a vapor) are limited or approach a maximum in mass flow velocity or rate, i.e., lbs/sec or lbs/hr through a pipe depending upon the specific upstream or starting pressure. This maximum rate of flow cannot be exceeded regardless of how much the downstream pressure is further reduced [3]. To determine the actual velocity in a pipe, calculate by

$$v = \frac{3.06\,W\,\bar{V}}{d^2}$$ or use Figure 2-34.

This maximum velocity of a compressible fluid in a pipe is limited by the velocity of propagation of a pressure wave that travels at the speed of sound in the fluid [3]. This speed of sound is specific for each individual gas or vapor or liquid and is a function of the ratio of specific heats of the fluid. The pressure reduces and the velocity increases as the fluid flows downstream through the pipe, with the maximum velocity occurring at the downstream end of the pipe. When, or if, the pressure drop is great enough, the discharge or exit or outlet velocity will reach the velocity of sound for that fluid.

If the outlet or discharge pressure is lowered further, the pressure upstream at the origin will not detect it because the pressure wave can only travel at sonic velocity. Therefore, the change in pressure downstream will not be detected upstream. The excess pressure drop obtained by lowering the outlet pressure after the maximum discharge has been reached takes place beyond the end of the pipe [3]. This pressure is lost in shock waves and turbulence of the jetting fluid. See References 12, 13, 24, and 15 for further expansion of shock waves and detonation waves through compressible fluids.



Based on Babcock Formula: $P = 0.000131\left(1+\frac{3.6}{d}\right)\frac{w^2L}{\rho d^5}$

Figure 2-32. Steam flow chart. (By permission, Walworth Co. Note: use for estimating only (this author).)

The maximum possible velocity of a compressible fluid in a pipe is sonic (speed of sound) velocity, as:

$$\text{or, } \Delta P / 100 \text{ ft} = \frac{0.000001959f\,(q'_h)^2 S_g^{\,2}}{d^5 p} \qquad (2\text{-}78)$$

where
- k = ratio of specific heat for gas or vapor at constant pressure
- R = individual gas constant = MR/M = 1544/M
- M = molecular weight
- MR = universal gas constant = 1544
- T = temperature of gas, R, = (460 + °F)
- P′ = pressure, Psi abs (Psia)
- v_s = sonic or critical velocity of flow of a gas, ft/sec
- $\overline{V}_1$ = specific volume of fluid, cu ft/lb at T and P′
- g = acceleration of gravity = 32.2 ft/per/sec

Thus the maximum flow in a pipe occurs when the velocity at the exit becomes sonic. The sonic location may be other than the exit, can be at restrictive points in the system, or at control/safety relief valves.

Shock waves travel at supersonic velocities and exhibit a near discontinuity in pressure, density, and tempera-

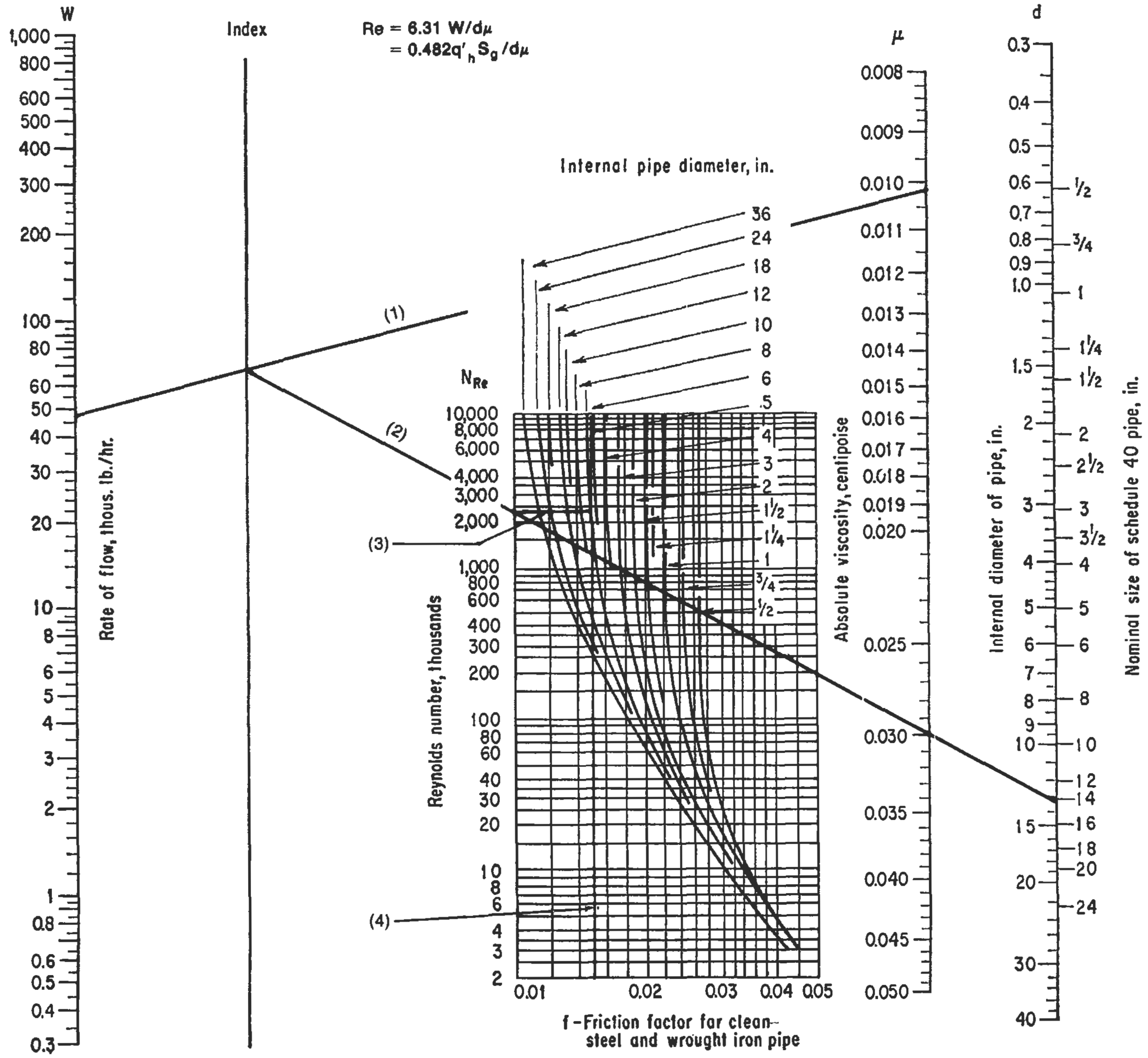


Figure 2-33. Reynolds number for compressible flow, steel pipe. By permission, Crane Co., *Technical Paper #410*, Engineering Div., 1957. Also see 1976 edition.

ture, and a great potential exists for damage from such waves [15]. A discussion of shock waves is beyond the scope of this chapter.

Velocity considerations are important in rotating or reciprocating machinery systems, because, if the compressible fluid velocity exceeds the speed of sound in the fluid, shock waves can be set up and the results of such conditions are much different than the velocities below the speed of sound. The ratio of the actual fluid velocity to its speed of sound is called the Mach number [38].

The velocity of sound at 68°F in air is 1126 ft/sec.

For any gas, the speed of sound is:

$$v_s = \sqrt{kgp''/\rho}, \text{ ft/sec} \tag{2-86}$$

(equation continued on page 113)

$$v = \frac{3.06\, W\bar{V}}{d^2} = \frac{3.06\, W}{d^2 \rho}$$

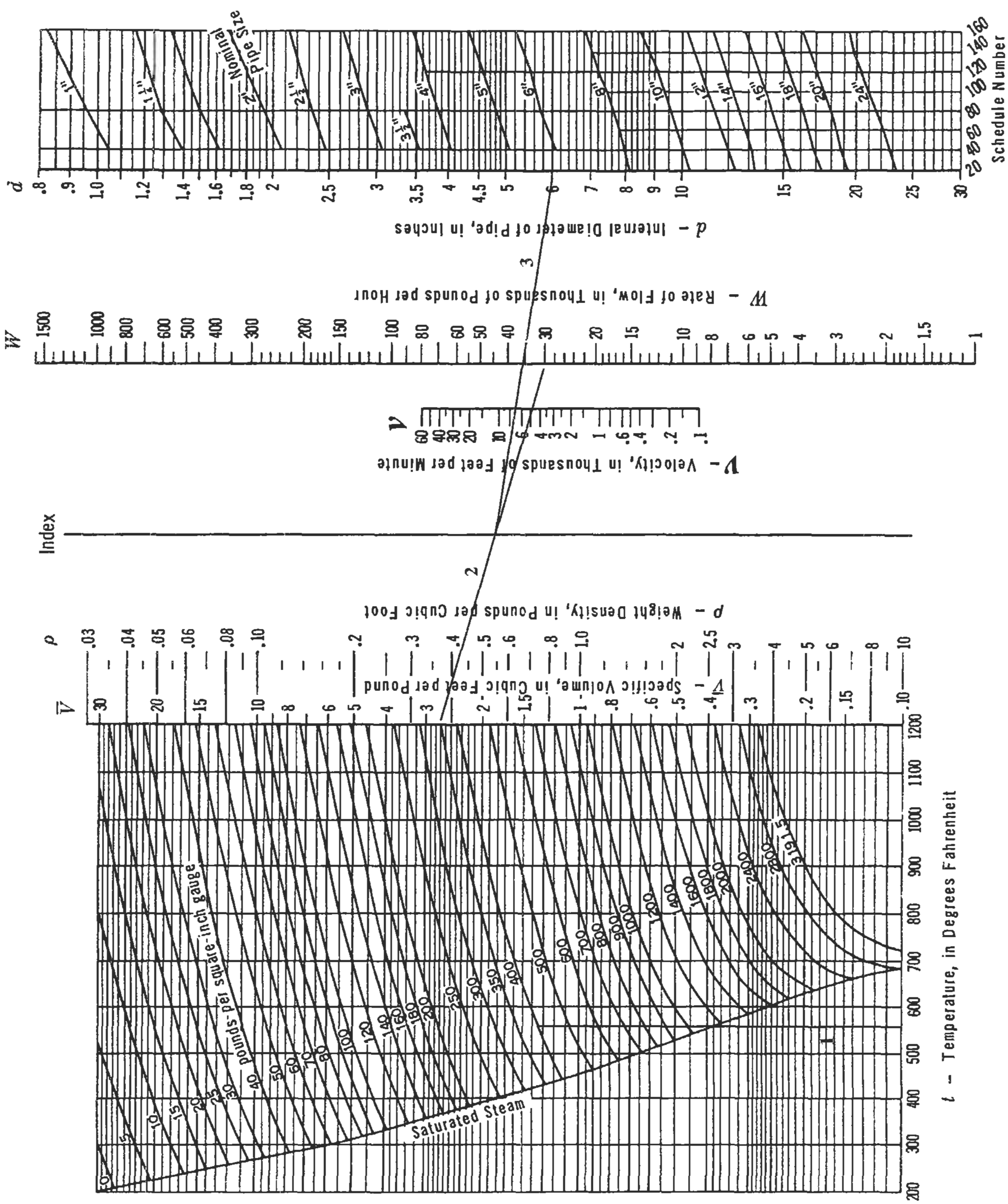


Figure 2-34. Pressure drop in compressible flow lines. By permission, Crane Co., *Technical Paper #410,* Engineering Div., 1957. Also see 1976 edition.

SHEET NO. ______

LUDWIG CONSULTING ENGINEERS

LINE SIZE SHEET

By ______ Job No. ______

Date ______ Charge No. ______

Line No. LP - 61 Flow Sheet Drawing No. ______

Line Description Vent through Exchanger for Tower T - 3

Fluid in line N_2 + Hydrocarbon Temperature 140 °F

GPM (Calc.) ______ GPM (des.) ______ Pressure 5.3 psig

CFM (Calc.) 2060 CFM (des.) 2270 Sp. Gr. 0.975

Lbs./hr.(Calc.) 10,841 Lbs./hr.(des.) 12,000 Sp. Vol. 11.3 cu.ft./lb.

Recommended Velocity ______ fp ______ Viscosity 0.019 cp

Straight pipe, fittings, valves expansion, contraction, etc.			
Item	No.	Unit Eq. Ft.	Total Eq. Ft.
St. Line			5
Gate V.	1	11	11
	1	6	6
Tee - SO	1	50	50
		Total	72

Item	Pressure Drop in-
Pipe & Equivalent	0.0617
Orifice	
Motor Valve	
Miscellaneous	
Exchanger drop	1.50
Total	1.56

Cross-sect. area, 10'' pipe=0.547 sq.ft.

Velocity = 2270/0.547 = 4150 Feet/Min.

Estimated line size 10'' (existing)

Actual Velocity 4150 fp

Unit Loss per 100 ft. 0.0857 psi

Total head loss in feet of liquid ______

Total pressure drop in psi 1.56

Selected pipe size 10'' Material & Weight Schedule 40, Steel

Calculations: $Re = \frac{6.31\ W}{d\ \mu} = \frac{(6.31)(12{,}000)}{(10.02)(0.019)} = 3.98 \times 10^5$

$f = 0.0158 \qquad \rho = 1/\bar{V}$

$\Delta P/100\ \text{feet} = \frac{(0.000336)(f)(W)^2}{d^5 \rho} = \frac{(0.000336)(0.0158)(12{,}000)^2}{(10.02)^5\ (1/11.3)}$

= 0.0857 Psi/100 equivalent feet of pipe(as pipe,fittings,valves, etc)

ΔP Total (friction) = (0.0857/100)(72) = 0.0617 Psi

Checked by: ______ Date: ______

Figure 2-35. Example of pressure drop for a vapor system, Example 2-8.

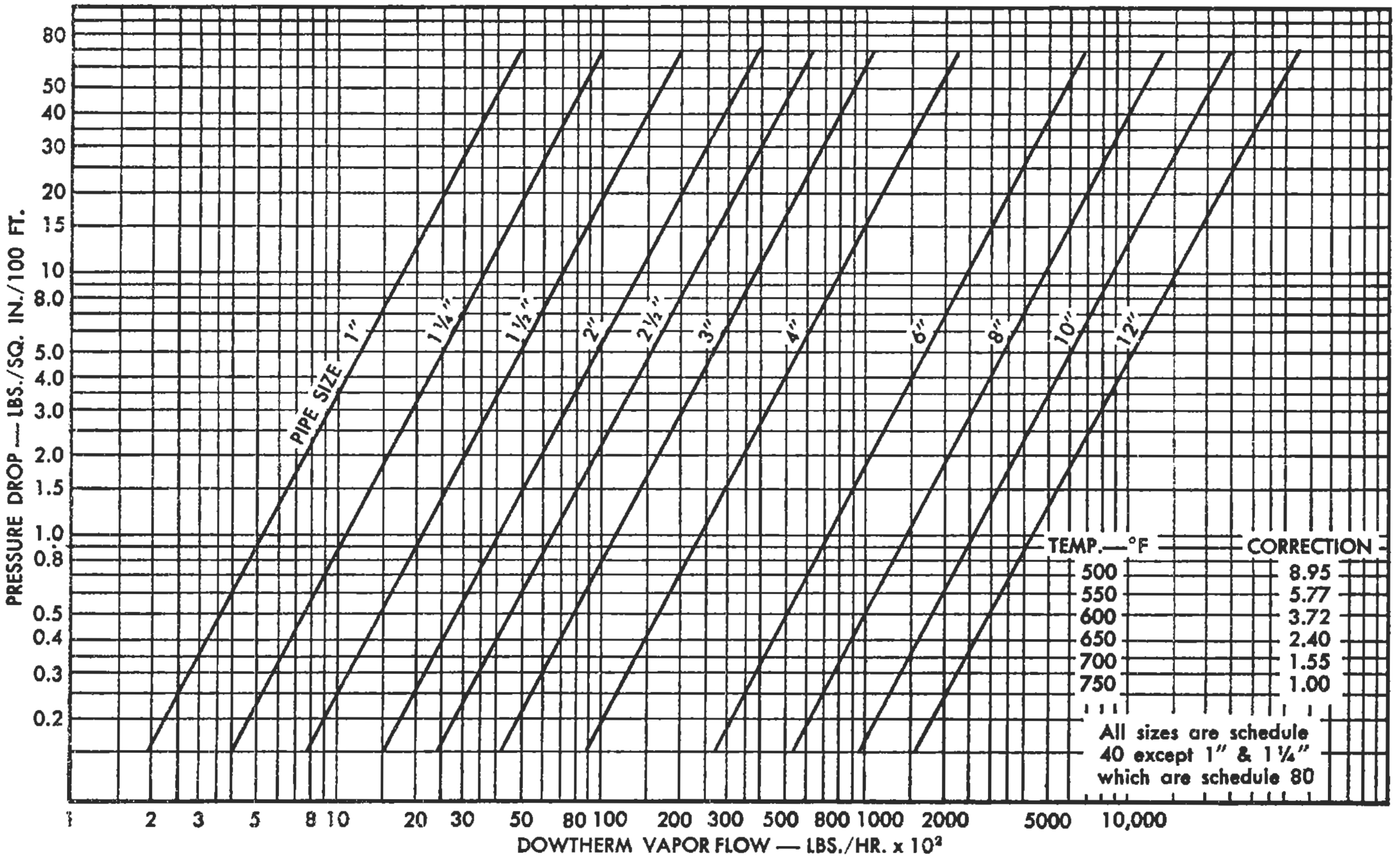


Figure 2-36. Pressure drop, Dowtherm "A"® vapor in steel pipe. By permission, Struthers Wells Corp., Bull. D-45.

(equation continued from page 110)

k = ratio of specific heat of gas, at constant pressure to that at constant volume, = c_p/c_v. See Table 2-14

g = 32.2 ft/sec squared

p″ = pressure, pounds per sq ft, abs (Psf abs) (note units)

ρ = the specific weight, lb/cu ft (see Appendix) at T and p″

This sonic velocity occurs in a pipe system in a restricted area (for example, valve, orifice, venturi) or at the outlet end of pipe (open-ended), as long as the upstream pressure is high enough. The physical properties in the above equations are at the point of maximum velocity.

For the discharge of compressible fluids from the end of a short piping length into a larger cross section, such as a larger pipe, vessel, or atmosphere, the flow is considered adiabatic. Corrections are applied to the Darcy equation to compensate for fluid property changes due to the expansion of the fluid, and these are known as Y net expansion factors [3]. The corrected Darcy equation is:

For valves, fittings, and pipe (vapors/gases):

$$w_s = 0.525\, Y\, d_i^2 \sqrt{\Delta P / (K\overline{V}_1)},\ \text{lbs/sec} \qquad (2\text{-}87)$$

For nozzles and orifices (vapors/gases):

$$w_s = 0.525\, Y\, d_i^2\, C' \sqrt{\frac{\Delta P}{\overline{V}_1}} \qquad (2\text{-}88)$$

For valves, fittings, and pipe (liquids):

$$w_s = 0.525\, d_i^2 \sqrt{\frac{\Delta P\,(\rho_1)}{K}} \qquad (2\text{-}89)$$

For nozzles and orifices (liquids):

$$w_s = 0.525\, d_i^2\, C' \sqrt{\Delta p\,(\rho_1)} \qquad (2\text{-}90)$$

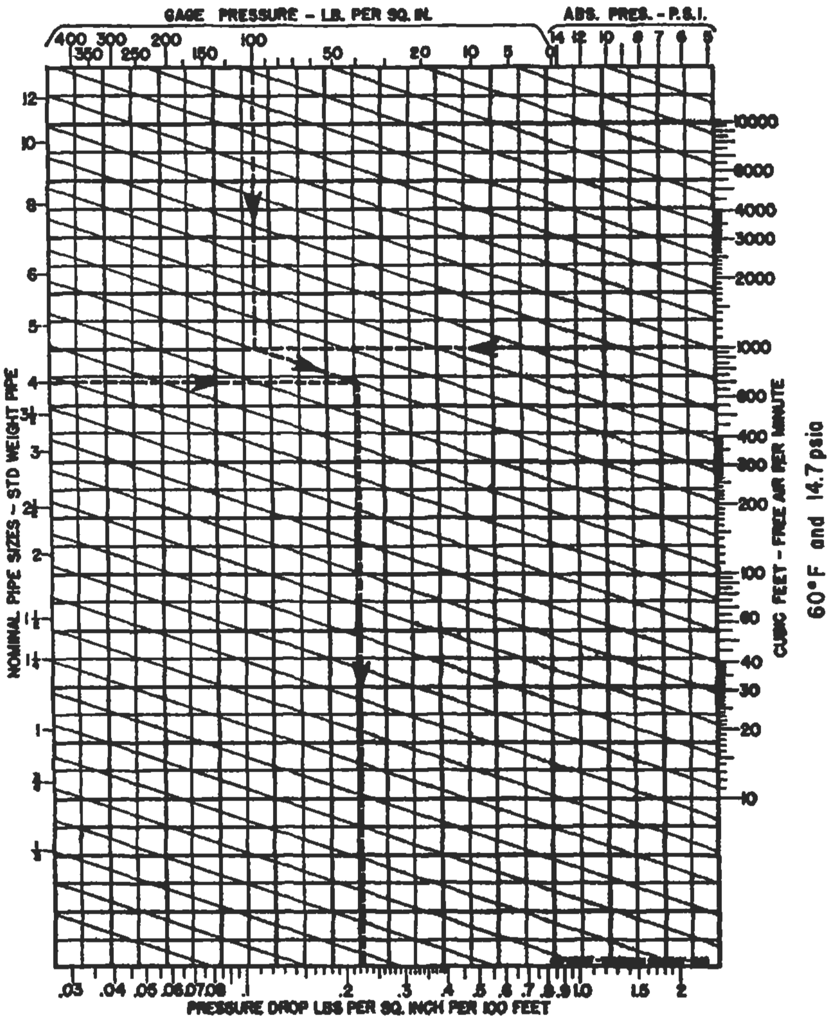
where $\overline{V}_1$ = upstream specific volume of fluid, cu ft/lbs

w_s = rate of flow, lbs/sec

ΔP = pressure drop across the system, psi (inlet-discharge)

K = total resistance coefficient of pipe, valves, fittings, and entrance and exit losses in the line

Fluid Flow



Based on Harris Formula, $\Delta p = \dfrac{0.1025\, LV}{d^{5.31}\, R_c}$, psi

L = Pipe Length, feet

R_c = Ratio of Compression (from free air) at Entrance of Pipe

d = I.D., Inches

V = Air Flow, cu. ft./sec. (free air)

Figure 2-37. Compressed air flow chart. By permission, Walworth Co. Note: use for estimating only (this author).

Y = net expansion factor for compressible flow through orifices, nozzles, and pipe [3] (see Figures 2-38A and 2-38B)

ΔP = pressure drop ratio in ΔP/P′, used to determine Y from Figures 2-38A and 2-38B. The ΔP is the difference between the inlet pressure and the pressure in the area of larger cross section.

d_i = pipe inside diameter, in.

C′ = flow coefficient for orifices and nozzles (Figures 2-17 and 2-18)

For example, for a line discharging a compressible fluid to atmosphere, the ΔP is the inlet gauge pressure or the difference between the absolute inlet pressure and atmospheric pressure absolute. When $\Delta P/P_1'$ falls outside the limits of the K curves on the charts, sonic velocity occurs at the point of discharge or at some restriction within the pipe, and the limiting value for Y and ΔP must be determined from the tables on Figure 2-38A, and used in the velocity equation, v_s, above [3].

Table 2-14
Typical Ratios of Specific Heats, k

Compound	$k = c_p/c_v$
Air	1.40
Ammonia	1.29
Argon	1.67
Carbon Dioxide	1.28
Carbon Monoxide	1.41
Ethylene	1.22
Hydrochloric acid	1.40
Hydrogen	1.40
Methane	1.26
Methyl Chloride	1.20
Nitrogen	1.40
Oxygen	1.40
Sulfur dioxide	1.25

Figures 2-38A and 2-38B are based on the perfect gas laws and for sonic conditions at the outlet end of a pipe. For gases/vapors that deviate from these laws, such as steam, the same application will yield about 5% greater flow rate. For improved accuracy, use the charts in Figures 2-38A and 2-38B to determine the downstream pressure when sonic velocity occurs. Then use the fluid properties at this condition of pressure and temperature in:

$$v_s = \sqrt{kgRT},\ \text{ft/sec} = (kg\ (144)P'\overline{V})^{1/2} \qquad (2\text{-}85)$$

to determine the flow rate at this condition from:

$$v = q/A = 183.3\ q/d^2 = 0.0509\ W/(d^2)(\rho) \qquad (2\text{-}91)$$

d = internal diameter of pipe, in.
A = cross section of pipe, sq ft
q = cu ft/sec *at* flowing conditions
T = temperature, R
k = ratio of specific heats
P' = pressure, psi abs
W = flow, lbs/hr
v = velocity, mean or average, ft./sec

These conditions are similar to flow through orifices, nozzles, and venturi tubes. Flow through nozzles and venturi devices is limited by the critical pressure ratio, r_c = downstream pressure/upstream pressure at sonic conditions (see Figure 2-38C).

For nozzles and venturi meters, the flow is limited by critical pressure ratio and the minimum value of Y to be used.

For flow of gases and vapors through nozzles and orifices:

$$q = YC'A\sqrt{\frac{2g\ (144)\Delta P}{\rho}},\ \text{cu ft/sec flow} \qquad (2\text{-}48)$$

where β = ratio of orifice throat diameter to inlet diameter
C' = flow coefficient for nozzles and orifices (see Figures 2-17 and 2-18), when used as per ASME specification for differential pressure
ρ = fluid density, lb/cu ft
A = cross-sectional flow area, sq ft

Note : the use of C' eliminates the calculation of velocity of approach. The flow coefficient C' is $C' = C_d/\sqrt{1 - \beta^4} > \forall$
C_d = discharge coefficient for orifices or nozzles [3].

For compressible fluids flowing through nozzles and orifices use Figures 2-17 and 2-18, using h_L or ΔP as differential static head or pressure differential across taps located one diameter upstream at 0.5 diameters downstream from the inlet face of orifice plate or nozzle, when values of C are taken from Figures 2-17 and 2-18 [3]. For any fluid:

$$q = C'A\ ([2g\ (144)\ \Delta P]/\rho)^{1/2},\ \text{cu ft/sec flow} \qquad (2\text{-}48)$$

Note for liquids ΔP is upstream gauge pressure.

For estimating purposes for *liquid* flow with viscosity similar to water through orifices and nozzles, the following can be used [53]:

$$Q = 19.636\ C'd_1^{\ 2}\sqrt{h}\sqrt{\frac{1}{1 - \left(\frac{d_o}{d_i}\right)^4}}$$

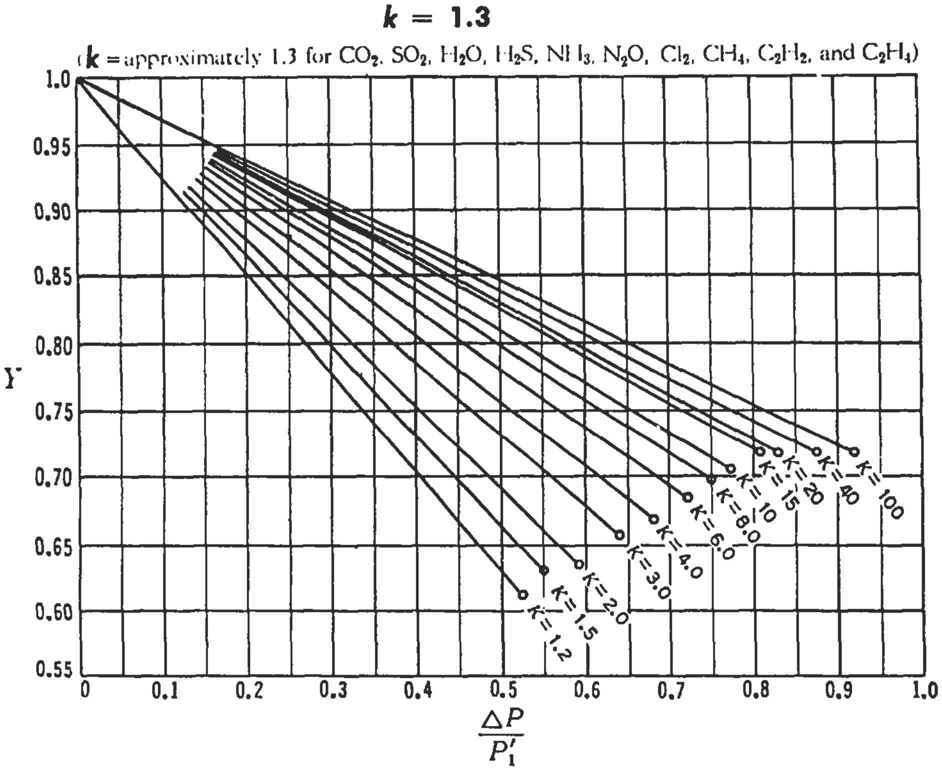
$$\text{where } \frac{d_o}{d_i} \text{ is greater than 0.3} \qquad (2\text{-}92)$$

$$Q = 19.636\ C'd_0^{\ 2}\sqrt{h} \text{ where } \frac{d_o}{d_i} \text{ is less than 0.3} \qquad (2\text{-}93)$$

$$\text{or [3], } W = 157.6\ d_0^{\ 2}C'\sqrt{h_L\rho^2}$$
$$= 1891\ d_0^{\ 2}C'\sqrt{\Delta P\rho} \qquad (2\text{-}94)$$

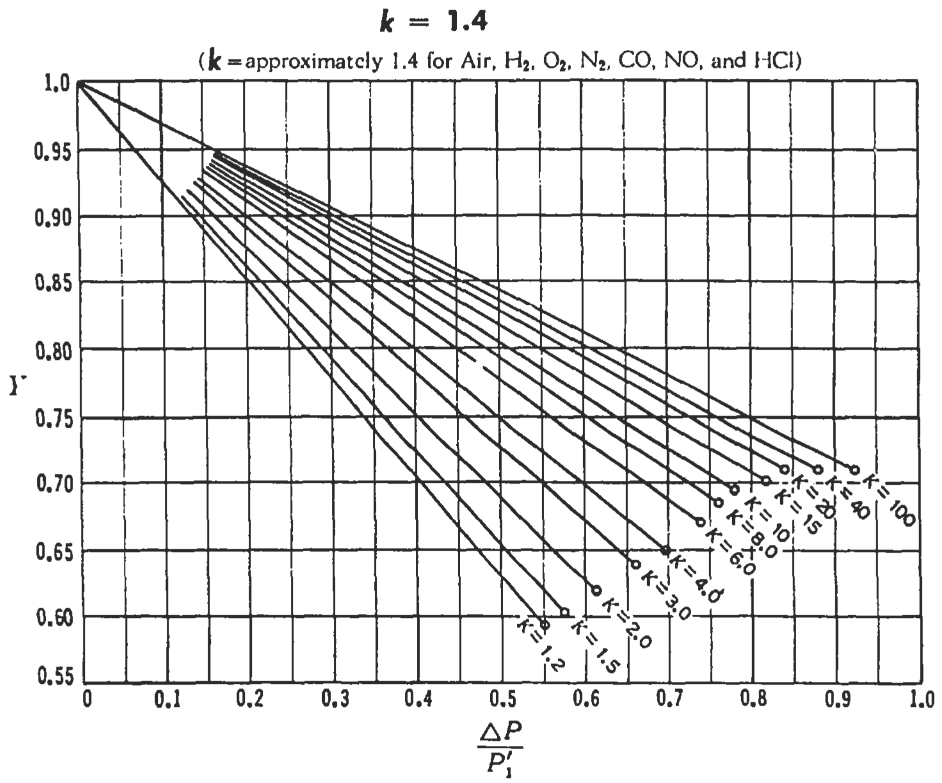
where Q = liquid flow, gpm
d_o = diameter of orifice or nozzle opening, in.
d_i = pipe inside diameter in which orifice or nozzle is installed, in.
h'_L = differential head at orifice, ft liquid
C' = flow coefficient (see Figure 2-39 for water and Figure 2-18 and 2-19 for vapors or liquids)

(text continued on page 118)



Limiting Factors For Sonic Velocity
k = 1.3

K	$\frac{\Delta P}{P'_1}$	Y
1.2	.525	.612
1.5	.550	.631
2.0	.593	.635
3	.642	.658
4	.678	.670
6	.722	.685
8	.750	.698
10	.773	.705
15	.807	.718
20	.831	.718
40	.877	.718
100	.920	.718



Limiting Factors For Sonic Velocity
k = 1.4

K	$\frac{\Delta P}{P'_1}$	Y
1.2	.552	.588
1.5	.576	.606
2.0	.612	.622
3	.662	.639
4	.697	.649
6	.737	.671
8	.762	.685
10	.784	.695
15	.818	.702
20	.839	.710
40	.883	.710
100	.926	.710

Figure 2-38A. Net expansion factor, Y, for compressible flow through pipe to a larger flow area. By permission, Crane Co., *Technical Paper #410*, Engineering Div., 1957. Also see 1976 edition.

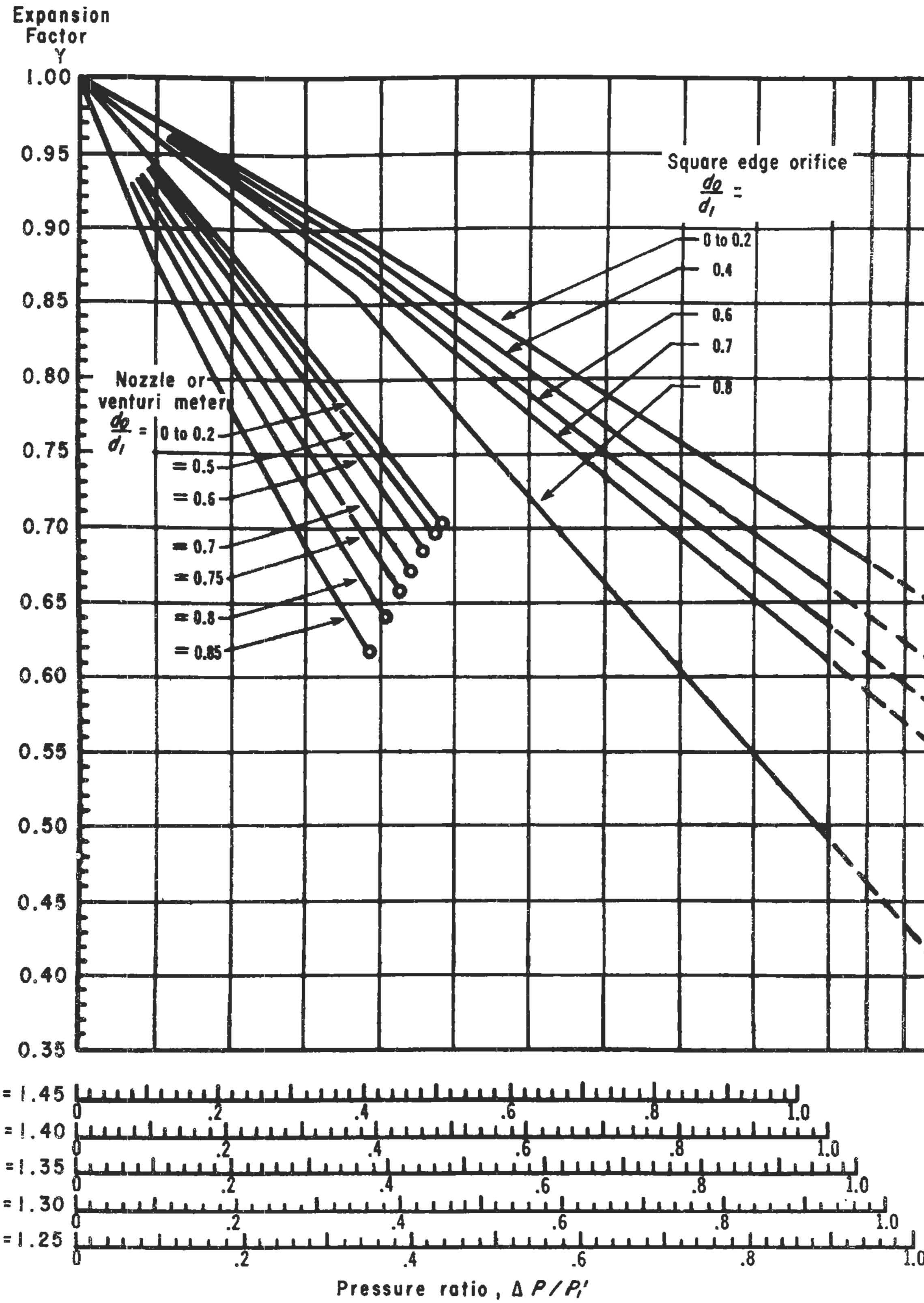
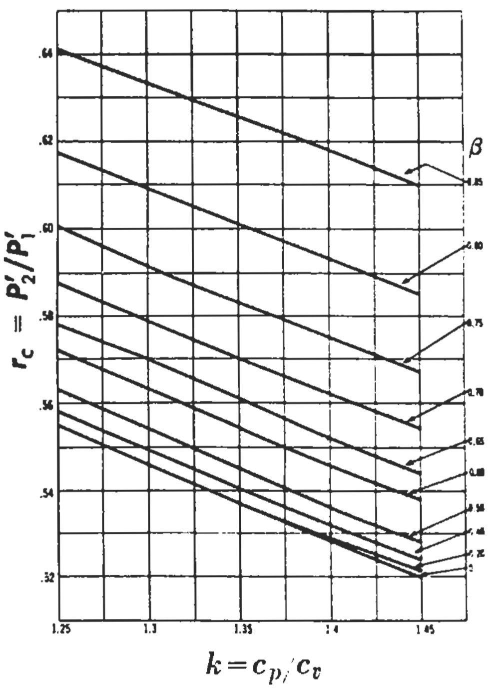


Figure 2-38B. Net expansion factor, Y, for compressible flow through nozzles and orifices. By permission, Crane Co., *Technical Paper #410*, Engineering Div., 1957. Also see 1976 edition and *Fluid Meters, Their Theory and Application, Part 1,* 5th Ed., 1959 and R. G. Cunningham, Paper #50-A-45, American Society of Mechanical Engineers.

(text continued from page 115)

q = cu ft/sec at flowing conditions (Figure 2-37) Coefficient from Reference [22] for liquids discharge

r_c = critical pressure ratio for compressible flow, = P'_2/P'_1



P′ = psia

B = ratio small-to-large diameter in orifices and nozzles, and contractions or enlargements in pipes

Figure 2-38C. Critical Pressure Ratio, r_c, for compressible flow through nozzles and venturi tubes. By permission, Crane Co., *Technical Paper #410,* 1957. Also see 1976 edition. See note at Figure 2-18 explaining details of data source for chart. Note: P′ = psia β= ratio of small-to-large diameter in orifices and nozzles, and contractions or enlargements in pipes.

Flow of gases and vapors (compressible fluids) through nozzles and orifices. (For flow field importance see References [31]). From [3]:

$$q = YC'A\sqrt{\frac{2g\,(144)\Delta P}{\rho}},\ \text{cu ft/sec}$$
(at flowing conditions) (2-48)

Y = net expansion factor from Figures 2-38A or 2-38B
ΔP = differential pressure (equal to inlet gauge pressure when discharging to atmosphere)
ρ = weight density of fluid, lbs/cu ft at flowing conditions
A = cross section of orifice or nozzle, sq ft
C′ = flow coefficient from Figures 2-38A or 2-38B

$$\text{or, } W = 1891\ Yd_1^{\,2}C'\sqrt{\frac{\Delta P}{\overline{V}_1}},\ \text{lbs/hr} \quad (2\text{-}95)$$

where d_o = internal diameter of orifice, in
$\overline{V}_1$ = specific volume of fluid, cu ft/lb

$$\text{or, } q' = 11.30\ Yd_0^{\,2}C'\sqrt{\frac{\Delta P\ P_1'}{T_1\ S_g}},\ \text{cu ft/sec}$$
at 14.7 psia and 60° F (2-96)

where S_g = Sp Gr gas relative to air, = mol wt. gas/29
T_1 = absolute temperature, °R
P'_1 = pressure, psi abs

Procedure

A. How to determine pipe size for given capacity and pressure drop.

1. Assume a pipe diameter, and calculate velocity in feet/second using the given flow.
2. Calculate sonic velocity for fluid using Equations 2-84 or 2-85.

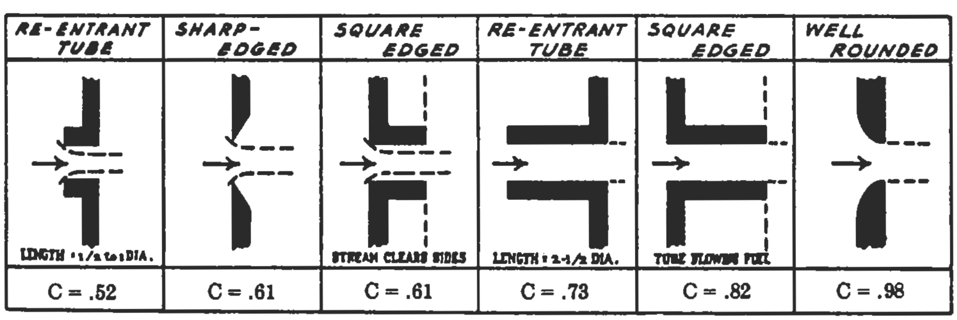


Figure 2-39. Discharge coefficients for liquid flow. By permission, *Cameron Hydraulic Data,* Ingersoll-Rand Co., Washington, N.J., 1979.

3. If sonic velocity of step 2 is greater than calculated velocity of step 1, calculate line pressure drop using usual flow equations. If these velocities are equal, then the pressure drop calculated will be the maximum for the line, using usual flow equations. If sonic velocity is less than the velocity of step 1, reassume line size and repeat calculations.

B. How to determine flow rate (capacity) for a given line size and fixed pressure drop.

This is also a trial and error solution following the pattern of (A), except capacities are assumed and the pressure drops are calculated to find a match for the given conditions of inlet pressure, calculating back from the outlet pressure.

C. How to determine pressure at inlet of pipe system for fixed pipe size and flow rate.

1. Determine sonic velocity at outlet conditions and check against a calculated velocity using flow rate. If sonic is the lower, it must be used as limiting, and capacity is limited to that corresponding to this velocity.
2. Using the lower velocity, and corresponding capacity, calculate pressure drop by the usual equations. For greater accuracy start at the outlet end of the line, divide it in sections using the physical properties of the system at these points, backing up to the inlet end of the line for the friction loss calculations. This procedure is recommended particularly for steam turbine and similar equipment exhausting to atmosphere or vacuum. The pressure at the inlet of the line is then the sum of the discharge or outlet line pressure and all the incremental section pressure losses. In the case of a turbine, this would set its outlet pressure, which would be higher than the pressure in the condenser or exhaust system.

Example 2-10: Gas Flow Through Sharp-edged Orifice

A 1″—Schedule 40 pipe is flowing methane at 40 psig and 50°F. The flange taps across the orifice (0.750 inch diameter) show a 3 psi pressure differential. Determine the flow rate through the orifice.

Solution:

CH_4; Sp Gr = Sg = 0.553

Gas constant = R = 96.4

Ratio Sp. ht. = k = 1.26

Absolute system pressure = P = 40 + 14.7 = 54.7 psia

$\Delta P/P_1 = 3.0/54.7 = 0.0549$

Pipe ID = 1.049 in.

$d_o/d_1 = 0.750/1.049 = 0.7149$

From Figure 2-38, Y = 0.97; from Figure 2-18.

$$C' \text{ (assumed turbulent)} = \frac{C_d}{[1 - (d_o/d_1)^4]^{1/2}} \tag{2-47}$$

where C_d = orifice discharge coefficient, uncorrected for velocity of approach

$C' = 0.74$ at est. $R_e \geqq 2000$

Temperature = 460 + 50 = 510°F

$$\text{Density} = \rho = \frac{144P}{RT} = \frac{144\ (54.7)}{(96.4)\ (510)}$$
$$= 0.1602 \text{ lb/cu ft}$$

$$W = 1891\, Y d_o^2 C\, (\Delta P \rho)^{1/2} \tag{2-95}$$
$$W = 1891\ (0.97)\ (0.750)^2\ 0.74\ [(3)\ (0.1602)]^{1/2}$$
$$W = 529.2 \text{ lbs/hr methane}$$

Check assumed R_e to verify turbulence; if not in reasonable agreement, recalculate C′ and balance of solution, checking:

Viscosity of methane = 0.0123 centipoise

$= 6.31\ W/d\mu$

$= 6.31\ (502)/(0.750)\ (0.0123)$

$R_e = 343{,}373$

This is turbulent and satisfactory for the assumption. For helpful quick reference for discharge of air through an orifice, see Table 2-12B.

Example 2-11: Sonic Velocity

Water vapor (4930 lbs/hr) is flowing in a 3-inch line at 730°F. The outlet pressure is less than one half the inlet absolute pressure. What is maximum flow that can be expected?

$c_p/c_v = 1.30$

MW vapor = 18.02

$v_s = [(1.30)\ (32.2)\ (1544/18.02)\ (730 + 460)]^{1/2}$

$= 2{,}065$ ft/sec

Cross section of 3-inch pipe = 0.0513 sq ft

Volume flow = (2,065) (0.0513) = 105.7 cu ft/sec

Vapor density = 4930/(3600) (105.7) = 0.01295 lb/cu ft

Pressure at end of line
= 0.01295 (379/18.02) (14.7) (1190/520)
= 9.16 psia (below atmos.)

Friction Drop for Compressible Natural Gas in Long Pipe Lines

Tests of the U.S. Department of the Interior, Bureau of Mines, reported in Monograph 6 *Flow of Natural Gas Through High-Pressure Transmission Lines* [43] indicate that the Weymouth formula gives good results on flow measurements on lines 6 inches in diameter and larger when operating under steady flow conditions of 30 to 600 psig.

Long gas transmission lines of several miles length are not considered the same as process lines inside plant connecting process equipment where the lengths usually are measured in feet or hundreds of feet. Some plants will transfer a manufactured gas, such as oxygen, carbon dioxide, or hydrogen, from one plant to an adjacent plant. Here the distance can be from one to fifteen miles. In such cases, the previously discussed flow relations for compressible gases can be applied in incremental segments, recalculating each segment, and then the results can be checked using one of the formulas that follow. However, there are many variables to evaluate and understand in the Weymouth, Panhandle, Panhandle-A and modifications as well as other flow relationships. Therefore, they will be presented for reference. However, the engineer should seek out the specialized flow discussions on this type of flow condition. The above mentioned equations are derived somewhat empirically for the flow of a natural gas containing some entrained liquid (perhaps 5% to 12%), and the results vary accordingly, even though they are not two-phase flow equations.

Table 2-15 [15] tabulates the transmission factors of the various equations. Most of these are established as correction factors to the correlation of various test data.

Dunning [40] recommends this formula (from Reference [43]) for 4 to 24-inch diameter lines with specific gravity of gas near 0.60, and actual mean velocities from 15 to 30 feet per second at temperature near 60°F.

The Bureau of Mines report states that minor corrections for bends, tees, and even compressibility are unnecessary due to the greater uncertainties in actual line conditions. Their checks with the Weymouth relation omitted these corrections. The relation with pressure base of 14.4 psia is to be used with the Bureau of Mines multipliers [43].

Table 2-15
Dry-Gas Flow Transmission Factors

Title	Transmission Factor $(\sqrt{1/f})$	Ref.*
Weymouth	$11.2D^{0.167}$	
Blasius	$3.56Re^{0.125}$	
Panhandle A	$6.87Re^{0.073}$	
Modified Panhandle	$16.5Re^{0.0196}$	
Smooth pipe law (Nikuradse)	$4 \log (Re\sqrt{f}) - 0.4$	
Rough pipe law (Nikuradse)	$4 \log \frac{(D)}{(2\varepsilon)} + 3.48$	
Colebrook	$4 \log \frac{D}{2\varepsilon} + 3.48 - 4 \log \left[1 + 9.35 \frac{D}{2Re\sqrt{f}}\right]$	

Note: D = inches
*See listing of source references in Reference [15]. By permission, Hope, P. M. and Nelson, R. G., "Fluid Flow, Natural Gas," McKetta, J. J. Ed., Encyclopedia of Chemical Processing and Design, vol. 22, 1985, M. Dekker, p. 304 [15].

$$q_h \text{ (at 14.4 psia \& 60°F)} = 36.926\, d^{2.667} \left[\frac{P_1^2 - P_2^2}{L_m}\right]^{1/2} \text{ scfh} \quad (2\text{-}97)$$

q'_h (at 14.4 psia and 60°F)

$$= 28.0\, d^{2.667} \left[\frac{P_1^2 - P_2^2}{S_g L_m}\left(\frac{520}{T}\right)\right]^{1/2}, \text{ scfh (Ref. 8)} \quad (2\text{-}98)$$

Weymouth's formula [57] has friction established as a function of diameter and may be solved by using alignment charts.

The Weymouth formula is also expressed (at standard conditions) as:

$$q_d = 433.49E\,(T_s/P_s)\,[P'^2_1 - P'^2_2]/S_g T_1 L_m Z]^{1/2} d^{2.667} \quad (2\text{-}99)$$

E = transmission factor, usually taken as: $1.10 \times 11.2\, d^{0.167}$ (omit for pipe sizes smaller than 24 in.)
d = pipe I.D., in.
T_s = 520°R
P_s = 14.7 psia
T_1 = flowing temperature of gas, °R
q_d = cu ft/day gas at std conditions of P_s and T_s
P'_1 = inlet pressure, psia
P'_2 = outlet pressure, psia

Z = compressibility factor
L_m = pipe length, miles

or from Reference [3]:

$$q_h = 28.0\, d^{2.667} \{[((P'_1)^2 - (P'_2)^2)/S_g L_m][520/T]\}^{1/2} \quad (2\text{-}100)$$

Example 2-12: Use of Base Correction Multipliers

Tables 2-16, 2-17, 2-18, and 2-19 are set up with base reference conditions. In order to correct or change any base condition, the appropriate multiplier(s) must be used.

A flow of 5.6 million cu ft/day has been calculated using Weymouth's formula [57], with these conditions: measuring base of 60°F and 14.4 psia; flowing temperature of 60°F, and specific gravity of 0.60. Suppose for comparison purposes the base conditions must be changed to: measuring base of 70°F and 14.7 psia; flowing temperature of 80°F, and specific gravity of 0.74.

Multipliers from the tables are:

Pressure base:	0.9796
Temperature base:	1.0192
Specific gravity base:	0.9005
Flowing temperature base:	0.9813

New base flow
= (5,600,000) (0.9796) (1.0192) (0.9005) (0.9813)
= 4,940,000 cu ft/day

Panhandle-A Gas Flow Formula [3]

This formula is considered to be slightly better than the ±10 percent accuracy of the Weymouth formula.

$$q_{ds} = 435.87\, E(T_s / P_s)^{1.07881} \left[\frac{P_1^2 - P_2^2}{S_g^{0.8539} T L_m Z}\right]^{0.5394} d^{2.6182} \quad (2\text{-}101)$$

or E = 0.92, usually 0.8539

$$P_1^2 - P_2^2 = \left[\frac{S_g^{0.4606}}{435.87\,(T_s/P_s)^{1.07881}\, d^{2.6182}}\right]^{1.8539} T L_m\, q^{1.8539} \quad (2\text{-}102)$$

where T = gas flowing temperature, °R = 460°F + t
E = efficiency factor for flow, use 1.00 for new pipe without bends, elbows, valves and change of pipe diameter or elevation
0.95 for very good operating conditions
0.92 for average operating conditions
0.85 for poor operating conditions

For bends in pipe add to length [38]:

Bend Radius	Add*, as pipe diameters, d_e
1 Pipe dia.	17.5
1.5 Pipe dia.	10.4
2 Pipe dia.	9.0
3 Pipe dia.	8.2

*These must be converted to the unit of length used in the formula.

If a line is made up of several different sizes, these may be resolved to one, and then the equation solved once for this total equivalent length. If these are handled on a per size basis, and totaled on the basis of the longest length of one size of line, then the equivalent length, L_e, for any size d, referenced to a basic diameter, d_e.

$$L_e = L_m (d_e/d)^{4.854} \quad (2\text{-}103)$$

where L_m is the length of pipe of size d to be used.
L_e is the equivalent length of pipe size d, length L_m after conversion to basis of reference diameter, d_e.

The calculations can be based on diameter d_e and a length of all the various L_e values in the line plus the length of line of size d_e, giving a total equivalent length for the line system.

Modified Panhandle Flow Formula [15]

$$q_{DS} = 737.2\, E\,(T_o/P_o)^{1.02} [\{P_1^2 (1 + 0.67\, ZP_1) - P_2^2 (1 + 0.67\, ZP_2)\}/T\, L_m\, G^{0.961}]^{0.51} (d)^{2.53} \quad (2\text{-}104)$$

where L_m = miles length
d = inside diameter, in.
T = flowing temperature, R
Z = gas deviation, compressibility factor
T_o = base temperature, (520 R)
G = gas specific gravity
Z = compressibility correction term
P = pressure, psi, absolute
P_o = base pressure, (14.73 psi, absolute)
E = "efficiency factor," which is really an adjustment to fit the data
f = fanning friction factor
q_{DS} = flow rate, SCF/day

American Gas Association (AGA) Dry Gas Method

See Reference [16] AGA, Dry Gas Manual. Some tests indicate that this method is one of the most reliable above a fixed Reynolds number.

Complex Pipe Systems Handling Natural (or similar) Gas

The method suggested in the Bureau of Mines Monograph No. 6 [43] has found wide usage, and is outlined here using the Weymouth Formula as a base.

1. Equivalent lengths of pipe for different diameters

$$L_1 = L_2 (d_1/d_2)^{16/3} \tag{2-105}$$

where L_1 = the equivalent length of any pipe of length L_2 and diameter, d_2, in terms of diameter, d_1.

$$d_1 = d_2 (L_1/L_2)^{3/16} \tag{2-106}$$

where d_1 = the equivalent diameter of any pipe of a given diameter, d_2, and length, L_2, in terms of any other length, L_1.

2. Equivalent diameters of pipe for parallel lines

$$d_o = (d_1^{8/3} + d_2^{8/3} \ldots + d_n^{8/3})^{3/8} \tag{2-107}$$

where d_o is the diameter of a single line with the same delivery capacity as that of the individual parallel lines of diameters d_1, d_2 . . . and d_n. Lines of same length.

This value of d_o may be used directly in the Weymouth formula.

Example 2-13: Series System

Determine the equivalent length of a series of lines: 5 miles of 14-in. (13.25-in. I.D.) connected to 3 miles of 10-in. (10.136-in. I.D.) connected to 12 miles of 8-in (7.981-in. I.D.).

Select 10-in. as the base reference size.

The five-mile section of 14-in. pipe is equivalent to:

$$L_1 = 5(10.136/13.25)^{5.33} = 1.195 \text{ miles of 10-in.}$$

The 12 mile section of 8-in. is equivalent to:

$$L_1 = 12(10.136/7.981)^{5.33} = 42.8 \text{ miles of 10-in.}$$

Total equivalent length of line to use in calculations is:

$$1.195 + 3.0 + 42.8 = 46.995 \text{ miles of 10-in. (10.136-in. I.D.).}$$

An alternate procedure is to calculate (1) the pressure drop series-wise one section of the line at a time, or (2) capacity for a fixed inlet pressure, series-wise.

Example 2-14: Looped System

Determine the equivalent length of 25 miles of 10-in. (10.136-in. I.D.) which has a parallel loop of 6 miles of 8-in. (7.981-in. I.D.) pipe tied in near the midsection of the 10-in. line.

Figure the looped section as parallel lines with 6 miles of 8-in. and 6 miles of 10-in. The equivalent diameter for one line with the same carrying capacity is:

$$d_o = [(7.981)^{8/3} + (10.136)^{8/3}]^{3/8} = 11.9\text{-in.}$$

This simplifies the system to one section 6 miles long of 11.9-in. I.D. (equivalent) pipe, plus one section of 25 minus 6, or 19 miles of 10-in. (10.136-in. I.D.) pipe.

Now convert the 11.9-in. pipe to a length equivalent to the 10-in. diameter.

$$L_1 = 6(10.136/11.9)^{5.33} = 2.58 \text{ miles}$$

Total length of 10-in. pipe to use in calculating capacity is 19 + 2.58 = 21.58 miles.

By the principles outlined in the examples, gas pipe line systems may be analyzed, paralleled, cross-tied, etc.

Example 2-15: Parallel System: Fraction Paralleled

Determine the portion of a 30-mile, 18-in. (17.124-in. I.D.) line which must be paralleled with 20-in. (19.00-in. I.D.) pipe to raise the total system capacity 1.5 times the existing rate, keeping the system inlet and outlet conditions the same.

$$x = \frac{(q_{da}/q_{db})^2 - 1}{\dfrac{1}{[1 + (d_b/d_a)^{2.667}]^2} - 1} \tag{2-108}$$

For this example, $q_{db} = 1.5\, q_{da}$

$$x = \frac{(1/1.5)^2 - 1}{\left[\dfrac{1}{[1 + (19.00/17.124)^{2.667}]^2} - 1\right]} = 0.683$$

This means 68.3 percent of the 30 miles must be parallel with the new 19-in. I.D. pipe.

Table 2-16
Pressure-Base Multipliers For Quantity*

$$\text{Multiplier} = \frac{14.4}{\text{New pressure base, Lbs./sq. in. abs.}}$$

New Pressure Base, Lbs./sq.in. abs.	Multiplier
12.00	1.2000
13.00	1.1077
14.00	1.0286
14.40	1.0000
14.65	0.9829
14.7	0.9796
14.9	0.9664
15.4	0.9351
16.4	0.8780

*By permission, Johnson, T. W. and Berwald, W. B., *Flow of Natural Gas Through High Pressure Transmission Lines, Monograph No. 6,* U.S. Dept. of Interior, Bureau of Mines, Washington, DC.

Table 2-17
Temperature-Base Multipliers For Quantity*

$$\text{Multiplier} = \frac{460 + \text{new temperature base, }^\circ\text{F.}}{460 + 60}$$

New Temperature Base, ° F.	Multiplier
45	0.9712
50	0.9808
55	0.9904
60	1.0000
65	1.0096
70	1.0192
75	1.0288
80	1.0385
85	1.0481
90	1.0577
95	1.0673
100	1.0769

*By permission, Johnson, T. W. and Berwald, W. B., *Flow of Natural Gas Through High Pressure Transmission Lines, Monograph No. 6,* U.S. Dept. of Interior, Bureau of Mines, Washington, D.C.

Table 2-18
Specific Gravity Multipliers For Quantity*

$$\text{Multiplier} = \left[\frac{0.600}{\text{actual Specific Gravity}}\right]^{1/2}$$

Specific Gravity	0	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08	0.09
0.5	1.0954	1.0847	1.0742	1.0640	1.0541	1.0445	1.0351	1.0260	1.0171	1.0084
0.6	1.0000	0.9918	0.9837	0.9759	0.9682	0.9608	0.9535	0.9463	0.9393	0.9325
0.7	0.9258	0.9193	0.9129	0.9066	0.9005	0.8944	0.8885	0.8827	0.8771	0.8715
0.8	0.8660	0.8607	0.8554	0.8502	0.8452	0.8402	0.8353	0.8305	0.8257	0.8211
0.9	0.8165	0.8120	0.8076	0.8032	0.7989	0.7947	0.7906	0.7865	0.7825	0.7785
1.0	0.7746	0.7708	0.7670	0.7632	0.7596	0.7559	0.7524	0.7488	0.7454	0.7419
1.1	0.7385	0.7352	0.7319	0.7287	0.7255	0.7223	0.7192	0.7161	0.7131	0.7101

*By permission, Johnson, T. W. and Berwald, W. B., *Flow of Natural Gas Through High Pressure Transmission Lines, Monograph No. 6,* U.S. Dept. of Interior, Bureau of Mines, Washington, DC.

Table 2-19
Flowing-Temperature Multipliers For Quantity*

$$\text{Multiplier} = \left[\frac{460 + 60}{460 + \text{actual flowing temperature}}\right]^{1/2}$$

Temp. °F.	0	1	2	3	4	5	6	7	8	9
..	1.0632	1.0621	1.0609	1.0598	1.0586	1.0575	1.0564	1.0552	1.0541	1.0530
10	1.0518	1.0507	1.0496	1.0485	1.0474	1.0463	1.0452	1.0441	1.0430	1.0419
20	1.0408	1.0398	1.0387	1.0376	1.0365	1.0355	1.0344	1.0333	1.0323	1.0312
30	1.0302	1.0291	1.0281	1.0270	1.0260	1.0249	1.0239	1.0229	1.0219	1.0208
40	1.0198	1.0188	1.0178	1.0167	1.0157	1.0147	1.0137	1.0127	1.0117	1.0107
50	1.0098	1.0088	1.0078	1.0068	1.0058	1.0048	1.0039	1.0029	1.0019	1.0010
60	1.0000	0.9990	0.9981	0.9971	0.9962	0.9952	0.9943	0.9933	0.9924	0.9915
70	0.9905	0.9896	0.9887	0.9877	0.9868	0.9859	0.9850	0.9841	0.9831	0.9822
80	0.9813	0.9804	0.9795	0.9786	0.9777	0.9768	0.9759	0.9750	0.9741	0.9732
90	0.9723	0.9715	0.9706	0.9697	0.9688	0.9680	0.9671	0.9662	0.9653	0.9645

*By permission, Johnson, T. W. and Berwald, W. B., *Flow of Natural Gas Through High Pressure Transmission Lines, Monograph No. 6,* U.S. Dept. of Interior, Bureau of Mines, Washington, DC.

Parallel System: New Capacity after Paralleling

Solve this relation, rearranged conveniently to [43]:

$$q_{db} = \frac{q_{da}}{\left\{x\left[\frac{1}{[1+(d_b/d_a)^{2.667}]^2} - 1\right] + 1\right\}^{1/2}} \quad (2\text{-}109)$$

Two-phase Liquid and Gas Flow

The concurrent flow of liquid and gas in pipe lines has received considerable study [33], [35], [37], [41]. However, pressure drop prediction is not extremely reliable except for several gas pipe line conditions. The general determinations of pressure drop for plant process lines can only be approximated.

The latest two-phase flow research and design studies have broadened the interpretation of some of the earlier flow patterns and refined some design accuracy for selected situations. The method presented here serves as a fundamental reference source for further studies. It is suggested that the designer compare several design concept results and interpret which best encompasses the design problem under consideration. Some of the latest references are included in the Reference Section. No one reference has a solution to all two-phase flow problems.

If two-phase flow situations are not recognized, pressure drop problems may develop which can prevent systems from operating. It requires very little percentage of vapor, *generally* above 7% to 8%, to establish volumes and flow velocities that must be solved by two-phase flow analysis. The discharge flow through a pressure relief valve on a process reactor is often an important example where two-phase flow exists, and must be recognized for its back pressure impact.

Flow Patterns

Six or seven types of flow patterns (Figure 2-40) are usually considered in evaluating two-phase flow. Only one type can exist in a line at a time, but as conditions change (velocity, roughness, elevation, etc.) the type may also change. The unit pressure drop varies significantly between the types. Figure 2-40 illustrates the typical flow regimes recognized in two-phase flow.

Figure 2-41 [17] typically represents a graphical illustration of the various flow patterns of Figure 2-40 as the two-phase mixture flows through the piping. Long gas transport lines may have hydrocarbon or other liquids form (condense) as the fluid flows, and this becomes a real problem for offshore or long buried onshore raw gas transmission (see section dealing with calculation methods).

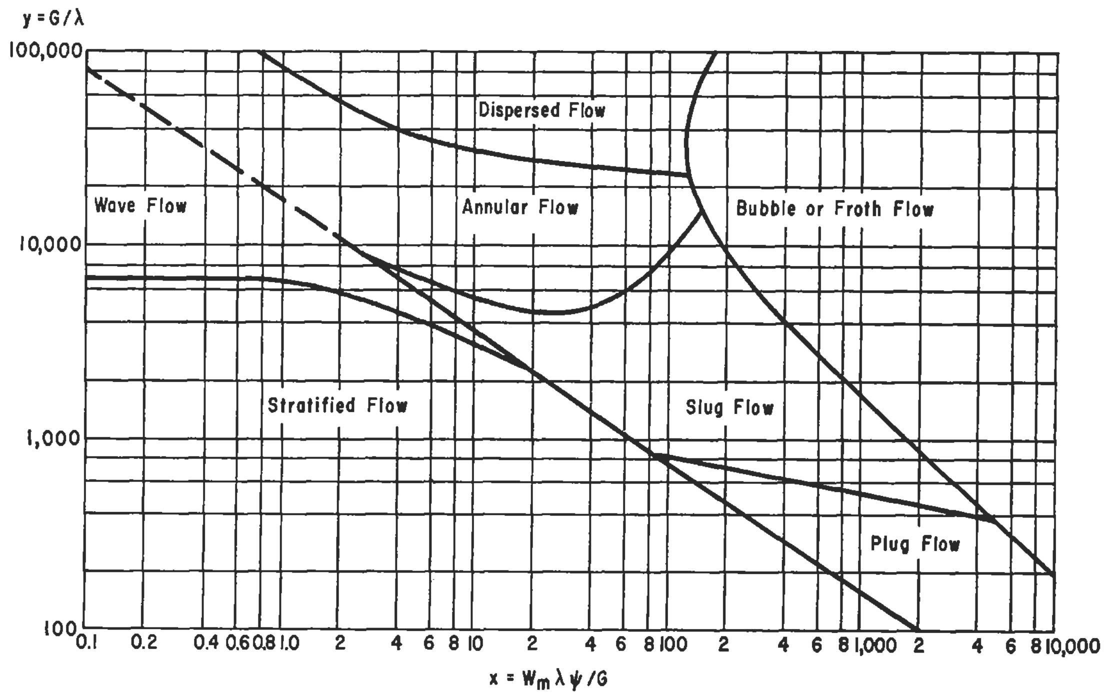


Figure 2-40. Flow patterns for horizontal two-phase flow. (Based on data from 1-in., 2-in., and 4-in., pipe). By permission, O. Baker, *Oil and Gas Journal,* Nov. 10, 1958, p. 156.

Type of Flow For Horizontal Pipes

Bubble or Froth:	Bubbles dispersed in liquid
Stratified:	Liquid and gas flow in stratified layers
Wave:	Gas flows in top of pipe section, liquid in waves in lower section
Slug:	Slugs of gas bubbles flowing through the liquid
Annular:	Liquid flows in continuous annular ring on pipe wall, gas flows through center of pipe
Plug:	Plugs of liquid flow followed by plugs of gas
Dispersed:	Gas and liquid dispersed

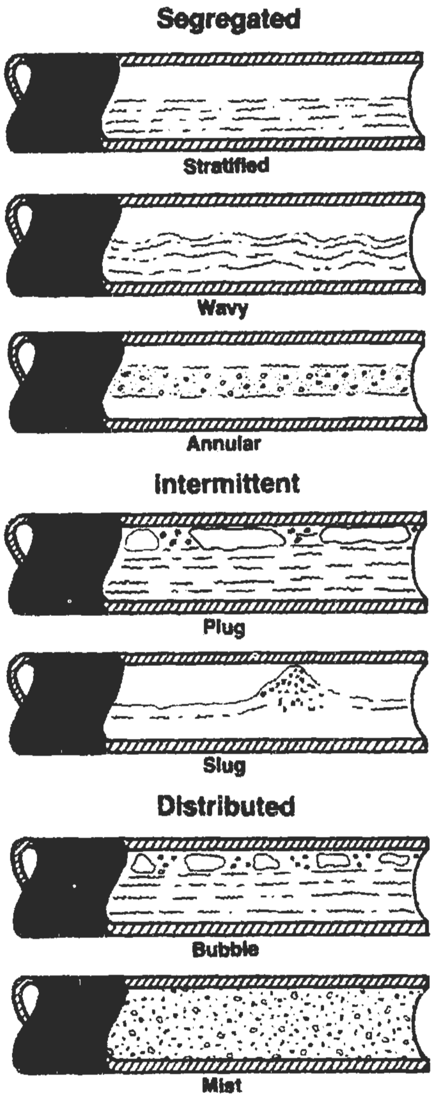


Figure 2-41. Representative forms of horizontal two-phase flow patterns; same as indicated in Figure 2-40. By permission, Heim, H., *Oil and Gas Journal,* Aug. 2, 1982, p. 132.

Total System Pressure Drop

The pressure drop for a system of horizontal and vertical (or inclined) pipe is the sum of the horizontal pressure drop plus the additional drop attributed to each vertical rise, regardless of initial and final elevations of the line [33].

$$\Delta P_{TPh} = \Delta P_{PT} \text{ (horizontal pipe)} + nhF_e\, \rho_L/144 \qquad (2\text{-}110)$$

A. To determine most probable type of two-phase flow using Figure 2-40.

1. Calculate $W_m\, \lambda\psi/G$
2. Calculate G/λ
3. Read intersection of ordinate and abscissa to identify probable type of flow. Since this is not an exact, clear-cut position, it is recommended that the adjacent flow types be recorded also. Note: See Example 2-16 for definitions of λ and ψ.

B. Calculate the separate liquid and gas flow pressure drops.

1. For general process application both ΔP_L and ΔP_g may be calculated by the general flow equation:
   ΔP_L or ΔP_g (using proper values respectively)

$$= \frac{3.36 fLW^2 (10^{-6})}{d^5 \rho} \qquad (2\text{-}111)$$

where f is obtained from Reynolds-Friction Factor chart (Figure 2-3) for an assumed line size, d.

2. For gas transmission, in general form [33]

$$\Delta P_g = \frac{(q_{d\,14.65})\, LS_g\, TZf}{20{,}000\, d^5 P_{avg}} \qquad (2\text{-}112)$$

where $q_{d\,14.65}$ is the thousands of standard cubic feet of gas per day, measured at 60°F and 14.65 psia, and P_{avg} is the average absolute pressure in the pipe system between inlet and outlet. This is an estimated value and may require correction and recalculation of the final pressure drop if it is very far off.

For oil flow in natural gas transmission lines [33]

$$\Delta P_L = \frac{f L Q^2{}_b \rho}{181{,}916\, d^5} \qquad (2\text{-}113)$$

3. Calculate

$$X = (\Delta P_L/\Delta P_g)^{1/2} \quad (2\text{-}114)$$

4. Calculate θ for types of flow selected from Figure 2-40 [33].

Type Flow	Equation for Φ_{GTT}
Froth or Bubble	$\Phi = 14.2\ X^{0.75}/W_m^{0.1}$
Plug	$\Phi = 27.315\ X^{0.855}/W_m^{0.17}$
Stratified	$\Phi = 15{,}400\ X/W_m^{0.8}$
Slug	$\Phi = 1{,}190\ X^{0.185}/W_m^{0.5}$
Annular*	$\Phi = (4.8 - 0.3125d)\ X^{0.348 - 0.021d}$

*Set d = 10 for any pipe larger than 10-in.
$X = [\Delta P_{Liq.}/\Delta P_{gas}]^{1/2}$

5. Calculate two-phase pressure drop, horizontal portions of lines. For all types of flow, except wave and fog or spray:

$$\Delta P_{TP} = \Delta P_G \phi^2_{GTT}, \text{ psi per foot} \quad (2\text{-}115)$$

For wave [52].

$$\Delta P_{TP} = f_{TP}\ (G'_g)^2/193.2\ _d\rho_g, \text{ psi/foot} \quad (2\text{-}116)$$

where

$$f_{TP} = 0.0043\ (W_m \mu_L/G\mu_g)^{0.214} \quad (2\text{-}117)$$

6. Total two-phase pressure drop, including horizontal and vertical sections of line. Use calculated value times 1.1 to 2.0, depending upon critical nature of application.

$$\Delta P_{TPh} = \Delta P_{TP}\ L + n\ h\ F_e \rho_L/144 \quad (2\text{-}118)$$

where ρ_L is the density, lb/cu ft, of the liquid flowing in the line, and F_e, elevation factor using gas velocity, v.

$$F_e = 0.00967\ W_m^{0.5}/v^{0.7}, \text{ for } v > 10 \quad (2\text{-}119)$$

$$\text{or as an alternate: } F_e = 1.7156\ V_g^{-0.702} \quad (2\text{-}120)$$

Use Figure 2-42 for v less than 10. Most gas transmission lines flow at from 1–15 ft/sec.

For fog or spray type flow, Baker [33] suggests using Martinelli's correlation and multiplying results by two [46].

(a) For gas pipe line flow, the values of ϕ_{GTT} may be converted to "efficiency E" values and used to calculate the flow for the horizontal portion using a fixed allowable pressure drop in the general flow equation [33]. The effect of the vertical component must be added to establish the total pressure drop for the pumping system.

$$q_{d\,14.65} = \left[\frac{38.7744\ T_s}{1000\ P_s}\ \frac{(P_1^2 - P_2^2)\ d^5}{L_m\ S_g\ TZ}\right]^{0.5} \left(\frac{E}{f_g}\right)^{0.5} \quad (2\text{-}121)$$

where 14.65 refers to reference pressure P_s.

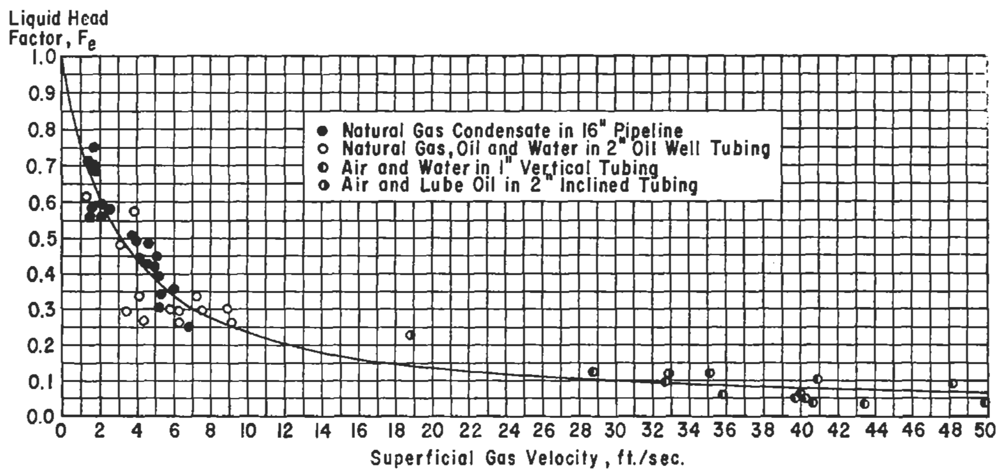


Figure 2-42. Estimating pressure drop in uphill sections of pipeline for two-phase flow. By permission, O. Flanigan, *Oil and Gas Journal*, Mar. 10, 1958, p. 132.

or

$$\Delta P_{TP} = \frac{(q_{d14.65})^2 LS_g TZ}{20,000\, d^5 P_{avg}}\left(\frac{f_g}{E^2}\right) \quad (2\text{-}122)$$

where $E = 1/\phi_{GTT}$

(b) For the Panhandle equation, Baker [33] summarizes:

$$q_{d14.65} = 0.43587\left(\frac{T_s}{P_s}\right)^{1.07881}\left(\frac{P_1^2 - P_2^2}{ZTL_m}\right)^{0.5394}\left(\frac{d^{2.618}}{S_g^{0.4606}}\right)(E) \quad (2\text{-}123)$$

where E (Panhandle) $= 0.9/\phi_{GTT}^{1.077}$

Example 2-16: Two-phase Flow

A liquid-vapor mixture is to flow in a line having 358 feet of level pipe and three vertical rises of 10 feet each plus one vertical rise of 50 feet. Evaluate the type of flow and expected pressure drop.

Vapor = 3,000 lbs/hr
Liquid = 1,000 lbs/hr
Density: lbs/cu ft; Vapor = 0.077
Liquid = 63.0
Viscosity, centipoise; Vapor = 0.00127
Liquid = 1.0
Surface tension liquid = 15 dynes/cm
Pipe to be schedule 40, steel
Use maximum allowable vapor velocity = 15,000 ft/min.

1. Determine probable types of flow:

$$\lambda = [(\rho_g/0.075)(\rho_L/62.3)]^{0.5} = \left[\left(\frac{0.077}{0.075}\right)\left(\frac{63.0}{62.3}\right)\right]^{0.5}$$

$\lambda = 1.017$

$$\psi = (73/\gamma)[\mu_L(62.3/\rho_L)^2]^{1/3} = \left(\frac{73}{15}\right)\left[1.0\left(\frac{62.3}{63.0}\right)^2\right]^{1/3}$$

$\psi = 4.86$

Try 3-in. pipe, 3.068-in. I.D., cross-section area = 0.0513 sq. ft.

$W_m = 1{,}000/0.0513 = 19{,}494$ lbs/hr (sq ft)

$G = 3{,}000/0.0513 = 58{,}482$ lbs/hr (sq ft)

$W_m\lambda\psi/G = 19{,}494\ (1.017)\ (4.86)/58{,}482 = 1.641$

$G/\lambda = 58{,}482/1.017 = 57{,}500$

Reading Figure 2-40 type flow pattern is probably annular, but could be wave or dispersed, depending on many undefined and unknown conditions.

2. Liquid Pressure drop

$$\Delta P_L = 3.36\, fLW^2\, (10^{-6})/d^5\rho \quad (2\text{-}124)$$

Determine R_e for 3-in. pipe:
From Figure 2-11; $\varepsilon/d = 0.0006$ for steel pipe

$$v = \frac{1000}{63\ (3600)\ (0.0513)} = 0.086 \text{ ft/sec}$$

$\mu_e = 1$ cp/1488 = 0.000672 lbs/ft sec
$D = 3.068/12 = 0.2557$ ft
$\rho = 63.0$
$R_e = D\,v\rho/\mu_e = 0.2557\ (0.086)\ (63.0)/0.000672$
$R_e = 2060$ (this is borderline, and in critical region)

Reading Figure 2-3, approximate f = 0.0576
Substituting:

$\Delta P_L = 3.36\ (10^{-6})\ (0.0576)\ (1000)^2\ (1 \text{ foot})/(3.068)^5\ (63)$
$= 1.1\ (10^{-5})$ psi/foot

Gas pressure drop

$$v = \frac{3000}{0.077\ (3600)\ (0.0513)} = 211 \text{ ft/sec}$$

$\mu_e = 0.00127/1488 = 0.000000854$ lbs/ft sec

$R_e = Dv\,\rho/\mu_e = 0.2557\ (211)\ (0.077)/0.000000854$
$= 4{,}900{,}000$

Reading Figure 2-3, f = 0.0175

$\Delta P_G = 3.36\ (10^{-6})(0.0175)(1 \text{ foot})(3000)^2/(3.068)^5(0.077)$
$= 0.0254$ psi/foot

3. $X = (\Delta P_L/\Delta P_g)^{1/2} = (1.1\ (10^{-5})/2.54 \times 10^{-2})^{1/2}$
$= 2.10\ (10^{-2})$

4. For annular flow:

$\Phi_{GTT} = (4.8 - 0.3125d)\ X^{0.343 - 0.021d}$
$= [4.8 - 0.3125\ (3.068)]\,(2.10 \times 10^{-2})^{0.343 - 0.021\,(3.068)}$
$= 1.31$

5. Two-phase flow for horizontal flows:

$$\Delta P_{TP} = \Delta P_G \Phi^2_{GTT} = (0.0254)\ (1.31)^2 = 0.0438 \text{ psi/ft}$$

6. $F_e = 0.00967\ (W_m)^{0.5}/v^{0.7}$
$= 0.00967\ (19{,}494)^{0.5}/(211)^{0.7}$
$= 0.032$

Vertical elevation pressure drop component:

$= n\ h\ F_e \rho_L/144 = [(3)\ (10) + (1)\ (50)](0.032)(63)/144$
$= 1.125$ psi total

Total:

$\Delta P_{TPh} = (0.0438)\ (358) + 1.125$
$= 16.7$ psi, total for pipe line

Because these calculations are somewhat uncertain due to lack of exact correlations, it is best to calculate pressure drop for other flow patterns, and apply a generous safety factor to the results.

Table 2-20 gives calculated results for other flow patterns in several different sizes of lines.

Table 2-20
Two-Phase Flow Example

Pipe I.D. Inches	Horizontal Flow Pattern Annular Psi/Ft.	Stratified Psi/Ft.	Wave Psi/Ft.	Elevation Factor, F_e	Ft./sec., Gas Vel.
3.068	0.0438	0.000367	0.131	0.032	210.9
4.026	0.0110	0.000243	0.0336	0.0465	122.5
6.065	0.00128	0.000131	0.00434	0.0826	53.9
7.981	0.00027	0.000087	0.00110	0.121	31.1
10.020	0.000062	0.000062	0.00035	0.166	19.7

Pressure Drop in Vacuum Systems

Vacuum in process systems refers to an *absolute* pressure that is less than or below the local barometric pressure at the location. It is a measure of the degree of removal of atmospheric pressure to some level between atmospheric barometer and absolute vacuum (which cannot be attained in an absolute value in the real world), but is used for a reference of measurement. In most situations, a vacuum is created by pumping air out of the container (pipe, vessels) and thereby lowering the pressure. See Figure 2-1 to distinguish between vacuum gauge and vacuum absolute.

This method [54] is for applications involving air or steam in cylindrical piping under conditions of (a) turbulent flow, (b) sub-atmosphere pressure, (c) pressure drop is limited to 10% of the final pressure (see comment to follow), and (d) the lower limit for application of the method is

$$W/d\ 7\ \ 20 \qquad (2\text{-}125)$$

where W is the flow rate in lbs/hr and d is the inside pipe diameter in inches. If the above ratio is less than 20, the flow is "streamlined" and the data does not apply.

If the pressure drop is greater than 10% of the final pressure, the pipe length can be divided into sections and the calculations made for each section, maintaining the same criteria of (c) and (d) above.

Method [54]

The method solves the equation (see Figure 2-43)

$$\Delta P_{vac} = \frac{(F_1 C_{D1} C_{T1}) + (F_2 C_{D2} C_{T2})}{p_1} \qquad (2\text{-}126)$$

where ΔP_{vac} = pressure drop, in. water/100 ft of pipe
p_1 = initial pressure, inches mercury absolute
F_1 = base friction factor, Figure 2-43
F_2 = base friction factor, Figure 2-43
C_{T1} = temperature correction factor, Figure 2-43
C_{T2} = temperature correction factor, Figure 2-43
C_{D1} = diameter correction factor, Figure 2-43
C_{D2} = diameter correction factor, Figure 2-43

Example 2-17: Line Sizing for Vacuum Conditions

Determine the proper line size for a 350 equivalent feet vacuum jet suction line drawing air at 350°F, at a rate of 255 lbs/hr with an initial pressure at the source of 0.6 in. Hg. Abs. Assume 10-in. pipe reading Figure 2-43. Note: watch scales carefully.

$F_1 = 0.0155$
$F_2 = 0.071$
$C_{D1} = 0.96$
$C_{D2} = 0.96$
$C_{T1} = 1.5$
$C_{T2} = 1.67$
$\Delta P_{vac} = [(0.0155)\ (0.96)\ (1.5) + (0.071)\ (0.96)\ (1.67)]/0.6$
$= (0.02232 + 0.1138)/0.6$
$= 0.2269$ in. water/100 ft.

Total line pressure drop:

$$\Delta P_{vac} = \left(\frac{0.2269}{100}\right)(350) = 0.794 \text{ in. water (for } 350')$$
$$= (0.794/13.6) = 0.0584 \text{ in. Hg}$$

Final calculated pressure = 0.6 + 0.0584 = 0.6584 in. Hg
10% of 0.658 = 0.0658 in. Hg

Therefore the system is applicable to the basis of the method, since the calculated pressure drop is less than 10% of the final pressure, and w/d = 25.5, which >20.

Low Absolute Pressure Systems for Air [54]

For piping with air in streamline flow at absolute pressures in the range between 50 microns and 1 millimeter of mercury, the following is a recommended method. Calculation procedures in pressure regions below atmospheric are very limited and often not generally applicable to broad interpretations.

For this method to be applicable, the pressure drop is limited to 10% of the final pressure.

Method [54]

Refer to Figure 2-44 for low pressure friction factor and air viscosity of Figure 2-45 to correspond to Figure 2-44.

$$P_1' - P_2' = \frac{4fL\rho v^2}{2gD(144)}, \text{ psi} \qquad (2\text{-}127)$$

where P'_1 = upstream static pressure, psi abs.
P'_2 = downstream static pressure, psi abs.
f = friction factor, from Figure 2-44.
L = length of pipe (total equivalent), ft, incl. valves and fittings
ρ = average density, lbs/cu ft
v = average velocity, ft/sec
g = acceleration due to gravity, 32.17 ft/sec-sec
D = inside diameter of pipe, ft
μ = abs. viscosity of air, lbs/ft-sec

Vacuum for other Gases and Vapors

Ryans and Roper categorize [18] vacuum in process systems as:

Category	Absolute Vacuum (Absolute Pressure)
Rough vacuum	760 torr to 1 torr
Medium vacuum	1 to 10^{-3} torr
High vacuum	10^{-3} to 10^{-7} torr
Ultra high vacuum	10^{-7} torr and below

The majority of industrial chemical and petrochemical plants' vacuum operations are in the range of 100 microns to 760 torr. This is practically speaking the rough vacuum range noted above. For reference:

1 torr = 1 mm mercury (mmHg)
1 in. mercury (in. Hg) = 25.4 torr
1 micron (μm Hg) = 0.0010 torr
For other conversions, see Appendix.

In general, partially due to the size and cost of maintaining vacuum in a piping system, the lines are not long (certainly not transmissions lines), and there is a minimum of valves, fittings, and bends to keep the resistance to flow low.

The procedure recommended by Reference [18] is based on the conventional gas flow equations, with some slight modifications. The importance in final line size determination is to determine what is a reasonable pressure loss at the absolute pressure required and the corresponding pipe size to balance these. In some cases a trial/error approach is necessary.

Method [18], by permission:

1. Convert mass flow rate to volumetric flow rate, q_m.

$$q_m = W(359/M)(760/P_t)(T/(32+460))(1/60), \text{ cu ft/min} \qquad (2\text{-}128)$$

where P_t = pressure, torr
T = temperature, °R
W = mass flow, lbs/hr
M = molecular weight

2. Calculate section by section from the process vessel to the vacuum pump (point of lowest absolute pressure).
3. Assume a velocity, v, ft/sec consistent with Figure 2-46. Use Table 2-21 for short, direct connected connections to the vacuum pump. Base the final specifications for the line on pump specifications. Also the diameter of the line should match the inlet connection for the pump. General good practice indicates that velocities of 100 to 200 ft/sec are used, with 300 to 400 ft/sec being the upper limit for the rough vacuum classification.

Sonic velocity, $v_s = (kg\,[1544/M]\,T)^{1/2}$, ft/sec.
Use v from Figure 2-46, and q_m from Equation 2-128.

4. Determine pipe diameter, D, ft,

$$D = 0.146\sqrt{q_m/v} \qquad (2\text{-}129)$$

Round this to the nearest standard pipe size. Recalculate v based on actual internal diameter of the line.

(text continued on page 132)

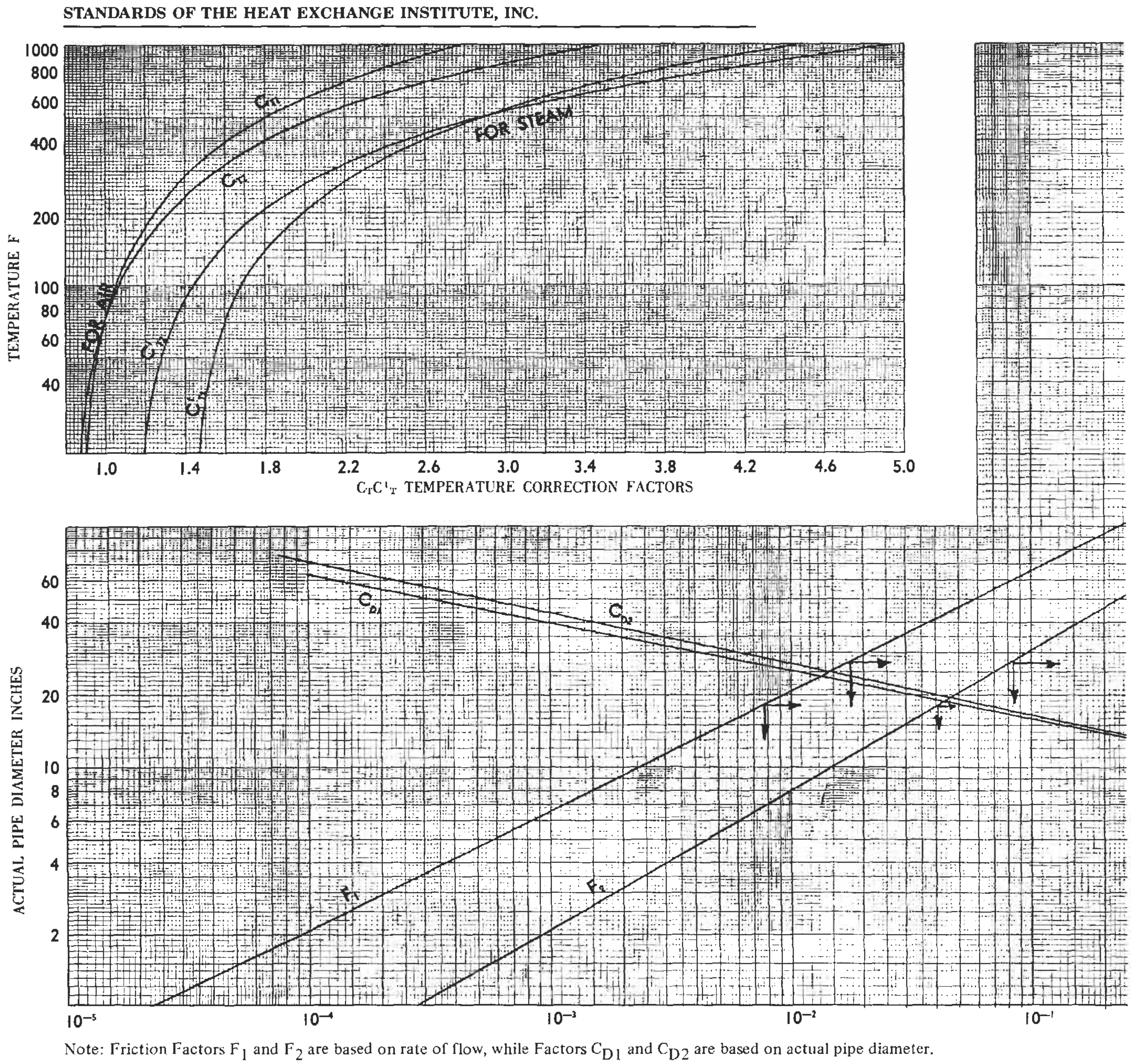
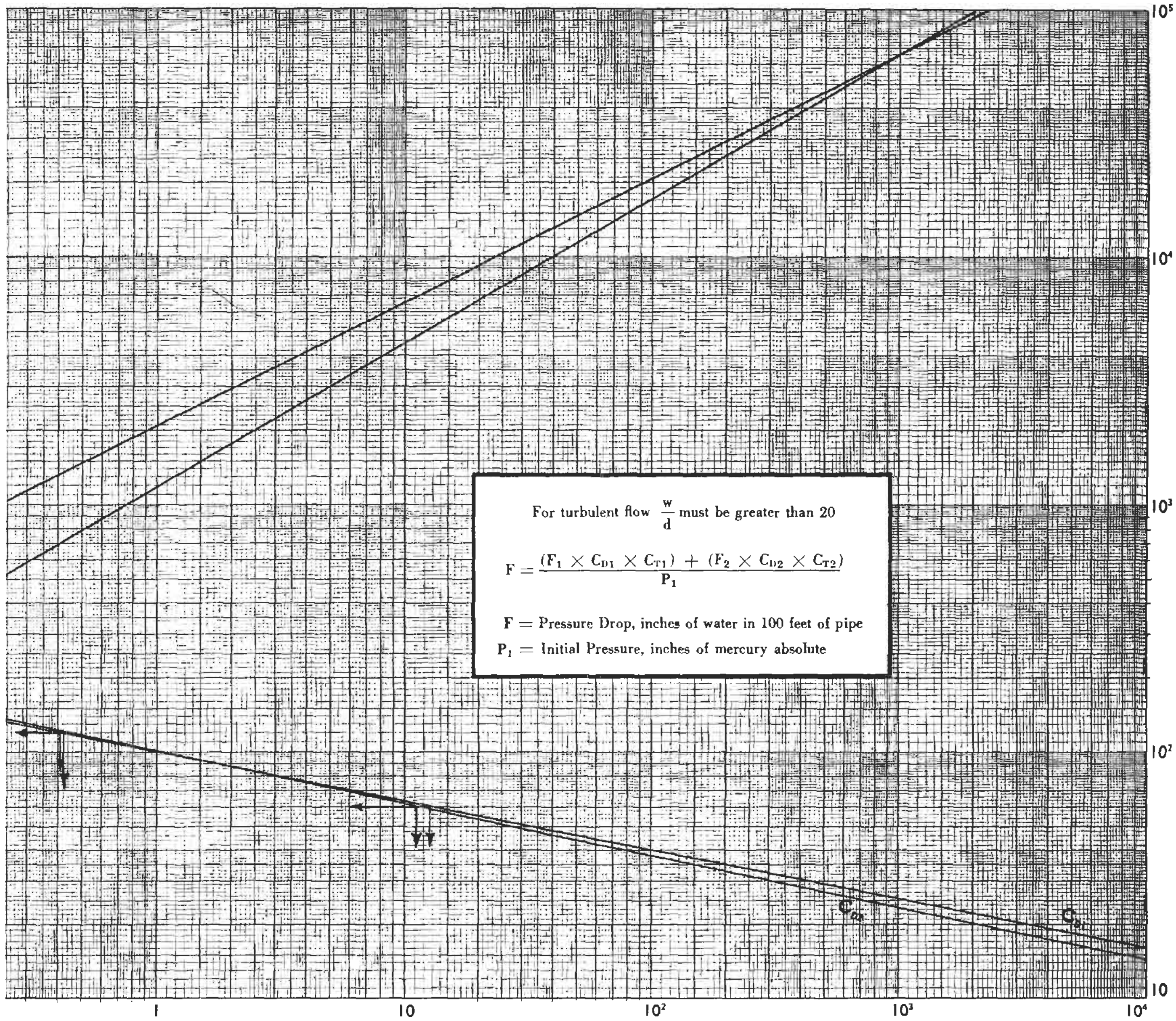


Figure 2-43. Evaluation curves for friction losses of air and steam flowing turbulently in commercial pipe at low pressures. By permission, *Standards for Steam Jet Ejectors*, 4th Ed., Heat Exchange Institute, 1988.

STEAM JET VACUUM SYSTEMS



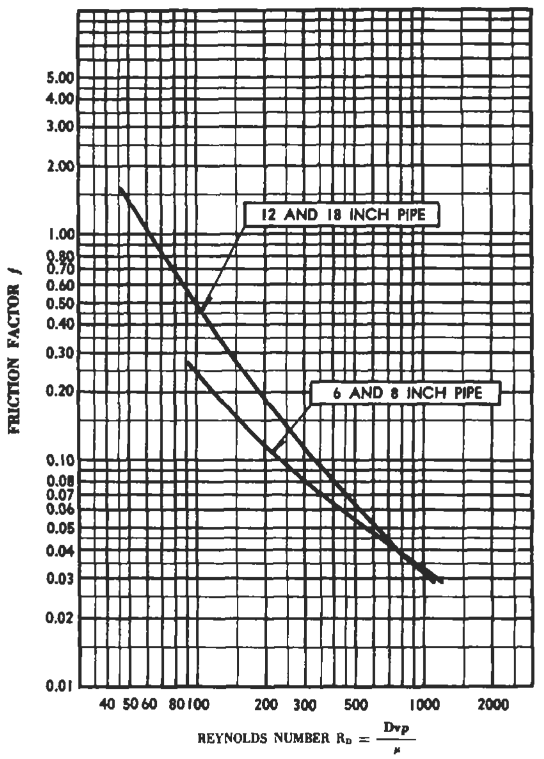


Figure 2-44. Friction factor for streamlined flow of air at absolute pressures from 50 microns Hg. to 1 mm Hg. By permission, *Standards for Steam Jet Ejectors,* 3rd. Ed., Heat Exchange Institute, 1956 [54] and *Standards for Steam Jet Vacuum Systems,* 4th Ed., 1988. Note: f on same basis as Figure 2-3 [58].

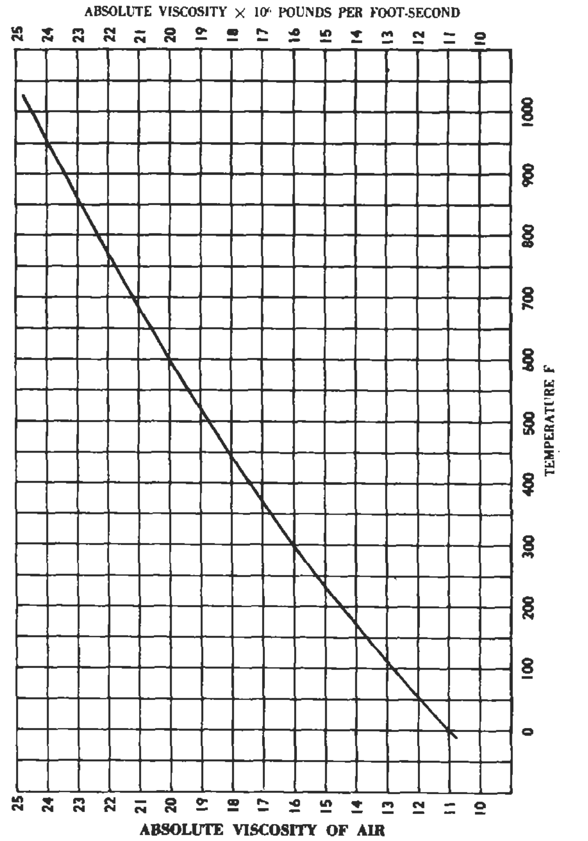


Figure 2-45. Absolute viscosity of air. By permission, *Standards for Steam Jet Ejectors,* 3rd Ed., Heat Exchange Institute, 1956 [54]; also, *Standards for Steam Jet Vacuum Systems,* 4th Ed., 1988 [58].

(text continued from page 129)

5. Determine Reynolds Number, R_e.

$$R_e = \rho D v / \mu_e \qquad (2\text{-}15)$$

ρ = density, lb/cu ft at flowing conditions
D = pipe inside diameter, ft
v = vapor velocity (actual), ft/sec
μ_e = viscosity of vapor, lb/ft-sec

6. Determine friction factor, f, from Moody Friction Factor Charts, Figure 2-3.

or, calculate for turbulent flow using Blausius' equation [18]:

$$f = 0.316/(R_e)^{1/4}, \text{ for } R_e < 2.0 \times 10^5$$

7. Tabulate the summation of equivalent lengths of straight pipe, valves, fittings, entrance/exit losses as presented in earlier sections of this chapter.

8. Calculate the pressure drop for the specific line section (or total line) from:

$$\Delta P_T = 0.625\ \rho_i f\ L q_m^2/d^5, \text{ torr} \qquad (2\text{-}130)$$

$$\text{or, } = 4.31\ \rho_i f\ L v^2/2gd, \text{ torr} \qquad (2\text{-}130A)$$

where ρ = density, lb/cu ft
d = pipe inside diameter, in.
q_m = volumetric flowrate, cu ft/min
f = friction factor, (Moody) Figure 2-3
ΔP_T = pressure drop, torr

Calculate: $\rho_i = P_i M / 555 T_i$, lb/cu ft (2-131)

P_i = pressure, torr
M = average molecular weight of mixture flowing
T_i = temperature, °R

9. If the calculated pressure drop does not exceed the maximum given in Figure 2-47, use this calculated value to specify the line. If the ΔP exceeds the limit

Table 2-21
Criteria for Sizing Connecting Lines in Vacuum Service

Vacuum pump	Assumed flow velocity, ft/s
Steam jet:	
System pressure, torr	
0.5–5	300
5–25	250
25–150	200
150–760	150
Liquid ring pump:	
Single-stage*	100
Two-stage	150
Rotary piston:	
Single-stage	50
Two-stage	25
Rotary vane:†	
Single-stage	200
Two-stage	400
Rotary blowers:	
Atmospheric discharge	50
Discharging to backing pump	100

*Assumes the pump features dual inlet connections and that an inlet manifold will be used.
†Based on rough vacuum process pumps. Use 25 ft/s for high vacuum pumps.
By permission, Ryans, J. L. and Roper, D. L., *Process Vacuum System Design and Operation*, McGraw-Hill Book Co. Inc., 1986 [18].

of Figure 2-47, increase the pipe size and repeat the calculations until an acceptable balance is obtained. For initial estimates, the authors [18] recommend using 0.6 times the value obtained from Figure 2-47 for an acceptable pressure loss between vessel and the pump.

The suction pressure required *at the vacuum pump* (in absolute pressure) is the actual process equipment operating pressure minus the pressure loss between the process equipment and the source of the vacuum. *Note that absolute pressures must be used* for these determinations and not gauge pressures. Also keep in mind that the absolute pressure at the vacuum pump must always be a lower absolute pressure than the absolute pressure at the process.

Pipe Sizing for Non-Newtonian Flow

Non-Newtonian fluids vary significantly in their properties that control flow and pressure loss during flow from the properties of Newtonian fluids. The key factors influencing non-Newtonian fluids are their shear thinning or thickening characteristics and time dependency of viscosity on the stress in the fluid.

Most conventional chemical and petrochemical plants do not process many, if any, non-Newtonian fluids. However, polymers, grease, heavy oils, cellulose compounds, paints, fine chalk suspensions in water, some asphalts, and other materials do exhibit one type or another of the characteristics of non-Newtonians, classified as:

- Bingham plastics
- Dilatant
- Pseudoplastic
- Yield pseudoplastics

Solving these classes of flow problems requires specific data on the fluid, which is often not in the public literature, or requires laboratory determinations using a rotational viscometer. The results do not allow use of the usual

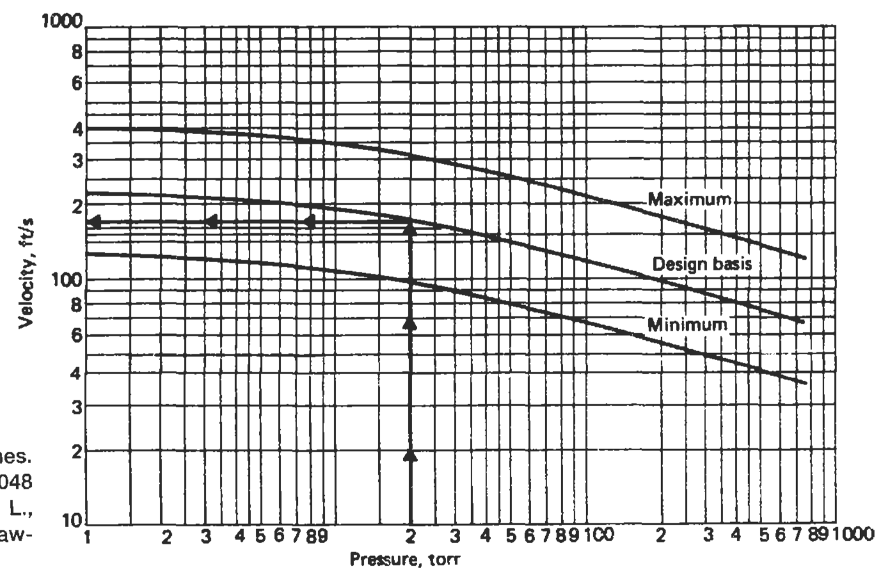


Figure 2-46. Typical flow velocities for vacuum lines. Note: 1 torr = 1.33 mb = 133.3 Pa. 1.0 ft/sec = 0.3048 m/sec. By permission, Ryans, J. L. and Roper, D. L., *Process Vacuum System Design & Operation*, McGraw-Hill Book Co., Inc., 1986 [18].

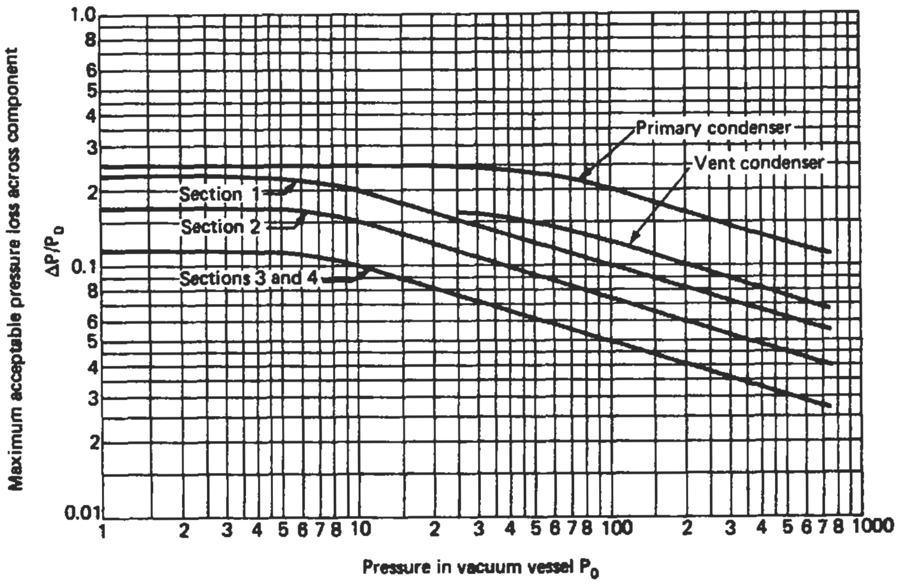


Figure 2-47. Acceptable pressure losses between the vacuum vessel and the vacuum pump. Note: reference sections on figure to system diagram to illustrate the sectional type hook-ups for connecting lines. Use 60% of the pressure loss read as acceptable loss for the system from process to vacuum pump, for initial estimate. P = pressure drop (torr) of line in question; P_o = operating pressure of vacuum process equipment, absolute, torr. By permission, Ryans, J. L. and Roper, D. L., *Process Vacuum System Design & Operation*, McGraw-Hill Book Co., Inc., 1986 [18].

Fanning or Moody friction charts and are beyond the scope of this chapter. Design literature is very limited, with some of the current available references being Sultan [21], Bird *et al.* [22], Cheremisinoff, N. P. and Gupta [14], Perry *et al.* [5], and Brodkey and Hershey [23].

Slurry Flow in Process Plant Piping

Most industrial process plants have from none to a few slurry flow lines to transport process fluids. The more common slurry lines discussed in the literature deal with long transmission lines for coal/water, mine tailings/water, limestone/water, wood pulp-fibers/water, gravel/water, and others. These lines usually can be expected to have flow characteristics somewhat different than in-plant process slurries. Considerable study has been made of the subject, with the result that the complexity of the variables make correlation of all data difficult, especially when dealing with short transfer lines. For this reason, no single design method is summarized here, but rather reference is given to the methods that appear most promising (also see Reference [30]).

Derammelaere and Wasp [25] present a design technique that ties into their classification of slurries as heterogeneous and homogeneous (Figures 2-48 and 2-49). This method uses the Fanning friction factor and conventional equations for pressure drop. The recommended design slurry velocities range from 4 to 7 ft/sec. Pipe abrasion can be a problem for some types of solids when the velocity approaches 10 ft/sec. For velocities below 4 ft/sec there can be a tendency for solids to settle and create blockage and plugging of the line.

The concentration of the solids in the slurry determines the slurry rheology or viscosity. This property is used as the viscosity factor in the pressure drop calculations. The two principal classifications are [25]:

1. Newtonian slurries are simple rheological property viscosities, and can be treated as true fluids as long as the flowing velocity is sufficient to prevent the dropout of solids. For this type of slurry, the viscosity = μ.
2. Bingham-plastic slurries require a shear stress diagram showing shear rate vs. shear stress for the slurry in order to determine the coefficient of rigidity, η, which is the slope of the plot at a particular concentration. This is laboratory data requiring a rheometer. These are usually fine solids at high concentrations.

Reference [25] has two practical in-plant design examples worked out.

The pressure drop design method of Turian and Yuan [24] is the development of the analysis of a major literature data review. The method categorizes slurry flow regimes similar in concept to the conventional multiregime diagram for two-phase flow, Figure 2-50. Their friction factor correlations are specific to the calculated flow regime. See Figure 2-51 for one of four typical plots in the original reference.

Example calculations are included, and Figure 2-52 illustrates the effect of pipe size on the placement of the flow regime.

Pressure Drop for Flashing Liquids

Steam is the most common liquid that is flashed in process plants, but of course, it is not the only one as many processes utilize flash operations of pure com-

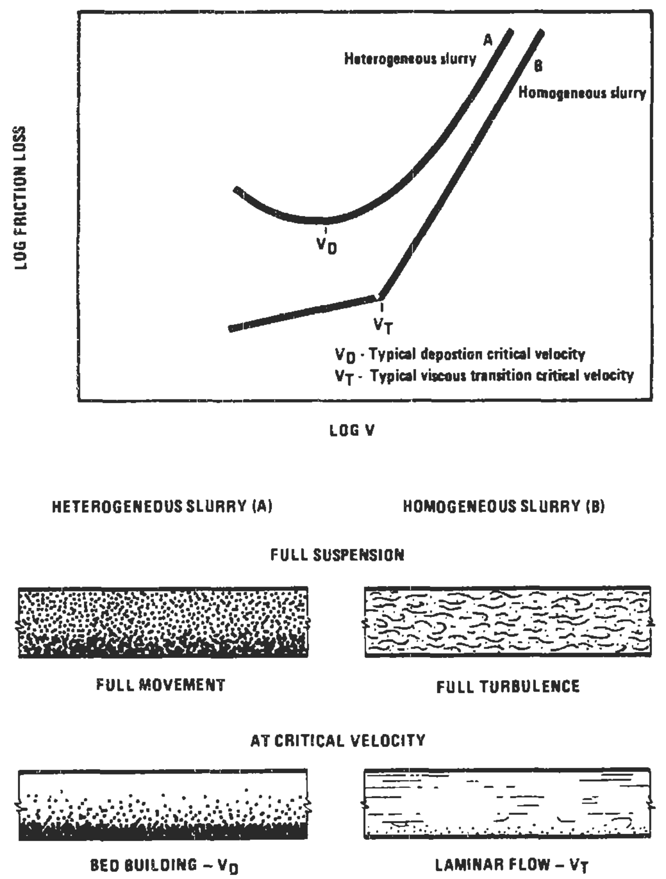


Figure 2-48. Critical velocity characteristics depend on whether slurry is heterogeneous or homogeneous. By permission, Derammelaere, R. H. and Wasp, E. J., "Fluid Flow, Slurry Systems and Pipelines," *Encyclopedia of Chemical Processing and Design,* J. McKetta, Ed., M. Dekker, vol. 22, 1985 [25].

pounds as well as mixtures. Although this presentation is limited to steam, the principles apply to other materials.

Steam condensate systems often are used to generate lower pressure steam by flashing to a lower pressure. When this occurs, some steam is formed and some condensate remains, with the relative quantities depending upon the pressure conditions. Figure 2-53 is a typical situation.

Percent incoming condensate flashed to steam:

$$\% \text{ flash} = \frac{(h_1 - h_2)100}{L_v} \tag{2-132}$$

where h_1 = enthalpy of liquid at higher pressure, Btu/lb
h_2 = enthalpy of liquid at lower or flash pressure, Btu/lb
L_v = latent heat of evaporation of steam at flash pressure, Btu/lb

Example 2-18: Calculation of Steam Condensate Flashing

There are 79,500 lbs/hr of 450 psig condensate flowing into a flash tank. The tank is to be held at 250 psig, generating steam at this pressure. Determine the quantity of steam produced.

Enthalpy of liquid at 450 psig = 441.1 Btu/lb
Enthalpy of liquid at 250 psig = 381.6 Btu/lb
Latent heat of vaporization at 250 psig = 820.1 Btu/lb

$$\% \text{ flash into steam} = \frac{441.1 - 381.6}{820.1}(100) = 7.25\%$$

Steam formed = (0.0725) (79,500) = 5,763 lbs/hr
Condensate formed = 79,500 − 5,763 = 73,737 lbs/hr

Sizing Condensate Return Lines

Steam condensate lines usually present a two-phase flow condition, with hot condensate flowing to a lower pressure through short and long lines. As the flow progresses down the pipe, the pressure falls and flashing of condensate into steam takes place continuously. For small lengths with low pressure drops, and the outlet end being within a few pounds per square inch of the inlet, the flash will be such a small percent that the line can often be sized as an all liquid line. However, caution must be exercised as even 5% flashing can develop an important impact on the pressure drop of the system.

Calculation of condensate piping by two-phase flow techniques is *recommended;* however, the tedious work per line can often be reduced by using empirical methods and charts. Some of the best are proprietary and not available for publication; however, the Sarco method [42] has been used and found to be acceptable, provided no line less than 1½″ is used regardless of the chart reading. Under some circumstances, which are too random to properly describe, the Sarco method may give results too small by possibly a half pipe size. Therefore, latitude is recommended in selecting either the flow rates or the pipe size.

Design Procedure Using Sarco Chart [42]

1. Establish upstream or steam pressure from which condensate is being produced and discharged into a return line through steam traps, or equivalent, psig.
2. Establish the steam condensate load or rate in lbs/hr flow.
3. Establish the pressure of the condensate return line, psig.
4. The method is based on an allowable 5,000 ft/min velocity in the return line (mixture).

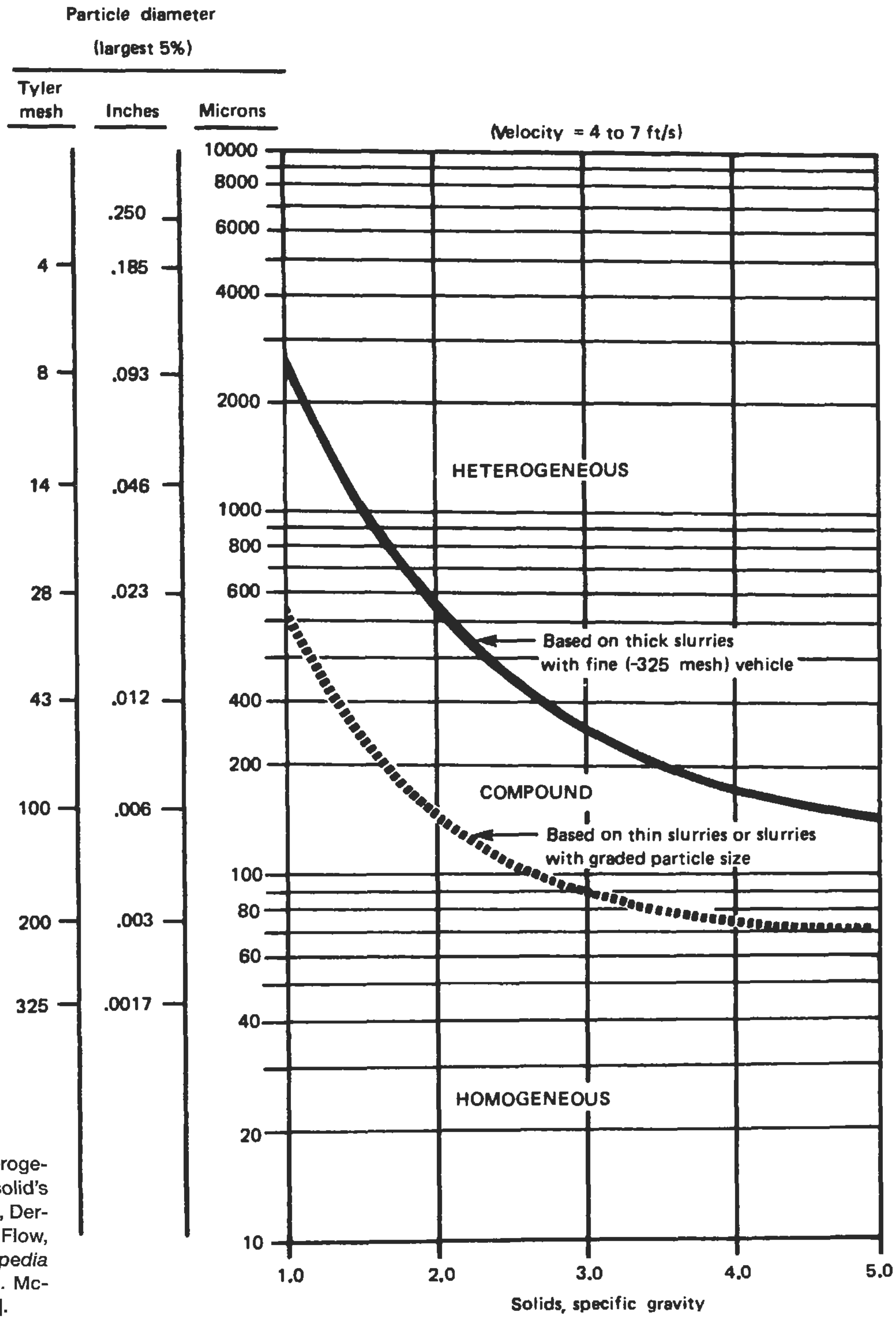


Figure 2-49. Slurry flow regime (heterogeneous, homogeneous) is a function of solid's size and specific gravity. By permission, Derannelaere, R. H. and Wasp, E. J., "Fluid Flow, Slurry Systems and Pipelines," *Encyclopedia of Chemical Processing and Design,* J. McKetta, Ed., M. Dekker, vol. 22, 1985 [25].

5. Calculate *load factor.*

$$= \frac{5{,}000\ (100)}{\text{Condensate Rate, lbs/hr}} = \frac{500{,}000}{C} \qquad (2\text{-}133)$$

6. Establish condensate receiver (or flash tank) pressure, psig.
7. Referring to Figure 2-54, enter at steam pressure of (1) above, move horizontally to condensate receiver pressure of (6) above, and then up vertically to the "factor scale."

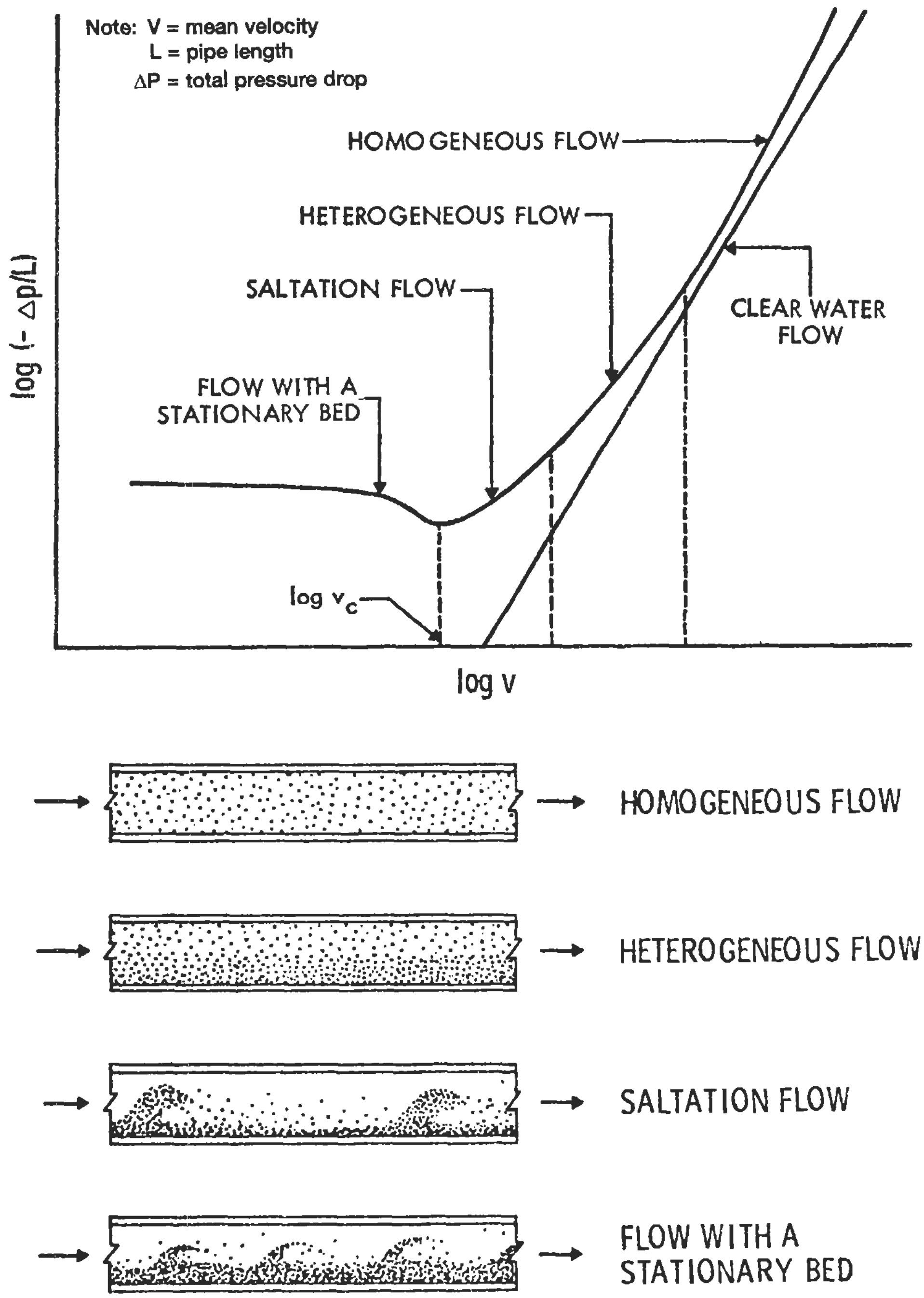


Figure 2-50. Representative plot of pressure drop for slurry flow. By permission, Turian, R. M. and Yuan, T. F., "Flow of Slurries in Pipelines," *A.I.Ch.E. Journal*, vol. 23, 1977, p. 232–243.

8. Divide the load factor (step 1) by the value from the "factor scale" of (7) above, obtain ft/min/(100 lb/hr load).

9. Enter chart on horizontal velocity line, go vertically up to the steam pressure of (1) above, and read pipe size to the next largest size if the value falls between two pipe sizes.

10. For pipe sizes larger than 3-in., follow the steps (1) thru (8) above. Then enter the vertical scale at the steam pressure of (1) above, and more to the 3-in. pipe size and down to the horizontal velocity scale.

11. Divide the result of step 8 above by the result of step (10).

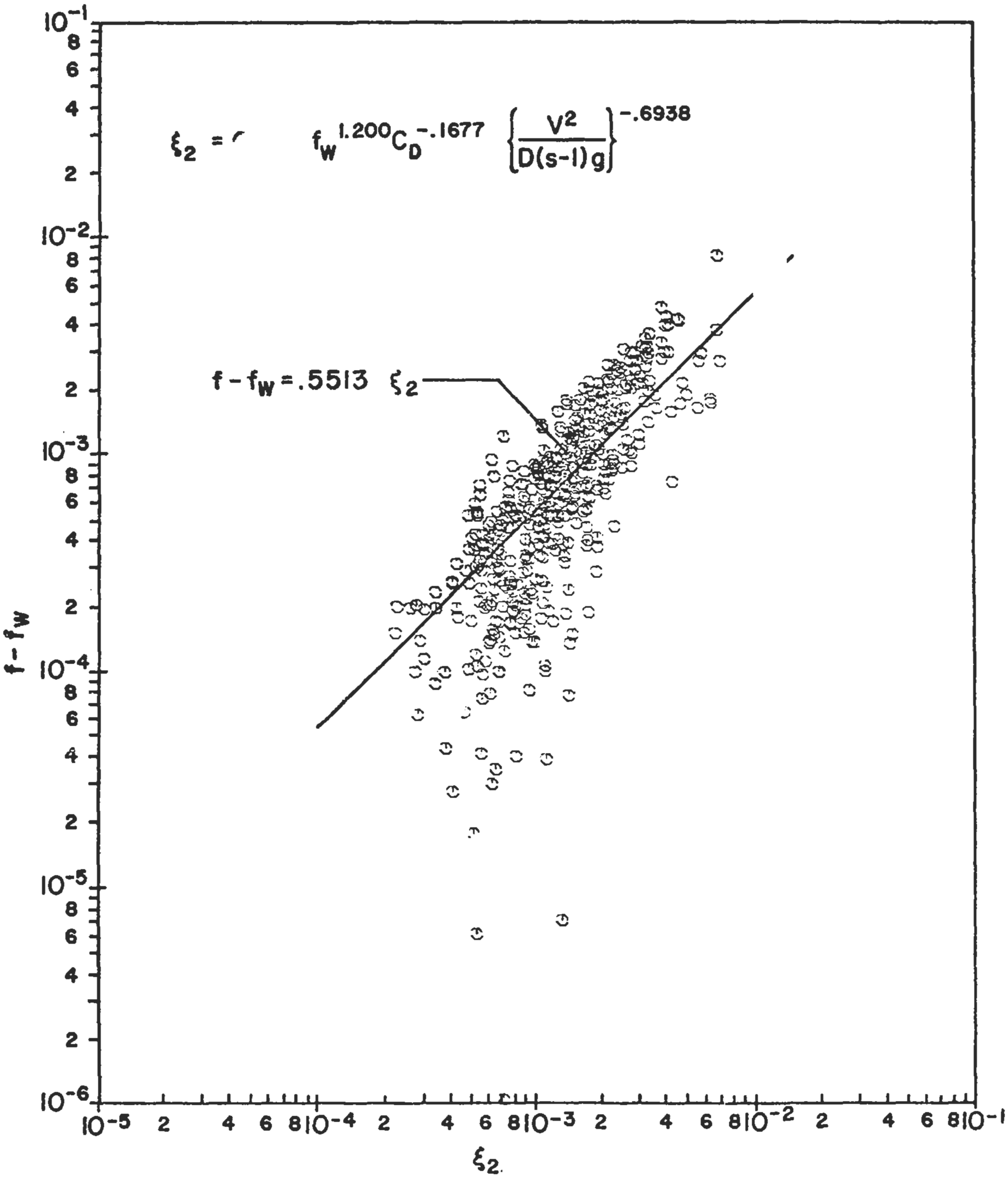


Figure 2-51. Friction factor correlation for slurry flow in heterogeneous flow regime. By permission, Turian, R. M. and Yuan, T. F., "Flow of Slurries in Pipelines," *A.I.Ch.E. Journal*, vol. 23, 1977, p. 232–243.

12. Refer to the large pipe multipliers shown in the table on the chart, and select the pipe size whose factor is equal to or smaller than the result of step (11) above. This is the pipe size to use, provided a sufficient factor of safety has been incorporated in the data used for the selection of pipe size.

13. Calculation of "factor scale" for receiver pressures different than those shown on chart:

$$\text{factor} = \frac{36.2\ (\bar{V})\ (h_p - h_r)}{L_v\ (h_p - 180)} \qquad (2\text{-}134)$$

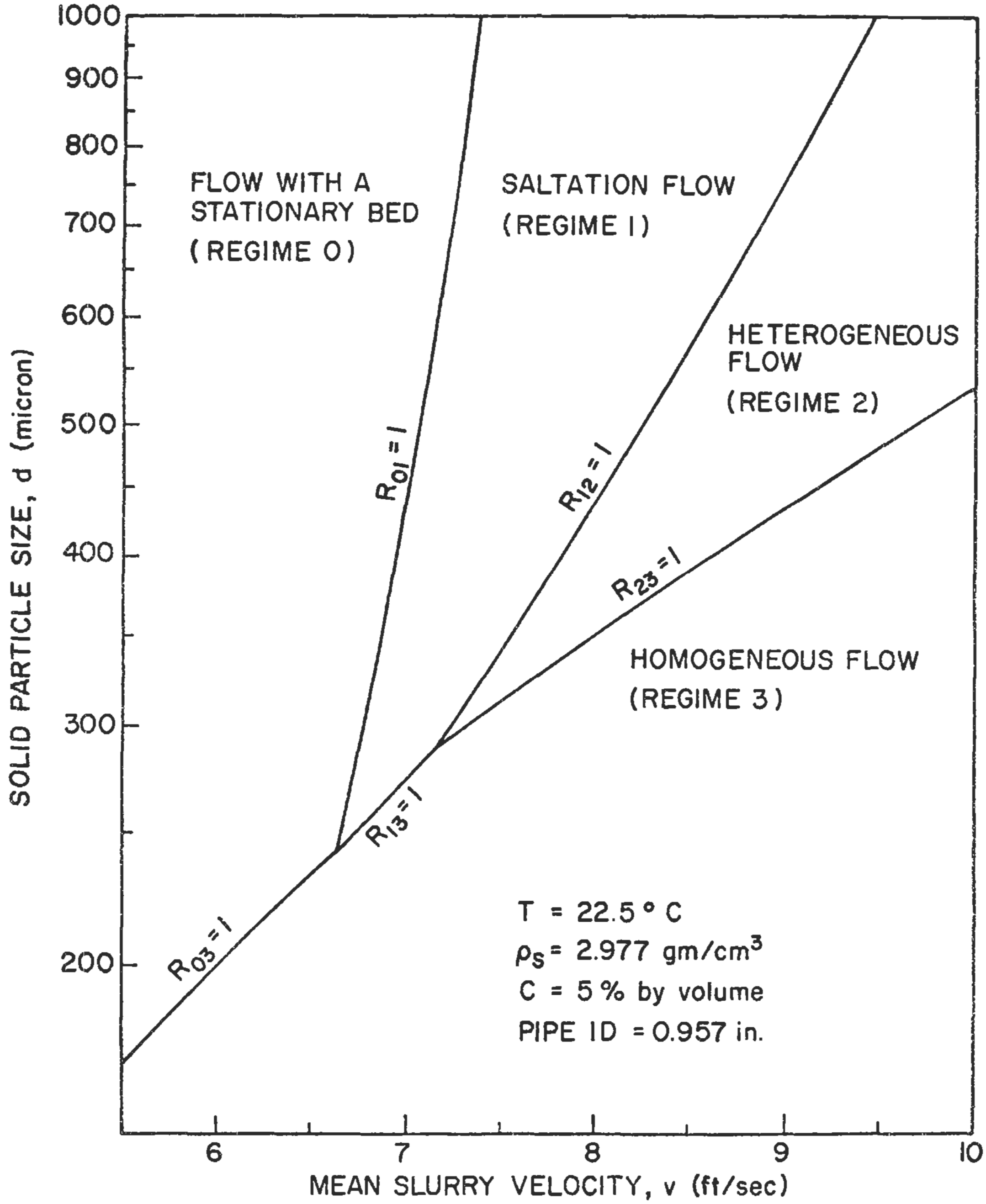


Figure 2-52. Flow regime diagram for solid-water flow in 1-in. PVC pipe. By permission, Turian, R. M. and Yuan, T. F., "Flow of Slurries in Pipelines," *A.I.Ch.E. Journal*, vol. 23, 1977, p. 232–243.

where $\overline{V}$ = specific volume of steam at return line pressure, cu ft/lb

h_p = enthalpy of liquid at supply steam pressure, Btu/lb

h_r = enthalpy of liquid at return line pressure, Btu/lb

L_v = latent heat of evaporation at return line pressure, Btu/lb

Use the factor so calculated just as if read from the chart, i.e., in step (8) above.

Example 2-19: Sizing Steam Condensate Return Line

A 450 psig steam system discharges 9,425 lbs/hr of condensate through traps into a return condensate line. The return header is to discharge into a flash tank held at 90

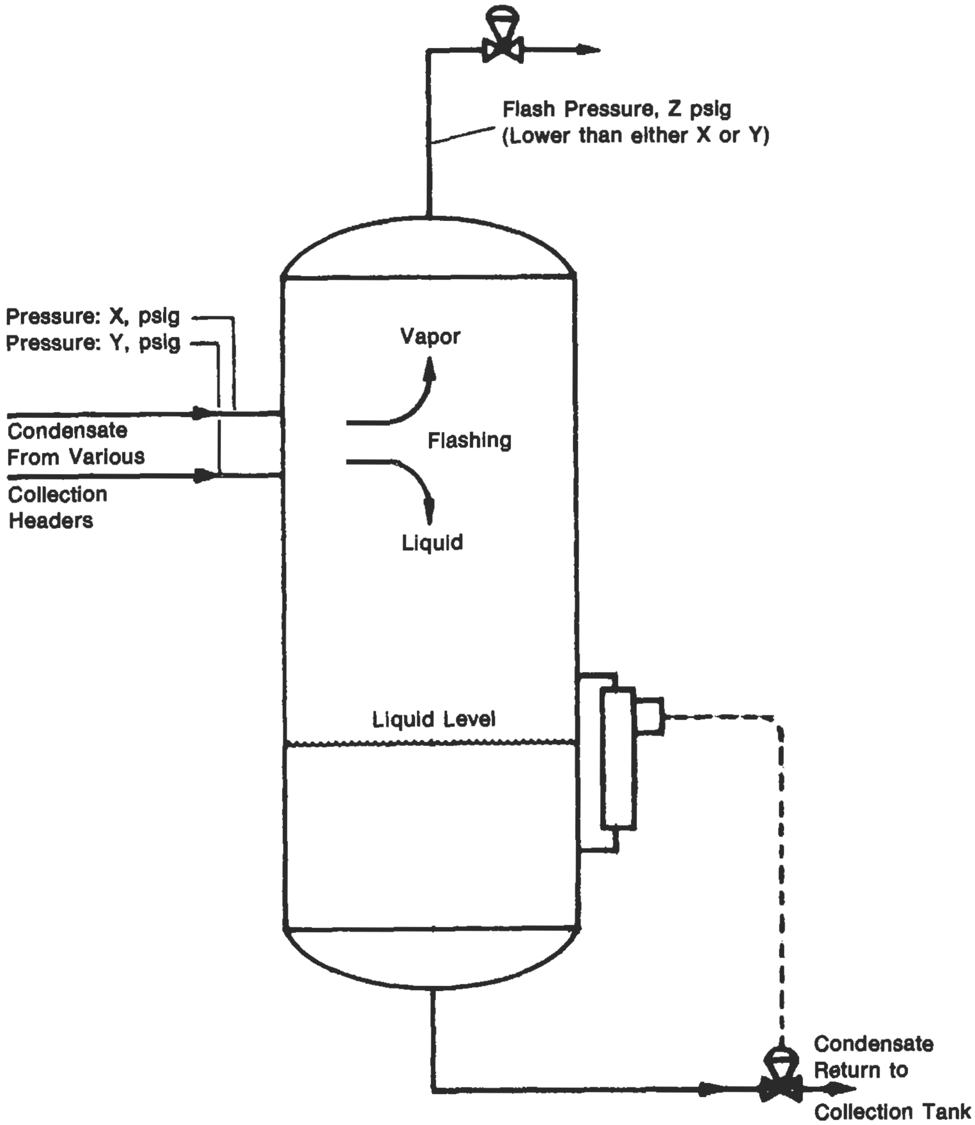


Figure 2-53. Typical steam condensate flashing operation.

psig. The calculated total equivalent length of pipe, valves, and fittings is 600 feet.

Using the Sarco chart, Figure 2-54, determine the recommended line size for the return line.

1. Upstream steam pressure = 450 psig
2. Condensate load = 9,425 lbs/hr
3. Return line pressure = 90 psig
4. Use the Sarco recommendation of 5,000 ft/min
5. Load factor

$$= \frac{(5{,}000)\ (100)}{9{,}425} = 53.0$$

6. Receiver pressure = 90 psig
7. Refer to Figure 2-54 and note that required receiver pressure is not shown, so calculate "factor scale" by previous formula:

Data: h_p = 441 Btu/lb at 450 psig
h_r = 302 Btu/lb at 90 psig
L_r = 886 Btu/lb at 90 psig
$\overline{V}$ = 4.232 cu ft/lb at 90 psig

$$\text{“factor scale” value} = \frac{36.2\ (4.232)\ (441 - 302)}{886\ (441 - 180)} = 0.092$$

8. $$\text{Ft/min/100\#/hr} = \frac{53}{0.092} = 576$$

VELOCITY AT PIPE EXIT WHEN DISCHARGING CONDENSATE AT SATURATION TEMPERATURES FROM VARIOUS PRESSURES TO ATMOSPHERE AT A RATE OF 100 POUNDS/HR.

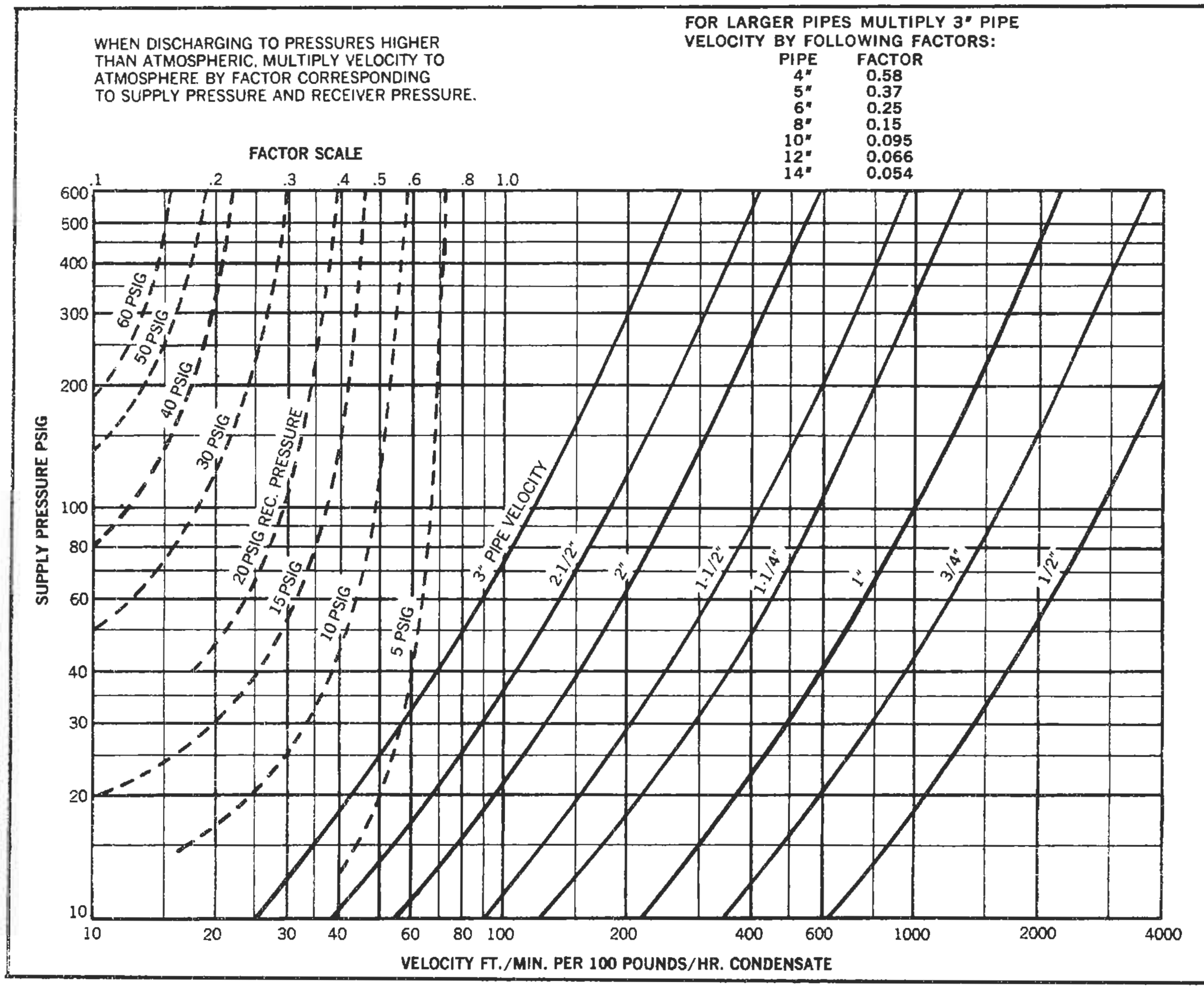


Figure 2-54. Sarco flashing steam condensate line sizing flow chart. By permission, Spirax-Sarco, Inc., Allentown, Pa. [59].

9. Read Chart: At 450 psig and 576, the line size shows just under 2-in. Recommend use 2-in.

Because flashing steam-condensate lines represent two-phase flow, with the quantity of liquid phase depending on the system conditions, these can be designed following the previously described two-phase flow methods. An alternate by Ruskin [28] uses the concept but assumes a single homogeneous phase of fine liquid droplets dispersed in the flashed vapor. Pressure drop was calculated by the Darcy equation:

$$\Delta P = 0.000336\ (f\ W^2)/d^5\ (\rho),\ \text{psi/100 ft} \qquad (2\text{-}55A)$$

Friction factor was calculated:

$$f = 0.25\ [-\log\ (0.000486/d)]^{-2.0} \qquad (2\text{-}135)$$

for complete turbulence in steel pipe. For large pressure drops through the transmission system, the line should be broken into increments of length for successive pressure drop calculations over the length, and the pressure drops summed to equal the total available/required.

The procedure for using the convenient chart Figure 2-55 [28] is, for example:

Step 1: Enter the figure at 600 psig below the insert near the right-hand side, and read down to the 200-psig end-pressure.

(text continued on page 153)

Figure 2-55. Flashing steam condensate line sizing chart. By permission, Ruskin, R. P., "Calculating Line Sizes for Flashing Steam Condensate," *Chem. Eng.*, Aug. 18, 1985, p. 101.

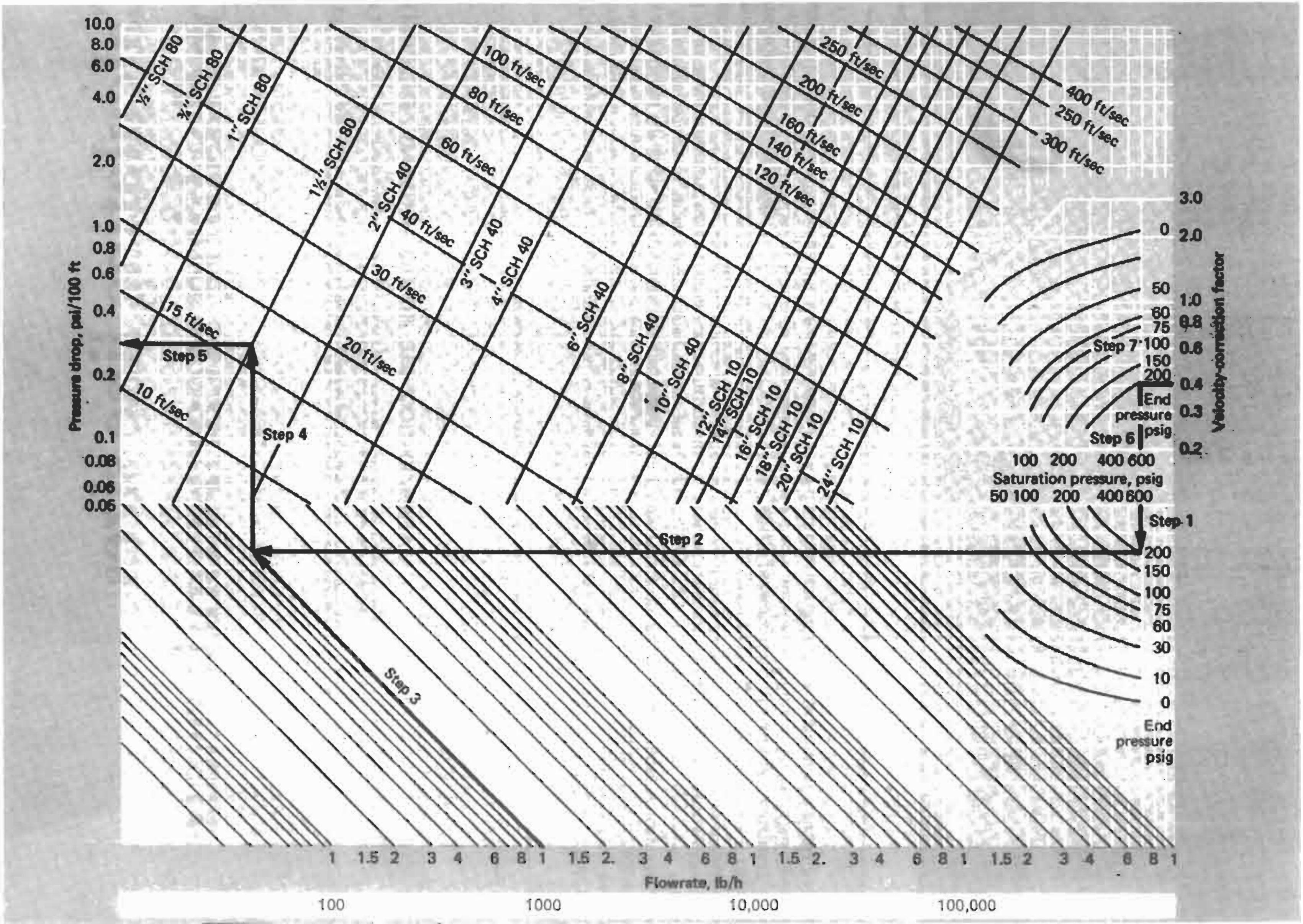


Table 2-22
Cameron Hydraulic Data*

Friction losses in pipes carrying water

Among the many empirical formulae for friction losses that have been proposed that of Williams and Hazen has been most widely used. In a convenient form it reads:

$$f = .2083\left(\frac{100}{C}\right)^{1.85} \frac{q^{1.85}}{d^{4.8655}}$$

in which

f = friction head in ft of liquid per 100 ft of pipe (if desired in lb per sq in. multiply f × .433 × sp gr)
d = inside dia of pipe in inches
q = flow in gal per min
C = constant accounting for surface roughness

This formula gives accurate values only when the kinematic viscosity of the liquid is about 1.1 centistokes or 31.5 SSU, which is the case with water at about 60F. But the viscosity of water varies with the temperature from 1.8 at 32F to .29 centistokes at 212F. The tables are therefore subject to this error which may increase the friction loss as much as 20% at 32F and decrease it as much as 20% at 212F. Note that the tables may be used for any liquid having a viscosity of the same order as indicated above.

Values of C for various types of pipe are given below together with the corresponding multiplier which should apply to the tabulated values of the head loss, f, as given on pages 29 to 48.

TYPE OF PIPE	VALUES OF C: Range — High = best, smooth, well laid — Low = poor or corroded	Average value for good, clean, new pipe	Commonly used value for design purposes
Cement—Asbestos	160–140	150	140
Fibre	—	150	140
Bitumastic-enamel-lined iron or steel centrifugally applied	160–130	148	140
Cement-lined iron or steel centrifugally applied	—	150	140
Copper, brass, lead, tin or glass pipe and tubing	150–120	140	130
Wood-stave	145–110	120	110
Welded and seamless steel	150–80	140	100
Continuous-interior riveted steel (no projecting rivets or joints)	—	139	100
Wrought-iron	150–80	130	100
Cast-iron	150–80	130	100
Tar-coated cast-iron	145–80	130	100
Girth-riveted steel (projecting rivets in girth seams only)	—	130	100
Concrete	152–85	120	100
Full-riveted steel (projecting rivets in girth and horizontal seams)	—	115	100
Vitrified	—	110	100
Spiral-riveted steel (flow with lap)	—	110	100
Spiral-riveted steel (flow against lap)	—	100	90
Corrugated steel	—	60	60

Value of C	150	140	130	120	110	100	90	80	70	60
Multiplier to correct tables	.47	.54	.62	.71	.84	1.0	1.22	1.59	1.93	2.57

Friction Losses In Pipe; C = 100

1/8 Inch

FLOW US gal per min	STANDARD WT STEEL .269″ inside dia: Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	EXTRA STRONG STEEL .215″ inside dia: Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
0.1	.565	.00	1.75	.884	.01	5.21
0.2	1.13	.02	6.31	1.77	.05	18.8
0.3	1.69	.04	13.4	2.65	.11	39.8
0.4	2.26	.08	22.8	3.54	.19	67.7
0.5	2.83	.12	34.4	4.42	.30	102
0.6	3.39	.18	48.2	5.32	.44	147
0.7	3.95	.24	64.1	6.29	.61	191
0.8	4.52	.32	82.0	7.08	.78	244
0.9	5.08	.40	102	7.96	.98	303
1.0	5.65	.50	124	8.84	1.21	369

1/4 Inch

FLOW US gal per min	STANDARD WT STEEL .364″ inside dia: Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	EXTRA STRONG STEEL .302″ inside dia: Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
0.4	1.23	.02	5.22	1.79	.05	13.0
0.6	1.85	.05	11.1	2.69	.11	27.4
0.8	2.47	.09	18.8	3.59	.20	46.7
1.0	3.08	.15	28.5	4.48	.31	70.6
1.2	3.71	.21	39.9	5.38	.45	98.9
1.4	4.33	.29	53.0	6.27	.61	132
1.6	4.94	.38	67.9	7.17	.80	168
1.8	5.55	.48	84.4	8.07	1.01	209
2.0	6.17	.59	103	8.96	1.25	254
2.5	7.71	.92	155	11.2	1.95	385

3/8 Inch

FLOW US gal per min	STANDARD WT STEEL .493″ inside dia: Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	EXTRA STRONG STEEL .423″ inside dia: Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
0.8	1.35	.03	4.30	1.83	.05	9.07
1.0	1.68	.04	6.50	2.28	.08	13.7
1.5	2.52	.10	13.8	3.43	.18	29.0
2.0	3.36	.18	23.4	4.57	.32	49.4
2.5	4.21	.28	35.4	5.71	.51	74.6
3.0	5.05	.40	49.6	6.85	.73	105
3.5	5.89	.54	66.0	8.00	.99	139
4.0	6.73	.70	84.5	9.14	1.30	178
5.0	8.41	1.10	134	11.4	2.0	269
6.0	10.1	1.58	179	13.7	2.9	377

*By permission G. V. Shaw and A. W. Loomis *Cameron Hydraulic Data*, 11th Edition, Ingersoll-Rand Co., 1942 [53].

Table 2-18: Cameron Hydraulic Data (cont)

Friction Losses In Pipe; C = 100

½ Inch

FLOW	Standard Wt Steel			Extra Strong Steel			Double Extra Strong Steel		
	.622″ inside dia			.546″ inside dia			.252″ inside dia		
US gal per min	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
0.5	.528	.00	.582	.686	.01	1.10	3.22	.16	47.2
1.0	1.06	.02	2.10	1.37	.03	3.96	6.44	.64	170
1.5	1.58	.04	4.44	2.06	.07	8.38	9.66	1.45	361
2.0	2.11	.07	7.57	2.74	.12	14.3	12.9	2.59	614
2.5	2.64	.11	11.4	3.43	.18	21.6	16.1	4.03	928
3.0	3.17	.16	16.0	4.11	.26	30.2			
3.5	3.70	.21	21.3	4.80	.36	40.2			
4.0	4.23	.28	27.3	5.48	.47	51.4			
4.5	4.75	.35	33.9	6.17	.59	64.0			
5.0	5.28	.43	41.2	6.86	.73	77.7			
5.5	5.81	.52	49.2	7.54	.88	92.7			
6.0	6.34	.62	57.8	8.23	1.05	109			
6.5	6.87	.73	67.0	8.91	1.23	126			
7.0	7.39	.85	76.8	9.60	1.43	145			
7.5	7.92	.97	87.3	10.3	1.6	165			
8.0	8.45	1.11	98.3	11.0	1.9	185			
8.5	8.98	1.25	110	11.6	2.1	207			
9.0	9.51	1.4	122	12.3	2.4	231			
9.5	10.0	1.6	135	13.0	2.6	255			
10	10.6	1.7	149	13.7	2.9	280			

¾ Inch

FLOW	Standard Wt Steel			Extra Strong Steel			Double Extra Strong Steel		
	.824″ inside dia			.742″ inside dia			.434″ inside dia		
US gal per min	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
1.5	.903	.01	1.13	1.11	.02	1.88	3.25	.16	25.6
2.0	1.20	.02	1.93	1.48	.03	3.21	4.34	.29	43.6
2.5	1.51	.04	2.91	1.86	.05	4.85	5.42	.46	65.9
3.0	1.81	.05	4.08	2.23	.08	6.79	6.51	.66	92.3
3.5	2.11	.07	5.42	2.60	.11	9.03	7.60	.90	123
4.0	2.41	.09	6.94	2.97	.14	11.6	8.68	1.17	157
4.5	2.71	.11	8.63	3.34	.17	14.4	9.77	1.48	195
5	3.01	.14	10.5	3.71	.21	17.5	10.9	1.8	238
6	3.61	.20	14.7	4.45	.31	24.5	13.0	2.6	333
7	4.21	.28	19.6	5.20	.42	32.6	15.2	3.6	443
8	4.82	.36	25.0	5.94	.55	41.7	17.4	4.7	567
9	5.42	.46	31.1	6.68	.69	51.8	19.5	5.9	704
10	6.02	.56	37.8	7.42	.86	63.0	21.7	7.3	856
11	6.62	.68	45.1	8.17	1.04	75.1			
12	7.22	.81	53.0	8.91	1.23	88.3			
13	7.82	.95	61.5	9.63	1.44	102			
14	8.43	1.10	70.5	10.4	1.7	117			
16	9.63	1.44	90.2	11.9	2.2	150			
18	10.8	1.8	112	13.4	2.8	187			
20	12.0	2.2	136	14.8	3.4	227			

Friction Losses In Pipe; C = 100

1 Inch

FLOW	Standard Wt Steel			Extra Strong Steel			Double Extra Strong Steel		
	1.049″ inside dia			.957″ inside dia			.599″ inside dia		
US gal per min	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
2	.742	.01	.595	.892	.01	.930	2.28	.08	11.7
3	1.11	.02	1.26	1.34	.03	1.97	3.42	.18	24.8
4	1.49	.03	2.14	1.79	.05	3.28	4.56	.32	42.2
5	1.86	.05	3.24	2.23	.08	5.07	5.69	.50	63.8
6	2.23	.08	4.54	2.68	.11	7.10	6.83	.72	69.4
8	2.97	.14	7.73	3.57	.20	12.1	9.11	1.29	152
10	3.71	.21	11.7	4.46	.31	18.3	11.4	2.0	230
12	4.46	.31	16.4	5.36	.45	25.6	13.7	2.9	322
14	5.20	.42	21.8	6.25	.61	34.0	15.9	3.9	429
16	5.94	.55	27.9	7.14	.79	43.6	18.2	5.1	549
18	6.68	.69	34.7	8.03	1.00	54.2	20.5	6.5	682
20	7.43	.86	42.1	8.92	1.24	65.8	22.7	8.0	829
22	8.17	1.04	50.2	9.82	1.50	78.5	25.1	9.8	989
24	8.91	1.23	59.0	10.7	1.8	94.4			
26	9.66	1.45	68.4	11.6	2.1	107			
28	10.4	1.7	78.5	12.5	2.4	123			
30	11.1	1.9	89.2	13.4	2.8	139			
35	13.0	2.6	119	15.6	3.8	185			
40	14.9	3.5	152	17.9	5.0	237			
45	16.7	4.3	189	20.1	6.3	295			

1¼ Inch

FLOW	Standard Wt Steel			Extra Strong Steel			Double Extra Strong Steel		
	1.380″ inside dia			1.278″ inside dia			.896″ inside dia		
US gal per min	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
4	.86	.01	.564	1.00	.02	.821	2.04	.06	4.63
5	1.07	.02	.853	1.25	.02	1.24	2.54	.10	6.98
6	1.29	.03	1.20	1.50	.04	1.74	3.05	.14	9.78
7	1.50	.04	1.59	1.75	.05	2.31	3.56	.20	13.0
8	1.72	.05	2.04	2.00	.06	2.96	4.07	.26	16.7
10	2.15	.07	3.08	2.50	.10	4.47	5.09	.40	25.2
12	2.57	.10	4.31	3.00	.14	6.26	6.11	.58	35.3
14	3.00	.14	5.73	3.50	.19	8.33	7.12	.79	46.9
16	3.43	.18	7.34	4.00	.25	10.7	8.14	1.03	60.0
18	3.86	.23	9.13	4.50	.31	13.3	9.16	1.30	74.7
20	4.29	.29	11.1	5.00	.39	16.1	10.2	1.6	90.7
25	5.36	.45	16.8	6.25	.61	24.9	12.7	2.5	137
30	6.43	.64	23.5	7.50	.87	34.1	15.3	3.6	192
35	7.51	.88	31.2	8.75	1.19	45.4	17.8	4.9	255
40	8.58	1.14	40.0	10.0	1.6	58.1	20.4	6.5	327
50	10.7	1.8	60.4	12.5	2.4	87.8	25.4	10.0	494
60	12.9	2.6	84.7	15.0	3.5	123	30.5	14.5	692
70	15.0	3.5	114	17.5	4.8	164	35.6	19.7	921
80	17.2	4.6	144	20.0	6.2	209			
90	19.3	5.8	179	22.5	7.9	260			

Friction Losses In Pipe; C = 100
1½ Inch

FLOW US gal per min	Standard Wt Steel 1.610″ inside dia			Extra Strong Steel 1.500″ inside dia			Double Extra Strong Steel 1.100″ inside dia		
	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
4	.63	.01	.267	.73	.01	.376	1.35	.03	1.70
5	.79	.01	.403	.91	.01	.569	1.69	.04	2.57
6	.95	.01	.565	1.09	.02	.797	2.03	.06	3.60
7	1.10	.02	.751	1.27	.03	1.06	2.36	.09	4.79
8	1.26	.02	.962	1.45	.03	1.36	2.70	.11	6.14
9	1.42	.03	1.20	1.63	.04	1.69	3.04	.14	7.63
10	1.58	.04	1.45	1.82	.05	2.05	3.38	.18	9.27
12	1.89	.06	2.04	2.18	.07	2.87	4.05	.25	13.0
14	2.21	.08	2.71	2.54	.10	3.82	4.73	.35	17.3
16	2.52	.10	3.47	2.90	.13	4.89	5.40	.45	22.1
18	2.84	.13	4.31	3.27	.17	6.08	6.08	.57	27.5
20	3.15	.15	5.24	3.63	.20	7.39	6.75	.71	33.4
22	3.47	.19	6.25	3.99	.25	8.82	7.43	.86	39.9
24	3.78	.22	7.34	4.36	.30	10.4	8.10	1.02	46.8
26	4.10	.26	8.51	4.72	.35	12.0	8.78	1.20	54.3
28	4.41	.30	9.76	5.08	.40	13.8	9.45	1.39	62.3
30	4.73	.35	11.1	5.45	.46	15.7	10.1	1.6	70.8
32	5.04	.39	12.5	5.81	.52	17.6	10.8	1.8	79.8
34	5.36	.45	14.0	6.17	.59	19.7	11.5	2.1	89.2
36	5.67	.50	15.5	6.54	.66	21.9	12.2	2.3	99.2
38	5.99	.56	17.2	6.90	.74	24.2	12.8	2.5	110
40	6.30	.62	18.9	7.26	.82	26.7	13.5	2.8	121
42	6.62	.68	20.7	7.63	.90	29.2	14.2	3.1	132
44	6.93	.75	22.5	7.99	.99	31.8	14.9	3.5	144
46	7.25	.82	24.5	8.35	1.08	34.5	15.6	3.8	156
48	7.57	.89	27.1	8.72	1.18	37.3	16.2	4.1	169
50	7.88	.97	28.5	9.08	1.28	40.3	16.9	4.4	182
55	8.67	1.17	34.0	9.99	1.55	49.0	18.6	5.4	217
60	9.46	1.39	40.0	10.9	1.8	56.4	20.3	6.4	255
65	10.2	1.6	46.4	11.8	2.2	65.4	21.9	7.5	296
70	11.0	1.9	53.2	12.7	2.5	75.0	23.6	8.7	339
75	11.8	2.2	60.4	13.6	2.9	85.3	25.3	9.9	386
80	12.6	2.5	68.1	14.5	3.3	96.1	27.0	11.3	435
85	13.4	2.8	76.2	15.4	3.7	107	28.7	12.8	486
90	14.2	3.1	84.7	16.3	4.1	119	30.4	14.4	540
95	15.0	3.5	93.6	17.2	4.6	132	32.1	16.0	597
100	15.8	3.9	103	18.2	5.1	145	33.8	17.8	657
110	17.3	4.7	123	20.0	6.2	173			
120	18.9	5.6	144	21.8	7.4	203			
130	20.5	6.5	167	23.6	8.7	236			
140	22.1	7.6	192	25.4	10.0	271			
150	23.6	8.7	218	27.2	11.5	308			
160	25.2	9.9	245	29.0	13.1	346			
170	26.8	11.2	275	30.9	14.8	387			
180	28.4	12.5	305	32.7	16.6	431			

Friction Losses In Pipe; C = 100
2 Inch

FLOW US gal per min	Standard Wt Steel 2.067″ inside dia			Extra Strong Steel 1.939″ inside dia			Double Extra Strong Steel 1.503″ inside dia		
	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
5	.48	.00	.120	.54	.00	.163	.90	.01	.563
6	.57	.01	.167	.65	.01	.229	1.09	.02	.789
7	.67	.01	.223	.76	.01	.304	1.27	.03	1.05
8	.77	.01	.285	.87	.01	.389	1.45	.03	1.34
9	.86	.01	.355	.98	.01	.484	1.63	.04	1.67
10	.96	.01	.431	1.09	.02	.588	1.81	.05	2.03
12	1.15	.02	.604	1.30	.03	.824	2.17	.07	2.85
14	1.34	.03	.803	1.52	.04	1.10	2.53	.10	3.78
16	1.53	.04	1.03	1.74	.05	1.40	2.89	.13	4.85
18	1.72	.05	1.28	1.96	.06	1.74	3.25	.16	6.02
20	1.91	.06	1.55	2.17	.07	2.12	3.62	.20	7.32
22	2.10	.07	1.85	2.39	.09	2.53	3.98	.25	8.73
24	2.29	.08	2.18	2.61	.11	2.97	4.34	.29	10.3
26	2.49	.10	2.52	2.83	.12	3.44	4.70	.34	11.9
28	2.68	.11	2.89	3.04	.14	3.95	5.06	.40	13.6
30	2.87	.13	3.29	3.26	.17	4.49	5.43	.46	15.5
35	3.35	.17	4.37	3.80	.22	5.97	6.33	.62	20.6
40	3.82	.23	5.60	4.35	.29	7.64	7.23	.81	26.4
45	4.30	.29	6.96	4.89	.37	9.50	8.14	1.03	32.8
50	4.78	.36	8.46	5.43	.46	11.5	9.04	1.27	39.9
55	5.26	.43	10.1	5.98	.56	13.7	9.95	1.54	47.5
60	5.74	.51	11.9	6.52	.66	16.2	10.9	1.8	54.6
65	6.21	.60	13.7	7.06	.77	18.8	11.8	2.2	64.8
70	6.69	.70	15.8	7.61	.90	21.5	12.7	2.5	74.3
75	7.17	.80	17.9	8.15	1.03	24.5	13.6	2.9	84.4
80	7.65	.91	20.2	8.69	1.17	27.6	14.5	3.3	95.2
85	8.13	1.03	22.6	9.03	1.27	30.8	15.4	3.7	106
90	8.61	1.15	25.1	9.78	1.49	34.3	16.3	4.1	118
95	9.08	1.28	27.7	10.3	1.6	37.9	17.2	4.6	131
100	9.56	1.42	30.5	10.9	1.8	41.6	18.1	5.1	144
110	10.5	1.7	36.4	12.0	2.2	49.7	19.9	6.2	172
120	11.5	2.1	42.7	13.0	2.6	58.3	21.7	7.3	201
130	12.4	2.4	49.6	14.1	3.1	67.7	23.5	8.6	234
140	13.4	2.8	56.9	15.2	3.6	77.6	25.3	9.9	268
150	14.3	3.2	64.7	16.3	4.1	88.4	27.1	11.4	305
160	15.3	3.6	72.8	17.4	4.7	99.3	28.9	13.0	343
170	16.3	4.1	81.4	18.5	5.3	111	30.7	14.6	384
180	17.2	4.6	90.5	19.6	6.0	124	32.5	16.4	427
190	18.2	5.1	100	20.6	6.6	137	34.4	18.4	471
200	19.1	5.7	110	21.7	7.3	150	36.2	20.4	518
220	21.0	6.8	131	23.9	8.9	179	39.8	24.6	618
240	22.9	8.2	154	26.1	10.6	210	43.4	29.3	726
260	24.9	9.6	179	28.3	12.4	244	47.0	34.3	842
280	26.8	11.2	205	30.4	14.4	280			
300	28.7	12.8	233	32.6	16.5	318			

Friction Losses In Pipe; C = 100
2½ Inch

FLOW	Standard Wt Steel			Extra Strong Steel			Double Extra Strong Steel		
	2.469″ inside dia			2.323″ inside dia			1.771″ inside dia		
US gal per min	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
8	.54	.00	.120	.61	.01	.163	1.04	.02	.605
10	.67	.01	.182	.76	.01	.244	1.30	.03	.914
12	.80	.01	.254	.91	.01	.342	1.56	.04	1.28
14	.94	.01	.338	1.06	.02	.455	1.82	.05	1.70
16	1.07	.02	.433	1.21	.02	.583	2.08	.07	2.18
18	1.21	.02	.538	1.36	.03	.724	2.34	.09	2.71
20	1.34	.03	.654	1.51	.04	.880	2.73	.12	3.30
22	1.47	.03	.780	1.67	.04	1.050	2.87	.13	3.93
24	1.61	.04	.917	1.82	.05	1.23	3.13	.15	4.62
26	1.74	.05	1.06	1.97	.06	1.43	3.39	.18	5.35
28	1.88	.05	1.22	2.12	.07	1.64	3.65	.21	6.14
30	2.01	.06	1.39	2.27	.08	1.86	4.00	.25	6.98
35	2.35	.09	1.84	2.65	.11	2.48	4.56	.32	9.28
40	2.68	.11	2.36	3.03	.14	3.17	5.21	.42	11.9
45	3.02	.14	2.93	3.41	.18	3.95	5.86	.53	14.8
50	3.35	.17	3.56	3.79	.22	4.79	6.51	.66	18.0
55	3.69	.21	4.24	4.16	.27	5.71	7.16	.80	21.4
60	4.02	.25	4.99	4.54	.32	6.72	7.81	.95	25.2
65	4.36	.30	5.79	4.92	.38	7.79	8.47	1.11	29.2
70	4.69	.34	6.64	5.30	.44	8.94	9.12	1.29	33.5
75	5.03	.39	7.55	5.68	.50	10.2	9.77	1.48	38.0
80	5.36	.45	8.50	6.05	.57	11.4	10.4	1.7	42.8
85	5.70	.50	9.51	6.43	.64	12.8	11.1	1.9	47.9
90	6.03	.57	10.6	6.81	.72	13.9	11.7	2.1	53.3
95	6.37	.63	11.7	7.19	.80	15.7	12.4	2.4	58.9
100	6.70	.70	12.8	7.57	.89	17.3	13.0	2.6	64.7
110	7.37	.84	15.3	8.33	1.08	20.6	14.3	3.2	77.2
120	8.04	1.00	18.0	9.08	1.28	24.2	15.6	3.8	90.7
130	8.71	1.18	20.9	9.84	1.50	28.1	16.9	4.4	105
140	9.38	1.37	23.9	10.6	1.7	32.2	18.2	5.1	121
150	10.0	1.6	27.3	11.3	2.0	36.7	19.5	5.9	137
160	10.7	1.8	30.7	12.1	2.3	41.2	20.8	6.7	154
170	11.4	2.0	34.3	12.9	2.6	46.1	22.2	7.7	173
180	12.1	2.3	38.1	13.6	2.9	51.3	23.4	8.5	192
190	12.7	2.5	42.1	14.4	3.2	56.7	24.7	9.5	212
200	13.4	2.8	46.3	15.1	3.5	62.3	26.1	10.6	233
220	14.7	3.4	55.3	16.7	4.3	74.3	28.7	12.8	278
240	16.1	4.0	66.4	18.2	5.1	87.3	31.3	15.2	327
260	17.4	4.7	75.3	19.7	6.0	101	33.9	17.9	379
280	18.8	5.5	86.3	21.2	7.0	116	36.5	20.7	436
300	20.1	6.3	98.1	22.7	8.0	132	39.1	23.8	494
350	23.5	8.6	130	26.5	10.9	175	45.6	32.3	657
400	26.8	11.2	167	30.3	14.3	225	52.1	42.2	841
450	30.2	14.2	208	34.1	18.1	279			
500	33.5	17.4	252	37.9	22.3	340			

Friction Losses in Pipe; C = 100
3 Inch

FLOW	Cast Iron			Std Wt Steel			Extra Strong Steel			Double Extra Strong Steel		
	3.0 inside dia			3.068″ inside dia			2.900″ inside dia			2.300″ inside dia		
US gal per min	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
10	.45	.00	.070	.43	.00	.063	.52	.00	.083	.77	.01	.256
15	.68	.01	.149	.65	.01	.134	.78	.01	.176	1.16	.02	.543
20	.91	.01	.254	.87	.01	.227	1.04	.02	.299	1.54	.04	.924
25	1.13	.02	.383	1.09	.02	.344	1.30	.03	.452	1.98	.06	1.40
30	1.36	.03	.537	1.30	.03	.481	1.56	.04	.633	2.32	.08	1.96
35	1.59	.04	.714	1.52	.04	.640	1.82	.05	.842	2.70	.11	2.60
40	1.82	.05	.914	1.74	.05	.820	2.08	.07	1.08	3.09	.15	3.33
45	2.04	.06	1.14	1.95	.06	1.02	2.34	.09	1.34	3.47	.19	4.14
50	2.27	.08	1.38	2.17	.07	1.24	2.60	.11	1.63	3.86	.23	5.03
55	2.50	.10	1.64	2.39	.09	1.47	2.86	.13	1.94	4.25	.28	5.99
60	2.72	.12	1.94	2.60	.11	1.74	3.12	.15	2.28	4.63	.33	7.05
65	2.95	.14	2.24	2.82	.12	2.01	3.38	.18	2.65	5.02	.39	8.18
70	3.18	.16	2.57	3.04	.14	2.31	3.64	.21	3.04	5.40	.45	9.38
75	3.40	.18	2.92	3.25	.16	2.62	3.90	.24	3.45	5.79	.52	10.7
80	3.63	.20	3.30	3.47	.19	2.96	4.16	.27	3.89	6.18	.59	12.0
85	3.86	.23	3.69	3.69	.21	3.31	4.42	.30	4.35	6.56	.67	13.4
90	4.09	.26	4.10	3.91	.24	3.67	4.68	.34	4.83	6.95	.75	14.9
95	4.31	.29	4.53	4.12	.26	4.06	4.94	.38	5.34	7.34	.84	16.5
100	4.54	.32	4.98	4.34	.29	4.47	5.21	.42	5.87	7.72	.93	18.1
110	4.99	.39	5.94	4.77	.35	5.33	5.73	.51	7.01	8.49	1.12	21.6
120	5.45	.46	6.98	5.21	.42	6.26	6.25	.61	8.23	9.27	1.34	25.4
130	5.90	.54	8.09	5.64	.49	7.26	6.77	.71	9.54	10.0	1.6	29.5
140	6.35	.63	9.28	6.08	.57	8.32	7.29	.83	10.9	10.8	1.8	33.8
150	6.81	.72	10.6	6.51	.66	9.48	7.81	.95	12.5	11.6	2.1	38.5
160	7.26	.82	11.9	6.94	.75	10.7	8.33	1.08	14.0	12.4	2.4	43.3
180	8.16	1.03	14.8	7.81	.95	13.2	9.37	1.36	17.4	13.9	3.0	52.6
200	9.08	1.28	18.0	8.68	1.17	16.1	10.4	1.7	21.2	15.4	3.7	65.4
220	9.99	1.55	21.4	9.55	1.42	19.2	11.5	2.1	25.3	17.0	4.5	78.0
240	10.9	1.8	25.2	10.4	1.7	22.6	12.5	2.4	29.7	18.5	5.3	91.6
260	11.8	2.2	29.2	11.3	2.0	26.2	13.5	2.8	34.4	20.1	6.3	106
280	12.7	2.5	33.5	12.2	2.3	30.0	14.6	3.3	39.5	21.6	7.3	122
300	13.6	2.9	38.0	13.0	2.6	34.1	15.6	3.8	44.8	23.2	8.4	138
320	14.5	3.3	42.8	13.9	3.0	38.4	16.7	4.3	50.5	24.7	9.5	156
340	15.4	3.7	47.9	14.8	3.4	43.0	17.7	4.9	56.5	26.3	10.7	175
360	16.3	4.1	53.3	15.6	3.8	47.8	18.7	5.4	62.8	27.8	12.0	194
380	17.3	4.6	58.9	16.5	4.2	52.8	19.8	6.1	69.4	29.3	13.3	214
400	18.2	5.1	64.7	17.4	4.7	58.0	20.8	6.7	76.3	30.9	14.8	236
420	19.1	5.7	70.8	18.2	5.1	63.5	21.9	7.5	83.5	32.4	16.3	258
440	20.0	6.2	77.2	19.1	5.7	69.2	22.9	8.2	91.1	34.0	18.0	281
460	20.9	6.8	83.8	20.0	6.2	75.3	23.9	8.9	98.9	35.5	19.6	305
480	21.8	7.4	90.7	20.8	6.7	81.3	25.0	9.7	107	37.1	21.4	330
500	22.7	8.0	97.8	21.7	7.3	87.7	26.0	10.5	115	38.6	23.2	356
550	25.0	9.7	117	23.9	8.9	105	28.6	12.7	138	42.5	28.1	425
600	27.2	11.5	137	26.0	10.5	123	31.2	15.1	162	46.3	33.3	499
650	29.5	13.5	159	28.2	12.4	143	33.8	17.8	187	50.2	39.0	579

Friction Losses In Pipe; C = 100
3½ Inch

FLOW	Cast Iron			Std Wt Steel			Extra Strong Steel			Double Extra Strong Steel		
	3.5 inside dia			3.548" inside dia			3.364" inside dia			2.728" inside dia		
US gal per min	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
15	.50	.00	.061	.49	.00	.058	.54	.00	.075	.82	.01	.208
20	.67	.01	.120	.65	.01	.112	.72	.01	.145	1.10	.02	.403
25	.84	.01	.181	.81	.01	.169	.90	.01	.220	1.37	.03	.609
30	1.00	.02	.254	.97	.01	.237	1.08	.02	.308	1.65	.04	.853
35	1.17	.02	.337	1.14	.02	.316	1.26	.02	.409	1.92	.06	1.13
40	1.34	.03	.432	1.30	.03	.404	1.44	.03	.524	2.20	.08	1.45
45	1.51	.04	.537	1.46	.03	.503	1.63	.04	.651	2.47	.10	1.81
50	1.67	.04	.653	1.62	.04	.611	1.80	.05	.791	2.74	.12	2.19
60	2.01	.06	.914	1.95	.06	.856	2.17	.07	1.11	3.29	.17	3.07
70	2.34	.09	1.22	2.27	.08	1.14	2.53	.10	1.47	3.84	.23	4.09
80	2.68	.11	1.56	2.60	.11	1.46	2.89	.13	1.89	4.39	.30	5.23
90	3.01	.14	1.94	2.92	.13	1.81	3.25	.16	2.35	4.94	.38	6.51
100	3.33	.17	2.35	3.25	.16	2.20	3.61	.20	2.85	5.49	.47	7.91
110	3.68	.21	2.81	3.57	.20	2.63	3.97	.24	3.40	6.04	.57	9.43
120	4.02	.25	3.30	3.99	.25	3.08	4.33	.29	4.00	6.59	.67	11.1
130	4.35	.29	3.82	4.22	.28	3.58	4.69	.34	4.64	7.13	.79	12.8
140	4.68	.34	4.38	4.54	.32	4.10	5.05	.40	5.32	7.68	.92	14.7
150	5.02	.39	4.99	4.87	.37	4.67	5.41	.45	6.05	8.23	1.05	16.8
160	5.35	.44	5.61	5.19	.42	5.25	5.78	.52	6.81	8.78	1.20	18.9
170	5.69	.50	6.28	5.52	.47	5.85	6.14	.59	7.61	9.33	1.35	21.1
180	6.02	.56	6.98	5.85	.53	6.53	6.50	.66	8.46	9.88	1.52	22.9
190	6.35	.63	7.71	6.17	.59	7.22	6.85	.73	9.35	10.4	1.7	25.9
200	6.69	.70	8.48	6.50	.66	7.94	7.22	.81	10.3	11.0	1.9	28.5
220	7.35	.84	10.1	7.14	.79	9.47	7.94	.98	12.3	12.1	2.3	34.0
240	8.03	1.00	11.9	7.79	.94	11.1	8.66	1.17	14.4	13.2	2.7	39.9
260	8.70	1.18	13.8	8.44	1.11	12.9	9.38	1.37	16.7	14.3	3.2	46.3
280	9.36	1.36	15.8	9.09	1.28	14.8	10.1	1.6	19.2	15.4	3.7	53.1
300	10.0	1.55	18.0	9.74	1.47	16.8	10.8	1.8	21.8	16.5	4.2	60.4
320	10.7	1.8	20.2	10.4	1.7	18.9	11.5	2.1	24.5	17.6	4.8	68.0
340	11.4	2.0	22.6	11.0	1.9	21.2	12.3	2.4	27.4	18.7	5.4	76.1
360	12.0	2.2	25.2	11.7	2.1	23.5	13.0	2.6	30.5	19.8	6.1	84.6
380	12.7	2.5	27.8	12.3	2.4	26.0	13.7	2.9	33.7	20.9	6.8	93.5
400	13.4	2.8	30.6	13.0	2.6	28.6	14.4	3.2	37.1	22.0	7.5	103
420	14.0	3.1	33.5	13.6	2.9	31.3	15.2	3.6	40.6	23.0	8.2	112
440	14.7	3.4	36.5	14.3	3.2	34.1	15.9	3.9	44.2	24.1	9.0	123
460	15.4	3.7	39.6	14.9	3.5	37.1	16.6	4.3	48.0	25.2	9.9	133
480	16.1	4.0	42.8	15.6	3.8	40.1	17.3	4.7	51.9	26.3	10.7	144
500	16.7	4.3	46.2	16.2	4.1	43.3	18.1	5.1	56.0	27.4	11.7	155
550	18.4	5.3	55.1	17.9	5.0	51.6	19.9	6.2	66.8	30.2	14.2	185
600	20.1	6.3	64.7	19.5	5.9	60.6	21.7	7.3	78.5	32.9	16.8	218
650	21.8	7.4	75.1	21.1	6.9	70.2	23.5	8.6	91.0	35.7	19.8	252
700	23.4	8.5	86.1	22.7	8.0	80.6	25.3	9.4	104	38.4	22.9	289
750	25.1	9.8	97.8	24.4	9.3	91.5	27.1	11.4	119	41.2	26.4	329
800	26.8	11.2	110	26.0	10.5	103	28.9	13.0	131	43.9	30.0	371
850	28.4	12.5	123	27.6	11.8	115	30.7	14.6	150	46.7	33.9	414

The National Bureau of Standards has recommended the elimination of this pipe size.

Friction Losses In Pipe; C = 100
4 Inch

FLOW	Cast Iron			Std Wt Steel			Extra Strong Steel			Double Extra Strong Steel		
	4.0 inside dia			4.026" inside dia			3.826" inside dia			3.152" inside dia		
US gal per min	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
20	.51	.00	.063	.50	.00	.061	.56	.00	.078	.82	.01	.199
30	.77	.01	.132	.76	.01	.128	.84	.01	.164	1.23	.02	.422
40	1.02	.02	.226	1.01	.02	.219	1.12	.02	.280	1.65	.04	.719
50	1.28	.03	.341	1.26	.03	.330	1.40	.03	.423	2.06	.07	1.09
60	1.53	.04	.477	1.51	.04	.463	1.67	.04	.593	2.47	.10	1.52
70	1.79	.05	.635	1.76	.05	.615	1.95	.06	.789	2.88	.13	2.02
80	2.04	.06	.813	2.02	.06	.788	2.23	.08	1.01	3.29	.17	2.59
90	2.30	.08	1.01	2.27	.08	.980	2.51	.10	1.26	3.70	.21	3.22
100	2.55	.10	1.23	2.52	.10	1.19	2.79	.12	1.53	4.11	.26	3.92
110	2.81	.12	1.47	2.77	.12	1.42	3.07	.15	1.82	4.52	.32	4.67
120	3.06	.15	1.72	3.02	.14	1.67	3.35	.17	2.14	4.94	.38	5.49
130	3.32	.17	2.00	3.28	.17	1.93	3.63	.20	2.48	5.35	.44	6.36
140	3.57	.20	2.29	3.53	.19	2.22	3.91	.24	2.84	5.76	.52	7.30
150	3.83	.23	2.61	3.78	.22	2.53	4.19	.27	3.24	6.17	.59	8.31
160	4.08	.26	2.93	4.03	.25	2.84	4.47	.31	3.64	6.58	.67	9.34
170	4.34	.29	3.28	4.29	.29	3.18	4.75	.35	4.07	6.99	.76	10.5
180	4.60	.33	3.64	4.54	.32	3.53	5.02	.39	4.52	7.40	.85	11.6
190	4.86	.37	4.03	4.79	.36	3.90	5.30	.44	5.00	7.82	.95	12.8
200	5.11	.41	4.43	5.05	.40	4.29	5.58	.48	5.50	8.23	1.05	14.1
220	5.62	.49	5.28	5.55	.48	5.12	6.14	.59	6.56	9.05	1.27	16.8
240	6.13	.58	6.21	6.05	.57	6.01	6.70	.70	7.70	9.87	1.51	19.8
260	6.64	.69	7.20	6.55	.67	6.97	7.26	.82	8.93	10.7	1.8	22.9
280	7.15	.79	8.25	7.06	.77	8.00	7.82	.95	10.2	11.5	2.1	26.3
300	7.66	.91	9.38	7.57	.89	9.09	8.38	1.09	11.6	12.3	2.4	29.9
320	8.17	1.04	10.6	8.07	1.01	10.2	8.94	1.24	13.1	13.2	2.7	33.7
340	8.68	1.17	11.8	8.58	1.14	11.5	9.50	1.40	14.7	14.0	3.0	37.7
360	9.19	1.31	13.1	9.08	1.28	12.7	10.0	1.6	16.3	14.8	3.4	41.9
380	9.70	1.46	14.5	9.59	1.43	14.1	10.6	1.7	18.0	15.6	3.8	46.3
400	10.2	1.6	16.0	10.1	1.6	15.5	11.2	1.9	19.8	16.5	4.2	50.9
420	10.7	1.8	17.5	10.6	1.7	16.9	11.7	2.1	21.7	17.3	4.7	55.7
440	11.2	1.9	19.0	11.1	1.9	18.5	12.3	2.3	23.6	18.1	5.1	60.7
460	11.7	2.1	20.7	11.6	2.1	20.0	12.8	2.5	25.7	18.9	5.6	65.9
480	12.3	2.3	22.4	12.1	2.3	21.7	13.4	2.8	27.8	19.7	6.0	71.3
500	12.8	2.5	24.1	12.6	2.5	23.4	14.0	3.0	30.0	20.6	6.6	76.9
550	14.0	3.0	28.8	13.9	3.0	27.9	15.3	3.6	35.7	22.6	7.9	91.7
600	15.3	3.6	33.8	15.1	3.5	32.8	16.7	4.3	42.0	24.7	9.5	108
650	16.6	4.3	39.2	16.4	4.2	38.0	18.1	5.1	48.7	26.7	11.1	125
700	17.9	5.0	45.0	17.6	4.8	43.6	19.5	5.9	55.8	28.8	12.9	143
750	19.2	5.7	51.1	18.9	5.6	49.5	20.9	6.8	63.4	30.8	14.7	163
800	20.4	6.5	57.6	20.2	6.3	55.8	22.3	7.7	71.5	32.9	16.8	183
850	21.7	7.3	64.4	21.4	7.1	62.4	23.7	8.7	79.9	35.0	19.0	205
900	23.0	8.2	71.6	22.7	8.0	69.3	25.1	9.8	88.9	37.0	21.3	228
950	24.3	9.2	79.1	24.0	9.0	76.6	26.5	10.9	98.2	39.1	23.8	252
1000	25.5	10.1	87.0	25.2	9.9	84.3	27.9	12.1	108	41.1	26.3	277
1100	28.1	12.3	104	27.7	11.9	101	30.7	14.6	129	45.2	31.7	331

Friction Losses In Pipe; C = 100
5 Inch

FLOW	Cast Iron			Std Wt Steel			Extra Strong Steel			Double Extra Strong Steel		
	5.0 inside dia			5.047" inside dia			4.813" inside dia			4.063" inside dia		
US gal per min	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
30	.49	.00	.045	.48	.00	.043	.53	.00	.054	.75	.01	.124
40	.65	.01	.076	.64	.01	.073	.71	.01	.092	1.00	.02	.212
50	.82	.01	.115	.80	.01	.110	.88	.01	.139	1.24	.02	.320
60	.98	.01	.161	.96	.01	.154	1.06	.02	.194	1.49	.03	.448
70	1.14	.02	.214	1.12	.02	.205	1.23	.02	.258	1.74	.05	.596
80	1.31	.03	.275	1.28	.03	.262	1.41	.03	.330	1.99	.06	.763
90	1.47	.03	.341	1.44	.03	.326	1.59	.04	.411	2.24	.08	.948
100	1.63	.04	.415	1.60	.04	.396	1.76	.05	.499	2.49	.10	1.15
120	1.96	.06	.581	1.92	.06	.555	2.11	.07	.700	2.98	.14	1.61
140	2.29	.08	.773	2.24	.08	.739	2.47	.09	.931	3.48	.19	2.15
160	2.61	.11	.990	2.56	.10	.946	2.82	.12	1.19	3.98	.25	2.75
180	2.94	.13	1.23	2.88	.13	1.18	3.17	.16	1.48	4.48	.31	3.41
200	3.27	.16	1.50	3.20	.16	1.43	3.52	.19	1.80	4.98	.39	4.15
220	3.59	.20	1.78	3.52	.20	1.70	3.88	.23	2.15	5.47	.47	4.95
240	3.92	.24	2.10	3.85	.23	2.00	4.23	.28	2.52	5.97	.55	5.82
260	4.25	.28	2.43	4.17	.27	2.32	4.58	.33	2.92	6.47	.65	6.75
280	4.58	.33	2.79	4.49	.31	2.66	4.94	.38	3.35	6.97	.76	7.74
300	4.90	.38	3.17	4.81	.36	3.03	5.29	.43	3.81	7.46	.87	8.79
320	5.23	.43	3.57	5.13	.41	3.41	5.64	.49	4.29	7.96	.98	9.91
340	5.56	.48	3.99	5.45	.46	3.81	5.99	.56	4.80	8.46	1.11	11.1
360	5.89	.54	4.44	5.77	.52	4.24	6.35	.63	5.34	8.96	1.25	12.3
380	6.22	.60	4.90	6.09	.58	4.68	6.70	.70	5.90	9.45	1.39	13.6
400	6.54	.66	5.39	6.41	.64	5.15	7.05	.77	6.49	9.95	1.54	15.0
420	6.87	.73	5.90	6.73	.70	5.64	7.40	.85	7.10	10.4	1.7	16.4
440	7.20	.81	6.43	7.05	.77	6.14	7.76	.94	7.74	10.9	1.8	17.8
460	7.52	.88	6.98	7.38	.85	6.67	8.11	1.02	8.40	11.4	2.0	19.4
480	7.85	.96	7.55	7.70	.92	7.22	8.46	1.11	9.09	11.9	2.2	21.0
500	8.17	1.04	8.15	8.02	1.00	7.79	8.82	1.21	9.81	12.4	2.4	22.6
550	8.99	1.26	9.72	8.82	1.21	9.28	9.70	1.46	11.7	13.7	2.9	27.0
600	9.80	1.49	11.7	9.62	1.49	10.9	10.6	1.7	13.7	14.9	3.5	31.7
650	10.6	1.7	13.2	10.4	1.7	12.6	11.5	2.1	15.9	16.2	4.1	36.8
700	11.4	2.0	15.2	11.2	1.9	14.5	12.3	2.4	18.3	17.4	4.7	42.2
750	12.3	2.4	17.2	12.0	2.2	16.5	13.2	2.7	20.8	18.7	5.4	47.9
800	13.1	2.7	19.4	12.8	2.5	18.6	14.1	3.1	23.4	19.9	6.2	54.0
850	13.9	3.0	21.7	13.6	2.9	20.8	15.0	3.5	26.2	21.1	6.9	60.4
900	14.7	3.4	24.2	14.4	3.2	23.1	15.9	3.9	29.1	22.4	7.8	67.1
950	15.5	3.7	26.7	15.2	3.6	25.5	16.7	4.3	32.2	23.6	8.7	75.9
1000	16.3	4.1	29.4	16.0	4.0	28.1	17.6	4.8	35.4	24.9	9.6	81.6
1100	18.0	5.0	35.0	17.6	4.8	33.5	19.4	5.8	42.2	27.4	11.7	97.3
1200	19.6	6.0	41.1	19.2	5.7	39.3	21.1	6.9	49.5	29.8	13.8	114
1300	21.2	7.0	47.7	20.8	6.7	45.6	22.9	8.2	57.4	32.3	16.2	133
1400	22.9	8.1	54.7	22.4	7.8	52.3	24.7	9.5	65.9	34.8	18.8	152
1500	24.5	9.3	62.2	24.0	9.0	59.4	26.4	10.8	74.8	37.3	21.6	173
1600	26.1	10.6	70.1	25.6	10.2	66.9	28.2	12.4	84.3	39.8	24.6	195
1700	27.8	12.0	78.4	27.2	11.5	74.9	30.0	14.0	94.4	42.3	27.8	218

Friction Losses In Pipe; C = 100
6 Inch

FLOW	Cast Iron			Std Wt Steel			Extra Strong Steel			Double Extra Strong Steel		
	6.0 inside dia			6.065" inside dia			5.761" inside dia			4.897" inside dia		
US gal per min	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
50	.57	.01	.047	.56	.00	.045	.62	.01	.058	.85	.01	.127
60	.68	.01	.066	.67	.01	.063	.74	.01	.081	1.02	.02	.178
70	.79	.01	.088	.78	.01	.084	.86	.01	.108	1.19	.02	.237
80	.91	.01	.113	.89	.01	.107	.98	.01	.138	1.36	.03	.304
90	1.02	.02	.141	1.00	.02	.133	1.11	.02	.171	1.53	.04	.378
100	1.13	.02	.171	1.11	.02	.162	1.23	.02	.208	1.70	.05	.459
120	1.36	.03	.239	1.33	.03	.227	1.48	.03	.292	2.04	.06	.643
140	1.59	.04	.318	1.56	.04	.302	1.72	.05	.388	2.38	.09	.855
160	1.82	.05	.408	1.78	.05	.387	1.97	.06	.497	2.72	.12	1.10
180	2.04	.06	.507	2.00	.06	.481	2.22	.08	.618	3.06	.15	1.36
200	2.27	.08	.616	2.22	.08	.584	2.46	.09	.751	3.40	.18	1.65
220	2.50	.10	.735	2.44	.09	.697	2.71	.11	.895	3.74	.22	1.97
240	2.72	.12	.863	2.67	.11	.819	2.96	.14	1.03	4.08	.26	2.32
260	2.95	.14	1.00	2.89	.13	.950	3.20	.16	1.22	4.42	.30	2.69
280	3.18	.16	1.15	3.11	.15	1.09	3.45	.19	1.40	4.77	.35	3.08
300	3.40	.18	1.30	3.33	.17	1.24	3.69	.21	1.59	5.11	.41	3.50
320	3.64	.21	1.47	3.56	.20	1.39	3.94	.24	1.79	5.45	.46	3.95
340	3.86	.23	1.64	3.78	.22	1.56	4.19	.27	2.00	5.79	.52	4.42
360	4.08	.26	1.83	4.00	.25	1.73	4.43	.31	2.23	6.13	.58	4.91
380	4.31	.29	2.02	4.22	.28	1.92	4.68	.34	2.46	6.47	.65	5.43
400	4.55	.32	2.22	4.44	.31	2.11	4.93	.38	2.71	6.81	.72	5.97
450	5.11	.41	2.76	5.00	.39	2.62	5.54	.48	3.36	7.66	.91	7.42
500	5.68	.50	3.36	5.56	.48	3.19	6.16	.59	4.09	8.51	1.13	9.02
550	6.25	.61	4.00	6.11	.58	3.80	6.77	.71	4.88	9.37	1.36	10.8
600	6.81	.72	4.70	6.66	.69	4.46	7.39	.85	5.73	10.2	1.6	12.6
650	7.38	.85	5.45	7.22	.81	5.17	8.00	.99	6.64	11.1	1.9	14.6
700	7.95	.98	6.25	7.78	.94	5.93	8.63	1.16	7.62	11.9	2.2	16.8
750	8.52	1.13	7.10	8.34	1.08	6.74	9.24	1.33	8.66	12.8	2.5	19.1
800	9.08	1.28	8.00	8.90	1.23	7.60	9.85	1.51	9.75	13.6	2.9	21.5
850	9.65	1.45	8.95	9.45	1.39	8.50	10.5	1.7	10.9	14.5	3.3	24.1
900	10.2	1.6	9.95	10.0	1.6	9.44	11.1	1.9	12.1	15.3	3.7	26.7
950	10.8	1.8	11.0	10.5	1.7	10.2	11.7	2.1	13.4	16.2	4.1	29.6
1000	11.4	2.0	12.1	11.1	1.9	11.5	12.3	2.4	14.7	17.0	4.5	32.5
1100	12.5	2.4	14.4	12.2	2.3	13.7	13.5	2.8	17.6	18.7	5.4	38.8
1200	13.6	2.9	16.9	13.3	2.7	16.1	14.8	3.4	20.7	20.4	4.5	45.5
1300	14.8	3.4	19.7	14.4	3.2	18.6	16.0	4.0	23.9	22.1	7.6	52.8
1400	15.9	3.9	22.5	15.6	3.8	21.4	17.2	4.6	27.5	23.8	8.8	60.6
1500	17.0	4.5	25.6	16.7	4.3	24.3	18.5	5.3	31.2	25.5	10.1	68.8
1600	18.2	5.1	28.9	17.8	4.9	27.4	19.7	6.0	35.2	27.2	11.5	77.5
1700	19.3	5.8	32.3	18.9	5.6	30.6	20.9	6.8	39.4	28.9	13.0	86.8
1800	20.4	6.5	35.9	20.0	6.2	34.0	22.2	7.7	43.7	30.6	14.6	96.4
1900	21.5	7.2	39.7	21.1	6.9	37.6	23.4	8.4	48.3	32.3	16.2	107
2000	22.7	8.0	43.6	22.2	7.7	41.4	24.6	9.4	53.1	34.0	18.0	117
2200	25.0	9.7	52.0	24.4	9.3	49.4	27.1	11.4	63.4	37.4	21.7	140
2400	27.2	11.5	61.1	26.7	11.1	58.0	29.6	13.6	74.5	40.8	25.9	164

Friction Losses In Pipe; C = 100
8 Inch

FLOW	Cast Iron			Std Wt Steel			Extra Strong Steel			Double Extra Strong Steel.		
	8.0 inside dia			7.981″ inside dia			7.625″ inside dia			6.875″ inside dia		
US gal per min	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
130	.83	.01	.069	.83	.01	.069	.91	.01	.087	1.12	.02	.143
140	.90	.01	.079	.90	.01	.079	.98	.01	.099	1.21	.02	.164
150	.96	.01	.089	.96	.01	.091	1.05	.02	.116	1.30	.03	.187
160	1.02	.02	.101	1.03	.02	.102	1.12	.02	.127	1.38	.03	.210
170	1.08	.02	.112	1.09	.02	.114	1.19	.02	.142	1.47	.03	.235
180	1.15	.02	.125	1.15	.02	.126	1.26	.02	.158	1.56	.04	.261
190	1.21	.02	.138	1.22	.02	.140	1.33	.03	.175	1.64	.04	.289
200	1.28	.03	.152	1.28	.03	.154	1.41	.03	.192	1.73	.05	.318
220	1.40	.03	.181	1.41	.03	.183	1.55	.04	.229	1.90	.06	.379
240	1.53	.04	.213	1.54	.04	.215	1.69	.04	.269	2.07	.07	.445
260	1.66	.04	.247	1.67	.04	.250	1.83	.05	.312	2.25	.08	.516
280	1.79	.05	.283	1.80	.05	.286	1.97	.06	.358	2.42	.09	.592
300	1.91	.06	.322	1.92	.06	.325	2.11	.07	.406	2.59	.10	.672
350	2.24	.08	.428	2.24	.08	.433	2.46	.09	.540	3.02	.14	.894
400	2.56	.10	.548	2.57	.10	.554	2.81	.12	.692	3.46	.19	1.14
450	2.87	.13	.681	2.88	.13	.689	3.16	.15	.860	3.89	.24	1.42
500	3.19	.16	.828	3.20	.16	.838	3.51	.19	1.05	4.32	.29	1.73
550	3.51	.19	.987	3.52	.19	.999	3.86	.23	1.25	4.75	.35	2.06
600	3.83	.23	1.16	3.85	.23	1.17	4.22	.28	1.46	5.19	.42	2.42
650	4.15	.27	1.34	4.17	.27	1.36	4.57	.32	1.70	5.62	.49	2.78
700	4.47	.31	1.54	4.49	.31	1.56	4.92	.38	1.95	6.05	.57	3.22
750	4.79	.36	1.75	4.81	.36	1.77	5.27	.43	2.21	6.49	.65	3.66
800	5.11	.41	1.97	5.13	.41	1.99	5.62	.49	2.49	6.91	.74	4.13
850	5.43	.46	2.21	5.45	.46	2.23	5.97	.55	2.79	7.35	.84	4.62
900	5.75	.51	2.46	5.77	.52	2.48	6.32	.62	3.10	7.78	.94	5.13
950	6.06	.57	2.71	6.09	.58	2.74	6.67	.69	3.43	8.21	1.05	5.67
1000	6.38	.63	2.98	6.41	.64	3.02	7.03	.77	3.77	8.64	1.16	6.24
1100	7.03	.77	3.56	7.05	.77	3.60	7.83	.95	4.49	9.50	1.40	7.44
1200	7.66	.91	4.18	7.69	.92	4.23	8.43	1.10	5.28	10.4	1.7	8.74
1300	8.30	1.07	4.85	8.33	1.08	4.90	9.13	1.30	6.12	11.2	2.0	10.4
1400	8.95	1.24	5.56	8.97	1.25	5.62	9.83	1.50	7.02	12.1	2.3	11.6
1500	9.58	1.43	6.32	9.61	1.44	6.39	10.5	1.7	7.98	13.0	2.6	13.2
1600	10.2	1.6	7.12	10.3	1.7	7.20	11.2	2.0	8.99	13.8	3.0	14.9
1800	11.5	2.1	8.85	11.5	2.1	8.95	12.6	2.5	11.2	15.6	3.8	18.6
2000	12.8	2.6	10.8	12.8	2.5	10.9	14.1	3.1	13.6	17.3	4.7	22.5
2200	14.1	3.1	12.8	14.1	3.1	13.0	15.5	3.7	16.6	19.0	5.6	26.8
2400	15.3	3.6	15.1	15.4	3.7	15.2	16.9	4.4	19.0	20.7	6.7	31.5
2600	16.6	4.3	17.5	16.7	4.3	17.7	18.3	5.2	22.1	22.5	7.9	36.5
2800	17.9	5.0	20.0	18.0	5.0	20.3	19.7	6.0	25.3	24.2	9.1	41.9
3000	19.1	5.7	22.8	19.2	5.7	23.0	21.1	6.9	28.8	25.9	10.4	47.6
3500	22.4	7.8	30.3	22.4	7.8	30.6	24.6	9.4	38.3	30.2	14.2	63.3
4000	25.6	10.2	38.8	25.6	10.2	39.2	28.1	12.3	49.0	34.6	18.6	81.0
4500	28.7	12.8	48.2	28.8	12.9	48.8	31.6	15.5	60.9	38.9	23.5	101
5000	31.9	15.8	58.6	32.0	15.9	59.3	35.1	19.1	74.0	43.2	29.0	122
5500	35.1	19.1	69.9	35.3	19.4	70.7	38.6	23.2	88.3	47.5	35.1	146

Friction Losses In Pipe; C = 100
10 Inch

FLOW	Cast Iron			Standard Wt Steel			Extra Strong Steel		
	10.0″ inside dia			10.02″ inside dia			9.750″ inside dia		
US gal per min	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
180	.74	.01	.042	.73	.01	.042	.77	.01	.048
200	.82	.01	.051	.81	.01	.051	.86	.01	.058
220	.90	.01	.061	.89	.01	.061	.95	.01	.069
240	.98	.01	.072	.98	.01	.071	1.03	.02	.081
260	1.06	.02	.083	1.06	.02	.083	1.12	.02	.094
280	1.14	.02	.096	1.14	.02	.095	1.20	.02	.108
300	1.22	.02	.109	1.22	.02	.108	1.29	.03	.123
350	1.43	.03	.144	1.42	.03	.143	1.50	.04	.163
400	1.63	.04	.185	1.63	.04	.183	1.72	.05	.209
450	1.84	.05	.230	1.83	.05	.228	1.93	.06	.260
500	2.04	.06	.280	2.04	.06	.277	2.15	.07	.316
550	2.24	.08	.333	2.24	.08	.330	2.36	.09	.377
600	2.45	.09	.392	2.44	.09	.388	2.58	.10	.443
650	2.65	.11	.454	2.64	.11	.450	2.79	.12	.514
700	2.86	.13	.521	2.85	.13	.516	3.01	.14	.589
800	3.26	.17	.667	3.25	.16	.660	3.46	.19	.754
900	3.67	.21	.829	3.66	.21	.821	3.87	.23	.938
1000	4.08	.26	1.01	4.07	.26	.998	4.30	.29	1.14
1100	4.49	.31	1.20	4.48	.31	1.19	4.73	.35	1.36
1200	4.90	.37	1.41	4.89	.37	1.40	5.16	.41	1.60
1300	5.31	.44	1.64	5.30	.44	1.62	5.59	.49	1.85
1400	5.71	.51	1.88	5.70	.50	1.86	6.01	.56	2.12
1500	6.12	.58	2.13	6.10	.58	2.11	6.44	.64	2.41
1600	6.53	.66	2.40	6.51	.66	2.38	6.88	.74	2.72
1700	6.94	.75	2.69	6.92	.74	2.66	7.30	.83	3.04
1800	7.35	.84	2.99	7.32	.83	2.96	7.74	.93	3.38
1900	7.76	.94	3.30	7.73	.93	3.27	8.16	1.03	3.74
2000	8.16	1.03	3.63	8.14	1.03	3.60	8.60	1.15	4.11
2200	8.98	1.25	4.33	8.95	1.24	4.29	9.45	1.39	4.90
2400	9.80	1.49	5.09	9.76	1.48	5.04	10.3	1.6	5.76
2600	10.6	1.7	5.90	10.6	1.7	5.84	11.2	1.9	6.67
2800	11.4	2.0	6.77	11.4	2.0	6.70	12.0	2.2	7.65
3000	12.2	2.3	7.69	12.2	2.3	7.61	12.9	2.6	8.70
3200	13.1	2.7	8.66	13.0	2.7	8.58	13.8	3.0	9.80
3400	13.9	3.0	9.69	13.8	3.0	9.60	14.6	3.3	11.0
3600	14.7	3.4	10.8	14.6	3.3	10.7	15.5	3.7	12.2
3800	15.5	3.7	11.9	15.5	3.7	11.8	16.3	4.1	13.5
4000	16.3	4.1	13.1	16.3	4.1	13.0	17.2	4.6	14.8
4500	18.4	5.3	16.3	18.3	5.2	16.1	19.3	5.8	18.4
5000	20.4	6.5	19.8	20.3	6.4	19.6	21.5	7.2	22.4
5500	22.4	7.8	23.6	22.4	7.8	23.4	23.6	8.7	26.7
6000	24.5	9.3	27.7	24.4	9.3	27.5	25.8	10.3	31.4
6500	26.5	10.9	32.1	26.4	10.8	31.8	27.9	12.1	36.4
7000	28.6	12.7	36.9	28.5	12.6	36.5	30.1	14.1	41.7
7500	30.6	14.6	41.9	30.5	14.5	41.5	32.2	16.1	47.4

Friction Losses In Pipe; C = 100
12 Inch

FLOW	Cast Iron 12.0" inside dia			Standard Wt Steel 12.000" inside dia			Extra Strong Steel 11.750" inside dia		
US gal per min	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
200	.57	.01	.021	.57	.01	.021	.59	.01	.023
250	.71	.01	.032	.71	.01	.032	.74	.01	.035
300	.85	.01	.045	.85	.01	.045	.89	.01	.050
350	.99	.02	.059	.99	.02	.059	1.03	.02	.066
400	1.14	.02	.076	1.14	.02	.076	1.18	.02	.084
450	1.28	.03	.095	1.28	.03	.095	1.33	.03	.105
500	1.42	.03	.115	1.42	.03	.115	1.48	.03	.128
550	1.56	.04	.137	1.56	.04	.137	1.63	.04	.152
600	1.70	.05	.161	1.70	.05	.161	1.77	.05	.179
700	1.99	.06	.214	1.99	.06	.214	2.07	.07	.238
800	2.27	.08	.275	2.27	.08	.275	2.37	.09	.304
900	2.56	.10	.341	2.56	.10	.341	2.66	.11	.378
1000	2.84	.13	.415	2.84	.13	.415	2.96	.14	.460
1100	3.12	.15	.495	3.12	.15	.495	3.25	.16	.548
1200	3.41	.18	.581	3.41	.18	.581	3.55	.20	.644
1300	3.69	.21	.674	3.69	.21	.674	3.84	.23	.747
1400	3.98	.25	.773	3.98	.25	.773	4.14	.27	.857
1500	4.26	.28	.878	4.26	.28	.878	4.44	.31	.973
1600	4.55	.32	.990	4.55	.32	.990	4.73	.35	1.10
1800	5.11	.41	1.23	5.11	.41	1.23	5.33	.44	1.36
2000	5.68	.50	1.50	5.68	.50	1.50	5.92	.54	1.66
2200	6.25	.61	1.78	6.25	.61	1.78	6.51	.66	1.98
2400	6.81	.72	2.10	6.81	.72	2.10	7.10	.78	2.32
2600	7.38	.85	2.43	7.38	.85	2.43	7.69	.92	2.69
2800	7.95	.98	2.78	7.95	.98	2.78	8.28	1.07	3.09
3000	8.52	1.13	3.17	8.52	1.13	3.17	8.88	1.23	3.51
3500	9.95	1.54	4.21	9.95	1.54	4.21	10.3	1.6	4.67
4000	11.4	2.0	5.39	11.4	2.0	5.39	11.8	2.2	5.97
4500	12.8	2.5	6.70	12.8	2.5	6.70	13.3	2.7	7.43
5000	14.2	3.1	8.15	14.2	3.1	8.15	14.8	3.4	9.03
5500	15.6	3.8	9.72	15.6	3.8	9.72	16.3	4.1	10.8
6000	17.0	4.5	11.4	17.0	4.5	11.4	17.7	4.9	12.6
6500	18.4	5.3	13.2	18.4	5.3	13.2	19.2	5.7	14.7
7000	19.9	6.2	15.2	19.9	6.2	15.2	20.7	6.7	16.8
7500	21.3	7.1	17.3	21.3	7.1	17.3	22.2	7.7	19.1
8000	22.7	8.0	19.4	22.7	8.0	19.4	23.7	8.7	21.5
8500	24.2	9.1	21.7	24.2	9.1	21.7	25.1	9.8	24.1
9000	25.6	10.2	24.2	25.6	10.2	24.2	26.6	11.0	26.8
9500	27.0	11.3	26.7	27.0	11.3	26.7	28.1	12.3	29.6
10000	28.4	12.5	29.4	28.4	12.5	29.4	29.6	13.6	32.5
11000	31.2	15.1	35.0	31.2	15.1	35.0	32.5	16.4	38.8
12000	34.1	18.1	41.2	34.1	18.1	41.2	35.5	19.6	45.6
13000	36.9	21.2	47.7	36.9	21.2	47.7	38.4	22.9	52.9
14000	39.8	24.6	54.7	39.8	24.6	54.7	41.4	26.6	60.6
15000	42.6	28.2	62.2	42.6	28.2	62.2	44.4	30.6	68.9

Friction Losses In Pipe; C = 100
14 Inch   16 Inch

FLOW	Cast Iron 14.0" inside dia			Steel 13.25" inside dia			FLOW	Cast Iron 16.0" inside dia			Steel 15.25" inside dia		
US gal per min	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	US gal per min	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
300	.63	.01	.021	.70	.01	.028	500	.80	.01	.028	.88	.01	.036
400	.84	.01	.036	.93	.01	.047	600	.96	.01	.040	1.05	.02	.050
500	1.04	.02	.054	1.16	.02	.071	700	1.12	.02	.053	1.23	.02	.067
600	1.25	.02	.076	1.40	.03	.100	800	1.28	.03	.068	1.41	.03	.086
700	1.46	.03	.101	1.63	.04	.132	900	1.44	.03	.084	1.58	.04	.106
800	1.67	.04	.130	1.86	.05	.170	1000	1.60	.04	.102	1.76	.05	.129
900	1.88	.05	.161	2.09	.07	.211	1200	1.92	.06	.143	2.11	.07	.181
1000	2.09	.07	.196	2.33	.08	.256	1400	2.24	.08	.191	2.46	.09	.241
1100	2.30	.08	.234	2.56	.10	.306	1600	2.56	.10	.250	2.81	.12	.308
1200	2.50	.10	.275	2.79	.12	.359	1800	2.87	.13	.304	3.16	.16	.383
1300	2.71	11	.318	3.02	.14	.416	2000	3.19	.16	.369	3.51	.19	.466
1400	2.92	.13	.365	3.26	.17	.477	2500	3.99	.25	.557	4.39	.30	.704
1500	3.13	.15	.415	3.49	.19	.542	3000	4.79	.36	.781	5.27	.43	.987
1600	3.34	.17	.478	3.72	.22	.611	3500	5.58	.48	1.06	6.15	.59	1.31
1700	3.54	.19	.523	3.95	.24	.684	4000	6.38	.63	1.33	7.03	.77	1.68
1800	3.75	.22	.581	4.19	.27	.760	4500	7.18	.80	1.65	7.91	.97	2.09
1900	3.96	.24	.643	4.42	.30	.840	5000	7.98	.99	2.01	8.79	1.2	2.54
2000	4.17	.27	.706	4.65	.34	.924	6000	9.58	1.43	2.82	10.5	1.7	3.56
2500	5.22	.42	1.07	5.81	.52	1.40	7000	11.2	1.9	3.75	12.3	2.4	4.73
3000	6.26	.61	1.50	6.98	.76	1.96	8000	12.8	2.5	4.79	14.1	3.1	6.06
3500	7.30	.83	1.99	8.15	1.03	2.60	9000	14.4	3.2	5.96	15.8	3.9	7.53
4000	8.34	1.08	2.55	9.31	1.35	3.32	10000	16.0	4.0	7.25	17.6	4.8	9.15
4500	9.4	1.37	3.17	10.5	1.7	4.13	11000	17.6	4.8	8.64	19.3	5.8	10.9
5000	10.4	1.7	3.85	11.6	2.1	5.03	12000	19.2	5.7	10.2	21.1	6.9	12.8
6000	12.5	2.4	5.39	14.0	3.0	7.05	13000	20.8	6.7	11.8	22.8	8.1	14.9
7000	14.6	3.3	7.17	16.3	4.1	9.38	14000	22.4	7.8	13.5	24.6	9.4	17.1
8000	16.7	4.3	9.18	18.6	5.4	12.0	15000	24.0	9.0	15.3	26.3	10.7	19.2
9000	18.8	5.5	11.4	20.9	6.8	14.9	16000	25.6	10.2	17.3	28.1	12.3	21.8
10000	20.9	6 8	13.9	23.3	8.4	18.1	18000	28.7	12.8	21.5	31.6	15.5	27.1
11000	23.0	8.2	16.5	25.6	10.2	21.6	20000	31.9	15.8	26.1	35.1	19.1	33.0
12000	25.0	9.7	19.4	27.9	12.1	25.4	22000	35.1	19.1	31.2	38.7	23.3	39.3
13000	27.1	11.4	22.5	30.2	14.2	29.5	24000	38.3	22.8	36.6	42.2	27.7	46.2
14000	29.2	13.3	25.9	32.6	16.5	33.8	25000	39.9	24.7	38.6	43.9	30.0	49.9
15000	31.3	15.2	29.4	34.9	18.9	38.4	26000	41.5	26.8	42.4	45.7	32.6	53.6
16000	33.4	17.3	33.1	37.2	21.5	43.3	28000	44.7	31.1	48.7	49.2	37.6	61.5
17000	35.4	19.5	37.0	39.5	24.2	48.4	30000	47.9	35.7	55.3	52.7	43.2	69.8
18000	37.5	21.6	41.2	41.9	27.3	53.8	32000	51.1	40.5	62.3	56.2	49.1	78.7
20000	41.7	27.0	50.0	46.5	33.6	65.4	34000	54.3	46	69.9	59.8	55.6	88.3
22000	45.9	32.7	59.7	51.2	40.7	78.0	36000	57.5	51	77.5	63.3	62.3	97.6
24000	50.0	38.9	70.1	55.8	48.4	91.6	38000	60.7	57	85.6	66.8	69.3	108

Friction Losses In Pipe; C = 100

18 Inch — 20 Inch

FLOW	Cast Iron			Steel			FLOW	Cast Iron			Steel		
	18.0″ inside dia			17.18″ inside dia				20.0″ inside dia			19.18″ inside dia		
U S gal per min	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	U S gal per min	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
500	.63	.01	.016	.69	.01	.020	800	.82	.01	.023	.89	.01	.028
600	.72	.01	.022	.83	.01	.028	1000	1.02	.02	.035	1.11	.02	.042
700	.89	.01	.030	.97	.01	.037	1200	1.23	.02	.048	1.33	.03	.059
800	1.01	.02	.038	1.11	.02	.048	1400	1.43	.03	.064	1.55	.04	.079
900	1.13	.02	.048	1.25	.02	.060	1500	1.64	.04	.082	1.78	.05	.101
1000	1.26	.02	.058	1.38	.03	.072	1800	1.83	.05	.102	2.00	.06	.126
1200	1.53	.04	.081	1.66	.04	.101	2000	2.04	.06	.125	2.22	.08	.153
1400	1.78	.05	.108	1.94	.06	.135	2500	2.55	.10	.188	2.78	.12	.231
1600	2.03	.06	.138	2.21	.08	.173	3000	3.06	.15	.264	3.33	.17	.323
1800	2.27	.08	.171	2.49	.10	.215	3500	3.57	.20	.351	3.89	.24	.430
2000	2.52	.10	.208	2.77	.12	.261	4000	4.08	.26	.449	4.45	.31	.551
2500	3.15	.15	.314	3.46	.19	.394	5000	5.10	.40	.679	5.55	.48	.832
3000	3.78	.22	.440	4.15	.27	.553	6000	6.13	.58	.951	6.67	.69	1.17
3500	4.41	.30	.586	4.85	.37	.735	7000	7.15	.79	1.26	7.78	.94	1.55
4000	5.04	.39	.750	5.54	.48	.941	8000	8.17	1.0	1.62	8.89	1.2	1.98
4500	5.67	.50	.932	6.23	.60	1.17	10000	10.2	1.6	2.45	11.1	1.9	3.00
5000	6.30	.62	1.13	6.92	.74	1.42	12000	12.3	2.4	3.43	13.3	2.7	4.20
6000	7.56	.89	1.59	8.31	1.1	1.99	14000	14.3	3.2	4.56	15.5	3.7	5.59
7000	8.83	1.2	2.11	9.70	1.5	2.65	15000	15.3	3.6	5.18	16.7	4.3	6.35
8000	10.1	1.6	2.70	11.1	1.9	3.39	16000	16.3	4.1	5.84	17.8	4.9	7.15
9000	11.3	2.0	3.36	12.5	2.4	4.22	18000	18.4	5.3	7.26	20.0	6.2	8.90
10000	12.6	2.5	4.08	13.8	3.0	5.12	20000	20.4	6.5	8.82	22.2	7.7	10.8
12000	15.3	3.6	5.72	16.6	4.3	7.18	22000	22.5	7.9	10.5	24.4	9.3	12.9
14000	17.8	4.9	7.61	19.4	5.8	9.55	24000	24.5	9.3	12.4	26.7	11.1	15.1
16000	20.3	6.4	9.74	22.1	7.6	12.2	25000	25.5	10.1	13.3	27.8	12.0	16.3
18000	22.7	8.0	12.1	24.9	9.6	15.2	26000	26.6	11.0	14.3	28.9	13.0	17.6
20000	25.2	9.9	14.7	27.7	11.9	18.5	28000	28.6	12.7	16.4	31.1	15.0	20.1
22000	27.7	11.9	17.6	30.3	14.3	22.0	30000	30.6	14.6	18.7	33.3	17.2	22.9
24000	30.6	14.6	20.6	33.2	17.1	25.9	32000	32.6	16.5	21.0	35.6	19.7	25.8
26000	32.8	16.7	23.9	36.0	20.1	30.0	34000	34.7	18.7	23.6	37.8	22.2	28.9
28000	35.5	19.6	27.4	38.8	23.4	34.4	35000	35.7	19.8	24.8	38.9	23.5	30.4
30000	37.8	22.2	31.2	41.5	26.8	39.1	36000	36.8	21.0	26.2	40.0	24.9	32.1
32000	40.6	25.6	35.1	44.3	30.5	44.1	38000	38.8	23.4	28.9	42.2	27.7	35.4
34000	42.8	28.5	39.4	47.1	34.5	49.4	40000	40.8	25.9	31.8	44.5	30.8	39.0
36000	45.4	32.0	43.7	49.9	38.7	54.8	45000	45.9	32.7	39.5	50.0	38.9	48.5
38000	47.9	35.7	48.3	52.6	43.0	60.6	50000	51.0	40	48.0	55.5	47.9	58.9
40000	50.4	39.5	53.1	55.4	47.1	66.6	55000	56.1	49	57.3	61.1	58	70.3
42000	52.9	43	58.1	58.2	52.6	72.9	60000	61.3	58	67.2	66.7	69	82.3
44000	55.4	48	63.3	61.0	57.8	79.4	65000	66.4	68	78.1	72.2	81	95.7
46000	58.0	52	68.7	63.7	63	86.3	70000	71.5	79	89.5	77.8	94	110

Friction Losses in Pipe; C = 100

24 in. inside dia					30 in. inside dia				
Discharge in U S gallons		Velocity feet per sec	Velocity head in ft	Head loss in feet per 100 ft	Discharge in U S gallons		Velocity feet per sec	Velocity head in ft	Head loss in feet per 100 ft
per min	per 24 hr				per min	per 24 hr			
350	505,000	.252	.00	.002	700	1,008,000	.322	.00	.002
700	1,008,000	.495	.00	.007	1000	1,440,000	.45	.00	.004
1000	1,440,000	.712	.01	.014	1300	1,872,000	.59	.01	.008
1400	2,016,000	.999	.01	.026	1700	2,448,000	.78	.01	.013
1700	2,448,000	1.21	.02	.038	2000	2,880,000	.91	.01	.017
2000	2,880,000	1.42	.03	.051	2400	3,456,000	1.095	.02	.023
2400	3,456,000	1.70	.045	.071	2700	3,888,000	1.23	.02	.030
2700	3,888,000	1.92	.06	.089	3100	4,464,000	1.43	.03	.039
3100	4,464,000	2.21	.08	.114	3400	4,896,000	1.55	.04	.046
3400	4,896,000	2.41	.09	.136	3800	5,472,000	1.74	.05	.057
3800	5,472,000	2.7	.113	.167	4100	5,904,000	1.87	.06	.065
4200	6,048,000	2.99	.13	.200	4500	6,480,000	2.05	.07	.077
4500	6,480,000	3.20	.16	.226	4800	6,912,000	2.19	.08	.087
4800	6,912,000	3.41	.18	.255	5200	7,488,000	2.37	.09	.101
5200	7,488,000	3.69	.21	.298	5500	7,920,000	2.51	.10	.112
5500	7,920,000	3.91	.24	.329	5900	8,496,000	2.69	.11	.127
5900	8,496,000	4.19	.27	.377	6200	8,928,000	2.83	.12	.139
6200	8,928,000	4.41	.32	.413	6900	9,936,000	3.12	.15	.170
6500	9,360,000	4.62	.33	.450	7600	10,944,000	3.47	.18	.203
6900	9,936,000	4.90	.37	.502	8300	11,952,000	3.79	.22	.240
7600	10,944,000	5.40	.50	.603	9000	12,960,000	4.10	.26	.278
8300	11,952,000	5.90	.54	.713	9700	13,968,000	4.42	.306	.319
9000	12,960,000	6.4	.64	.82	10000	14,400,000	4.56	.325	.337
9700	13,968,000	6.9	.74	.95	11000	15,840,000	5.01	.392	.401
10000	14,400,000	7.11	.79	1.00	12060	17,280,000	5.47	.47	.473
11000	15,840,000	7.82	.95	1.19	12500	18,000,000	5.70	.51	.51
12000	17,280,000	8.55	1.14	1.40	13000	18,720,000	5.94	.55	.55
12500	18,000,000	8.86	1.22	1.52	14000	20,160,000	6.4	.64	.63
13000	18,720,000	9.25	1.34	1.63	15000	21,600,000	6.85	.73	.71
14000	20,160,000	9.95	1.54	1.86	16000	23,040,000	7.30	.83	.81
15000	21,600,000	10.06	1.58	2.11	18000	25,920,000	8.20	1.05	1.00
16000	23,040,000	11.38	2.02	2.42	19000	27,360,000	8.67	1.17	1.11
18000	25,920,000	12.80	2.57	2.98	20000	28,800,000	9.12	1.30	1.22
19000	27,360,000	13.50	2.85	3.28	24000	34,600,000	10.09	1.89	1.73
20000	28,800,000	14.20	3.16	3.61	28000	40,320,000	12.75	2.46	2.27

Factor for correcting to other pipe sizes				Factor for correcting to other pipe sizes			
Dia in	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Dia in	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
23	1.089	1.186	1.230	29	1.078	1.161	1.199
22	1.190	1.416	1.527	28	1.148	1.318	1.399
21	1.306	1.706	1.916	27	1.235	1.524	1.670
				26	1.331	1.773	1.961

Friction Losses in Pipe; C = 100

36 in. inside dia

Discharge in U S gallons per min	Discharge in U S gallons per 24 hr	Velocity feet per sec	Velocity head in ft	Head loss in feet per 100 ft
1400	2,016,000	.44	.00	.004
1700	2,448,000	.53	.00	.005
2000	2,880,000	.63	.01	.007
2400	3,456,000	.75	.01	.010
2800	4,032,000	.88	.01	.015
3400	4,608,000	1.07	.02	.019
4000	5,040,000	1.26	.02	.026
4800	6,912,000	1.51	.04	.036
5600	8,054,000	1.76	.05	.048
6200	8,640,000	1.95	.06	.067
7000	10,080,000	2.20	.07	.072
7600	10,244,000	2.39	.09	.084
8300	11,952,000	2.61	.10	.098
9000	12,960,000	2.83	.12	.114
9700	13,968,000	3.05	.14	.131
10000	14,400,000	3.14	.16	.139
11000	15,840,000	3.46	.19	.164
12000	17,280,000	3.78	.22	.193
13000	18,720,000	4.09	.26	.226
14000	20,160,000	4.40	.30	.260
15000	21,600,000	4.71	.34	.294
16000	23,040,000	5.03	.39	.330
18000	25,920,000	5.66	.50	.412
19000	27,360,000	5.98	.56	.454
20000	28,800,000	6.30	.61	.504
21000	30,240,000	6.60	.67	.544
22000	31,680,000	6.92	.74	.590
23000	33,120,000	7.24	.81	.640
24000	34,560,000	7.55	.88	.695
26000	37,440,000	8.18	1.04	.806
28000	40,320,000	8.80	1.20	.935
30000	43,200,000	9.44	1.38	1.065
34000	48,960,000	10.70	1.77	1.340
38000	54,720,000	11.95	2.20	1.650
42000	60,480,000	13.20	2.70	1.990

Factor for correcting to other pipe sizes

Dia in	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
35	1.058	1.119	1.147
34	1.121	1.257	1.321
33	1.190	1.416	1.527
32	1.266	1.602	1.774

42 in. inside dia

Discharge in U S gallons per min	Discharge in U S gallons per 24 hr	Velocity feet per sec	Velocity head in ft	Head loss in feet per 100 ft
2000	2,880,000	.46	.003	.003
2500	3,600,000	.58	.005	.005
3000	4,320,000	.70	.007	.007
3500	5,040,000	.81	.010	.009
4000	5,760,000	.92	.013	.012
4500	6,480,000	1.04	.016	.015
5000	7,200,000	1.16	.021	.018
6000	8,640,000	1.39	.030	.025
7000	10,080,000	1.62	.041	.033
8000	11,520,000	1.85	.053	.043
9000	12,960,000	2.08	.067	.053
10000	14,400,000	2.31	.083	.065
12000	17,280,000	2.78	.120	.092
14000	20,160,000	3.24	.163	.122
16000	23,040,000	3.70	.212	.157
18000	25,920,000	4.16	.269	.194
20000	28,800,000	4.62	.331	.238
22000	31,680,000	5.10	.404	.282
24000	34,560,000	5.55	.477	.332
26000	37,440,000	6.02	.561	.383
28000	40,320,000	6.48	.651	.432
30000	43,200,000	6.94	.748	.501
32000	46,080,000	7.40	.850	.566
34000	48,960,000	7.86	.9	.632
36000	51,840,000	8.33	1.07	.702
38000	54,720,000	8.80	1.20	.778
40000	57,600,000	9.25	1.32	.855
42000	60,480,000	9.72	1.46	.936
44000	63,360,000	10.18	1.61	1.013
46000	66,240,000	10.63	1.76	1.100
48000	69,020,000	11.10	1.92	1.194
50000	72,000,000	11.58	2.08	1.280
52000	74,880,000	12.01	2.25	1.380
54000	77,760,000	12.49	2.41	1.490
56000	80,540,000	12.93	2.60	1.600

Factor for correcting to other pipe sizes

Dia in	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
41	1.049	1.101	1.124
40	1.102	1.216	1.268
39	1.160	1.345	1.434
38	1.222	1.492	1.627

Friction Losses in Pipe; C = 100

48 in. inside dia

Discharge in U S gallons per min	Discharge in U S gallons per 24 hr	Velocity feet per sec	Velocity head in ft	Head loss in feet per 100 ft
2000	2,880,000	.35	.002	.003
4000	5,760,000	.70	.008	.006
6000	8,640,000	1.05	.017	.012
8000	11,520,000	1.40	.030	.022
10000	14,400,000	1.76	.048	.034
12000	17,280,000	2.11	.069	.048
14000	20,160,000	2.47	.095	.064
16000	23,040,000	2.83	.124	.081
18000	25,920,000	3.18	.156	.101
20000	28,800,000	3.53	.193	.123
22000	31,680,000	3.89	.235	.147
24000	34,560,000	4.24	.279	.173
26000	37,440,000	4.60	.329	.201
28000	40,320,000	4.96	.383	.231
30000	43,200,000	5.32	.440	.262
32000	46,080,000	5.68	.500	.296
34000	48,960,000	6.03	.565	.331
36000	51,840,000	6.39	.632	.368
38000	54,720,000	6.75	.708	.406
40000	57,600,000	7.10	.782	.445
42000	60,480,000	7.44	.859	.487
44000	63,360,000	7.80	.943	.533
46000	66,240,000	8.16	1.04	.580
48000	69,020,000	8.51	1.12	.627
50000	72,000,000	8.87	1.22	.677
52000	74,880,000	9.22	1.32	.726
54000	77,760,000	9.58	1.42	.778
56000	80,540,000	9.94	1.54	.833
58000	83,520,000	10.28	1.64	.890
60000	86,400,000	10.63	1.76	.948
62000	89,280,000	10.99	1.87	1.070
64000	92,160,000	11.34	2.00	1.067
66000	95,040,000	11.70	2.12	1.130
68000	97,920,000	12.05	2.25	1.193
70000	100,800,000	12.41	2.39	1.258

Factor for correcting to other pipe sizes

Dia in	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
47	1.043	1.088	1.108
46	1.089	1.186	1.230
45	1.138	1.295	1.369
44	1.190	1.416	1.527

54 in. inside dia

Discharge in U S gallons per min	Discharge in U S gallons per 24 hr	Velocity feet per sec	Velocity head in ft	Head loss in feet per 100 ft
4000	5,760,000	.55	.005	.005
8000	11,520,000	1.11	.019	.012
12000	17,280,000	1.67	.043	.027
16000	23,040,000	2.23	.077	.046
20000	28,800,000	2.80	.122	.069
22000	31,680,000	3.07	.146	.083
24000	34,560,000	3.35	.174	.098
26000	37,440,000	3.63	.205	.113
28000	40,320,000	3.92	.238	.130
30000	43,200,000	4.20	.274	.148
32000	46,080,000	4.48	.311	.167
34000	48,960,000	4.76	.353	.186
36000	51,840,000	5.05	.395	.206
38000	54,720,000	5.33	.440	.227
40000	57,600,000	5.62	.490	.249
42000	60,480,000	5.89	.539	.272
44000	63,360,000	6.16	.590	.297
46000	66,240,000	6.44	.642	.323
48000	69,020,000	6.72	.701	.351
50000	72,000,000	7.00	.760	.380
52000	74,880,000	7.27	.820	.410
54000	77,760,000	7.56	.889	.441
56000	80,540,000	7.84	.952	.473
58000	83,520,000	8.12	1.02	.505
60000	86,400,000	8.40	1.10	.538
62000	89,280,000	8.68	1.17	.572
64000	92,160,000	8.96	1.25	.605
66000	95,040,000	9.24	1.32	.641
68000	97,920,000	9.52	1.40	.678
70000	100,800,000	9.80	1.49	.714
72000	103,680,000	10.07	1.57	.752
74000	106,560,000	10.35	1.66	.791
76000	109,440,000	10.62	1.76	.832
79000	112,320,000	10.91	1.85	.873
80000	115,200,000	11.19	1.94	.915

Factor for correcting to other pipe sizes

Dia in	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
53	1.038	1.078	1.095
52	1.078	1.163	1.202
51	1.121	1.257	1.321
50	1.166	1.360	1.452

Table 2-22: Cameron Hydraulic Data (concluded)

Friction Losses in Pipe; C = 100

60 in. inside dia				
Discharge in U S gallons per min	Discharge in U S gallons per 24 hr	Velocity feet per sec	Velocity head in ft	Head loss in feet per 100 ft
5000	7,200,000	.56	.005	.003
10000	14 400,000	1.12	.019	.011
15000	21,600,000	1.70	.045	.024
20000	28,800,000	2.26	.079	.042
25000	36,000,000	2.83	.124	.062
30000	43,200,000	3.40	.179	.088
32000	46,080,000	3.63	.205	.099
34000	48,960,000	3.86	.230	.111
36000	51,840,000	4.09	.259	.124
38000	54,720,000	4.32	.290	.137
40000	57,600,000	4.55	.320	.150
42000	60,480,000	4.78	.354	.164
44000	63,360,000	5.00	.387	.180
46000	66,240,000	5.22	.422	.196
48000	69,020,000	5.45	.460	.212
50000	72,000,000	5.68	.500	.229
52000	74,880,000	5.90	.540	.246
54000	77,760,000	6.12	.582	.263
56000	80,540,000	6.35	.626	.281
58000	83,520,000	6.58	.672	.299
60000	86,400,000	6.81	.720	.319
62000	89,280,000	7.03	.768	.339
64000	92,160,000	7.25	.819	.360
66000	95,040,000	7.49	.870	.381
68000	97,920,000	7.72	.925	.403
70000	100,800,000	7.95	.980	.425
72000	103,680,000	8.17	1.04	.447
74000	106,560,000	8.40	.10	.470
76000	109,440,000	8.62	1.15	.493
78000	112,320,000	8.86	1.22	.517
80000	115,200,000	9.06	1.28	.541
85000	122,400,000	9.64	1.44	.607
90000	129,600,000	10.20	1.61	.676
95000	136,800,000	10.78	1.80	.747
100000	144,000,000	11.36	2.00	.822

72 in. inside dia				
Discharge in U S gallons per min	Discharge in U S gallons per 24 hr	Velocity feet per sec	Velocity head in ft	Head loss in feet per 100 ft
10000	14,400,000	.78	.009	.005
20000	28,800,000	1.57	.038	.017
25000	36,000,000	1.97	.060	.026
30000	43,200,000	2.36	.086	.036
35000	50,400,000	2.76	.118	.048
40000	57,600,000	3.16	.154	.062
45000	64,800,000	3.54	.194	.077
50000	72,000,000	3.94	.240	.094
52000	74,880,000	4.09	.259	.100
54000	77,760,000	4.25	.280	.107
56000	80,540,000	4.41	.302	.114
58000	83,520,000	4.57	.324	.122
60000	86,400,000	4.73	.347	.130
62000	89,280,000	4.88	.370	.138
64000	92,160,000	5.04	.384	.146
66000	95,040,000	5.20	.420	.155
68000	97,920,000	5.36	.447	.164
70000	100,800,000	5.51	.473	.174
72000	103,680,000	5.67	.499	.183
74000	106,560,000	5.83	.528	.193
76000	109,440,000	5.99	.558	.203
78000	112,320,000	6.15	.588	.214
80000	115,200,000	6.31	.620	.225
82000	118,080,000	6.46	.650	.235
84000	120,960,000	6.62	.680	.245
86000	123,840,000	6.78	.712	.256
88000	126,720,000	6.93	.746	.266
90000	129,600,000	7.09	.780	.277
95000	136,800,000	7.49	.870	.306
100000	144,000,000	7.88	.965	.336
105000	151,200,000	8.28	1.06	.367
110000	158,400,000	8.67	1.16	.401
115000	165,600,000	9.05	1.27	.436
120000	172,800,000	9.45	1.38	.473
125000	180,000,000	9.85	1.51	.512

Factor for correcting to other pipe sizes				Factor for correcting to other pipe sizes			
Dia in	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Dia in	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
59	1.034	1.070	1.085	70	1.058	1.119	1.147
58	1.070	1.145	1.179	68	1.121	1.257	1.318
57	1.108	1.228	1.284	66	1.190	1.416	1.527
56	1.148	1.318	1.399	64	1.266	1.602	1.774

(text continued from page 141)

Step 2: Proceed left horizontally across the chart to the intersection, with:

Step 3: The 1,000-lb/h flowrate projected diagonally up from the bottom scale.

Step 4: Reading vertically up from this intersection, it can be seen that a 1-in. line will produce more than the allowed pressure drop, so a 1½-in. size is chosen.

Step 5: Read left horizontally to a pressure drop of 0.28 psi/100 ft on the left-hand scale.

Step 6: Note the velocity given by this line as 16.5 ft/s, then proceed to the insert on the right, and read upward from 600 psig to 200 psig to find the velocity correction factor as 0.41.

Step 7: Multiply 0.41 by 16.5 to get a corrected velocity of 6.8 ft/s.

The author has compared this method with Dukler [29] and others and reports good agreement for reasonably good cross section of flow regimes.

Nomenclature

A = Internal cross-section area for flow, sq ft; or area of orifice, nozzle, or pipe, sq ft.

a = Internal cross-section area for flow in pipe, sq. in.

a' = Fractional opening of control valve, generally assumed at 60% = 0.60

a_o = Orifice area, sq in.

a_w = Velocity of propagation of elastic vibration in the discharge pipe ft/sec = $4660/(1 + K_{hs}B_r)^{1/2}$

B = Base pressure drop for control valve from manufacturer, psi

B_r = Ratio of pipe diameter (ID) to wall thickness

C = Condensate, lbs/hr (Equation 2-133); or for pipe, Williams and Hazen constant for pipe roughness, (see Cameron Table 2-22 and Figure 2-24); or flow coefficient for sharp edged orifices

C' = Flow coefficient for orifices and nozzles which equal the discharge coefficient corrected for velocity of approach = $C_d/(1 - \beta^4)^{1/2}$

C' = C for Figures 2-17 and 2-18

$C' = c'$ = Orifice flow coefficient

C_d = Discharge coefficient for orifice and nozzles

C_{D1} = Diameter correction factor, vacuum flow, Figure 2-43

C_{D2} = Diameter correction factor, vacuum flow, Figure 2-43

C_v = Standard flow coefficient for valves; flow rate in gpm for 60°F water with 1.0 psi pressure drop across the valve, = $Q\{(\rho/62.4)\ (\Delta P)\}^{1/2}$

C'_v = Valve coefficient of flow, full open, from manufacturer's tables

C_{T1} = Temperature correction factor, vacuum flow, Figure 2-43

C_{T2} = Temperature correction factor, vacuum flow, Figure 2-43

C_1 = Discharge factor from chart in Figure 2-31

C_2 = Size factor from Table 2-11, use with equation on Figure 2-31

c_p/c_v = Ratio of specific heat at constant pressure to that at constant volume=k

D = Inside diameter of pipe, ft

D_H = Hydraulic diameter, ft

d = Inside diameter of pipe, in. = d_i

d_e = Equivalent or reference pipe diameter, in.

d_H = Hydraulic diameter, or equivalent diameter, in.

d_o = Orifice diameter, or nozzle opening, in.

d_{oo} = Diameter of a single line with the same delivery capacity as that of individual parallel lines d_1 and d_2 (lines of same length)

d_i = Inside diameter of pipe, in.

E = Gas transmission "efficiency" factor, varies with line size and surface internal condition of pipe

F = Factor in Babcock's steam flow equation

F_D = Friction pressure loss (total) at design basis, for a system, psi, for process equipment and piping, but excluding the control valve

F_e = Elevation factor for two-phase pipe line

F_M = Friction pressure loss (total) at maximum flow basis, for a system, psi

F_1 = Base friction factor, vacuum flow, Figure 2-43

F_2 = Base friction factor, vacuum flow, Figure 2-43

f = Friction factor, Moody or "regular" Fanning, see Note Figure 2-3

f_T = Turbulent friction factor, See Table 2-2

f_g = Moody or "regular" Fanning Friction for gas flow

f_{TP} = Two-phase friction for wave flow

$(1/f)^{1/2}$ = Gas transmission factor, or sometimes termed efficiency factor, see Table 2-15, f = Fanning friction factor

G = Mass flow rate of gas phase, pounds per hour per square foot of total pipe cross-section area

G' = Mass rate, lbs/(sec) (sq ft cross section)

GPM = Gallons per minute flow

g = Acceleration of gravity, 32.2 ft/(sec)^2

H = Total heat, Btu/lb

h = Average height of all vertical rises (or hills) in two-phase pipe line, ft

or, h = Static head loss, ft of fluid flowing

h_1 = Enthalpy of liquid at higher pressure, Btu/lb

h_2 = Enthalpy of liquid at lower or flash pressure, Btu/lb

$h_f = h_L$ = Loss of static pressure head due to friction of fluid flow, ft of liquid

h_p = Enthalpy of liquid at supply steam pressure, Btu/lb

h_r = Enthalpy of liquid at return line pressure, Btu/lb

$h_{L'}$ = Head at orifice, ft of liquid

h'_L = Differential static head or pressure loss across flange taps when C or C' values come from Figure 2-17 or Figure 2-18, ft of fluid

h_{wh} = Maximum pressure developed by hydraulic shock, ft of water (water hammer)

K = Resistance coefficient, or velocity head loss in equation, $h_L = Kv^2/2g$

K_d = Orifice or nozzle discharge coefficient

K_{hs} = Ratio of elastic modulus of water to that of the metal pipe material (water hammer)

k = Ratio of specific heat, c_p/c_v

L = Pipe, length, ft

L_e = Equivalent length of line of one size referenced to another size, miles, (or feet)

L_{eq} = Equivalent length of pipe plus equivalent length of fittings, valves, etc., ft.

L_m = Length of pipe, miles

L_v = Latent heat of evaporation of steam at flash pressure, Btu/lb

l = Horizontal distance from opening to point where flow stream has fallen one foot, in.

M = MW = molecular weight

MR = Universal gas constant

n = Number of vertical rises (or hills) in two-phase pipe line flow

or, n = Polytropic exponent in polytropic gas P-V relationship

P = Pressure, psig; or, pressure drop, P, pounds per square inch, Babcock Equation 2-82)

P_t = Absolute pressure, torr

ΔP_t = Pressure drop, torr

P' = Pressure, psi absolute (psia)

P_e = Total pressure at lower end of system, psig

P_{br} = Barometric pressure, psi absolute

P_S = Total pressure upstream (higher) of system, psig

P_s = Standard pressure for gas measurement, lbs/sq in. absolute, psia

p'' = Pressure, lbs/sq ft absolute; (in speed of sound equation, Equation 2-86), Note units.

p' = Gauge pressure, psig

or, P_1 = Initial pressure, in. of mercury absolute, vacuum system

ΔP = Pressure drop, lbs/sq in, psi; or static loss for flowing fluid, psi

ΔP_c = Pressure drop across a control valve, psi

ΔP_{vac} = Pressure drop in vacuum system due to friction, in. water/100 ft pipe

ΔP_{TPh} = Total two-phase pressure drop for system involving horizontal and vertical pipe, psi per foot of length

ΔP_{100} = Pressure drop, pounds per sq in per 100 ft of pipe or equivalent

Q = Flow rate, gallons per minute, gpm

Q_b = Flow rate, barrels/day

Q_D = Design flow rate, gpm, or ACFM

Q_M = Maximum flow rate, gpm, or ACFM

q = Flow rate at flowing conditions, cu ft/sec

q_d = Gas flow rate standard cubic feet per day, at 60°F and 14.7 psia (or 14.65 if indicated); or flow rate, cu ft/day at base conditions of T_s and P_s

q_{ds} = Gas flow at designated standard conditions, cu ft/day, cfD

q_h = Gas flow rate, cu ft/hr, at 60°F and 14.4 psiabs, (psia)

q' = Gas flow, cu ft/sec, at 14.7 psia and 60°F

q'_h = Flow rate at standard conditions (14.7 psia, and 60°F) cu ft/hr, SCFH

q_m = Flow rate cu ft/min

q'_m = Free air, cubic feet per minute @ 60°F and 14.7 psia

R = Individual gas constant = MR/M = 1544/M

R_e = Reynolds number, see Figure 2-3

R_H = Hydraulic radius, ft

R_c = Ratio of compression at entrance of pipe, Figure 2-37

r_c = Critical pressure ratio = P'_2/P'_1

S_g = Specific gravity of gas relative to air, (= ratio of molecular weight gas/29)

$S°$ = Degrees of superheat in a steam condition, degrees F above saturated (not the actual temperature)

s = Steam quality as percent dryness, fractional

SpGr = Specific gravity of fluid relative to water at same temperature

T = Absolute Rankin temperature, 460 + t, degrees R

T_s = Standard temperature for gas measurement, °R = 460 + t

T_1 = Average flowing temperature of gas, °R

t = Temperature, °F

t_s = Time interval required for the pressure wave to travel back and forth in a pipe, sec

V = Free air flow, cu ft/sec at 60°F and 14.7 psia

$\overline{V}$ = Specific volume of fluid, cu ft/lb

V' = Volume, cu ft

V_a = Volume, cu ft

v = Flow velocity (mean) or superficial velocity in pipe lines at flowing conditions for entire pipe cross section, ft/sec; or reduction in velocity, ft/sec (water hammer)

v_m = Mean velocity in pipe, at conditions of $\overline{V}$, ft/min

v_s = Sonic (critical) velocity in compressible fluid, ft/sec; or speed of sound, ft/sec

v_w = Reduction in velocity, ft/sec (actual flowing velocity, ft/sec)

W = Flow rate, lbs/hr

W_m = Mass flow rate of liquid phase, pounds per hour per square foot of total pipe cross-section area

W_t = Mass flow rate, lbs/hr/tube

w = Flow rate, lbs/min

w_s = Flow rate, lbs/sec; or sometimes, W_s

x = Fraction of initial line paralleled with new line

Y = Net expansion factor for compressible flow through orifices, nozzles, or pipe

Z = Compressibility factor for gases at average conditions, dimensionless. Omit for pressure under 100, psig

Greek Symbols

β = Ratio of internal diameter of smaller to large pipe sizes, or for orifices or nozzles, contractions or enlargements

γ' = Kinematic viscosity, sq ft/sec

γ = Surface tension of liquid, dynes/centimeter

ε = Roughness factor, effective height of pipe wall irregularities, ft, see Figure 2-11

θ = Angles of divergence or convergence in enlargements or contractions in pipe systems, degrees

λ = Two-phase flow term to determine probable type of flow = $[(\rho_g/0.075)\ (\rho_L/62.3)]^{1/2}$, where both liquid and gas phases are in turbulent flow (two-phase flow)

μ = Absolute viscosity, centipoise

μ_e = Absolute viscosity, lbs (mass)/(ft) (sec)

μ_g = Viscosity of gas or vapor phase, centipoise

μ_L = Viscosity of liquid phase, centipoise

ρ = Density of fluid, lbs/cu ft; or lb/gal, Eq. 2-113

Σ = Summation of items

ψ = Two-phase term = $(73/\gamma)\ [\mu_L\ (62.3/\rho_L)^2]^{1/3}$

ϕ = Equations for ϕ_{GTT} for two-phase pipe line flow

Subscripts

o = Base condition for gas measurement

1 = Initial or upstream or inlet condition, or i_i

2 = Second or downstream or outlet condition

a = Initial capacity or first condition

b = New capacity or second condition

g = Gas

L = Liquid

vc = Gradual contraction

VE = Gradual enlargement

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Chapter 3

Pumping of Liquids

Pumping of liquids is almost universal in chemical and petrochemical processes. The many different materials being processed require close attention to selection of materials of construction of the various pump parts, shaft sealing, and the hydraulics of the individual problems. A wide variety of types and sizes of pumps have been developed to satisfy the many special conditions found in chemical plant systems; however, since all of these cannot be discussed here, the omission of some does not mean that they may not be suitable for a service.

In general, the final pump selection and performance details are recommended by the manufacturers to meet the conditions specified by the process design engineer. It is important that the designer of the process system be completely familiar with the action of each pump offered for a service in order that such items as control instruments and valves may be properly evaluated in the full knowledge of the *system.*

This chapter presents information on rating, sizing, and specifying process pumps. The emphasis will be on centrifugal pumps, which are by far the most widely used in the process industries; however, applications of other types of pumps will also be discussed (see Table 3-1).

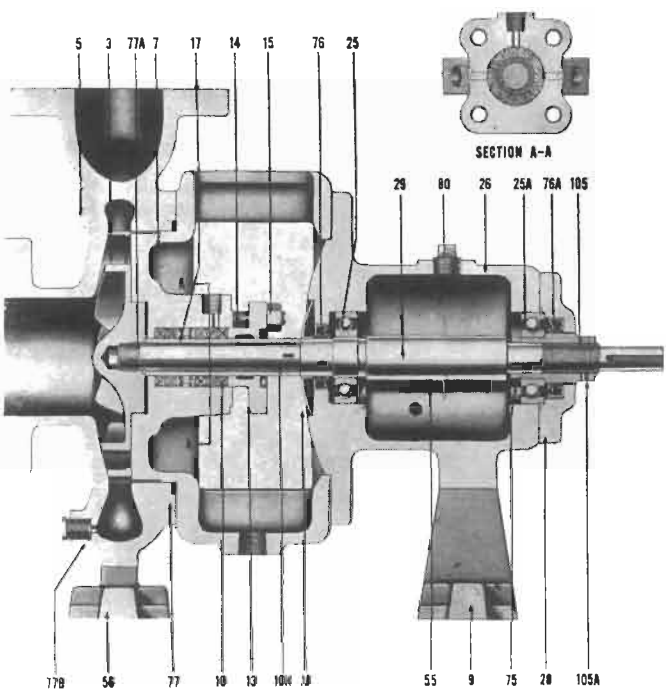
To properly accomplish a good and thorough rating/sizing of a centrifugal pump, the plant system designer should at a minimum:

1. understand the fundamentals of performance of the pump itself.
2. understand the mechanical details required for a pump to function properly in a system.
3. calculate the friction and any other pressure losses for each "side" of the pump, suction and discharge, (see Chapter 2).
4. determine the suction side and discharge side heads for the mechanical system connecting to the pump.
5. determine the important *available* net positive suction head ($NPSH_A$) for the pump suction side *mechanical* system, and compare this to the manufacturer's *required* net positive suction head ($NPSH_R$) by the pump itself. This requires that the designer make a tentative actual pump selection of one or more manufacturers in order to use actual numbers.
6. make allowable corrections to the pump's required NPSH (using charts where applicable) and compare with the available NPSH. The available must always be several feet greater than the corrected required.
7. make fluid viscosity corrections to the required performance if the fluid is more viscous than water.
8. examine specific speed index, particularly if it can be anticipated that future changes in the system may be required.
9. if fluid being pumped is at elevated temperature (usually above 90°F±), check temperature rise in pump, and the minimum flow required through the pump.
10. make pump brake horsepower corrections for fluids with a specific gravity different than water. Select actual driver (electric motor, usually) horsepower in order that horsepower losses between the driver and the pump shaft will still provide sufficient power to meet the pump's *input* shaft requirements.
11. if the pump has some unique specialty service or requirement, recognize these in the final sizing and selection. Consult a reliable manufacturer that produces pumps for the type of service and applications and have them verify the analysis of your system's application(s).

Table 3-1
General Types or Classification of Pumps

All types will not be treated in detail, but consideration of their particular features is important in many situations.

Centrifugal	Rotary	Reciprocating
1. Centrifugal	1. Cam	1. Piston
2. Propeller	2. Screw	2. Plunger
3. Mixed Flow	3. Gear	3. Diaphragm
4. Peripheral	4. Vane	
5. Turbine	5. Lobe	
6. Radial Flow	6. Piston	
7. Axial Flow	7. Flexible Rotor	



3	Impeller	26	Bearing Housing
5	Casing	28	Bearing End Cover
7	Back Head Cradle	29	Pump Shaft
9	Bearing Housing Foot	55	Oil Disc. (Flinger)
10	Shaft Sleeve	56	Casing Foot
10K	Shaft Sleeve Key	75	Retaining Ring
13	Stuffing Box Gland	76	Oil Seal—Front
14	Stuffing Box Gland Stud	76A	Oil Seal—Rear
15	Stuffing Box Gland Stud Nut	77	Gasket—Casing
		77A	Gasket—Sleeve
17	Seal Cage	77B	Gasket—Drain Plug
18	Splash Collar	80	Oil Vent
25	Shaft Bearing—Radial	105	Shaft Adjusting Sleeve
25A	Shaft Bearing—Thrust	105A	Sleeve Lock Nut

Figure 3-1. General Service Centrifugal Pump. (Courtesy Dean Brothers Pumps, Inc.)

The centrifugal pump (Table 3-2) develops its pressure by centrifugal force on the liquid passing through the pump and is generally applicable to high capacity, low to medium head installations. In order to satisfy pump discharge head (or pressure) requirements the unit may be a multistage (multiple impellers) instead of a single stage [28]. The conditions of pumping water vs. pumping hot light hydrocarbons require considerably different evaluation in pump design features for satisfactory operation, safety and maintenance.

The inline centrifugal process pump, Figure 3-3, is relatively new to general applications; however, it is finding many applications where space and installation costs are important. Each application must be carefully evaluated, as there are three basic types of pump construction to consider. Generally, for many applications the dimensions have been standardized through the American Voluntary Standard, ANSI, or API-610. The performance curves are typical of single stage centrifugal pumps.

The turbine is a special type of centrifugal pump (Figure 3-14) and has limited special purpose applications.

Pump Design Standardization

Certain pump designs have been standardized to aid manufacturer's problems, and to allow the owners to take advantage of standardization of parts and dimensions, and consequently maintain a more useful inventory. The standards are sponsored through the American National Standards Institute; however, many manufacturers also produce to the American Petroleum Institute and their own proprietary standards. These are special pumps that do not conform to all the standards, but are designed to accomplish specific pumping services.

The primary pump types for the chemical industry for horizontal and vertical inline applications have been standardized in ANSI B-123, ANSI Std # B73.1M for horizontal end suction centrifugal pumps, and ANSI B73.2M for vertical inline centrifugal pumps. The standards are in a continuous process of upgrading to suit requirements of industry and the manufacturers. The API-610 standard is primarily a heavy duty application, such as is used for the refinery and chemical industry requirements. This is the only true world pump [21] standard, although the International Organization for Standardization (ISO) is studying such an improved design [20].

The standards are important because they allow the dimensional interchangeability of pumps and shaft packing of different manufacturers, but only as long as the manufacturers conform to the standard.

(text continued on page 164)

A SECURE LUBRICATION

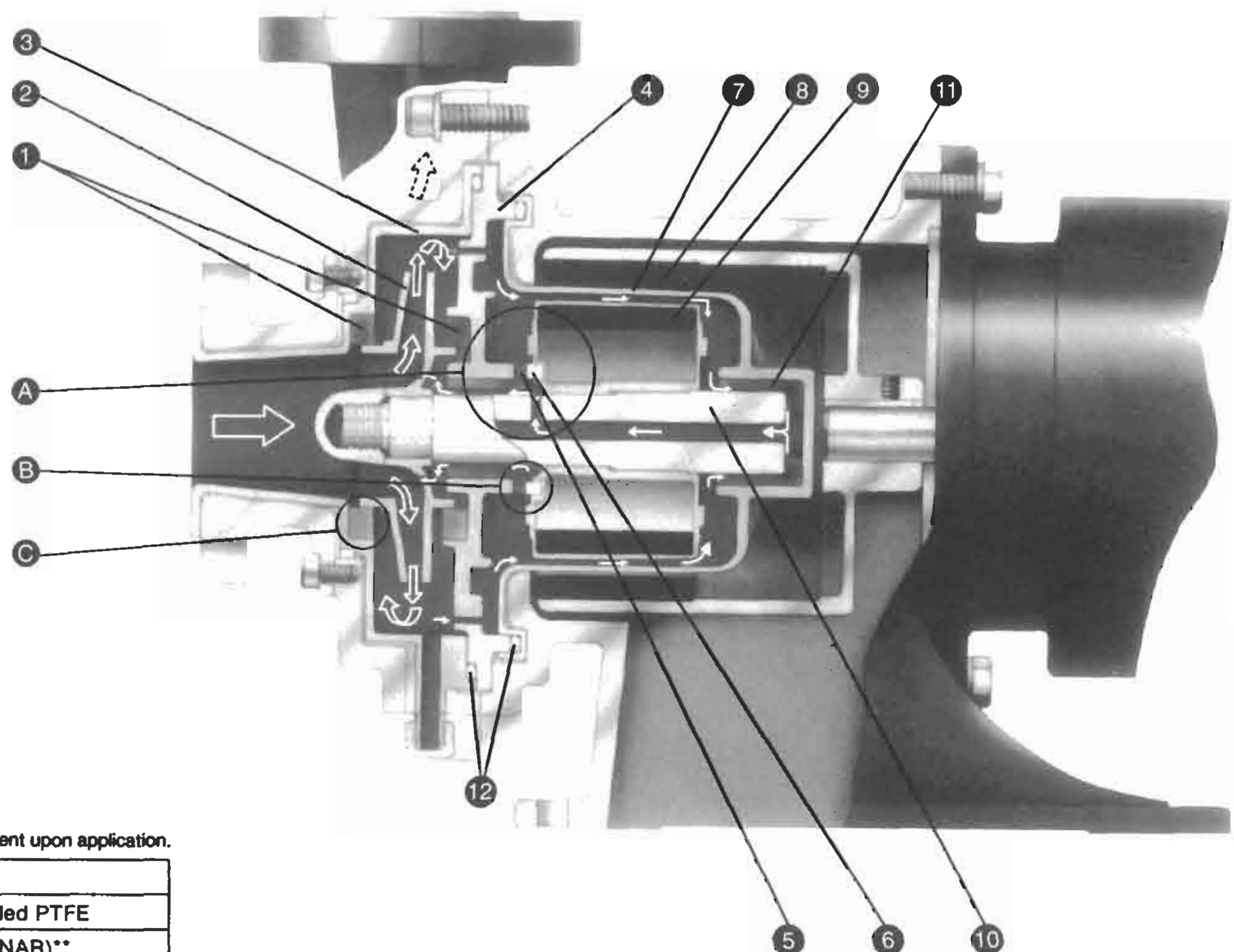
Liquid from the casing flows between the rear casing and the magnet lining which prevents clogging. The fluid is then forced to flow to the rear bushing then through the shaft to the front bushing. This flow guarantees perfect lubrication for the bushing.

B LONG LIFE THRUST RING

The thrust ring is connected to the magnet not to the impeller, which reduces its rotating velocity. The lower velocity increases the life of the thrust ring.

C NO ADHESIVE

All of the bushings as well as the liner ring and thrust ring are not attached by adhesives. Thus the parts are easily replaced and the parts are free from any weakness of the adhesive.

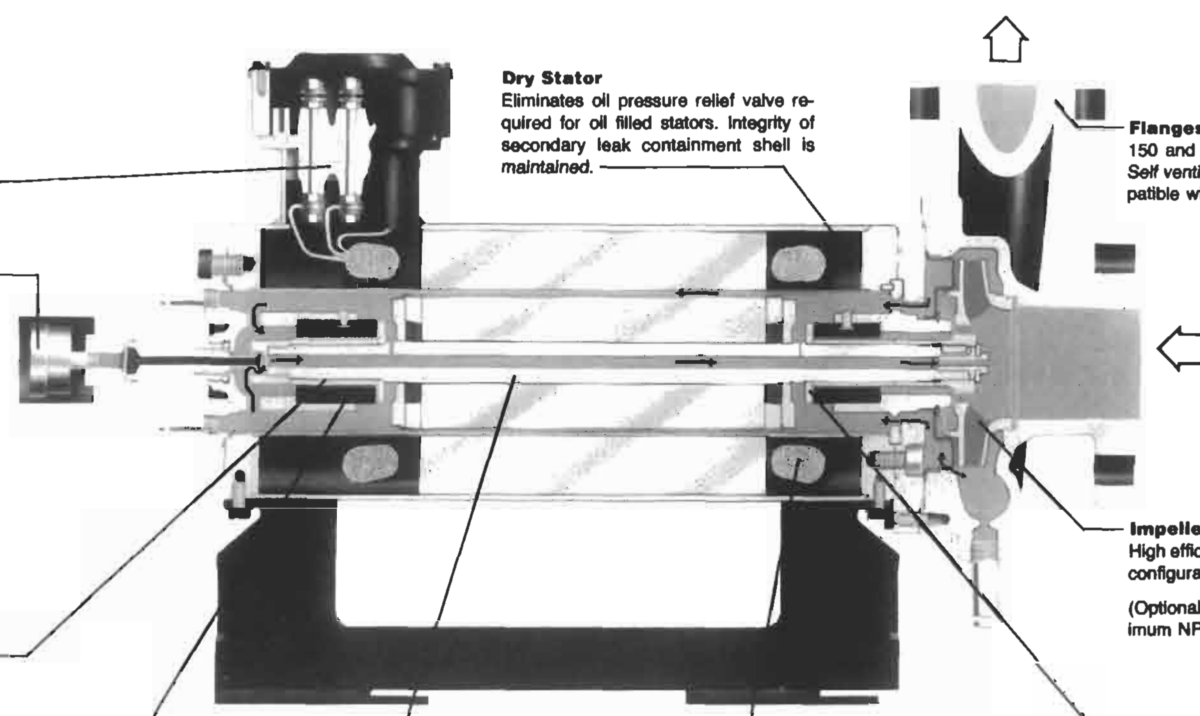


Standard materials: other materials available dependent upon application.

No.	Part Name	Material
1	Liner Ring	Carbon-filled PTFE
2	Impeller	PVDF (KYNAR)**
3	Casing	PVDF (KYNAR)
4	Bushing Plate	PVDF (KYNAR)
5	Front Bushing	Carbon-filled PTFE
6	Thrust Ring	CERAMIC
7	Rear Casing	Carbon-filled PVDF
8	Outer Magnet	
9	Inner Magnet	
10	Shaft	CERAMIC
11	Rear Bushing	Carbon-filled PTFE
12	"O" Ring	VITON

** Kynar is a registered trademark of Pennwalt Corp.

Figure 3-1A. Sealless Magnetic Drive Centrifugal Pump, no seals, no leakage, no coupling. Chemical resistance depends on materials of construction. See Table 3-2. (By permission, LaBour Pump Co.)



Motors

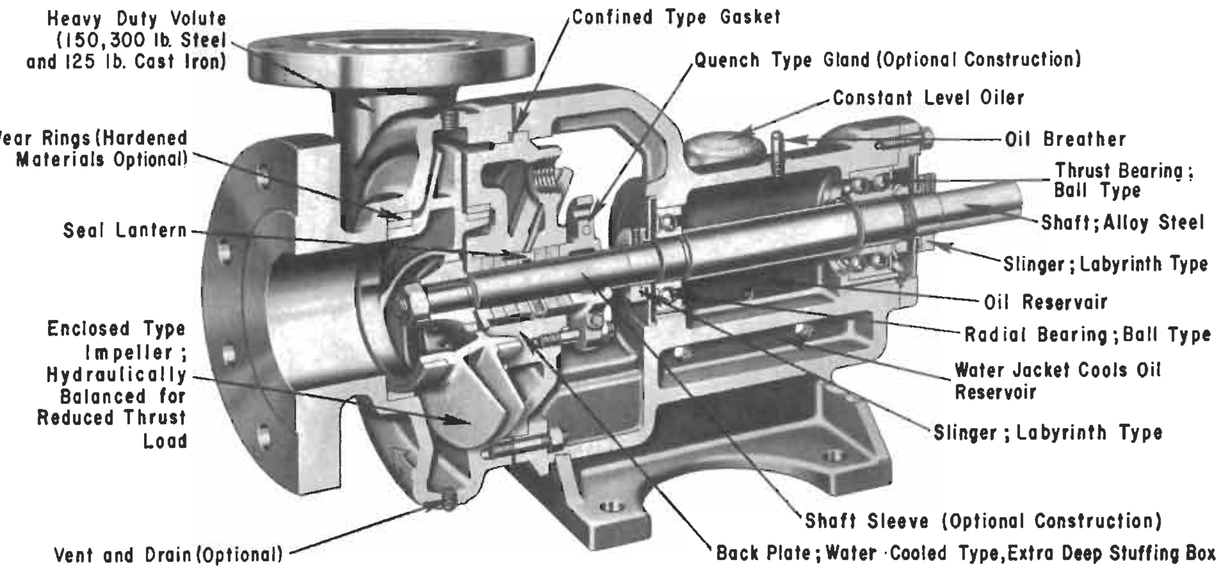
In the Sundyne Canned Motor design, the entire outside of the motor is enclosed in a secondary leakage containment shell or can. Primary leakage protection is provided by corrosion resistant liners which are seal welded and 100% leak checked to assure that pumped fluid does not contact the stator windings or rotor core. There is no shaft protrusion to seal and thus no seals to leak.

Pumped fluid is circulated inside the stator liner to cool the motor, and lubricate the bearings.

Motor windings and insulation systems are specially designed, developed and applied as an integral part of the pump so that design life is at least as great as for conventional air cooled motors. Winding temperature is primarily influenced by pumped fluid temperature and secondarily by use of cooling jacket. Fluid temperature is considered in pump application to assure full winding life. Thermostats are embedded in the hot spots of windings for shutdown in case of overheating.

Motors are suitable for use in general purpose areas and in Class I, Division 2, Group C and D areas for a wide range of pump fluid temperatures. For Class I, Division 1 Group C and D; U.L. listed, explosion proof motors are available.

Figure 3-1B. Sealless canned centrifugal pump, primary and secondary leakage containment. See Table 3-2. (Courtesy Sundstrand Fluid Handling Co.)



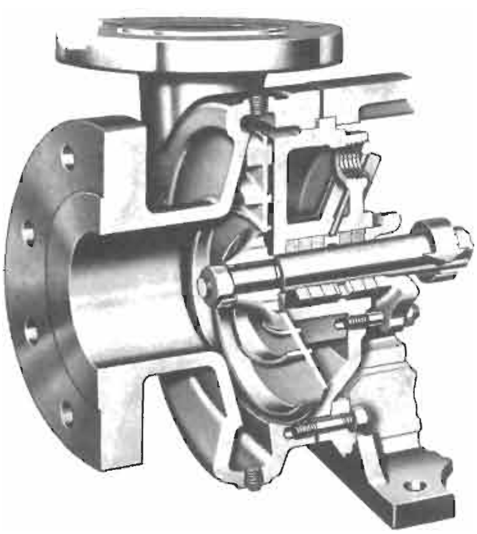


Figure 3-2. Cut-a-way section of single-stage pump. Part 1 (above) enclosed type impeller, Part 2 (lower left) open type impeller. (Courtesy Peerless Pump Div. FMC Corp.)

(text continued from page 161)

Basic Parts of a Centrifugal Pump

Table 3-3 is a quick reference as to the function of the basic parts.

Impellers

The three common types of impellers that impart the main energy to the liquid for process applications are (see Figure 3-15):

1. Fully enclosed—used for high head, high pressure applications.
2. Semi-enclosed—used for general purpose applications, has open vane tips at entrance to break up suspended particles and prevent clogging.
3. Open—used for low heads, suspended solids applications, very small flows.

Small radial vanes are usually provided on back shroud or plate of impeller to reduce the pressure on the stuffing box, and prevent suspended solids from entering the back side and possibly causing clogging.

The working or pumping vanes are backward in form relative to the impeller rotation.

These impellers are available in nearly any material of construction as well as rubber, rubber-lined, glass-lined,

Table 3-2
Approximate Capacity-Head Ranges For Centrifugal Pumps

Type	Max. GPM*	Max. Head* Ft.	Figure No.
Single Stage (H)	600	225	3-1 and 3-2
Canned, Sealless	2000	650	3-1A
Canned, Sealless	1000	800	3-1B
Single Stage (V)	>150	250+	3-3
Double Suction,			
Single Stage (H)	15,000	300	3-4 and 3-5
Multistage (H)	3,000	5,000	3-6, -7, and -8
Single and Multistage (V)			
a. Mixed Flow (V)	100,000	75	3-11
b. Axial Flow (V)	100,000	25	3-10
c. Centrifugal (V)	400±	5,750	3-9, -12, -13 and -14

*Not necessarily at same point.
(H) = Horizontal.
(V) = Vertical.

and plastic. The lined impellers are of the open or semi-open type.

Casing

The casing may be constructed of a wide variety of metals, as well as being lined to correspond to the material of the impeller. Operating pressures go to about 5000 psi for the forged or cast steel barrel-type designs. However, the usual process application is in the 75 psi to 1,000 psi range, the latter being in light hydrocarbon and similar high vapor pressure systems.

The removal of the casing parts is necessary for access to the impeller and often to the packing or seals. Some designs are conveniently arranged to allow dismantling the casing without removing the piping connections. There are proposed construction standards being considered which will allow easy maintenance of many of the types now being offered in a non-standard fashion.

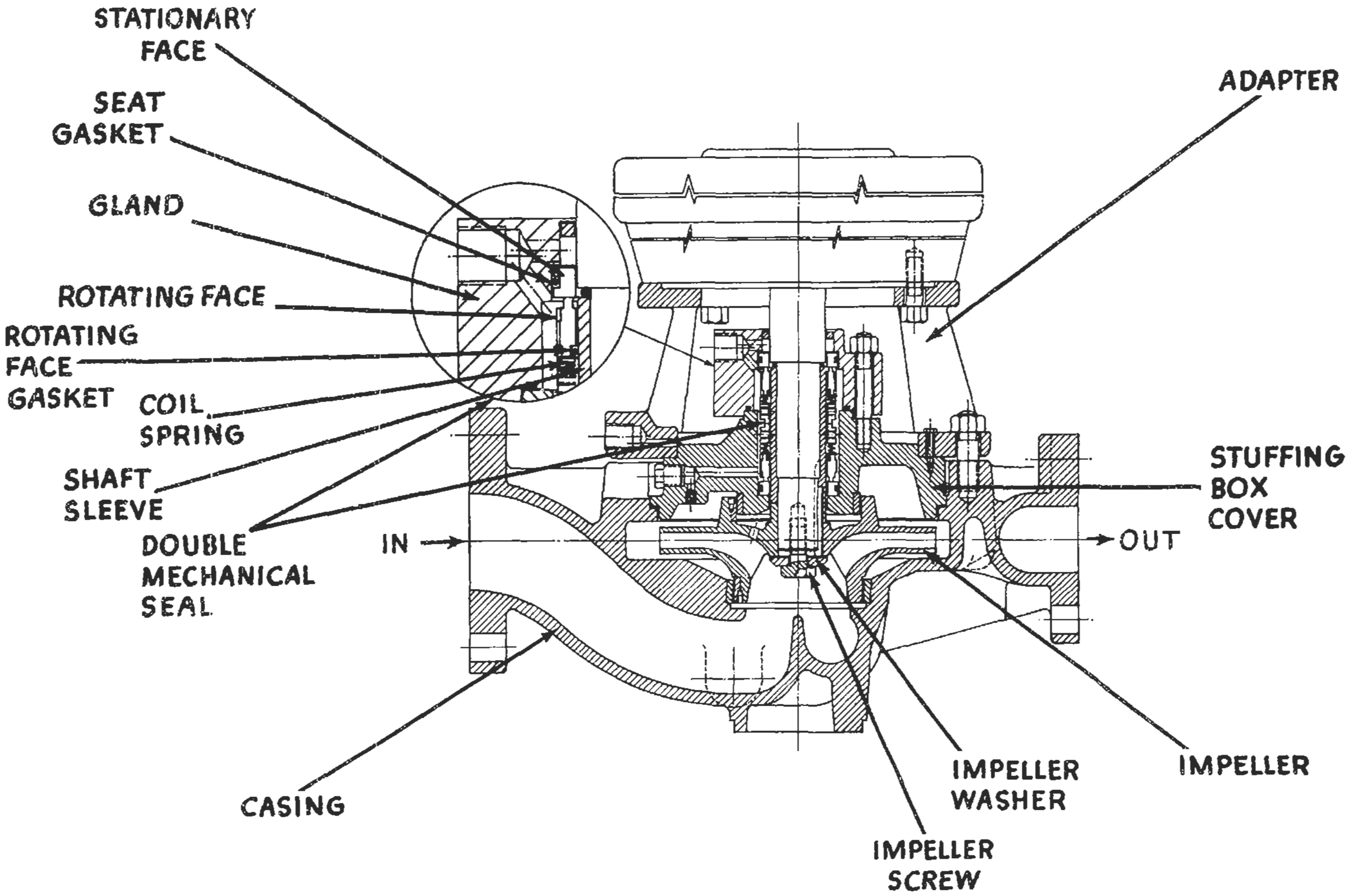


Figure 3-3. Cross-sectional view of a vertical in-line pump. (By permission, H. Knoll and S. Tinney, "Hydrocarbon Processing," May 1971, p. 131 and Goulds Pump, Inc. Mechanical seal and seal venting details courtesy Borg-Warner.)

Table 3-3
Basic Parts of a Centrifugal Pump

Part	Purpose
Impeller	Imparts velocity to the liquid, resulting from centrifugal force as the impeller is rotated.
Casing	Gives direction to the flow from the impeller and converts this velocity energy into pressure energy which is usually measured in feet of head.
Shaft	Transmits power from the driver to the impeller.
Stuffing box	This is a means of throttling the leakage which would otherwise occur at the point of entry of the shaft into the casing. Usually not a separate part, but rather made up of a group of small details, as "A" to "D".
(A) Packing	This is the most common means of throttling the leakage between the inside and outside of the casing.
(B) Gland	To position and adjust the packing pressure.
(C) Seal gage (also called water-seal or lantern ring	Provides passage to distribute the sealing medium uniformly around the portion of the shaft that passes through the stuffing box. This is very essential when suction lift conditions prevail to seal against in-leakage of air.
(D) Mechanical seal	Provides a mechanical sealing arrangement that takes the place of the packing. Basically it has one surface rotating with the shaft and one stationary face. The minutely close clearance between these two faces prevents leakage of liquid out or air in.
Shaft sleeve	Protects the shaft where it passes through the stuffing box. Usually used in pumps with packing but often eliminated if mechanical seals are employed.
Wearing rings	Keeps internal recirculation down to a minimum. Having these rings as replaceable wearing surfaces permits renewal of clearances to keep pump efficiencies high. On small types only one ring is used in the casing and on larger sizes, companion rings are used in the casing and on the impeller.
Wearing plates	With open type impellers or end clearance wearing fits, these perform the same purpose as wearing rings do with radial clearances.
Bearings	Accurately locate shaft and carry radial and thrust loads.
Frame	To mount unit rigidly and support bearings. In most single suction pumps this is a separate piece. In many double suction pumps, the support is through feet cast as part of the casing. In some special suction pumps, the feet are also part of the casing and the bearing assembly is overhung. With close coupled single suction types, this support is provided by the motor or by special supporting adapters.
Coupling	Connects the pump to the driver.

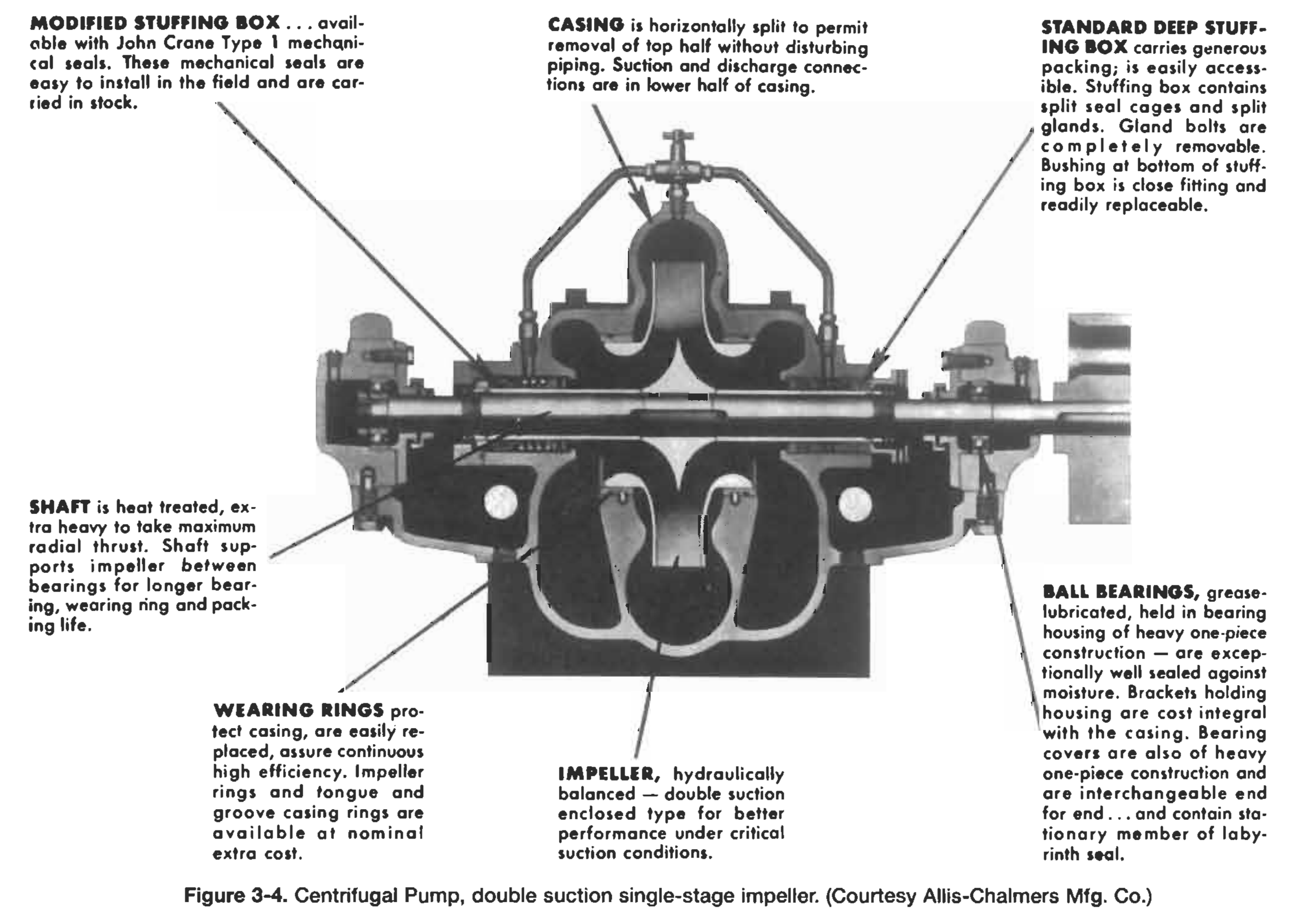


Figure 3-4. Centrifugal Pump, double suction single-stage impeller. (Courtesy Allis-Chalmers Mfg. Co.)

Shaft

Care should be given in selecting the shaft material. It must be resistant to the corrosive action of the process fluids, yet possess good strength characteristics for design. For some designs it is preferable to use a shaft sleeve of

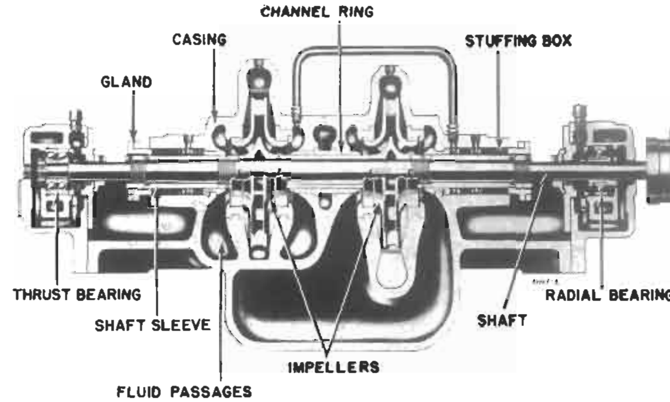


Figure 3-6. Cross-section horizontal two-stage horizontal split case centrifugal pump. (Courtesy Ingersoll-Rand Co.)

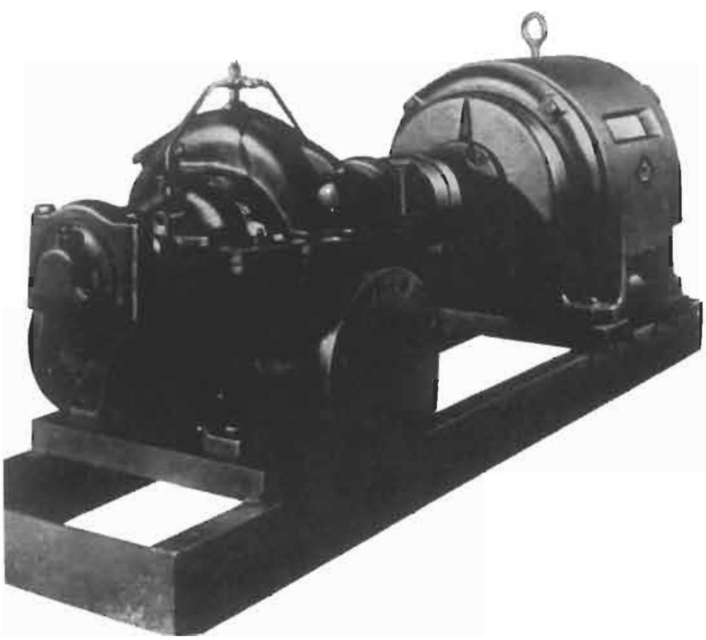


Figure 3-5. External view double suction single-stage pump. (Courtesy Allis-Chalmers Mfg. Co.)

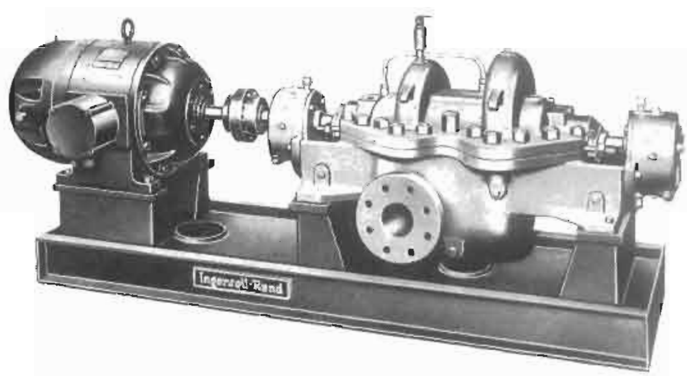


Figure 3-7. Exterior view of horizontal two-stage split case centrifugal pump. (Courtesy Ingersoll-Rand Co.)

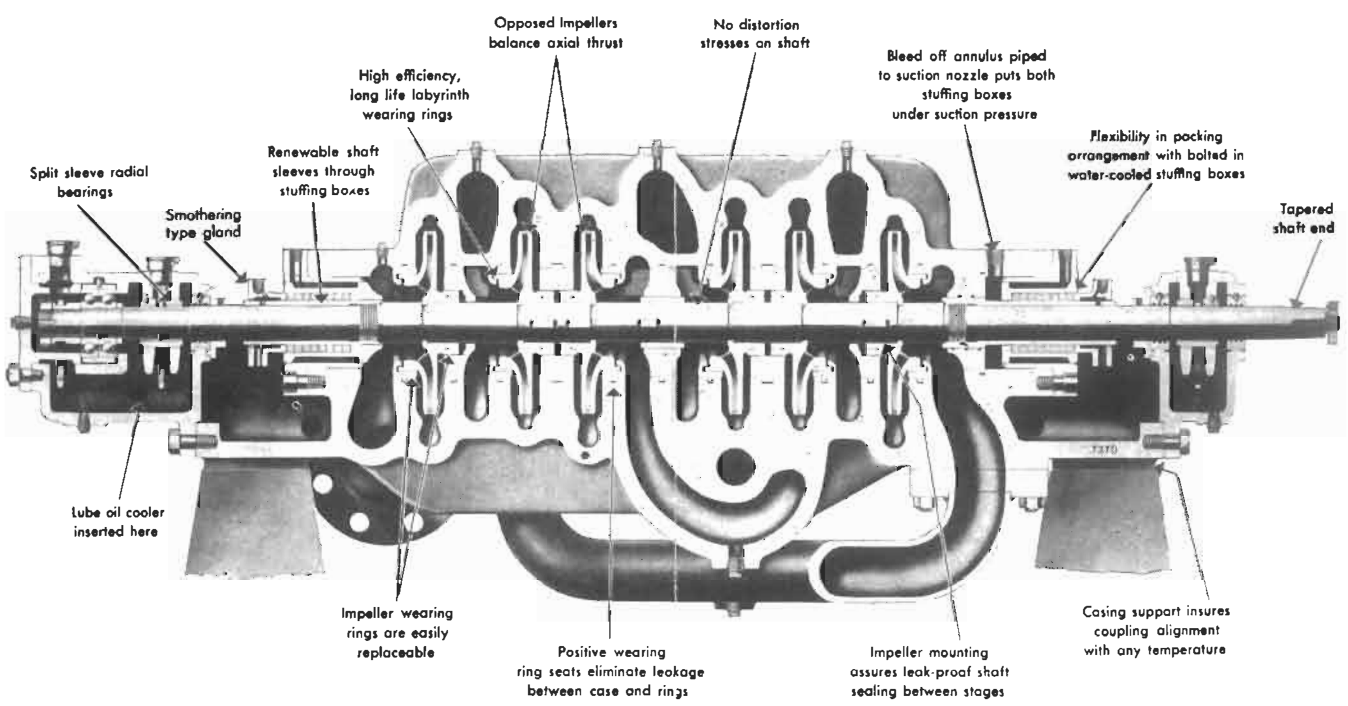
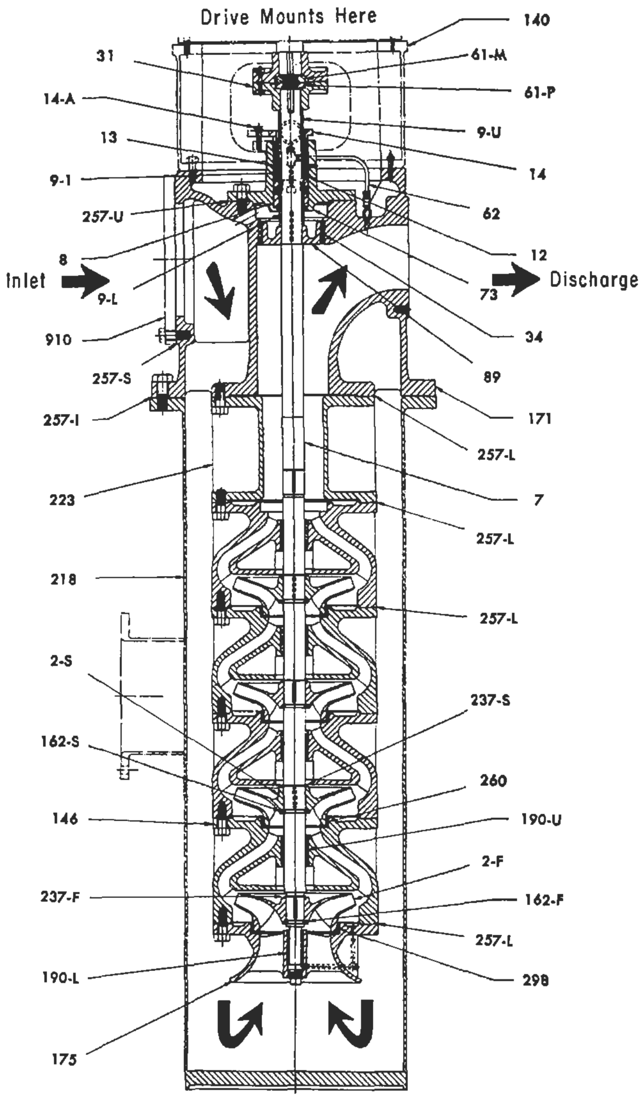


Figure 3-8. Refinery oil and boiler feed high pressure centrifugal pump. Courtesy Delaval Steam Turbine Co., currently Transamerica Delaval, Inc.)



PARTS LIST	
Cat. No.	Part Name
2-F	First Stage Impeller
2-S	Second Stage Impeller and Above
34	Nozzle Head Bushing
260	Diffuser Ring
298	Suction Bell Ring
8	Stuffing Box Bushing
9-L	Lower Shaft Sleeve
9-I	Intermediate Shaft Sleeve
9-U	Upper Shaft Sleeve
12	Packing
13	Seal Cage
14	Gland
14-A	Gland Bolt
31	Complete Coupling
61-P	Pump Half Coupling Lock Nut
61-M	Motor Half Coupling Lock Nut
62	Complete Piping
73	Stuffing Box
89	Balance Disk
140	Motor Support Column
146	Diffuser
162-F	1st. Stg. Impeller Retaining Collar
162-S	2nd Stg. and Above Impeller Retaining Collar
171	Nozzle Head
175	Suction Bell
190-L	Lower Sleeve Bearing
190-U	Upper Sleeve Bearing
218	Tank
223	Spacer Column
237-F	First Stage Snap Ring
237-S	Second Stage Snap Ring and Above
257-L	Lower Gasket
257-I	Intermediate Gasket
257-U	Upper Gasket
257-S	Blind Flange Gasket
7	Shaft with Keys
910	Blind Flange

Figure 3-9. Vertical multistage centrifugal pump with barrel casing. (Courtesy Allis-Chalmers Mfg. Co.)

the proper corrosion resistant material over the preferred structural shaft material. These sleeves may be metal, ceramic, rubber, etc., as illustrated in Figure 3-16.

Bearings

The bearings must be adequate to handle the shaft loads without excessive wear, provided lubrication is maintained. Usually this is not a point of real question provided the manufacturer has had experience in the type of loads imposed by the service conditions, and the responsibility for adequate design must be his.

In all cases, the bearings should be of the outboard type, that is, not in the process fluid, unless special conditions prevail to make this situation acceptable.

Packing and Seals on Rotating Shaft

Conventional soft or metallic packing in a stuffing box (Figure 3-17) is satisfactory for many low pressure, noncorrosive fluid systems. Special packings such as teflon, or mechanical seals are commonly used for corrosive fluids, since there can be leakage through the packing along the rotating shaft. However, for these conditions a mechanical seal is preferred. When the pressure becomes high (above about 50 psig) or the fluid is corrosive, additional means of sealing the shaft must be provided. Particular care must be taken in handling and using the mechanical seals, and these special instructions should be obtained from the seal manufacturer [27]. Generally speaking it is not wise to have the mechanical seal installed at the pump factory, as the slightest amount of grit on the faces can cause perma-

Figure 3-10. Vertical propeller type pump. (Courtesy Peerless Pump Div., FMC Corp.)

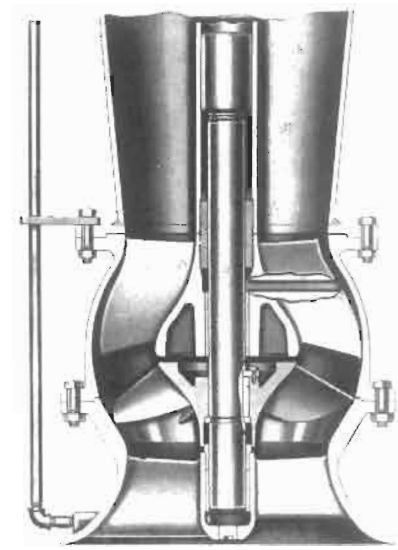


Figure 3-11. Vertical single-stage mixed flow type pump, liquid inlet and impeller. (Courtesy Peerless Pump Div., FMC Corp.)

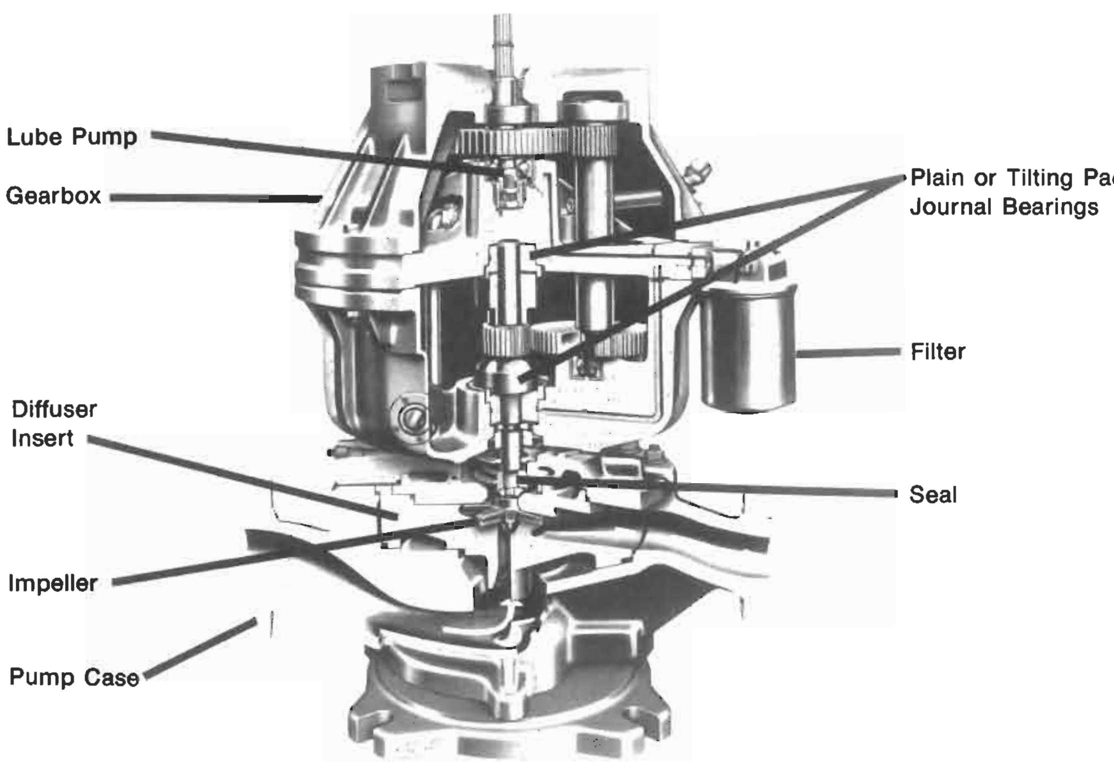


Figure 3-12. Single-stage, high speed (>5000 ft) centrifugal process pump. Pumps are high speed, gear driven, and especially suited for applications in light hydrocarbon liquid service. (Courtesy Sundstrand Fluid Handling, Inc.)

Impeller: Since the Sundyne straight, radial-bladed impeller does not require close running clearances, it eliminates the necessity for oversizing to compensate for performance deterioration common to conventional pumps. Clearance of the Sundyne impeller is 0.030 to 0.070 inch.

Inducer: An optional helical inducer is available to substantially reduce the pump NPSH requirement.

Figure 3-13. Impeller and inducer (optional) for pumps in Figure 3-12. (Courtesy Sundstrand Fluid Handling, Inc.)

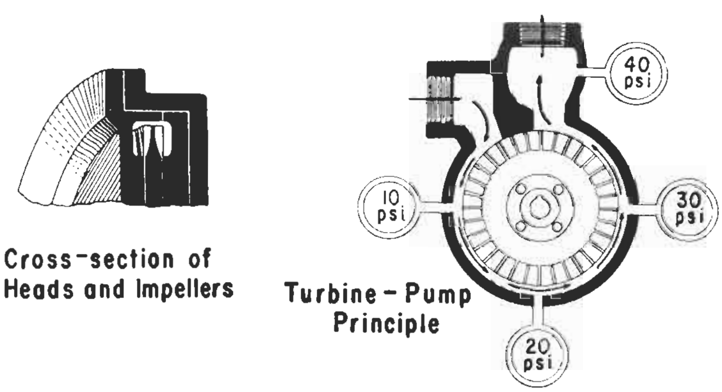
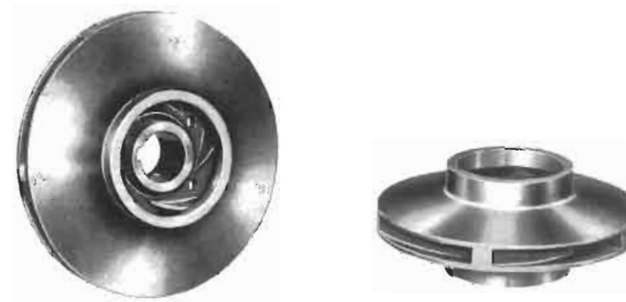


Figure 3-14. Turbine pump. (Courtesy Roth Pump Co.)

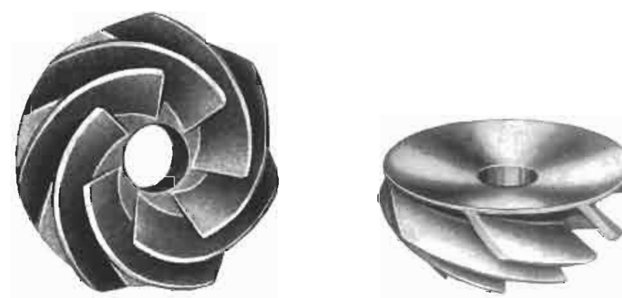
Enclosed single-suction impeller with sealing on suction and back sides. (Courtesy The Deming Co.)

Figure 3-15. Impeller types. Open impeller for corrosive or abrasive slurries and solids. (Courtesy Goulds Pumps, Inc.)

Figure 3-15. Continued.

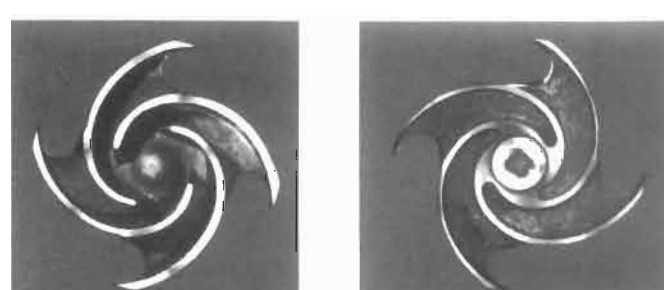


Enclosed double-suction impeller with sealing rings on both sides. (Courtesy The Deming Co.)

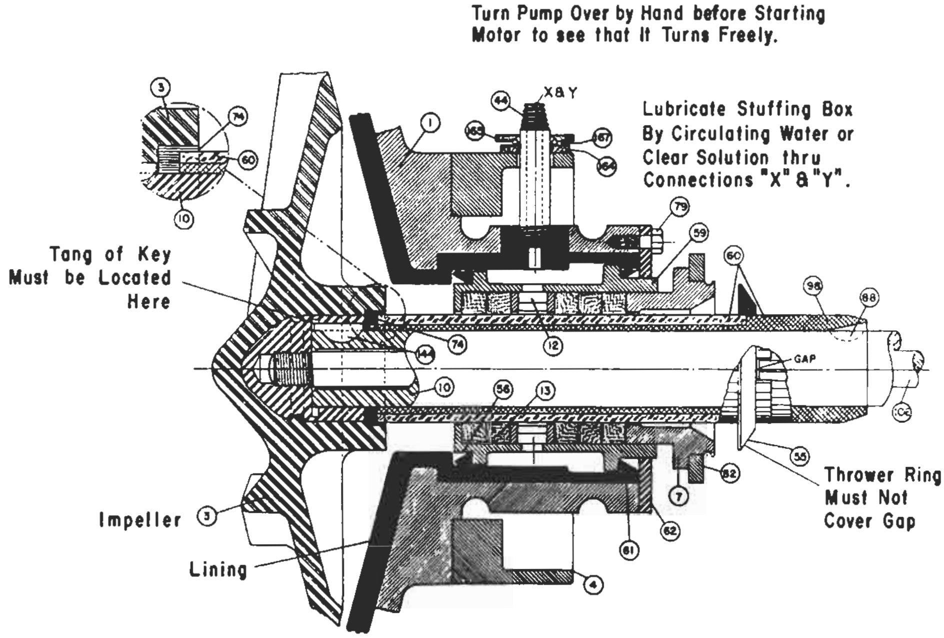


Mixed flow semi-enclosed impeller. (Courtesy The Deming Co.)

Semi-open or semi-enclosed impeller. (Courtesy Goulds Pumps Inc.)



Front Back



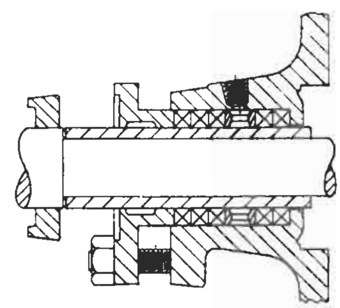
No.	Part Name
1.	Pump Casing
3.	Impeller
4.	Pump Frame
7.	Split Gland
10.	Shaft
12.	Lantern Ring
13.	Packing
44.	Nipple (2)
55.	Thrower Ring
56.	Rubber Ring
59.	Resilient Sleeve
60.	Shaft Sleeve
61.	Rubber Ring
62.	Retaining Ring
74.	Rubber Ring
79.	Capscrew (6)
82.	Gland Yoke
88.	Key
98.	Shaft Sleeve Ext'n.
106.	Tie Rod
144.	Key
164.	Washer (2)
165.	Retainer (2)
167.	Rubber Ring (2)

Figure 3-16. Stuffing box details lined pump with porcelain or teflon® shaft sleeve. (Courtesy Dorr-Oliver, Inc.)

nent damage or destruction on only one or two revolutions at pump speed. The seals should be inspected and cleaned immediately prior to initial start-up.

A mechanical seal system (see Figures 3-19A [23] and 3-19B [16]) contains a rotating element attached to the rotating shaft by set screws (or a clamp) that turns against a stationary unit set in the gland housing. The necessary continuous contact between the seal faces (see Figure 3-19A) is maintained by hydraulic pressure in the pump from the fluid being pumped and by the mechanical loading with springs or bellows. To seal the mechanical seal elements to the rotating shaft to prevent leakage along the shaft, two basic types of seals are used: (a) pusher type using springs and seal "O" rings, wedge rings, etc. and (b) non-pusher type using some form of bellows of elastomer or metal [24]. Also see Table 3-4.

The matching contact rubbing faces are made of dissimilar materials, precision finished to a mirror-like flat surface. There is little friction between these, and hence, they form a seal that is practically fluid tight. The rubbing materials may be some combination of low friction carbon, ceramics (aluminum oxide, silicon carbide), and/or tungsten. The choice of materials will depend on the service, as will the selection of the materials of construction for the other components, such as springs, "O" rings, other seal rings, and even the housing. The designer should consult the seal manufacturers for details of application not possible to include here.



Longitudinal section with Lantern Gland.

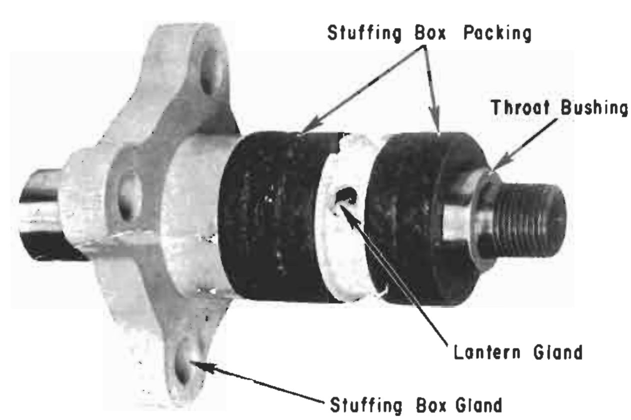


Figure 3-17. Packed stuffing box. (Courtesy Dean Brothers Pumps, Inc.)

The "single" mechanical seal is made of a rotating element fixed to the shaft (or shaft sleeve), and a stationary element fixed to the pump casing [16].

The "double" seal is for severe sealing problems where out-leakage to the environment cannot be tolerated and must be controlled. (See Figures 3-31C and 3-31D.) Depending upon the fluid's characteristics, the vent between the double seals (Figures 3-31A and B) may be purged with process liquid, or a different liquid or oil, or it may be connected to a seal pot and vent collection to prevent leakage to the air/environment. There are techniques for testing for leakage of the inner seal by measuring the vent space pressure through the seal liquid surge port. This should be essentially atmospheric (depending on the vent system backpressure). This allows detection before the leakage breaks through the outer seal.

Figure 3-18 illustrates a seal installed in a conventional stuffing box with cooling liquid flow path. Figures 3-18, 3-19A, 3-19B, 3-20 and 3-21 identify the fundamentals of mechanical seals, even though there are many specific designs and details. These various designs are attempts to correct operational problems or seal weaknesses when used under various conditions in the wide variety of process fluids.

The average unbalanced external seal is good for pressures of about 30 psig, while the balanced design will handle 150 psig. Special designs will handle much higher

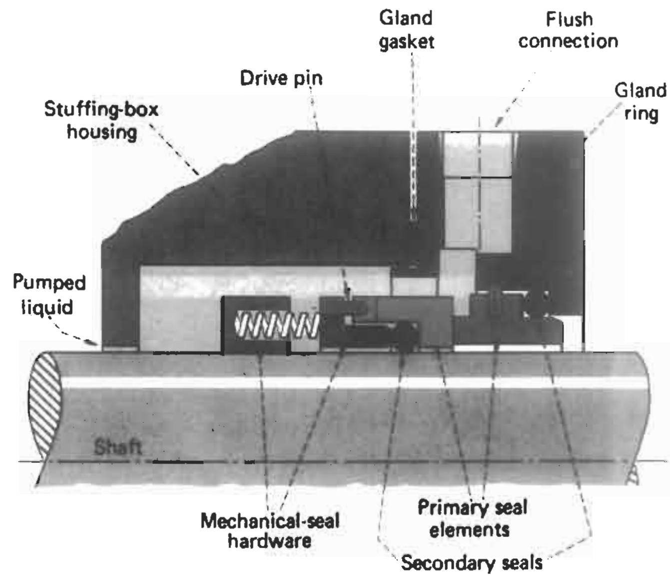


Figure 3-19A. Basic components of all mechanical seals. (By permission, Adams, W. H., *Chemical Engineering* Feb. 7, 1983, p. 48.)

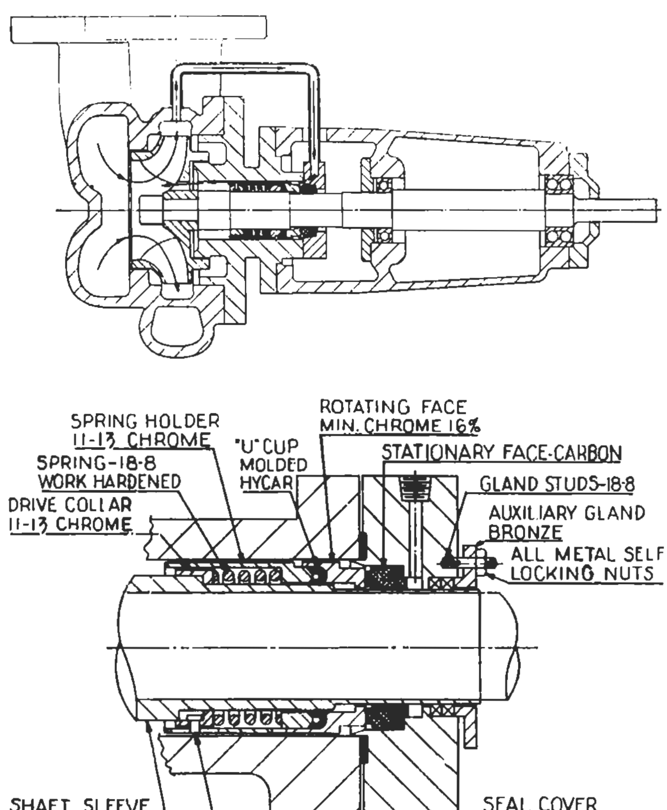


Figure 3-18. Typical single mechanical seal inside pump stuffing box. (Courtesy Borg-Warner Co.)

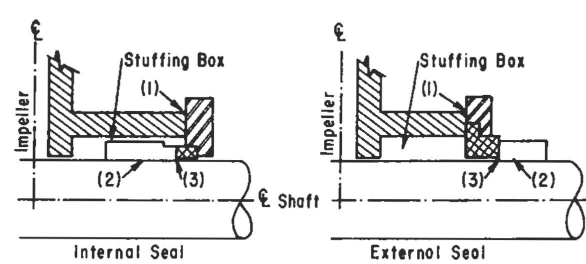


Figure 3-19B. The three sealing points in mechanical seals. (By permission, T. J. Sniffen, *Power and Fluids,* Winter 1958, Worthington Corp.)

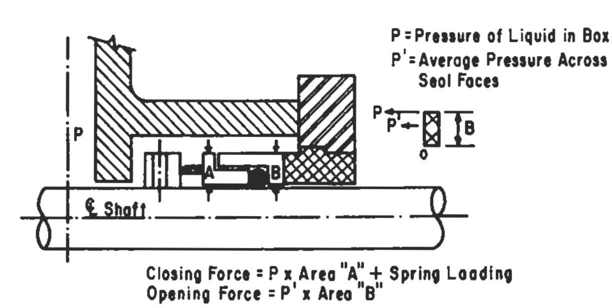


Figure 3-20. Area relationship for unbalanced seal construction. (By permission, T. J. Sniffen, *Power and Fluids,* Winter 1958, Worthington Corp.)

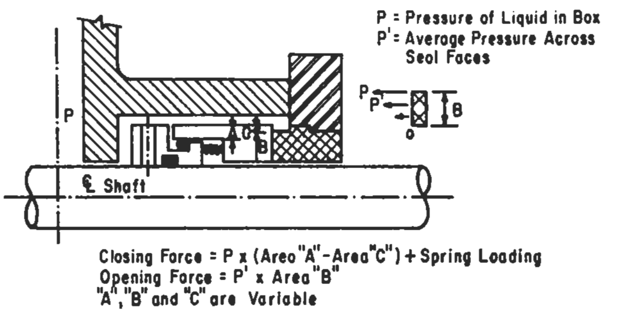


Figure 3-21. Area relationship for balanced seal construction. (By permission, T. J. Sniffen, *Power and Fluids,* Winter 1958, Worthington Corp.)

pressures. Actually the maximum operating process pressures are a function of the shaft speed and diameter for a given seal design fluid and fluid temperature.

Figure 3-22 is an outside balanced seal designed for vacuum to 150 psig and –40°F to +400°F. (See Table 3-4) The process fluid must be free of solids (as for practically

Table 3-4
Requirements for Mechanical Seal Installations

Feature	Description	Remarks	Fig. No.
Cooling	Water Jacketed Stuffing Box	Liquid must dead end in stuffing box	3–23
Cooling	Gland Plate	Efficient to cool contact faces	3–24
Lubrication	Dead-End	Good under vacuum, mild abrasives metal-metal, dry seals	3–25
Lubrication	Circulating	Good cooling of contact faces	3–26
Flushing	Inside Seal	Good for volatile liquids, sol'ns tending to crystallize, steam	3–27
Flushing	Outside Seal	Heating to prevent solidification	3–28
Quenching	Outside Seals (only)	For oxidizing and corrosive liquids, seal liquid washes process fluid, for high temp.	3–29
Vent and Drain	Inside Seal	Safety feature, for venting to flare, draining	3–30
Flushing	Double	Requires circulation system	3-31A
Flushing	Tandem	Requires circulation system	3-31B
Two Rotary	Double	For improved sealing	3-31C
Four Rotary	Double Tandem	For special sealing problem	3-31D

all mechanical seals) and must not attack the material of the O-ring shaft packing. Many other designs are available, and the manufacturers should be consulted for advice on specific sealing problems.

Centrifugal Pump Selection

The centrifugal pump is a versatile unit in the process plant, since its ease of control, non-pulsing flow, pressure limiting operation fits many small and large flow systems.

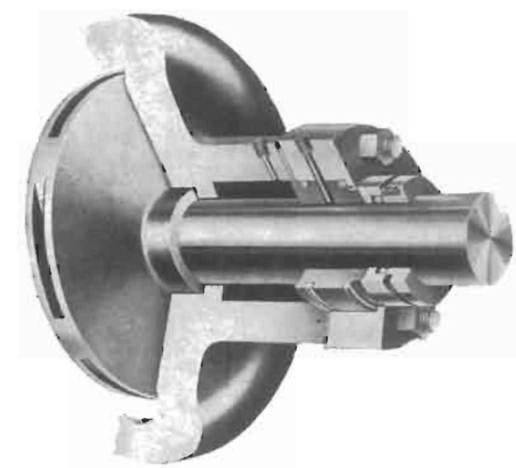


Figure 3-22. Outside balanced seal (single). (Courtesy Durametallic Corp.)

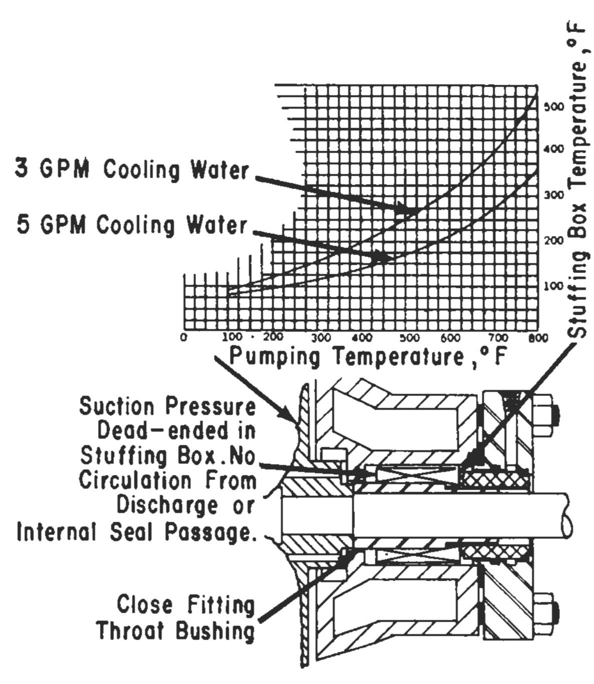


Figure 3-23. Water jacketed stuffing box. (By permission, H. P. Hummer and W. J. Ramsey, Bulletin SD, 752, O&T, Durametallic Corp.)

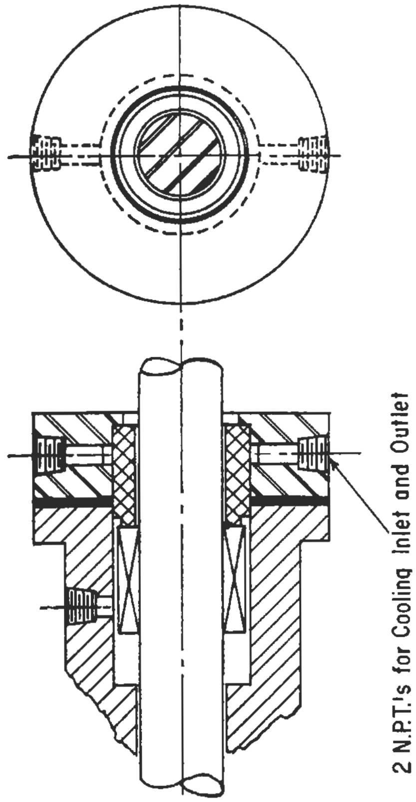


Figure 3-24. Gland plate cooling. (By permission, H. P. Hummer and W. J. Ramsey, Bulletin SD 752, O&T, Durametallic Corp.)

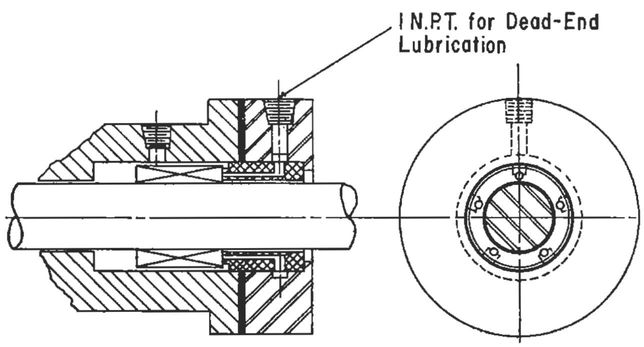


Figure 3-25. Dead-end lubrication. (By permission, H. P. Hummer and W. J. Ramsey. Bulletin SD 752, O&T, Durametallic Corp.)

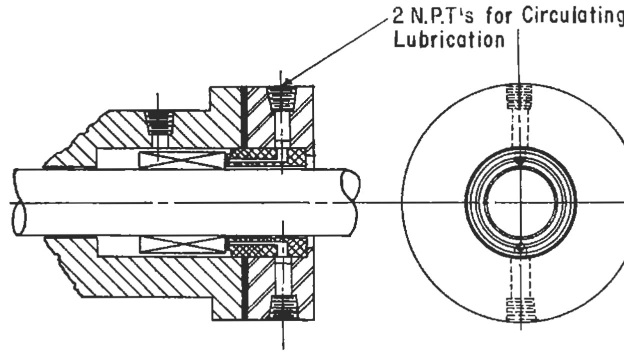


Figure 3-26. Circulating lubrication. (By permission, H. P. Hummer and W. J. Ramsey, Bulletin SD 752, O&T, Durametallic Corp.)

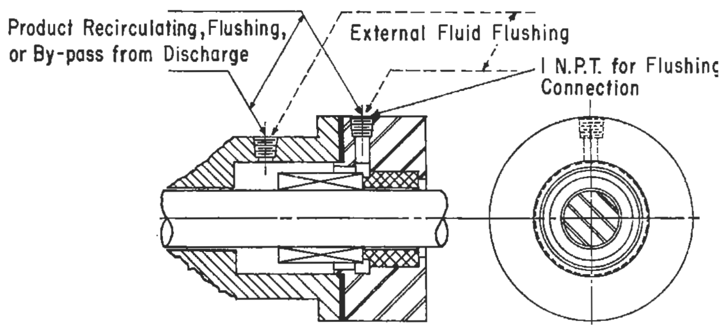


Figure 3-27. Flushing inside seal. (By permission, H. P. Hummer and W. J. Ramsey, Bulletin SD 752, O&T, Durametallic Corp.)

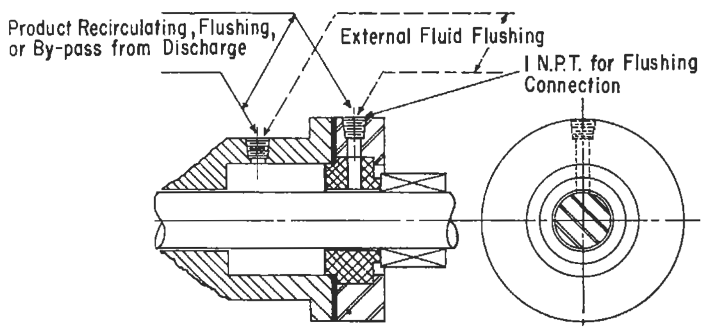


Figure 3-28. Flushing outside seal. (By permission, H. P. Hummer and W. J. Ramsey, Bulletin SD 752, O&T, Durametallic Corp.)

Generally speaking the centrifugal pump has these characteristics:

1. Wide capacity, pressure, and fluid characteristics range
2. Easily adapted to direct motor, V-belt or other drive
3. Relatively small ground area requirements
4. Relatively low cost
5. Difficult to obtain very low flows at moderate to high pressures
6. Develops turbulent conditions in fluids
7. Turbine type: (a) offers very high heads at low flows, (b) self-priming, (c) limited to very clean, non-abrasive fluids with limited physical properties, (d) clearances can be problem on assembly and maintenance.

Single-Stage (Single Impeller) Pumps

This type of pump (Figures 3-1, 3-2, 3-3) is the workhorse of the chemical and petrochemical industry. It also

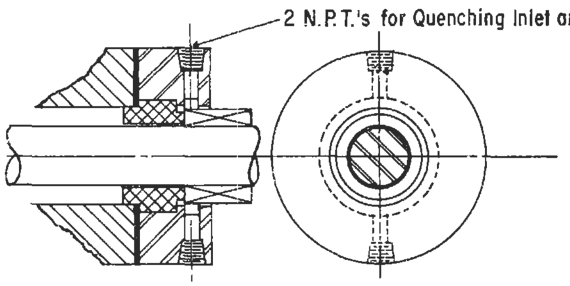


Figure 3-29. Quenching outside seal. (By permission, H. P. Hummer and W. J. Ramsey, Bulletin SD 752, O&T Durametallic Corp.)

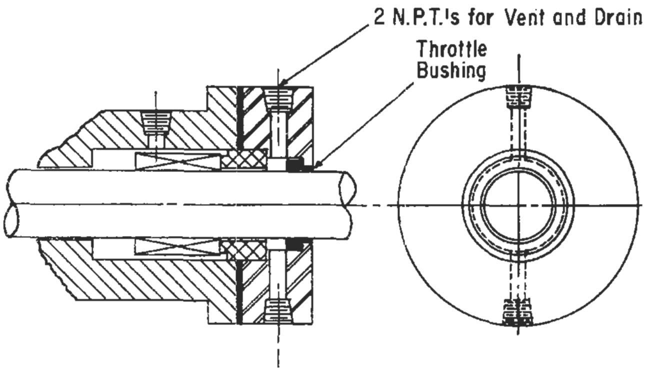


Figure 3-30. Vent and Drain. (By permission, H. P. Hummer and W. J. Ramsey, Bulletin SD 752, O&T, Durametallic Corp.)

serves important functions in petroleum refining and almost every industry handling fluids and slurries. Although the performance characteristics may vary for specific applications, the general fundamental features are the same especially for manufacturers who standardize to some extent through the Hydraulic Institute [17] and American National Standards Institute.

Figure 3-32 indicates the relative relationship for three of the centrifugal type pumps, with curves labeled "centrifugal" referring to the usual process (open or enclosed impeller) type unit. A similar set of curves is shown in Figure 3-33 for the turbine unit. Note that the flat head curve of the centrifugal unit has advantages for many process systems, giving fairly constant head over a wide range of flow. For some systems where changes in flow must be reflected by pressure changes, the turbine characteristic is preferred. The centrifugal impeller provides an ever rising horsepower requirement with increasing flow, while the horsepower of the turbine pump falls off with increasing flow (and decreasing head); hence it is "overloading" at low flows and must be operated with ample horsepower for these conditions.

The effects of impeller shape for the usual centrifugal process pump performance are given in Figure 3-34. The only part the process designer can play is in the selection of a manufacturer's performance curve to fit the control requirements of the system. If the curve is too steep, select an impeller of necessary basic characteristics to move the curve in the proper direction, providing the manufacturer has an impeller pattern to fit that pump casing, and with the improved physical dimensions. This may require changing the make of pump to obtain the necessary range and characteristic.

For conditions of (1) high suction side (or inlet) friction loss, from suction piping calculations or (2) low available Net Positive Suction Head (10 feet or less), a large open eye on the impeller inlet is necessary to keep the inlet velocity low. NPSH is discussed in a later section. The manufacturer should be given the conditions in order to properly appraise this situation.

In most instances the manufacturer has a series of impellers to use in one standard casing size. The impeller may be trimmed to proper diameter to meet head requirements and yet stay within the power range of a specified driver. It is not necessary to place a full size impeller in a casing unless the system requires this performance. It is good to know when larger impellers can be placed in the casing, and what their anticipated performance might be in order to adequately plan for future uses and changing loads on the pump.

Although the previous discussion has pertained to single impellers, the principles are the same for the multistage units (impellers in series in the casing) and the casing with double inlets. The latter pump is used for the higher flows, usually above 500 GPM, and this design serves to balance the inlet liquid load as it enters the impeller, or first stage (if more than one) from two sides instead of one as in the single impeller. The double suction pump has the liquid passages as a part of the casing, with still only one external suction piping connection.

The axial and mixed flow impellers are used primarily for very high capacities at relatively low heads as shown in Table 3-2. They are usually applied to services such as water distribution to a large system, waste water disposal, recirculating large process liquor flows, and the like.

Many applications can be handled either by a horizontal or a vertical pump. In the range usually associated with process plants and the associated services, Tables 3-5 and 3-6 are helpful guides in making the selection [12].

Pumps In Series

Sometimes it is advantageous or economical to use two or more pumps in series (one pump into and through the other) to reach the desired discharge pressure. In this situation the capacity is limited by the smaller capacity of any one of the pumps (if they are different) at its speed of operation. The total discharge pressure of the last pump is

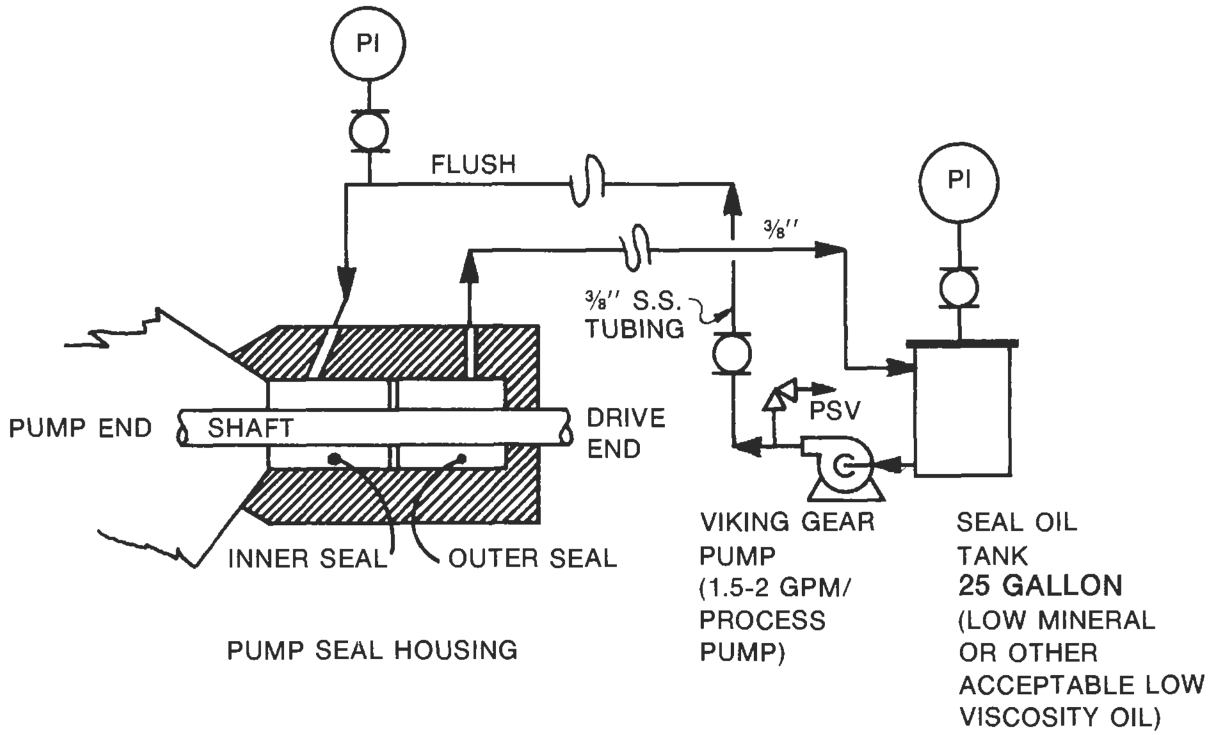


Figure 3-31A. Typical seal flush arrangement for double mechanical seals.

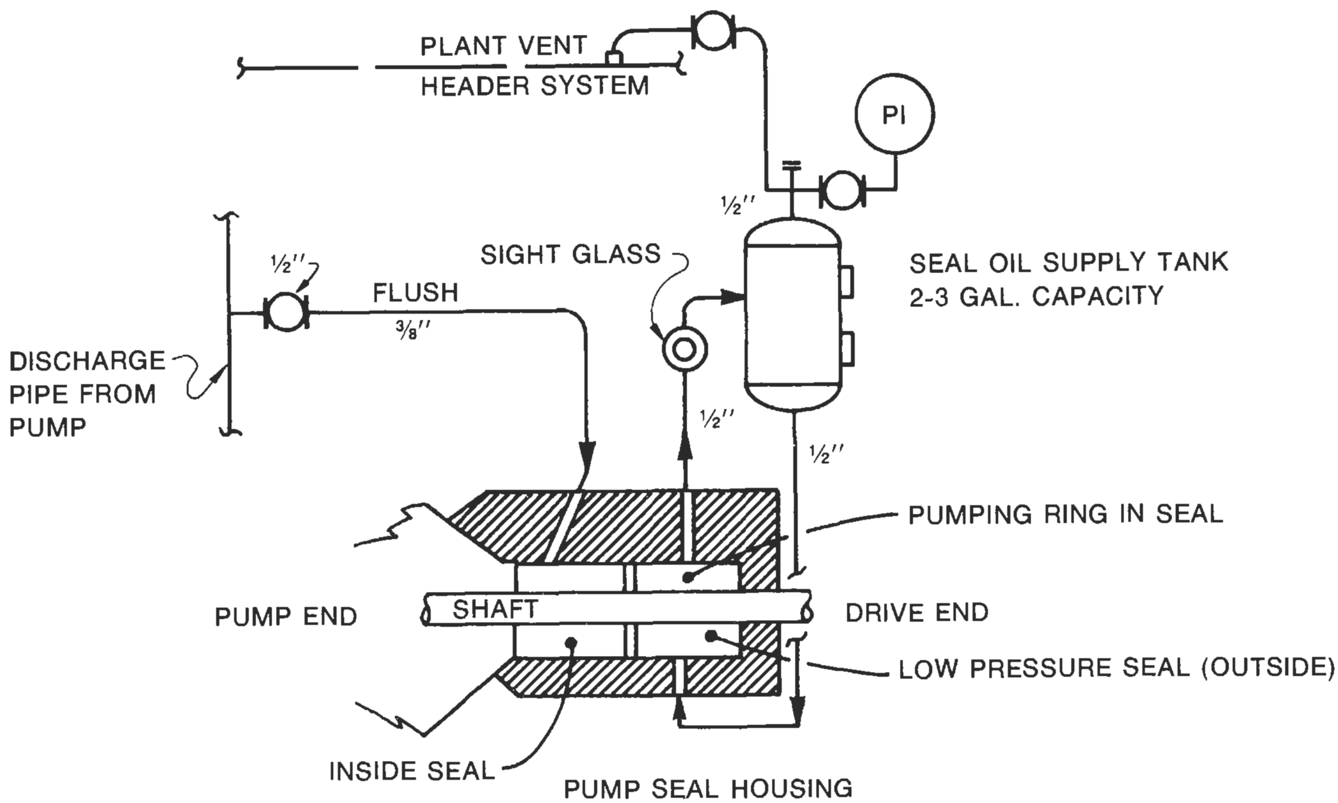


Figure 3-31B. Typical seal flush arrangement for tandem mechanical seals.

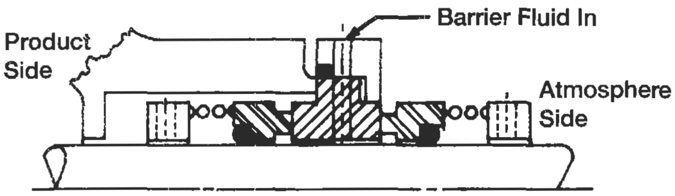


Figure 3-31C. Double mechanical seal, two rotary elements against common stationary. (By permission, Fischer, E. E., *Chem Processing*, Oct. 1983 [24].)

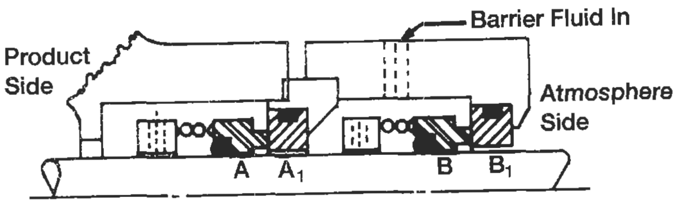


Figure 3-31D. Tandem double seal. (By permission, Fischer, E. E., *Chem. Processing*, Oct. 1983 [24].)

the sum of the individual discharge pressures of the individual pumps. For identical pumps, the capacity is that of one pump, and the discharge pressure of the last pump is the sum of the individual heads of each pump acting as a single unit. Thus, for two identical pumps the discharge head is twice that of the rated pressure of one pump at the designated flow rate (Figure 3-35). The pump casing of each stage (particularly the last) must be of sufficient pressure rating to withstand the developed pressure.

Pumps in Parallel

Pumps are operated in parallel to divide the load between two (or more) smaller pumps rather than a single large one, or to provide additional capacity in a system on short notice, or for many other related reasons. Figure 3-35 illustrates the operational curve of two identical pumps in parallel, each pump handling one half the capacity at the system head conditions. In the parallel arrangement of two or more pumps of the same or different characteristic curves, the capacities of each pump are added, at the head of the system, to obtain the delivery flow of the pump system. Each pump does not have to carry the same flow; but it will operate on its own characteristic curve, and must deliver the required head. At a common tie point on the discharge of all the pumps, the head will be the same for each pump, regardless of its flow.

The characteristic curves of each pump must be continuously rising (right to left) as shown for the single pump of Figure 3-35, otherwise with drooping or looped curves they may be two flow conditions for any one head,

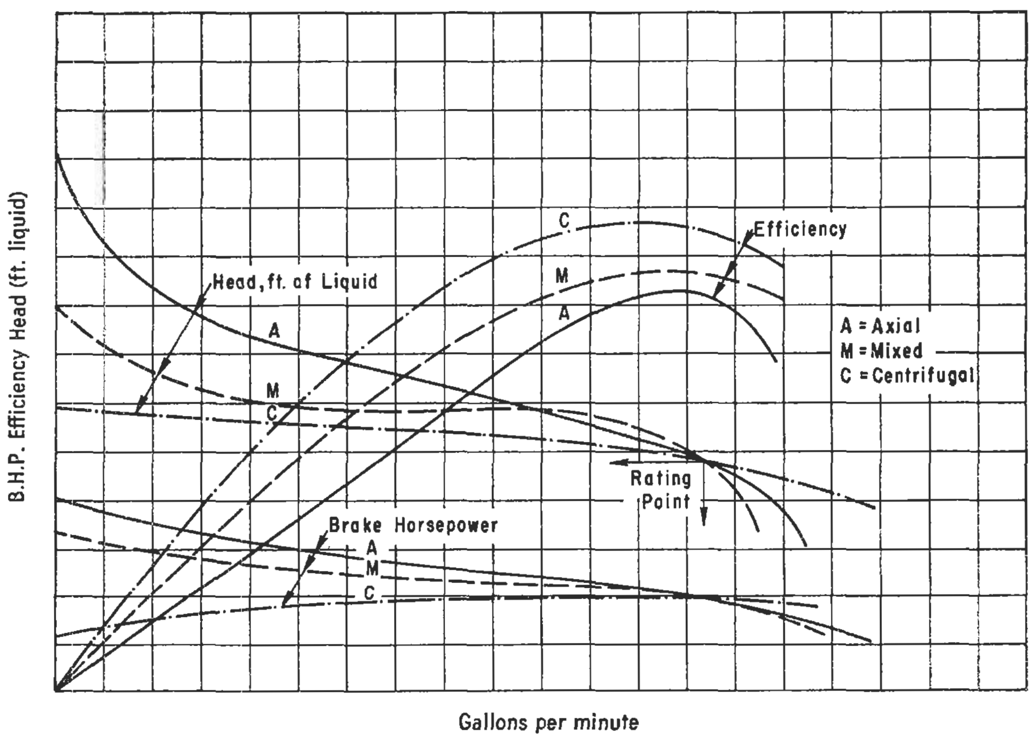


Figure 3-32. Comparison of impeller types for centrifugal pump performance. (Adapted by permission from *Pic-a-Pump*, Allis-Chalmers Mfg. Co.)

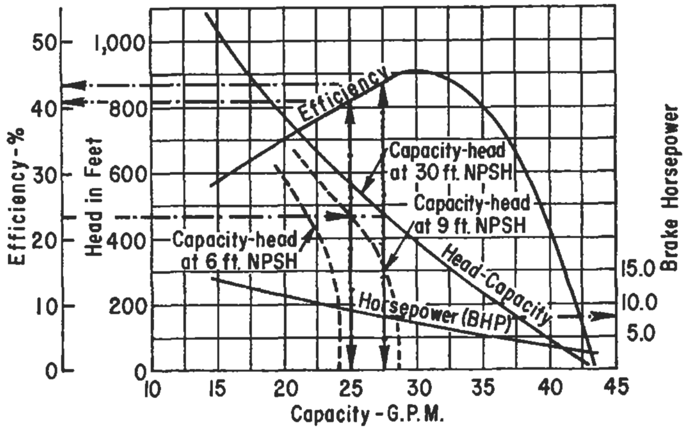


Figure 3-33. Performance of turbine type centrifugal pump. (Courtesy Roy E. Roth Co.)

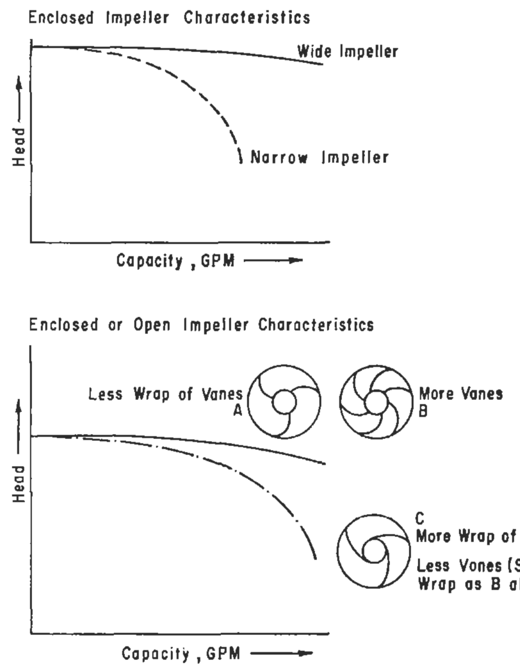
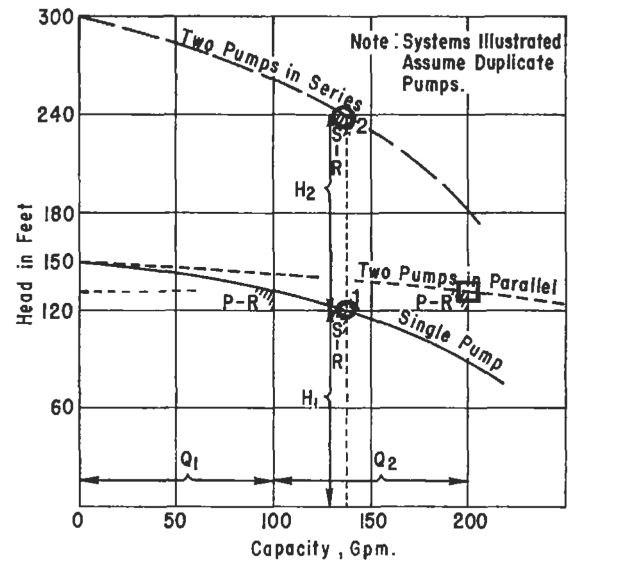


Figure 3-34. Impeller performance guide. Wrap refers to curvature of vanes on impeller. (Adapted by permission, *Pic-a-Pump,* Allis-Chalmers Mfg. Co.)

Table 3-5
Pump Selection Guide

Feature	Horizontal	Vertical
Space Requirements	Less head room	Less floor area, more head room.
NPSH	Requires more	Requires less
Priming	Required*	Usually not required
Flexibility (Relative to future changes)	Less	More
Maintenance	More accessible	Major work project
Corrosion and Abrasion	No great problem	Can be considerable problem
Cost	Less	More (requires more alloy to handle corrosive Fluid)

*For some conditions



Pump in Series: Q = Constant
H (Total) = $H_1 + H_2 + '''$
2○ = S-R denotes Series-Rating Point, Total
Pumps in Parallel: H = Constant
Q (Total) = $Q_1 + Q_2 + '''$ (at H for each Single Pump Curve)
□ = P-R denotes Parallel-Rating Point
1○ = Single pump rating

Figure 3-35. Operation curves of two duplicate centrifugal pumps in series and parallel.

and the pumps would "hunt" back and forth with no means to become stabilized.

Figures 3-36A, 3-36B, and 3-36C represent typical and actual performance curves showing discharge total head (*head* pressure at pump outlet connection for any fluid), required minimum water horsepower (for pumping water), and capacity or pumping volume of the pump (for any fluid) for several impeller diameters that would fit the same case (housing). In addition the important $NPSH_R$ (net positive suction head required by the pump) characteristics of the pump's design, impeller entrance opening and diameter, and the hydraulic operating efficiency of the pump at the fixed designated speed of the performance curves are shown on the chart. All of this performance is for one specific impeller diameter of the fixed rotating speed (rpm), and the fixed impeller design pattern proprietary to the manufacturer (number, shape and spacing of vanes, and wrap or curvature of vanes).

Note that Figure 3-36B plots the $NPSH_R$ curve for this "family" of impellers (different diameters, but exact same

Table 3-6

Type Selection Based on Liquid Handled

Liquid	Basic Pump Type	Type Impellers
Water and other clear non-corrosive liquids at cold or moderate temperatures.	Single or double suction	Closed except for very small capacities
Water above 250° F.	Single or double suction. This is usually boiler feed service at high pressures requiring multistage pumps.	Closed except for very small capacities
Hydrocarbons, hot	Single suction, often of the special type called refinery pumps, designed particularly for high temperature service.	Closed with large inlets.
Corrosives: Mildly acid or alkaline	Single or double suction	
Strongly acid or alkaline	Single or double suction with single suction probably less expensive if available for the rating.	Closed except for very small capacities or where liquid tends to form scale on surfaces of moving parts.
Hot corrosives	Single suction, with many refinery pump types also used here because of high temperatures and corresponding suction pressures.	
Water with solids in suspension: Fine abrasives	Single suction with end clearance wearing fits. If all particles pass through ⅛″ mesh screen, rubber lined pumps are available which will give many times the life of metal pumps, providing no chemical action or excessive temperature will deteriorate the rubber. Special rubber compounds can be applied to improve resistance to certain chemicals.	Open, which allows better application of the rubber, except in larger sizes. Also made in closed type.
Coarse abrasives	Single suction. Not available for full range of ratings, that is, small capacities not too easily obtained. Often have very large impellers operated at slow speeds for use when solids larger than 1″ diameter are the standard diet. This would be of the type called dredge pumps handling sizeable rocks.	Closed.
Pulpy solids such as paper stock	Single suction. Double suction only used on very slight solids concentrations and then with special end clearance wearing fits.	Closed. Open type used to be standard but change to end clearance wearing fits made closed impellers better suited.

design dimensions and features), while Figure 3-36A shows the $NPSH_R$ numbers printed at selected points on the curve.

Figure 3-36C illustrates the change in performance for the exact same pump, *same* impellers, but for different rotating speeds of 1750 and 3550 rpm. (Note that the respective motor designated standard speeds are 1800 and 3600 rpm, but the pump manufacturer cannot count on these speeds under load in order to provide performance information the customer needs for design of a system.)

Hydraulic Characteristics For Centrifugal Pumps

Capacity: the rate of liquid or slurry flow through a pump. This is usually expressed as gallons per minute (GPM) by pump manufacturers and design engineers in the chemical and petrochemical industries. A few convenient conversions are:

1 imperial gal/min = 1.201 U.S. GPM

1 barrel (42 gal)/day = 0.0292 U.S. GPM

For proper selection and corresponding operation, a pump capacity must be identified with the actual pumping temperature of the liquid in order to determine the proper power requirements as well as the effects of viscosity.

Figure 3-36A illustrates typical manufacturers' performance curves for centrifugal pumps as a function of capacity.

Pumps are normally selected to operate in the region of high efficiency, and particular attention should be given to avoiding the extreme right side of the characteristic curve where capacity and head may change abruptly.

Total Head: the pressure available at the discharge of a pump as a result of the change of mechanical input energy into kinetic and potential energy. This represents the total energy given to the liquid by the pump. Head, previously known as total dynamic head, is expressed as feet of fluid being pumped.

The *total head* read on the pump curve is the *difference* between the *discharge head* (the sum of the gauge reading on the discharge connection on the pump outlet, for most pumps corrected to the pump centerline, plus the velocity head at the point where the gauge is attached) and the *suction head* (the sum of the suction gauge reading corrected to the pump centerline and the velocity head at the point of attachment of the suction gauge) [25]. Note that the suction gauge reading may be positive or negative, and if negative, the discharge head minus a minus suction (termed lift) creates an additive condition. (See later discussion.)

This is shown on the curves of Figure 3-36A. *This head produced is independent of the fluid being pumped and is, there-*

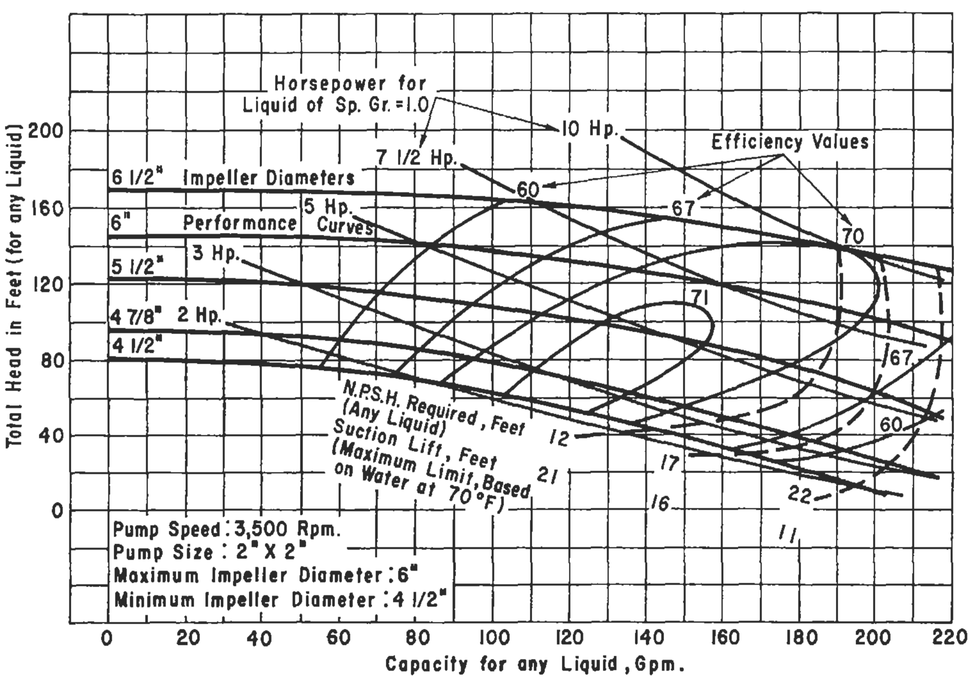


Figure 3-36A. Typical centrifugal pump curves. (Adapted by permission, Allis-Chalmers Mfg. Co.)

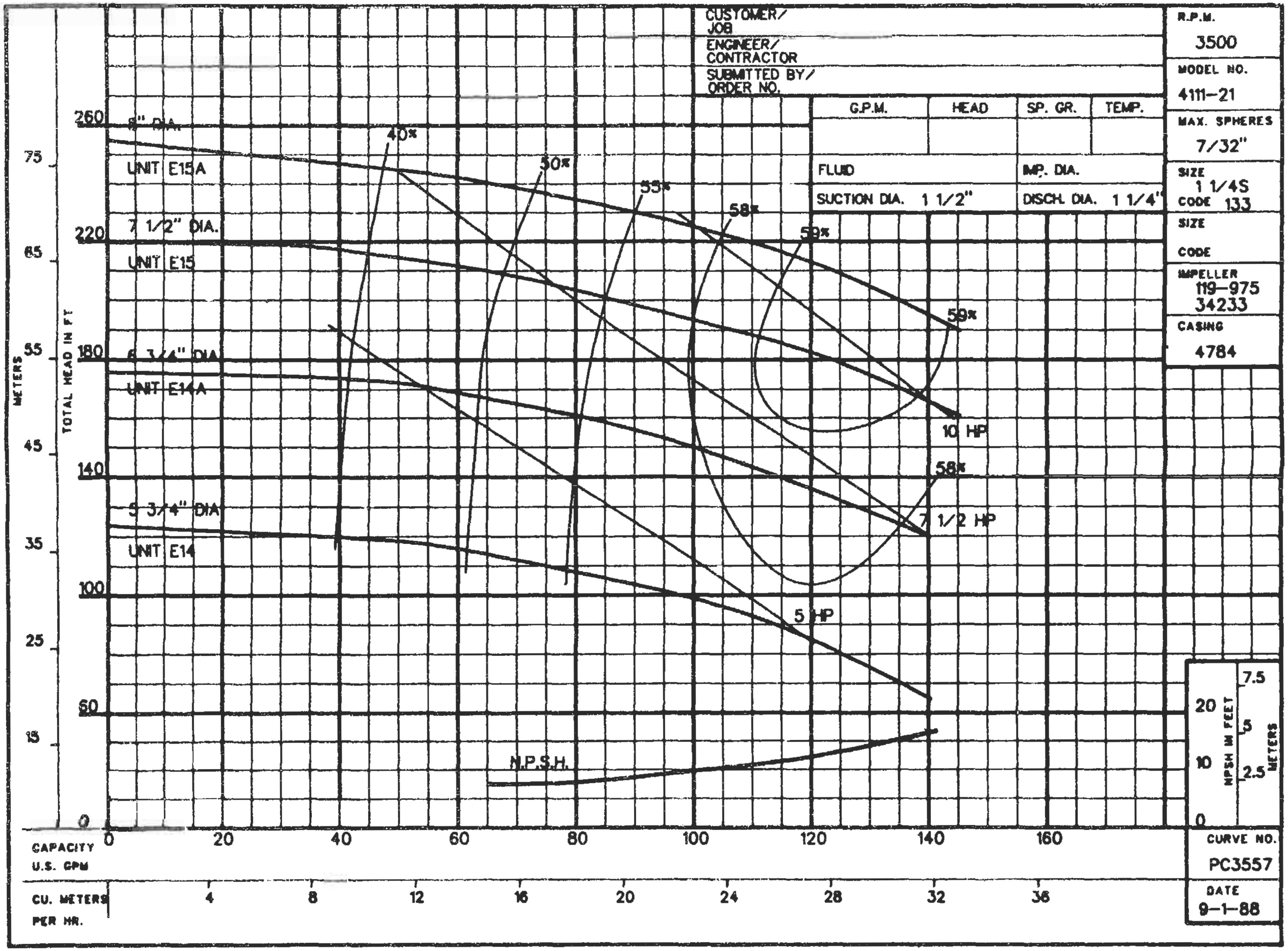
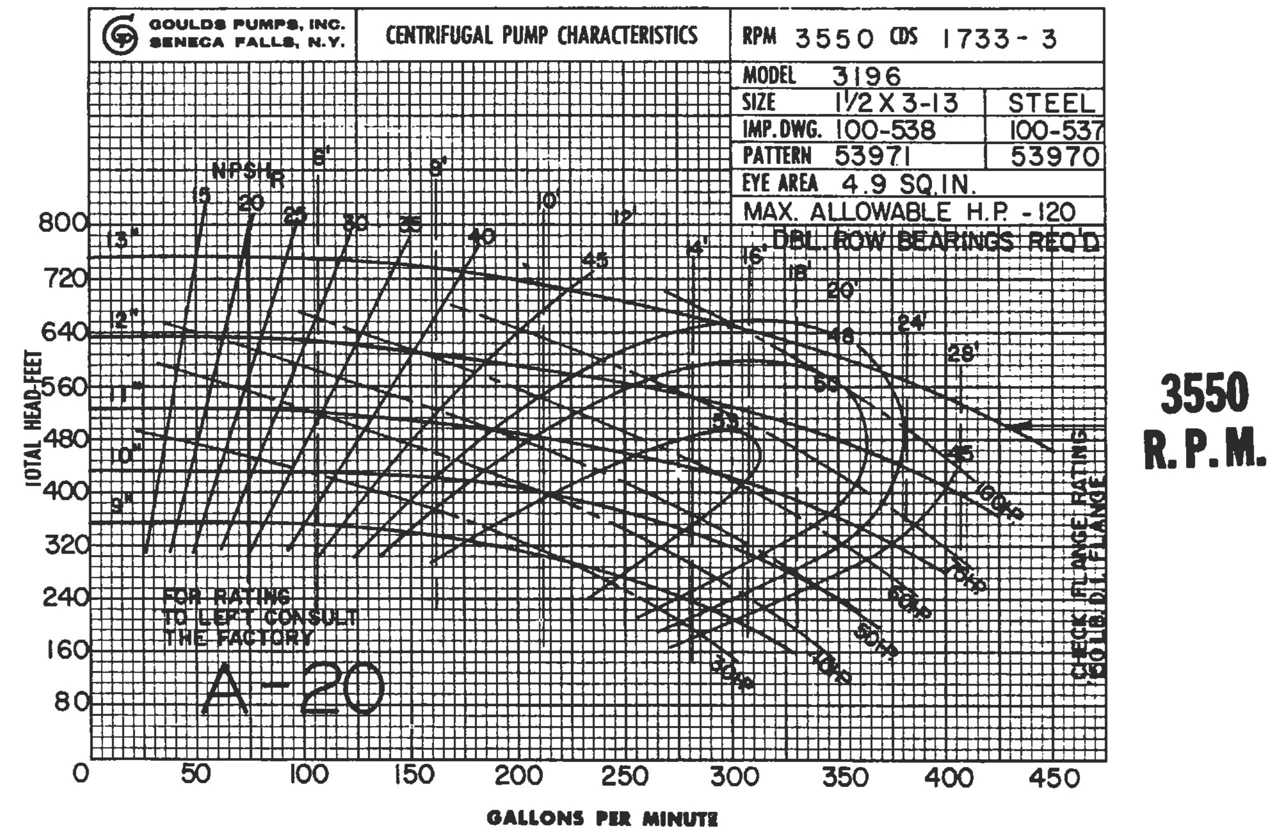


Figure 3-36B. Typical performance curve showing NPSH in convenient form. (By permission, Crane Co., Deming Pump Div.)



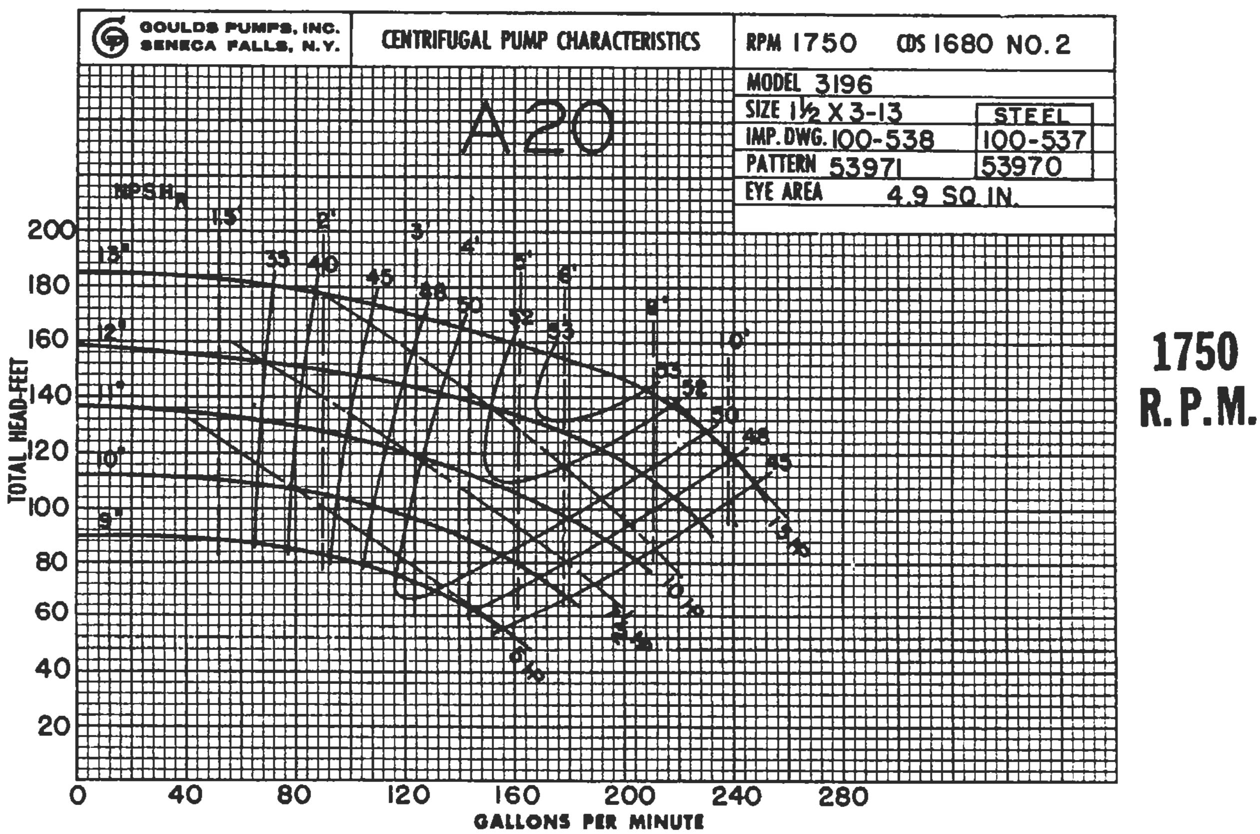


Figure 3-36C. Illustrates exact same pump casing and impellers at two different shaft speeds. (By permission, Goulds Pumps, Inc.)

fore, the same for any fluid through the pump at a given speed of rotation and capacity.

Through conversion, head may be expressed in units other than feet of fluid by taking the specific gravity of the fluid into account.

$$\text{(Head in feet)}, H = (\text{psi})(2.31)/\text{SpGr, for any fluid} \quad (3\text{-}1)$$

Note that psi (pounds per square inch) is pressure on the system and is not expressed as absolute unless the system is under absolute pressure. Feet are expressed as head, not head absolute or gauge (see later example). Note the conversion of psi pressure to feet of head pressure.

$$\text{or, (head in ft)}, H = (\text{psi})(144/\rho) \quad (3\text{-}2)$$

where ρ = fluid density, lb/cu ft

1 lb/sq in. = 2.31 ft of water at SpGr = 1.0
1 lb/sq in. = 2.31 ft of water/SpGr of liquid = ft liquid
1 in. mercury = 1.134 ft of water = 1.134/SpGr liquid, as ft liquid

For water, SpGr = 1.0 at 62°F, although for general use it can be considered 1.0 over a much wider range. For explanation of vacuum and atmospheric pressure, see Chapter 2.

Example 3-1: Liquid Heads

If a pump were required to deliver 50 psig to a system, for water, the feet of head on the pump curve must read,

2.31 (50) = 115.5 ft

For a liquid of SpGr 1.3, the ft of head on the pump curve must read, 115.5/1.3 = 88.8 ft of liquid.

For liquid of SpGr 0.86, the ft of head on the pump curve must read, 115.5/0.86 = 134.2 ft of liquid.

If a pump were initially selected to handle a liquid where SpGr = 1.3 at 88.8 ft, a substitution of light hydrocarbon where SpGr = 0.86 would mean that the head of liquid developed by the pump would still be 88.8 feet, but the pressure of this lighter liquid would only be 88.8/[(2.31)/(0.86)] or 44.8 psi. Note that for such a change in service, the impeller seal rings, packing (or mechanical seal) and pressure rating of casing must be evaluated to ensure proper operation with a very volatile fluid. For other examples, see Figure 3-37.

The total head developed by a pump is composed of the difference between the static, pressure and velocity heads plus the friction entrance and exit head losses for the suction and discharge sides of the pump. Refer to Figures 3-38 and 3-39.

$$H = h_d - h_s \quad (3\text{-}3)$$

The sign of h_s when a suction lift is concerned is negative, making $H = h_d - (-h_s) = h_d + h_s$

The three main components illustrated in the examples are (adapted from [5]):

1. Static head
2. Pressure head
3. Friction in piping, entrance and exit head losses

A pump is acted on by the total forces, one on the suction (inlet) side, the other on the discharge side. By subtracting (algebraically) all the suction side forces from the discharge side forces, the result is the net force that the pump must work against. However, it is extremely important to recognize the algebraic sign of the suction side components, that is, if the level of liquid to be lifted into the pump is below the pump centerline, its algebraic sign is negative (−). Likewise, if there is a negative pressure or vacuum on the liquid below the pump centerline, then this works against the pump and it becomes a negative (−). (See discussion to follow.)

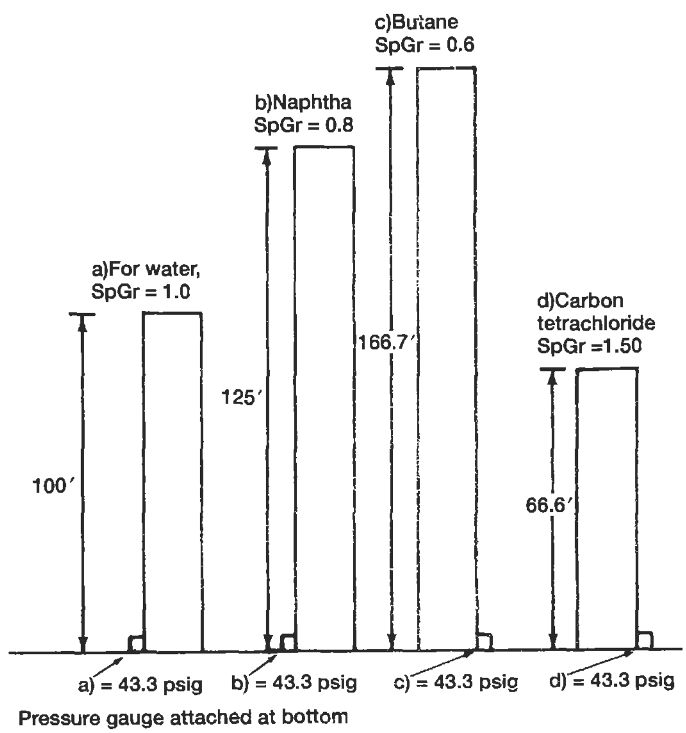


Figure 3-37. Comparison of columns of various liquids to register 43.3 psig on pressure gauge at bottom of column.

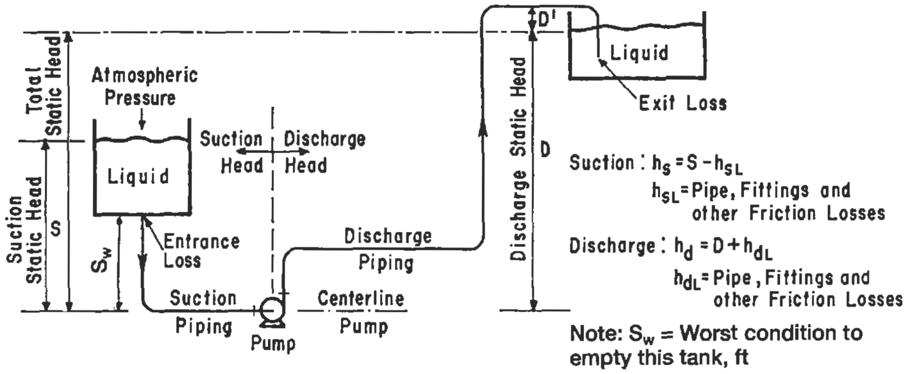


Figure 3-38. Suction head system.

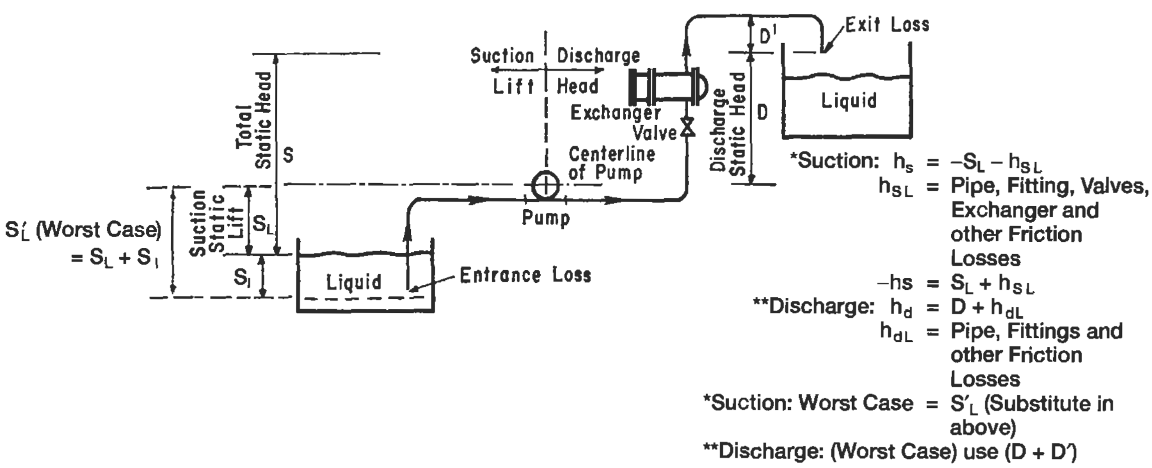


Figure 3-39. Suction lift system

Static Head

This is the overall height to which the liquid must be raised.

For Figure 3-40A

Discharge static head: H

Suction static head: L (actually −L)

Total system static head: H + L;
actually H − (−L) (3-4)

For Figure 3-40B

Discharge static head: H (from centerline of pump)

Suction static head: S, (actually +S)

Total system static head: H − S; or H− (+S) (3-5)

Pressure Head

For Figure 3-40C

Discharge pressure head	= 100 psig
Suction pressure head	= 0 psig
Total pressure head	= 100 − (+0) = 100 psig
	= 100(2.31)* = 231 ft of water

Note: The totals are differentials and neither gauge nor absolute values.

*Applies to water only. For the other fluids use appropriate specific gravity conversion.

For Figure 3-40D

Discharge pressure head	= 100 psig
Suction pressure head	= +50 psig (=64.7 psia)
Total pressure head	= 100 − (+50) = 50 psi not gauge or absolute = 50 (2.31) = 115.5 ft of water

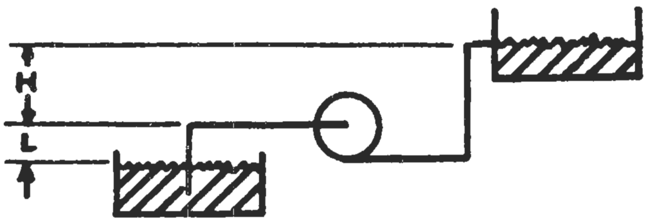


Figure 3-40A. Static head, overall = H + L. (Adapted by permission, *Centrifugal Pumps Fundamentals,* Ingersoll-Rand Co., Washington, N.J. 07882.)

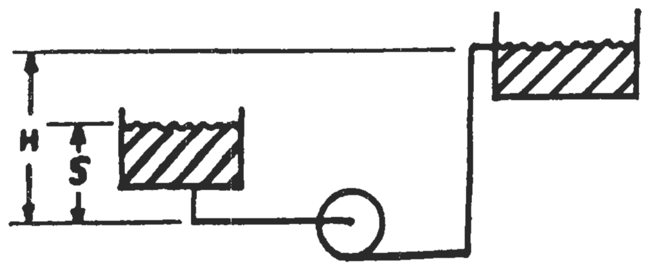
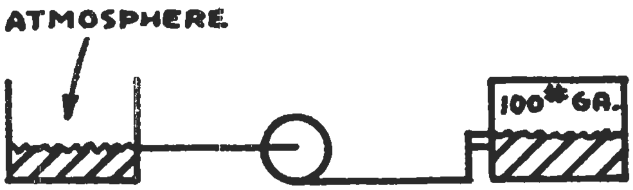


Figure 3-40B. Static head, overall = H − S. (Adapted by permission, *Centrifugal Pumps Fundamentals,* Ingersoll-Rand Co., Washington, N.J. 07882.)



The above examples purposely disregarded pressure head, friction, entrance, and exit head losses.

Figure 3-40C. Pressure head. (Adapted by permission, *Centrifugal Pumps Fundamentals,* Ingersoll-Rand Co., Washington, N.J. 07882.)

Note that both the discharge and suction pressures must be on the same base/units. These illustrations are for static head only, while overall the pump has to work against the static and the pressure heads. (To be discussed.)

For Figure 3-40E

Discharge pressure head	= 100 psig = 231 ft water (system fluid)
Discharge static head	= 50 ft
Total discharge head	= 231 + 50 = 281 ft (*Note that no flow friction losses or entrance/exit losses are included in this example)
Suction pressure head	= +50 psig = +115 ft water (system fluid)
Suction static head	= −10 ft
* Total suction head	= +115 + (−10) = +105 ft
* Total head on pump	= 281 − 105 = 176 ft fluid

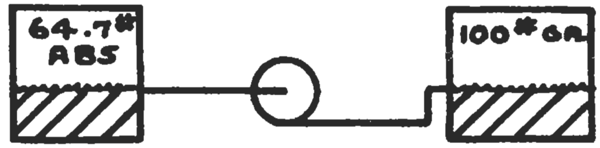


Figure 3-40D. Pressure head, positive suction. (Adapted by permission, *Centrifugal Pumps Fundamentals,* Ingersoll-Rand Co., Washington, N.J. 07882.)

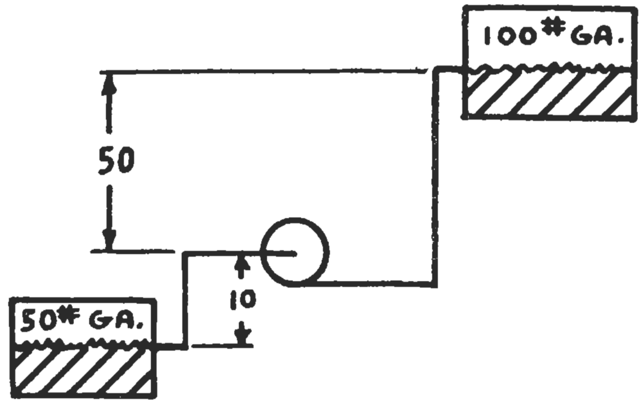


Figure 3-40E. Pressure head with negative suction. (Adapted by permission, *Centrifugal Pumps Fundamentals,* Ingersoll-Rand Co., Washington, N.J. 07882.)

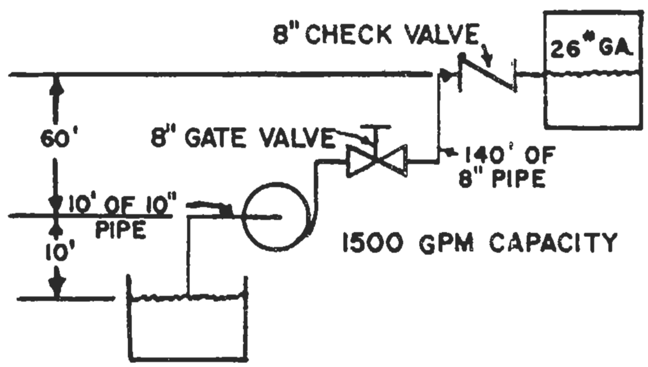


Figure 3-40F. Pumping arrangement for Example 3-2. (Adapted by permission, *Centrifugal Pumps Fundamentals,* Ingersoll-Rand Co., Washington, N.J. 07882.)

Friction Losses Due to Flow

Friction, entrance and exit heads, valve losses

These losses and calculation methods were presented in Chapter 2. Comments here will be limited. These losses are a function of the characteristics of the fluid flowing in the piping systems and the velocities of flow. Entrance and exit losses relate to the pipe and not the suction or discharge connections at the pump. Usually they are very small, but cannot be ignored without checking. Velocity heads at the pump connections are considered internal losses. These are handled by the manufacturer's design of the pump and are not considered with the external losses in establishing the pump heads.

Example 3-2: Illustrating Static, Pressure, and Friction Effects

Refer to Figure 3-40F for basis of the example.

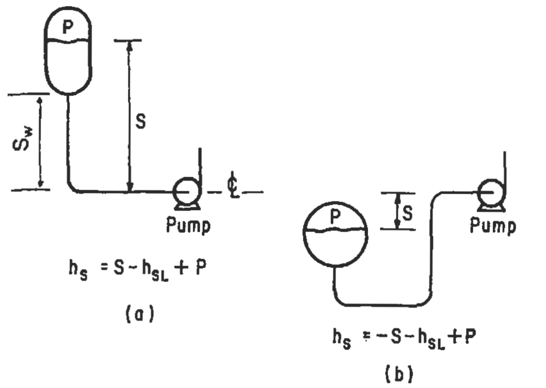
To aid in speed of computation, the friction figures are taken from the Cameron Hydraulic Tables in Chapter 2 and use water, which is suited to these tables, as an example fluid.

Discharge head = 60 ft		
Discharge pressure head = 26 psig (2.31 ft/psi)	=	60 ft gauge
Discharge friction and exit head (at pipe/tank):		
140 ft of 8-in. pipe: 6.32 ft/100(140)	=	8.8 ft
3 8-in. 90° ells: (6.32/100)(3)(20.2)	=	3.8
1 8-in. gate valve	=	0.3
1 8-in. check valve	=	3.3
*Exit loss: Assume 8-in. pipe = vel hd	=	1.4
Subtotal, ft	=	17.6
Total discharge head	=	137.6 ft
Suction static head (lift)	=	−10.0 ft
Suction pressure head 0, psig (atmos)	=	0.0
Suction friction and entrance head:		
10 ft of 10-in. pipe, (2.1 ft/100)(10)	=	0.2
1 10-in. suction 90° ell; (2.1/100)(25.3)	=	0.5
*Entrance loss: 10-in. pipe assume = vel head	=	0.6
Subtotal	=	−1.3
Total suction head = −10 + (−1.3)	=	−11.3 ft
Total pump head = 137.6 − (−11.3)	=	148.9 ft

*These are not velocity heads at pump connections, but are related to the piping connections. See earlier note in this regard.

Suction Head or Suction Lift, h_s

The total suction head, Figure 3-41, is the difference in elevation between the liquid on the pump suction side and the centerline of the pump (plus the velocity head). Note that the suction head is positive when above the



Note: When P is expressed in absolute pressure units, h_S will be in absolute units. If P is less than atmospheric pressure: P is (−) if expressed as a gauge reading and will be a negative feet of liquid. P is (+) if expressed in absolute units. The friction loss h_{SL} includes any entrance or exit losses and other such fittings in the system.

Figure 3-41. Typical suction systems. (Adapted by permission, Carter, R. and Karassik, "R.P.-477." Worthington Corp.)

pump centerline, and that it is decreased with an increase in friction losses through the suction piping system. Thus,

$$\text{total suction head (TSH)} = \text{static head} - h_{SL} \tag{3-6}$$

The total suction lift is defined as above except the level of the liquid is below the centerline of the pump or the head is below atmospheric pressure. Its sign is negative. Total Suction Lift (TSL) = static lift plus friction head losses.

In summary to clarify:

1. The pressure units (gauge or absolute) must be consistent for all components used in determining both suction side and discharge side conditions. Most designers use gauge as a reference, but this is not necessary.
2. Static head is positive pressure of fluid on pump suction above its centerline (S), (+).
3. Positive external pressure, P, on surface of fluid on pump suction is used as a positive integer, expressed as feet of fluid, (+).
4. Partial vacuum, P, on the surface of liquid is a negative pressure. As a *partial vacuum* expressed as a gauge reading as feet of liquid below atmospheric, the pressure is negative and would be designated by a minus (−) sign. A partial vacuum, P, expressed as absolute vacuum or *absolute pressure* would be designated by a positive (+) sign. It is essential to be consistent for all pressure units. If absolute units are used, the total suction head would be in absolute

units and the discharge head must be calculated in absolute units.

5. Suction lift is a negative suction head, S, used to designate a negative static condition on the suction of the pump (below atmospheric). The sign for suction head is positive (+), while its corresponding terminology of suction lift is negative (−), since the term "lift" denotes a negative condition. Note that the only difference in these terms is the difference in signs.

This applies because the total head for a pump is total discharge head a(+), minus (−) the [suction head, a(+)], or [suction lift, a(−)].

For general service the average centrifugal pump should lift about 15 feet of water on its suction side. However, since each process situation is different, it is not sufficient to assume that a particular pump will perform the needed suction lift. Actually, certain styles or models of a manufacturer's pumps are often specially adapted to high lift conditions. On the other hand it is unnecessary to select a high lift pump when pressure head or flooded suction conditions prevail. Proper evaluation of suction lift conditions cannot be over emphasized.

The theoretical maximum suction lift at sea level for water (14.7 psi) (2.31 ft/psi) = 34 ft. However, due to flow resistance, this value is never attainable. For safety, 15 feet is considered the practical limit, although some pumps will lift somewhat higher columns of water. When sealing a vacuum condition above a pump, or the pump pumps from a vessel, a seal allowance to atmosphere is almost always taken as 34 feet of water. High suction lift causes a reduction in pump capacity, noisy operation due to release of air and vapor bubbles, vibration and erosion, or pitting (cavitation) of the impeller and some parts of the casing. (The extent of the damage depends on the materials of construction.)

Discharge Head, h_d

The discharge head of a pump is the head measured at the discharge nozzle (gauge or absolute), and is composed of the same basic factors previously summarized; 1. static head 2. friction losses through pipe, fittings, contractions, expansions, entrances and exits 3. terminal system pressure.

Some typical discharge systems are given in Figure 3-42. General practice is to express the terminal discharge pressure, P, at a vessel as in Figure 3-42 in terms of gauge pressure, and hence P = 0 for atmospheric discharge. If P is less than atmospheric or otherwise expressed in absolute units, then it must be added as equivalent feet of liquid to the value of h_d ordinarily expressed as a gauge reading.

Figures 3-38 and 3-39 illustrate the use of siphon action in pump systems. Theoretically, the head in the siphon should be recoverable, but actually it may not, at least not equivalent foot for foot. Usually not more than 20 feet of siphon action can be included [4] even though 34 feet are theoretical at sea level. The siphon length is D′ in the figures [32]. For some systems the discharge head on the pump should be used as (D + D′), neglecting the siphon action. In any case, if air can be trapped in the loop, (and it usually can) it must be vented during start-up, otherwise the pump will be pumping against the head established using (D + D′). On start-up the flow can be gradually increased, making more head available from the pump to overcome the higher starting head of the system. This should not be overlooked nor underestimated in determining the specifications for the pump.

Velocity Head

Velocity head is the kinetic energy of a liquid as a result of its motion at some velocity, v. It is the equivalent head

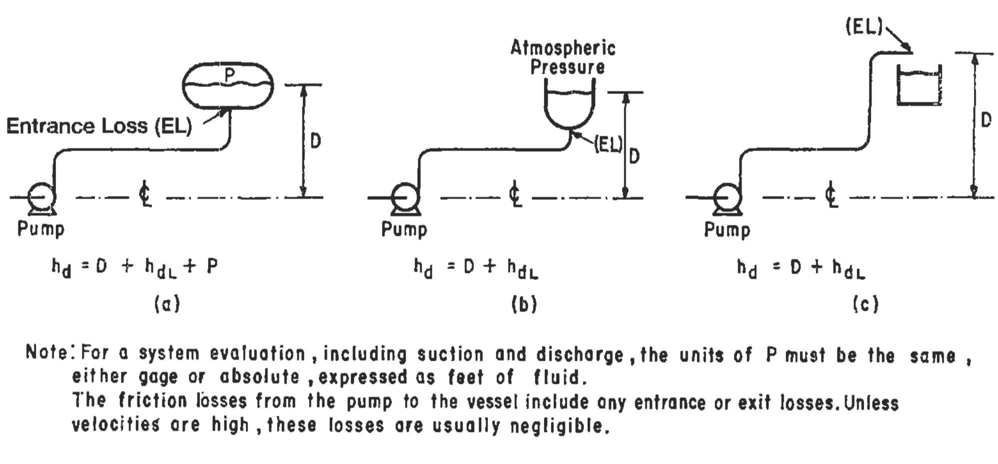


Figure 3-42. Typical discharge systems.

in feet through which water would have to fall to acquire the same velocity, expressed as foot-pounds per pound of liquid.

$$h_v = v^2/2\ g, \text{ feet of fluid} \tag{3-7}$$

where h_v = velocity head, ft
v = liquid velocity, ft/sec
g = acceleration of gravity, ft/sec-sec

As a component of both suction and discharge heads, velocity head is determined at the pump suction or discharge flanges respectively, and added to the gauge reading. The actual pressure head at any point is the sum of the gauge reading plus the velocity head, the latter not being read on the gauge since it is a kinetic energy function as contrasted to the measured potential energy. The values are usually (but not always) negligible. Present practice is for these velocity head effects at the pump suction and discharge connections to be included in the pump performance curve and pump design, and need not be actually added to the heads calculated external to the pump itself [5].

It is important to verify the effects of velocity head on the suction and discharge calculations for pump selection. In general, velocity head (kinetic energy) is smaller for high head pumps than for low head units. Sometimes the accuracy of all the other system calculations does not warrant concern, but for detailed or close calculations velocity head should be recognized. The actual suction or discharge head of a pump is the *sum* of the gauge reading from a pressure gauge at the suction or discharge and the velocity heads calculated at the respective points of gauge measurement.

Regardless of their density, all liquid particles moving at the same velocity in a pipe have the same velocity head [11]. The velocity head may vary across a medium to large diameter pipe. However, the average velocity of flow, that is, dividing the total flow as cu ft/sec by the cross-sectional area of the pipe is usually accurate enough for most design purposes.

Using the example of Reference [25], for a pump handling 1500 GPM, having a 6-inch discharge connection and 8-inch suction connection, the discharge velocity head is 4.5 ft and the suction is 1.4 ft, calculated as shown above. If the suction gauge showed 8.6 ft, the true head would be 8.6 + 1.4 = 10.0. If the discharge head showed 105.5 ft head, the true total head would be 105.5 + 4.5 = 110.0 ft, less (8.6 + 1.4) or 100 ft. The net true total head would be 110 ft − 10 ft = 100.0 ft. Looking only at the gauge readings, the difference would be 105.5 − 8.6 = 96.9 ft, giving an error of 3.1% of the total head. As an alternate example, if the discharge head were 45.5 ft, then the true total head = (45.5 + 4.5) − (8.6 + 1.4) = 40 ft, and the difference in gauge readings would be 45.5 − 8.6 = 36.9 ft, or an error of 7.8%.

Most designers ignore the effects of velocity head, but the above brief examples emphasize that the effect varies depending on the situation and the degree of accuracy desired for the head determinations.

Friction

The friction losses for fluid flow in pipe valves and fittings are determined as presented in Chapter 2. Entrance and exit losses must be considered in these determinations, but are not to be determined for the pump entrance or discharge connections into the casing.

NPSH and Pump Suction

Net positive suction head (in feet of liquid absolute) above the vapor pressure of the liquid at the pumping temperature is the absolute pressure available at the pump suction flange, and is a very important consideration in selecting a pump which might handle liquids at or near their boiling points, or liquids of high vapor pressures.

Do not confuse NPSH with suction head, as suction head refers to pressure above atmospheric [17]. If this consideration of NPSH is ignored the pump may well be inoperative in the system, or it may be on the border-line and become troublesome or cavitating. The significance of NPSH is to ensure sufficient head of liquid at the entrance of the pump impeller to overcome the internal flow losses of the pump. This allows the pump impeller to operate with a full "bite" of liquid essentially free of flashing bubbles of vapor due to boiling action of the fluid.

The pressure at any point in the suction line must never be reduced to the vapor pressure of the liquid (see Equation 3-6). Both the suction head and the vapor pressure must be expressed in feet of the liquid, and *must* both be expressed as gauge pressure or absolute pressure. Centrifugal pumps cannot pump any quantity of vapor, except possibly some vapor entrained or absorbed in the liquid, but *do not count on it.* The liquid or its gases must not vaporize in the eye/entrance of the impeller. (This is the lowest pressure location in the impeller.)

For low available NPSH (less than 10 feet) the pump suction connection and impeller eye may be considerably oversized when compared to a pump not required to handle fluid under these conditions. Poor suction condition due to inadequate available NPSH is one major contribution to cavitation in pump impellers, and this is a condition at which the pump cannot operate for very long without physical erosion damage to the impeller. See References [11] and [26].

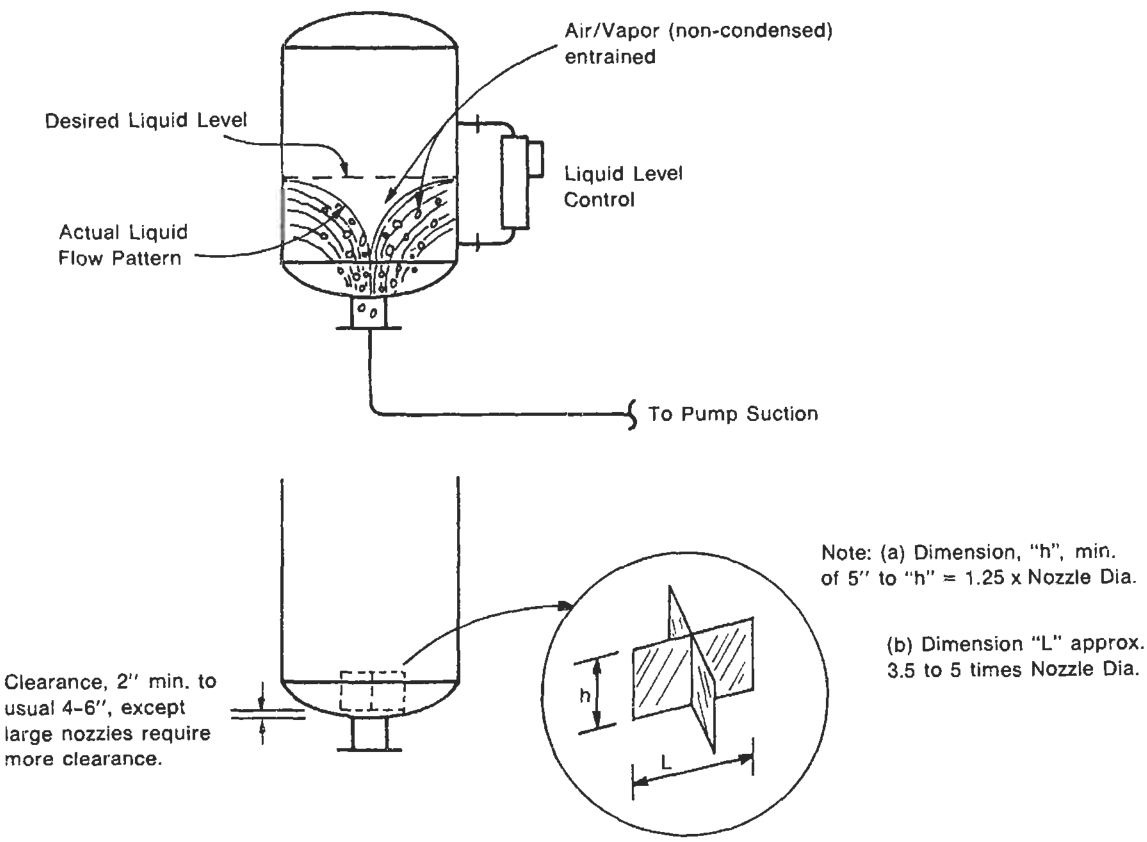
Cavitation of a centrifugal pump, or any pump, develops when there is insufficient NPSH for the liquid to flow into the inlet of the pump, allowing flashing or bubble formation in the suction system and entrance to the pump. Each pump design or "family" of dimensional features related to the inlet and impeller eye area and entrance pattern requires a specific minimum value of NPSH to operate satisfactorily without flashing, cavitating, and loss of suction flow.

Under cavitating conditions a pump will perform below its head-performance curve at any particular flow rate. Although the pump may operate under cavitation conditions, it will often be noisy because of collapsing vapor bubbles and severe pitting, and *erosion* of the impeller often results. This damage can become so severe as to completely destroy the impeller and create excessive clearances in the casing. To avoid these problems, the following are a few situations to watch:

1. Have $NPSH_A$ available *at least* 2 feet of liquid greater than the pump manufacturer requires under the worst possible operating conditions (see pump curves Figures 3-36A, B, C) with pump curve values for NPSH *expressed as feet of liquid handled*. These are the pump's required minimum $NPSH_R$. The pump's piping and physical external system provides the *available* $NPSH_A$.

$$NPSH_A \text{ must be} > NPSH_R \quad (3\text{-}8)$$

2. Internal clearance wear inside pump.
3. Plugs in suction piping system (screens, nozzles, etc.).
4. Entrained gas (non-condensable).
5. Deviations or fluctuations in suction side pressures, temperatures (increases), low liquid level.
6. Piping layout on suction, particularly tee-intersections, globe valves, baffles, long lines with numerous elbows.
7. Liquid vortexing in suction vessel, thus creating gas entrainment into suction piping. Figure 3-43 suggests a common method to eliminate suction vor-



(c) Bottom of vortex breaker may be attached to bottom or raised up 2" to 4". Vortex "cross" must be sturdy, welded of heavy plate (not light sheet metal). Vortex breaker must not restrict liquid flow into nozzle opening, but prevent swirling of liquid.

Figure 3-43. Liquid vortex in vessel and suggested design of vortex breaker.

texing. Since the forces involved are severe in vortexing, the vortex breaker must be of sturdy construction, firmly anchored to the vessel.

8. Nozzle size on liquid containing vessel may create severe problems if inadequate. Liquid suction velocities, in general, are held to 3–6.5 feet per second. Nozzle losses are important to recognize by identifying the exit design style (see Chapter 2). Usually, as a guide, the suction line is *at least* one pipe size larger than the pump suction nozzle.

The $NPSH_A$ *available* from or in the liquid system on the suction side of a pump is expressed (corrected to pump centerline) as:

$$NPSH_A = S + (p'_a - p'_{vp}) - h_{SL} \quad (3\text{-}9)$$

$$NPSH_A = S + (P_a - P_{vp})\ (2.31/SpGr) - h_{SL} \quad (3\text{-}10)$$

Where p'_a or P_a represent the absolute pressure in the vessel (or atmospheric) on the liquid surface on the suction side of the pump.

p'_{vp} or P_{vp} represent the absolute vapor pressure of the liquid at the pumping temperature.

h_{SL} is the suction line, valve, fitting and other friction losses from the suction vessel to the pump suction flange.

S may be (+) or (−) depending on whether static head or static lift is involved in the system.

This *available* value of $NPSH_A$ (of the system) must always be greater by a minimum of two feet and preferably three or more feet than the *required* NPSH stated by the pump manufacturer or shown on the pump curves in order to overcome the pump's internal hydraulic loss and the point of lowest pressure in the eye of the impeller. The NPSH required by the pump is a function of the physical dimensions of casing, speed, specific speed, and type of impeller, and must be satisfied for proper pump performance. The pump manufacturer must always be given complete suction conditions if he is to be expected to recommend a pump to give long and trouble-free service.

As the altitude of an installation increases above sea level, the barometric pressure, and hence p'_a or P_a decreases for any open vessel condition. This decreases the available NPSH.

Figure 3-36A represents a typical manufacturer's performance curve. The values of $NPSH_R$ given are the minimum values required at the pump suction. As mentioned, good practice requires that the $NPSH_A$ available be at least two feet of liquid above these values. It is important to recognize that the $NPSH_R$ and Suction Lift Values are for handling water at about 70°F. To use with other liquids it is necessary to convert to the equivalent water suction lift at 70°F and sea level.

Total Suction Lift (as water at 70°F) = $NPSH_A$ (calculated for fluid system) − 33 feet. The vapor pressure of water at 70°F is 0.36 psia.

Example 3-3: Suction Lift

What is the Suction Lift value to be used with the pump curves of Figure 3-36A, if a gasoline system calculates an NPSH of 15 feet *available:*

Total Suction Lift (as water) = 15 − 33 = −18 feet. Therefore, a pump must be selected which has a lift of at least 18 feet. The pump of Figure 3-36A is satisfactory using an interpolated Suction Lift line between the dotted curves for 16 feet and 21 feet of water. The performance of the pump will be satisfactory in the region to the left of the new interpolated 18-foot line. Proper performance should not be expected near the line.

If the previous system were at sea level, consider the same pump with the same system at an altitude of 6000 feet. Here the barometric pressure is 27.4 feet of water. This is 34 − 27.4 = 6.6 feet less than the sea level installation. The new $NPSH_A$ will be 15 ft − 6.6 ft = 8.4 feet *available.* Referring to the pump curve of Figure 3-36A it is apparent that this pump cannot do greater than 21 feet suction lift as water or 12 feet $NPSH_R$ of liquid (fluid).

Total Suction Lift as water = 8.4 − 33 = −24.6 feet. The pump curves show that 21 feet suction lift of water is all the pump can do, hence the 24.6 feet is too great. A different pump must be used which can handle this high a suction lift. Such a pump may become expensive, and it may be preferable to use a positive displacement pump for this high lift. Normally lifts are not considered reasonable if over 20 feet.

Example 3-4: NPSH Available in Open Vessel System at Sea level, Use Figure 3-38

Conditions: at sea level, atmospheric pressure, P_a = 14.7 psia.

Assume liquid is water at 85°F, vapor pressure = P_{vp} = 0.6 psia.

Assume tank liquid level is 10 feet above center line of pump, then S = +10 feet.

Assume that friction losses have been calculated to be 1.5 feet, h_{SL} = 1.5

Then: $NPSH_A$ available = $S + (P_a - P_{vp})(2.31/SpGr) - h_{SL}$

$$= +10 + (14.7 - 0.6)(2.31/0.997) - 1.5$$
$$= 41.2 \text{ ft (good)} \quad (3\text{-}10)$$

Note: For worst case, which is an empty tank, "S" becomes S_W on the diagram.

Example 3-5: NPSH Available in Open Vessel Not at Sea Level, Use Figure 3-39

Conditions: vessel is at altitude 1500 ft, where atmospheric pressure is 13.92 psia = p_a,

Liquid: water at 150°F, vapor pressure P_{vp} = 3.718 psia
SpGr = 0.982

Assume vessel liquid level is 12 ft below centerline of pump, $S_L = -12$.

Friction losses: assume calculated to be 1.1 ft of liquid.

Then: $NPSH_A$ available $= S + (P_a - P_{vp})(2.31/SpGr) - h_{SL}$
$= -12 + (13.92 - 3.718)(2.31/0.982) - 1.1$
$= +10.88$ ft (3-10)

The worst condition case should be calculated using S'_L, since this represents the maximum lift.

Example 3-6: NPSH Available in Vacuum System, Use Figure 3-41A

Conditions: vessel is liquid collector at 28 in. Hg Vacuum (referred to a 30 in. barometer). This is 30 − 28 = 2 in. Hg abs, or $P_a = [(14.7/30)](2) = 0.98$ psia.

Liquid: water at 101.2°F, vapor pressure = 0.98 psia.

Assume vessel liquid level is 5 feet above centerline of pump, $S = +5'$, worst case, $S_w = 2'$

Friction losses: assume to be 0.3 foot of liquid

Then: $NPSH_A$ available $= S + (P_a - P_{vp})(2.31/SpGr) - h_{SL}$
$= +5 + (0.98 - 0.98)(2.31/0.994) - 0.3$
$= +4.7$ ft (3-10)

Worst case = 1.7 (not practical design)

The pump selected for this application (water boiling at 0.98 psia) must have a required NPSH less than 4.7 ft, preferably about 3 to 3.5 ft. This is a difficult condition. If possible the vessel should be elevated to make more head (S) available, which will raise the available NPSH.

Example 3-7: $NPSH_A$ Available in Pressure System, Use Figure 3-41(b)

Conditions: vessel contains butane at 90°F and 60 psia system pressure. $P_a = 60$

Butane vapor pressure, P_{vp} at 90°F = 44 psia, SpGr = 0.58.

Assume liquid level is 8 feet below pump centerline, $S = -8$.

Friction losses: assume to be 12 ft of liquid.

Then: NPSH available $= S + (P_a - P_{vp})(2.31/SpGr) - h_{SL}$
$= -8 + (60 - 44)(2.31/0.58) - 12 = +43.8$ feet (3-10)

This presents no pumping problem.

Example 3-8: Closed System Steam Surface Condenser NPSH Requirements, Use Figure 3-44

This is a closed steam surface condenser system with condensate being pumped out to retreatment facilities. From the conditions noted on the diagram,

Friction loss in suction line side = 2.92 ft
Absolute pressure in condenser = p' = 1.5 in. Hg Abs
= 1.5(1.13 ft/in. Hg)
= 1.71 ft water
Water from steam tables at saturation = 1.5 in. Hg Abs @ 91.72°F
Vapor pressure, p'_{vp}, at 1.5 in. Hg Abs = 1.5(1.13)
= 1.71 ft water
$NPSH_A$ available $= +10 + (1.71 - 1.71) - 2.92$
$= +7.08$ ft

The suction head or lift for the pump (separate calculation from $NPSH_A$) is:

The 28.42 in. vacuum Hg (gauge) is equivalent to 1.5 in. Hg Abs

28.42 in. vacuum (1.137)	= 32.31 ft water
Static submergence	= 10.0 (see figure)
Friction/entrance losses	= 2.92 ft
Net static submergence	= 7.08 7.08 ft
Equivalent suction lift	= 25.23 ft [Note: 32.31 − 7.08]

(= vacuum effect less net submergence)

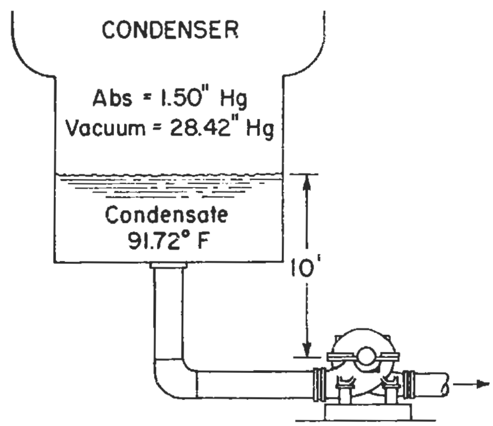


Figure 3-44. Surface condenser condensate removal. Closed system steam surface condenser NPSH requirements. (By permission, *Cameron Hydraulic Data,* 16th ed. Ingersoll-Rand Co., 1979, p. 1-12.)

Note that the equivalent suction lift must be added to the total discharge head for the pump system to obtain the total system head. Keep in mind that the work the pump must accomplish is overcoming the suction losses (+ or −) plus the discharge losses, that is, + discharge loss (all) − (+ if head, or − if lift on suction losses, all). Thus, the suction lift becomes a (−)(−) or a (+) to obtain the total system head. Keep in mind that a vacuum condition on the suction of a pump never helps the pump, but in effect is a condition that the pump must work to overcome.

Example 3-9: Process Vacuum System, Use Figure 3-45

For this process example, again using water for convenience, a low pressure, low temperature water is emptied into a vented vessel, and then pumped to the process at a location at about 3000 feet altitude (see Appendix A-6) where atmospheric pressure is approximately 13.2 psia. Water SpGr is at 200°F = 0.963.

Determine the $NPSH_A$ for pump:

$$NPSH_A = +S + (p_a - P_{vp})(2.31/SpGr) - h_{sl}$$
$$= +10 + (13.2 - 11.5)(2.3)/.963 - 1.0$$
$$NPSH_A = +13.07 \text{ ft available}$$

For hydrocarbons and water significantly above room temperatures, the Hydraulic Institute [17] recommends the use of a correction deduction as given in Figure 3-46. This indicates that the required NPSH as given on the pump curves can be reduced for conditions within the range of the curve based on test data.

If the pump given in the curve of Figure 3-36A were being used to pump butane at 90°F and 0.58 gravity, the correction multiplier from the NPSH curve is about 0.99 by interpolation. This means that the values of Figure 3-36A should be multiplied by 0.99 to obtain the actual NPSH the pump would require when handling a *hydrocarbon* of these conditions. The correction does not apply to other fluids.

If the system pressure were 46 psia, then $NPSH_A$ available = −8 + (46 − 44)(2.31/0.58) − 12 = −12 feet, and this is an impossible and unacceptable condition. This means liquid will flash in the line and in the impeller, and cannot be pumped. NPSH must *always* be positive in sign.

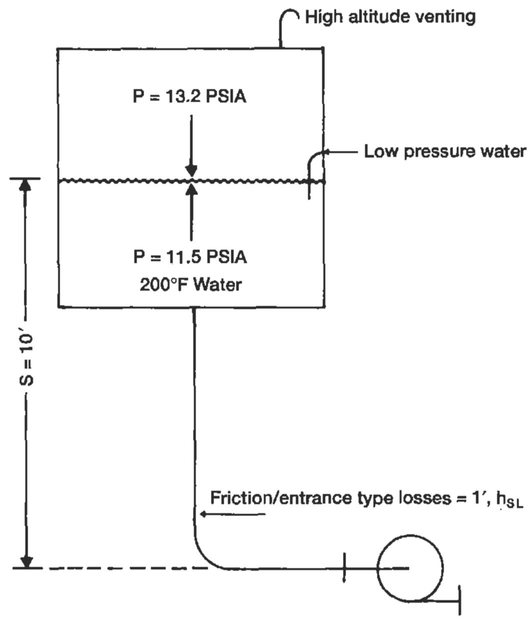


Figure 3-45. High altitude process vacuum system, NPSH requirements.

Reductions in $NPSH_R$

Limitations for use of the Hydraulic Institute NPSH reduction chart (Figure 3-46) are [17]:

1. NPSH reductions should be limited to 50% of the $NPSH_R$ required by the pump for cold water, which is the fluid basis of the manufacturer's $NPSH_R$ curves.
2. Based on handling pure liquids, without entrained air or other non-condensable gases, which adversely affect the pump performance.
3. Absolute pressure at the pump inlet must not be low enough to release non-condensables of (2). If such release can occur, then the $NPSH_R$ would need to be increased above that of the cold water requirements to avoid cavitation and poor pump performance.
4. For fluids, the worst actual pumping temperature should be used.
5. A factor of safety should be applied to ensure that NPSH does not become a problem.
6. Do not extrapolate the chart beyond NPSH reductions of 10 feet.

Example 3-10: Corrections to $NPSH_R$ for Hot Liquid Hydrocarbons and Water

In Figure 3-46, use the dashed example lines at a temperature of 55°F for propane [17], and follow the vertical line to the propane vapor pressure dashed line, which

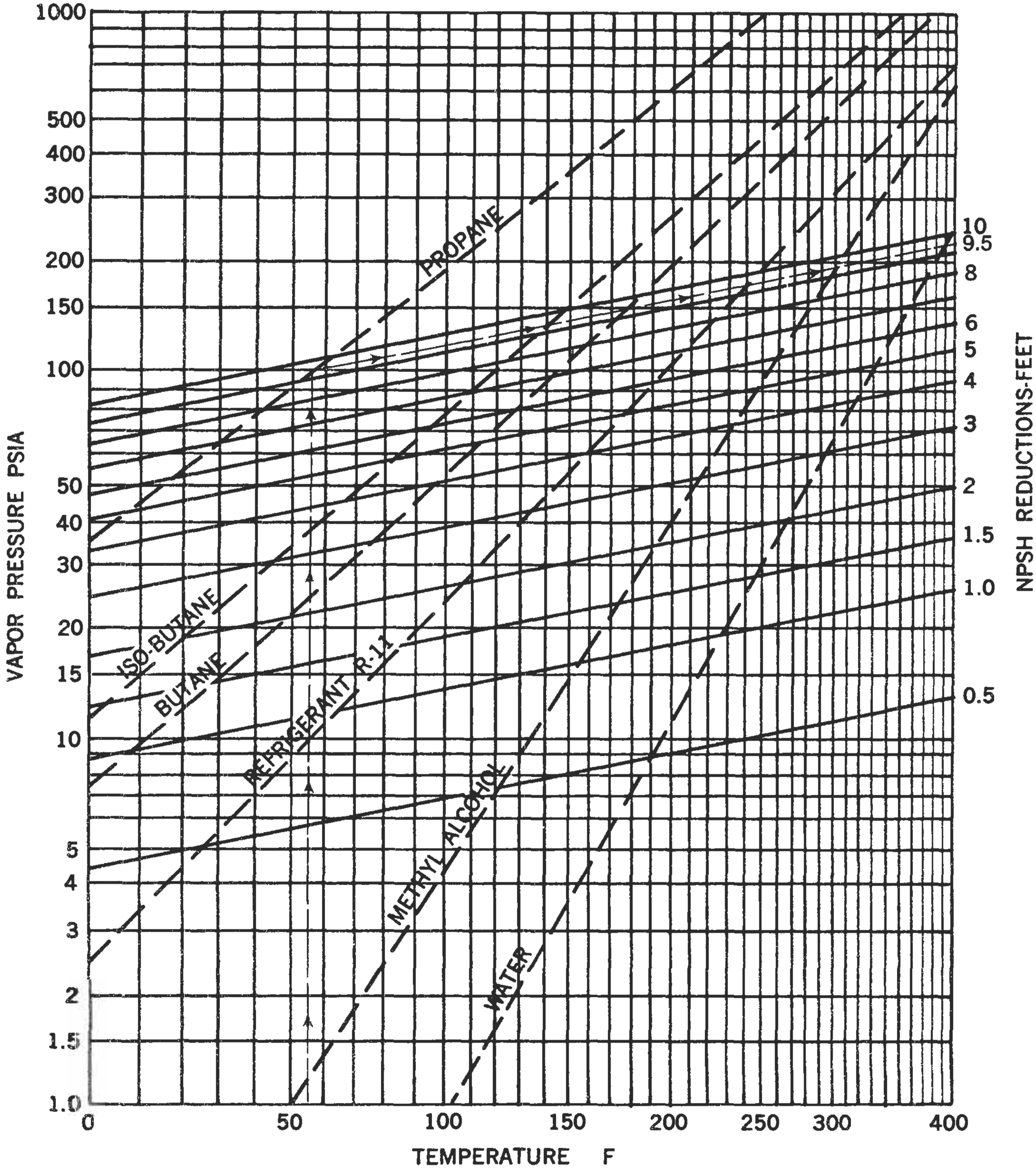


Figure 3-46. NPSH reductions for pumps handling hydrocarbon liquids and high temperature water. (Note: do not use for other fluids.) (By permission, *Hydraulic Institute Standards for Centrifugal, Rotary, and Reciprocating Pumps*, Hydraulic Institute, 13th ed., 1975.)

reads 100 psia vapor pressure. Then follow the slant lines (parallel) to read the scale for NPSH reductions, that is, feet at 9.5 ft.

Now the pump selected reads $NPSH_R$ on its pump performance curve of 12 feet for cold water service.

Now, ½ of 12 ft = 6 ft
Figure 3-46 reads = 9.5 ft reduction
Corrected value of $NPSH_R$ to use = 6 ft, since 9.5 ft is > ½ the cold water value

Example 3-11: Alternate to Example 3-10

Assume that a boiler feed water is being pumped at 180 °F. Read the chart in Figure 3-46 and the water vapor pressure curve, and follow over to read NPSH reduction = 0.45 feet. A pump selected for the service requires 6 feet cold water service $NPSH_R$:

½ of 6 = 3 ft
Value from chart for 180°F = 0.45 ft reduction
Then correct $NPSH_R$ to use = 6 ft − 0.45 ft = 5.55 ft required by the pump for this service

Specific Speed

The specific speed of a centrifugal pump correlates the basic impeller types as shown in Figure 3-47.

The formula for specific speed index number is:

$$N_s = n\sqrt{Q}/H^{3/4} \quad (3\text{-}11)$$

where: Q is the GPM capacity at speed n in rpm and head H.
H is the total head per stage, in feet.

The principle of dynamical similarity expresses the fact that two pumps geometrically similar to each other will have similar (not necessarily identical) performance characteristics. The three main characteristics of capacity, head, and rotative speed are related into a single term designated "specific speed" [25]. The expression for specific speed is the same whether the pump has a single or double suction impeller.

The principle significance of specific speed for the process engineer is to evaluate the expected performance of a second pump in a particular manufacturer's series while basing it on the known performance (or curve) at the point of optimum efficiency of a first and different size pump. In effect the performance of any impeller of a manufacturer's homologous series can be estimated from the known performance of any other impeller in the series, at the point of optimum efficiency. Figures 3-48 and 3-49 represent the standardized conditions of essentially all pump manufacturers.

A typical "operating specific speed" curve is shown in Figure 3-50 and represents a technique for plotting the specific speed on the operating performance curve. Figure 3-50 represents a 6-inch pump operating at 1760 rpm, with maximum efficiency at 1480 GPM and 132 feet head [25]. The operating specific speed is zero at no flow and increases to infinity at the maximum flow of 2270 gpm and zero head. Stable operations beyond about 1600–1700 gpm cannot be planned from such a curve with a sharp cutoff drop for head capacity.

"Type specific speed" is defined as that operating specific speed that gives the maximum efficiency for a specific pump and is the number that identifies the pump type [25]. This index number is independent of the rotative speed at which the pump is operating, because any change in speed creates a change in capacity in direct proportion and a change in head that varies as the square of the speed [25]. Practice is to "true type" the specific speed of the pump reasonably close to the conditions of maximum effi-

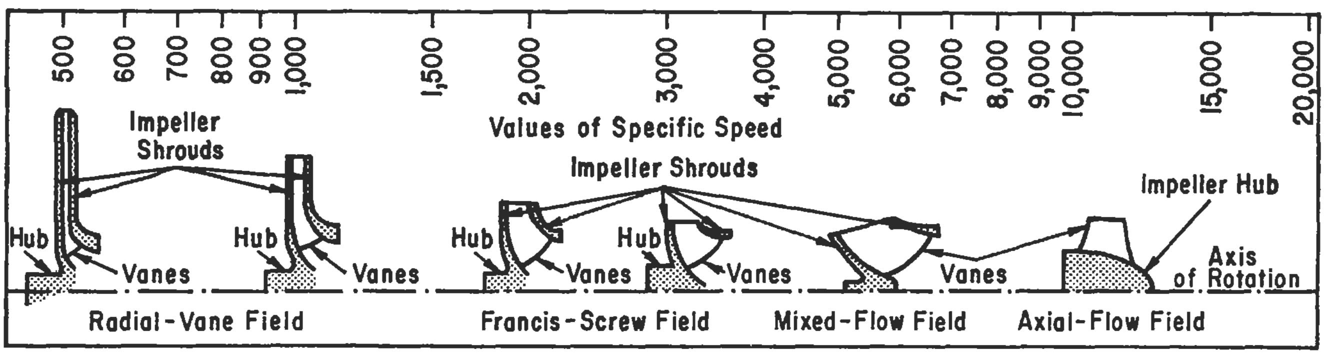


Figure 3-47. Impeller designs and corresponding specific speed range. (By permission, *Standards of the Hydraulic Institute*, 10th ed.) Also see [17], Hydraulic Institute, 13th ed., 1975.

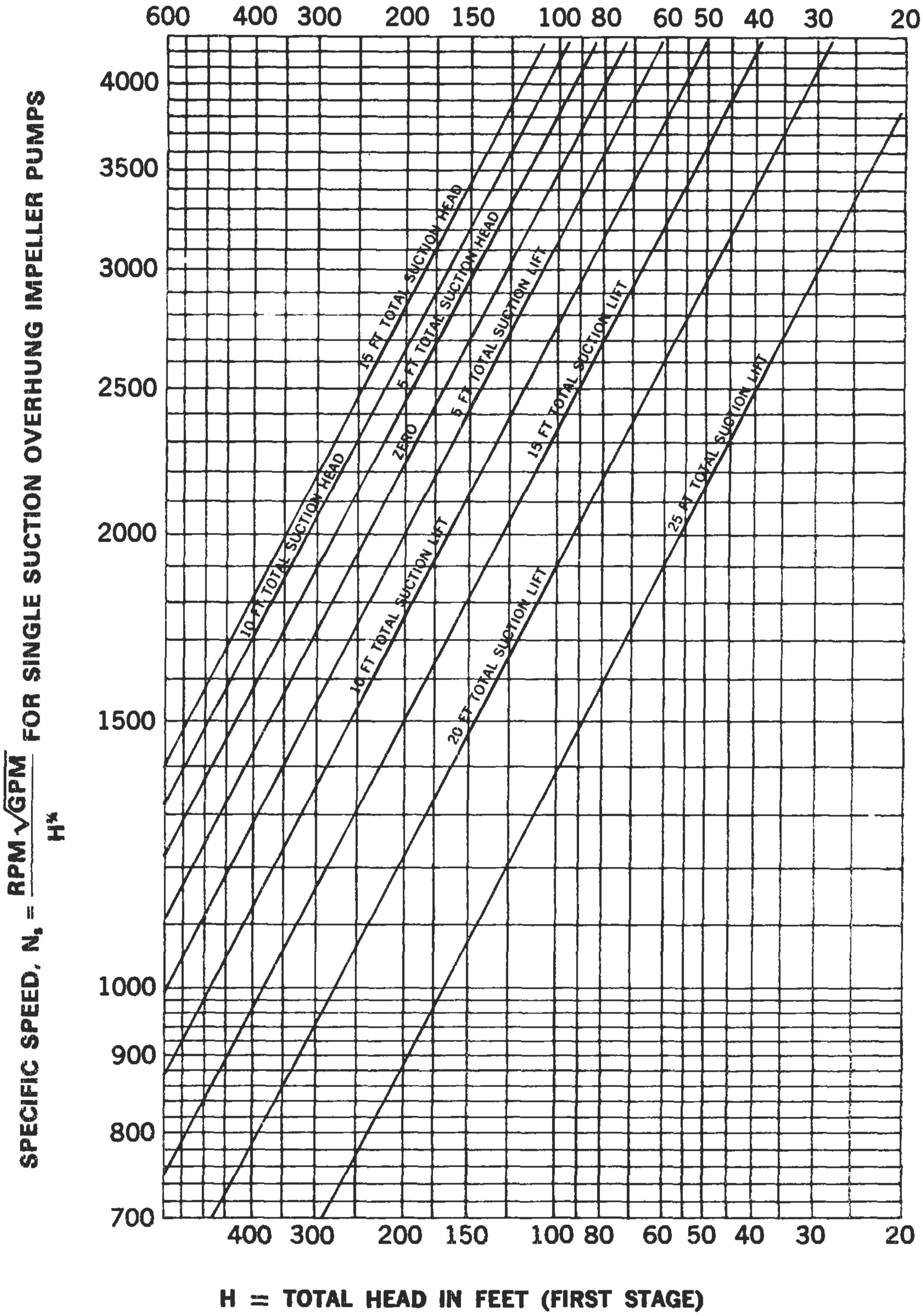


Figure 3-48. Upper limits of specific speeds for single suction overhung impeller pumps handling clear water at 85°F at sea level. (By permission, *Hydraulic Institute Standards for Centrifugal, Rotary, and Reciprocating Pumps,* Hydraulic Institute, 13th ed., 1975.)

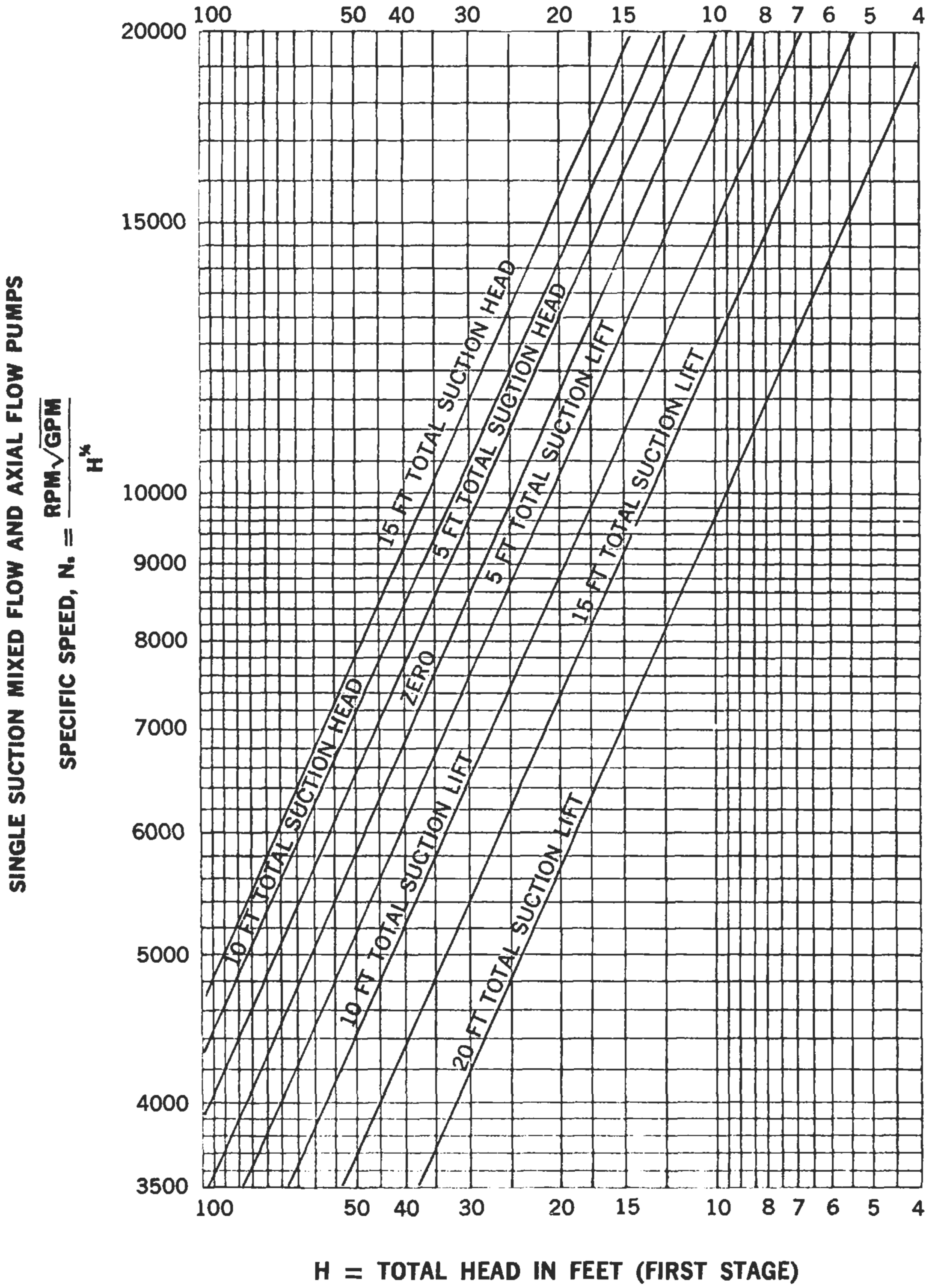


Figure 3-49. Upper limits of specific speeds for single suction, mixed and axial flow pumps handling clear water at 85°F at sea level. (By permission, *Hydraulic Institute Standards for Centrifugal, Rotary, and Reciprocating Pumps,* Hydraulic Institute, 13th ed., 1975 [17].)

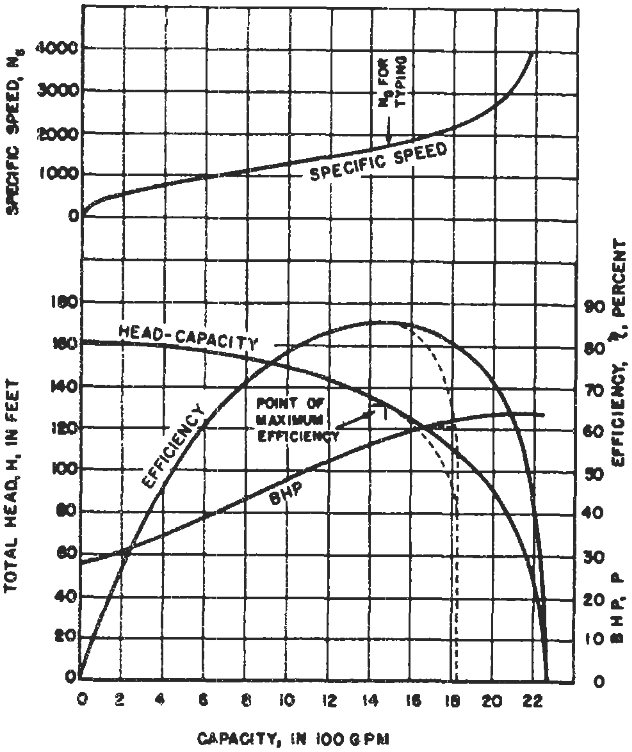


Figure 3-50. Typical centrifugal pump characteristic curve with auxiliary specific speed curve. Double-suction, single-stage, 6-in. pump, operating at 1760 rpm constant speed. (By permission, Karassik, I. and Carter, R., *Centrifugal Pumps,* McGraw-Hill Book Co., inc., 1960, p. 197.)

ciency. Figure 3-47 illustrates the range of typical specific speed index numbers for particular types of impellers.

Example 3-12: "Type Specific Speed"

In Figure 3-50, where the pump operates at 1760 rpm (a standard motor speed under load) and has maximum efficiency at 1480 GPM and 132 feet head, the "type" specific speed is

$$N_s = \left[\frac{n\sqrt{Q}}{H^{0.75}}\right] \tag{3-11}$$

$$N_s = \left[\frac{1760\sqrt{1480}}{(132)^{0.75}}\right] = 1740$$

Figure 3-47 indicates the general type of impeller installed.

The specific speed of a given type pump must not exceed the specific speed values presented by the Hydraulic Institute [17]. This is based on a known or fixed condition of suction lift, and relates speed, head and capacity. This index is a valuable guide in establishing the maximum suction lifts and minimum suction heads to avoid cavitation of the impeller with resultant unstable hydraulic performance and physical damage. For a given set of conditions on the suction and discharge of a pump, a slow rotative speed will operate safer at a higher suction lift than a pump of higher rotative speed.

Rotative Speed

The rotative speed of a pump is dependent upon the impeller characteristics, type fluid, NPSH available and other factors for its final determination. The most direct method is by reference to manufacturer's performance curves. When a seemingly reasonable selection has been made, the effect of this selected speed on the factors such as NPSH required, suction head or lift, fluid erosion and corrosion, etc., must be evaluated. For many systems these factors are of no concern or consequence.

Normal electric motor speeds run from the standard induction speeds for direct connection of 3600, 1800 and 1200 rpm to the lower speed standards of the synchronous motors, and then to the somewhat arbitrary speeds established by V-belt or gear drives. For some cases, the pump speed is set by the type of drivers available, such as a gasoline engine.

Electric motors in pump application never run at the "standard" rotative design speeds noted above, but rotate at about (with some deviation) 3450, 1750, and 1150 rpm, which are the speeds that most pump manufacturers use for their performance curves. If the higher numbers were used (motor designated or name plate) for pump performance rating, the pumps would not meet the expected performance, because the motors would not be actually rotating fast enough to provide the characteristic performance curves for the specific size of impeller.

Pumping Systems and Performance

It is important to recognize that a *centrifugal pump will operate only along its performance curve* [10, 11]. External conditions will adjust themselves, or must be adjusted in order to obtain stable operation. *Each pump operates within a system,* and the conditions can be anticipated if each component part is properly examined. The system consists of the friction losses of the suction and the discharge piping plus the total static head from suction to final discharge point. Figure 3-51 represents a typical system head curve superimposed on the characteristic curve for a 10 by 8-inch pump with a 12-inch diameter impeller.

Depending upon the corrosive or scaling nature of the liquid in the pipe, it may be necessary to take this condi-

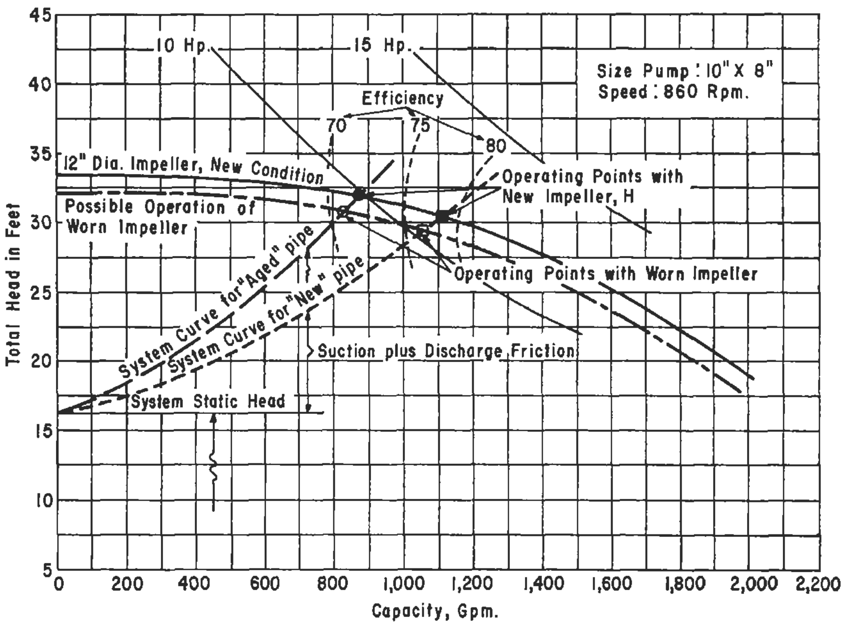


Figure 3-51. System head curves for single pump installation.

tion into account as indicated. Likewise, some pump impellers become worn with age due to the erosive action of the seemingly clean fluid and perform as though the impeller were slightly smaller in diameter. In erosive and other critical services this should be considered at the time of pump selection.

Considering Figure 3-39 as one situation which might apply to the system curve of Figure 3-51 the total head of this system is:

$$H = D + h_{DL} - (-S_L - h_{SL}) \qquad (3\text{-}12)$$

The values of friction loss (including entrance, exit losses, pressure drop through heat exchangers, control valves and the like) are h_{SL} and h_{DL}. The total static head is $D - S_L$, or $[(D + D') - (-S_L)]$ if siphon action is ignored, and $[(D + D') - (S'_L)]$ for worst case, good design practice.

Procedure:

1. Calculate the friction losses h_{SL} and h_{DL} for three or more arbitrarily chosen flow rates, but rates which span the area of interest of the system.
2. Add $[h_{SL} + h_{DL} + (D \mp S)]$ for each value of flow calculated. These are the points for the system head curve.
3. Plot the GPM values versus the points of step 2, above.
4. The intersection of the system curve with the pump impeller characteristic curve is the operating point corresponding to the total head, H. This point will change only if the external system changes. This may be accomplished by adding resistance by partially closing valves, adding control valves, or decreasing resistance by opening valves or making pipe larger, etc.

For the system of Figure 3-39, the total pumping head requirement is

$$H = (D + h_{DL}) - [-S_L + (-h_{SL})] = (D + h_{DL}) + (S_L + h_{SL}) \qquad (3\text{-}13)$$

The total static head of the system is $[D - (-S)]$ or $(D + S)$ and the friction loss is still $h_{DL} + h_{SL}$, which includes the heat exchanger in the system.

For a system made up of the suction side as shown in Figure 3-41(a) and the discharge as shown in Figure 3-42(a), the total head is

$$H = D + h_{DL} + P_1 - [+S - h_{SL} + P_2] \qquad (3\text{-}14)$$

where P_2 is used to designate a pressure different than P_1. The static head is $[(D + P_1) - (S + P_2)]$, and the friction head is $h_{DL} + h_{SL}$.

Figure 3-52 illustrates the importance of examining the system as it is intended to operate, noting that there is a wide variation in static head, and therefore there must be a variation in the friction of the system as the GPM delivered to the tank changes. It is poor and perhaps erroneous design to select a pump which will handle only the average conditions, e.g., about 32 feet total head. Many pumps might operate at a higher 70-foot head when selected for a lower GPM value; however, the flow rate might be unacceptable to the process.

Example 3-13: System Head Using Two Different Pipe Sizes in Same Line

The system of Figure 3-53 consists of the pump taking suction from an atmospheric tank and 15 feet of 6-inch pipe plus valves and fittings; on the discharge there is 20 feet of 4-inch pipe in series with 75 feet of 3-inch pipe plus a control valve, block valves, fittings, etc. The pressure of the discharge vessel (bubble cap distillation tower) is 15 psig. Using water as the liquid at 40°F

Suction: 6-inch pipe (using Cameron Tables—Table 2-22). To simplify calculations for greater accuracy, use detailed procedure of Chapter 2.

	Pipe or Fitting Loss	200 GPM	300 GPM
Loss, ft/100 ft		0.584	1.24
For 15 ft	15.0 ft		
Two, 90° ell, eq.	22.8		
Gate valve, open	3.2		
Total	41.0 ft	0.24 ft.	0.51 ft

The total suction head = h_s = +7 − 0.24 = +6.76 at 200 gpm

h_s = +7 − 0.51 = +6.49 at 300 GPM

Discharge:

	4 inch pipe		3 inch pipe	
	200 GPM	300 GPM	200 GPM	300 GPM
Loss, ft/100 ft	4.29	9.09	16.1	34.1
For 20 ft	20.0	20.0		
For 75 ft			75.0	75.0
Two, 3″, 90° ells, eq.			8	8
One, 4″, 90° ell, eq.	4.6	4.6		
One Gate Valve open			1.7 ft	1.7 ft
Total, equivalent ft	24.6	24.6	84.7	84.7
Friction loss, ft fluid	1.06	2.23	13.6	28.8
Control valve at 60% of total, ft fl	1.59	3.34	20.4	43.2
Total discharge friction loss, ft	2.65	5.57	34.0	72.0

Total static head = 45 − 7 + 15(2.31) = 72.6 ft, SpGr = 1.0

Composite head curve
at 200 GPM: head = 72.6 + 0.24 + 2.65 + 34.0 = 109.49 ft
at 300 GPM: head = 72.6 + 0.51 + 5.57 + 72.0 = 150.68 ft

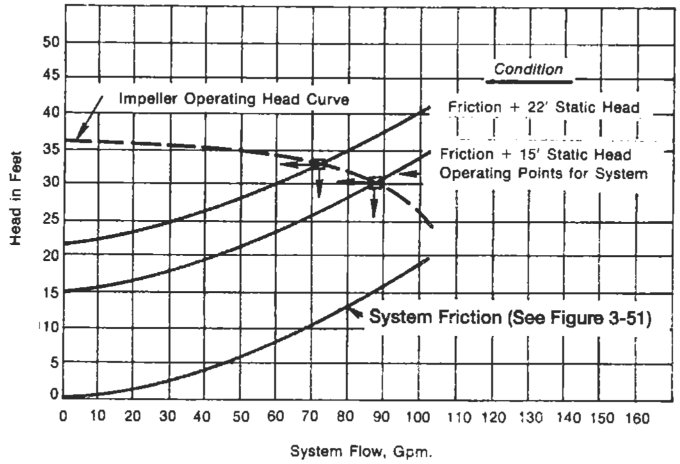


Figure 3-52. System head curves for variable static head.

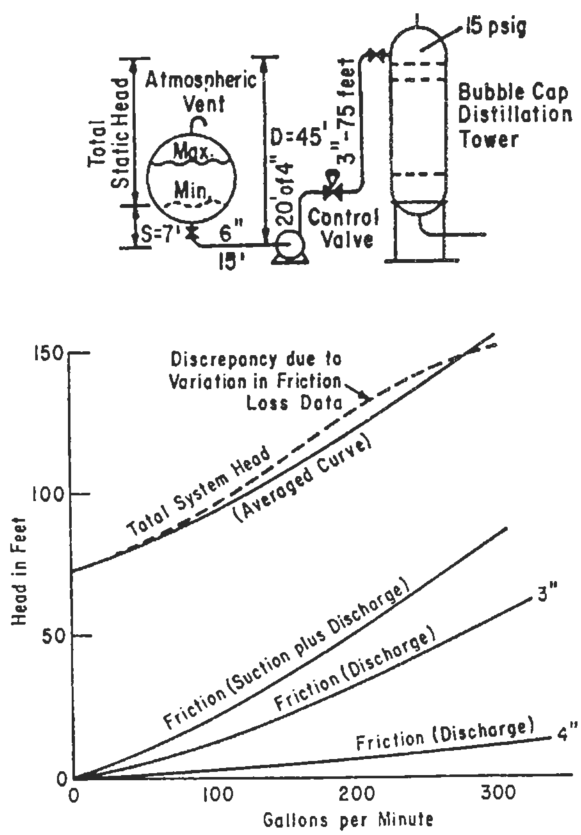


Figure 3-53. System head using two different pipe sizes in same line.

Total head on pump at 300 GPM =

$$H = 45 + 15(2.31) + 5.57 + 72.0 - 7 + 0.51 = 150.68 \text{ ft}$$

The head at 200 GPM (or any other) is developed in the same manner.

Example 3-14: System Head for Branch Piping with Different Static Lifts

The system of Figure 3-54 has branch piping discharging into tanks at different levels [13]. Following the diagram, the friction in the piping from point B to point C is represented by the line B-P-C. At point C the flow will all go to tank E unless the friction in line C-E exceeds the static lift, b, required to send the first liquid into D. The friction for the flow in line C-E is shown on the friction curve, as is the corresponding friction for flow through C-D. When liquid flows through both C-E and C-D, the combined capacity is the sum of the values of the individual curves read at constant head values, and given on curve (C-E) + (C-D). Note that for correctness the extra static head, b, required to reach tank D is shown with the friction head curves to give the *total* head above the "reference base." This base is an arbitrarily but conveniently selected point.

The system curves are the summation of the appropriate friction curves plus the static head, a, required to reach the base point. Note that the suction side friction is represented as a part of B-P-C in this example. It could be handled separately, but must be added in for any total curves. The final Total System curve is the friction of (B-P-C) + (C-E) + (C-D) plus the head, a. Note that liquid will rise in pipe (C-D) only to the reference base point unless the available head is greater than that required to flow through (C-E), as shown by following curve (B-P-C) + (C-E) + a. At point Y, flow starts in both pipes, at a rate corresponding to the Y value in GPM. The amounts flowing in each pipe under any head conditions can be read from the individual System Curves.

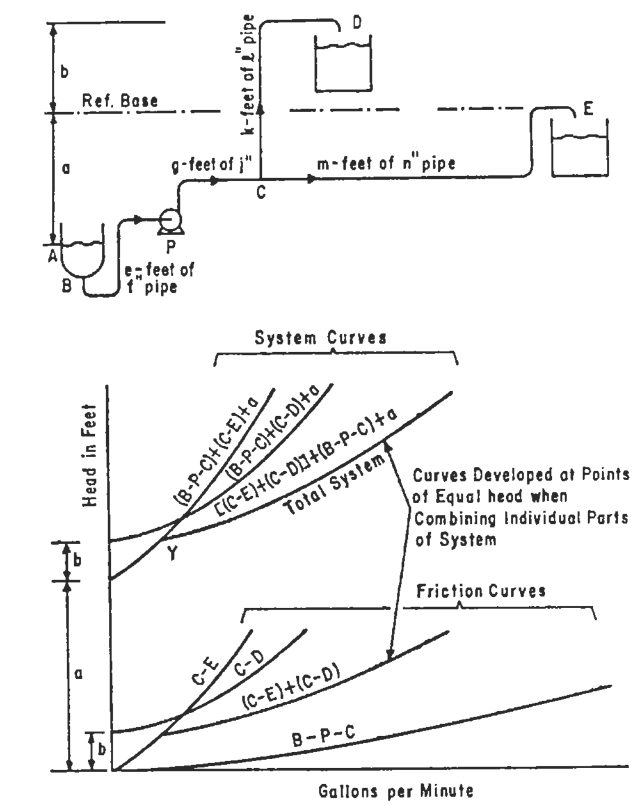


Figure 3-54. System head for branch piping with different static lifts.

The principles involved here are typical and may be applied to many other system types.

Relations Between Head, Horsepower, Capacity, Speed

Brake Horsepower Input at Pump

$$BHP = QH(SpGr)/(3960e) \quad (3\text{-}15)$$

where e is the pump efficiency, fraction.

Water or liquid horsepower [25]

$$whp = QH(SpGr)/3960 \quad (3\text{-}16)$$

The difference between the brake horsepower and the water or liquid horsepower is the pump efficiency. The requirement in either case is the horsepower input to the shaft of the pump. For that reason, the brake horsepower represents the power required by the pump, which must be transmitted from the driver through the drive shaft through any coupling, gear-box, and/or belt drive mechanism to ultimately reach the driven shaft of the pump. Therefore, the losses in transmission from the driver to the pump itself must be added to the input requirement of the driven pump and are not included in the pump's brake horsepower requirement.

$$\text{Pump efficiency [17]} = \frac{\text{liquid HP (energy delivered by pump to fluid)}}{\text{brake HP (energy to pump shaft)}} \quad (3\text{-}17)$$

$$\text{Overall efficiency [17]} = \frac{\text{WHP (energy delivered by pump to fluid)}}{\text{eHP (energy supplied to input side of pump's driver)}} \quad (3\text{-}18)$$

where eHP = electrical horsepower
WHP = liquid horsepower

For the rising type characteristic curve, the maximum brake horsepower required to drive the pump over the entire pumping range is expressed as a function of the brake horsepower at the point of maximum efficiency for any particular impeller diameter [7].

$$\text{BHP (max.)} = 1.18 \text{ (BHP at max. efficiency point)} \quad (3\text{-}19)$$

Unless specifically identified otherwise, *the BHP values read from a manufacturers performance curve* represent the power only for handling a fluid of viscosity about the same as water and a specific gravity *the same as water, i.e., SpGr = 1.0.* To obtain actual horsepower for liquids of specific gravity other than 1.0, the curve values must be multiplied by the gravity referenced to water. Viscosity corrections are discussed in another section. Good design must allow for variations in these physical properties.

Driver Horsepower

The driver horsepower must be greater than the calculated (or value read from curves) input BHP to the shaft of the pump. The mechanical losses in the coupling, V-belt, gear-box, or other drive plus the losses in the driver must be accounted for in order that the driver rated power output will be sufficient to handle the pump.

Best practice suggests the application of a non-overloading driver to the pump. Thus a motor rated equal to or greater than the maximum required BHP of the pump, assuming no other power losses, would be non-overloading over the entire pumping range of the impeller. It is important to examine the pump characteristic curve and follow the changes in power requirements before selecting a driver.

For example, referring to Figure 3-36A, if your pump were selected with a 6-inch diameter impeller for a rated normal pumping of 100 GPM, the pump would put out about 138 feet of head of *any fluid* (neglecting viscosity effects for the moment). The intersection of the 100-GPM vertical line with the 6-inch performance curve would indicate that 5.75 brake horsepower (hp) would be required *for water* (between 5 hp and 7.5 hp). Therefore, to be non-overloading (that is, the motor driver will not overheat or lose power) at this condition would require a 7.5 horsepower motor (if no other losses occur between driver and pump), because there is no standard motor for direct connected service between the standard 5 and 7.5 hp. Now, if you know or project that you may need at some time to pump 160 GPM of any fluid with this pump at 160 feet head, then (1) this pump could not be used because it will not physically take an impeller larger than 6.5-inch-diameter. However, recognizing this, (2) if you change the external physical piping, valves, etc., and reduce the head to fit the 6.5-inch impeller curve, at 160 GPM, you could handle 152 feet head (estimated from the curve for a 6.5-inch impeller).

This condition would require a brake horsepower from the pump curve between 7.5 and 10, that is, about 9.25 BHP for the pump's input shaft (for water calculates at 9.03 BHP), estimating the spread between 7.5 and 10. Thus a 10 hp (next standard size motor) would be required, and this would satisfy the original condition and the second condition for water. It would still be satisfactory for any fluid with a specific gravity < 1.0, but if pumping a liquid of 1.28 SpGr (ethyl chloride, for example), then (1), the original BHP would need to be 1.28(5.75) = 7.36 BHP, and (2), the second condition would require 1.28(9.25) = 11.84 BHP (calculates 11.56). Whereas, a 10-hp motor would be non-overloading for the water pumping case, it would require a 15-hp (next standard above a 10 hp) motor direct drive to satisfy the ethyl chloride case under the 160 GPM condition.

If you do not select a non-overloading motor, and variations in head and/or flow occur, the motor could overheat and stop operating. Study the pump-capacity curve shape to recognize the possible variations.

Important note: Any specific pump impeller operating in a physical (mechanical) system will only perform along its operating characteristic curve. If there is a change in the system flow characteristics (rate or friction resistance or pressure head), the performance will be defined by the new conditions and the pump performance will "slide" along its fixed curve. Thus, the designer cannot arbitrarily pick a point and expect the pump to "jump" to that point. Refer to Figure 3-36A. Using a 6-inch impeller curve, for example, the designer cannot make this pump operate at a point of 100 GPM and 150 feet head. This would require about a 6¼-inch diameter impeller. The 6-inch curve will only put out 138 feet (approx.) at the intersection of 100 GPM and the 6-inch curve.

A driver selected to just handle the power requirements of the design point (other than maximum) is usually a poor approach to economy. Of course, there are applications where the control system takes care of the possibilities of power overload.

Affinity Laws

The affinity laws relate the performance of a known pump along its characteristic curve to a new performance curve when the speed is changed. This would represent the same "family" of pump curves. As an example, see Figures 3-36A, B, and C.

1. For change in speed with a geometrically similar family of fixed impeller design, diameter and efficiency, the following conditions and characteristics change *simultaneously* [25]:

$$Q_2 = Q_1(n_2/n_1) \tag{3-20}$$

$$H_2 = H_1(n_2/n_1)^2 \tag{3-21}$$

$$(BHP)_2 = (BHP)_1(n_2/n_1)^3 \tag{3-22}$$

For a fixed speed [25]:

$$Q_2 = Q_1(d_2/d_1) \tag{3-23}$$

$$H_2 = H_1(d_2/d_1)^2 \tag{3-24}$$

$$(BHP)_2 = (BHP)_1(d_2/d_1)^3 \tag{3-25}$$

For geometrically similar impellers operating at the same specific speed, the affinity laws are [25,11]:

$$\frac{Q_2}{Q_1} = (n_2/n_1)(d_2/d_1)^3 \tag{3-26}$$

$$\frac{H_2}{H_1} = (n_2/n_1)^2(d_2/d_1)^2 \tag{3-27}$$

$$\frac{BHP_2}{BHP_1} = (n_2/n_1)^3(d_2/d_1)^5 \tag{3-28}$$

where: condition of subscript (2) represents the new noncavitating or desired condition, and condition of subscript (1) represents the condition for which a *set* of conditions are known.

These relations do not hold exactly if the ratio of speed change is greater than 1.5 to 2.0, nor do they hold if suction conditions become limiting, such as NPSH.

Figure 3-55 illustrates the application of these performance laws to the 1750 rpm curves (capacity, brake horsepower, and efficiency) of a particular pump to arrive at the 1450 rpm and 1150 rpm curves. Note that the key value is the constant efficiency of points (1) and (2). When the speed drops to 1450 rpm, capacity drops:

$Q_2 = 204(1450/1750) = 169$ GPM

The head also drops:

$H_2 = 64(1450/1750)^2 = 44$ ft

and:

$(BHP)_2 = 6.75(1450/1750)^3 = 3.84$ BHP

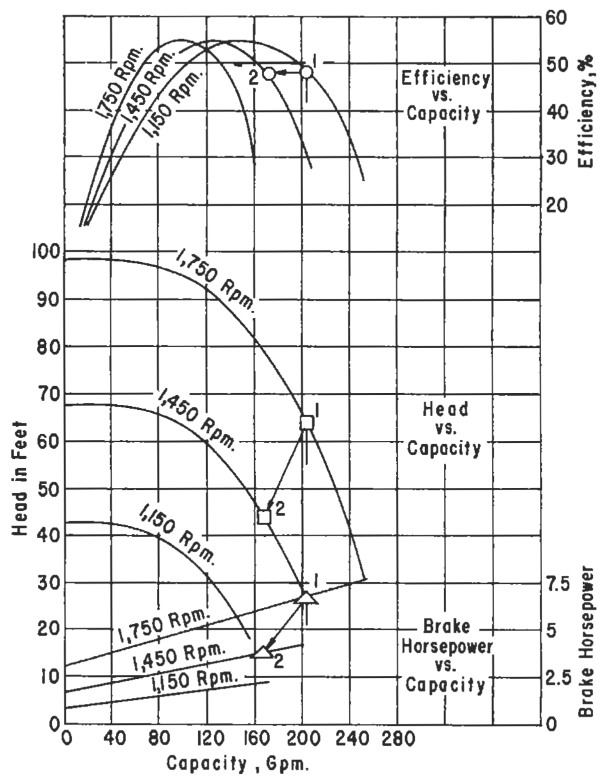


Figure 3-55. Relation of speed change to pump characteristics.

2. For changes (cut-down) in impeller diameter (not design) at fixed efficiency: [11]

$$Q_2 = Q_1\left(\frac{n_2}{n_1}\right)\left(\frac{d_2}{d_1}\right) \tag{3-29}$$

$$H_2 = H_1\left(\frac{n_2}{n_1}\right)^2\left(\frac{d_2}{d_1}\right)^2 \tag{3-30}$$

$$BHP_2 = BHP_1\left(\frac{n_2}{n_1}\right)^3\left(\frac{d_2}{d_1}\right)^3 \tag{3-31}$$

where d_1 is the original impeller diameter in inches, and subscript (2) designates the new or desired conditions corresponding to the new impeller diameter, d_2. All performance changes occur *simultaneously* when converting from condition (1) to condition (2), no single condition can be true unless related to its corresponding other conditions.

An impeller can be cut from one size down to another on a lathe, and provided the change in diameter is not greater than 20 percent, the conditions of new operation can be described by the type of calculations above. A cut to reach 75–80 percent of the original diameter may adversely affect performance by greatly lowering the efficiency [4].

Most standard pump curves illustrate the effect of changing impeller diameters on characteristic performance (Figure 3-36A). Note change as reflected in the different impeller diameters. However, the slight change in efficiency is not recorded over the allowable range of impeller change.

Recognizing the flexibility of the affinity laws, it is better to select an original pump impeller diameter that is somewhat larger than required for the range of anticipated performance, and then cut this diameter down after in-service tests to a slightly smaller diameter. This new performance can be predicted in advance. Once the impeller diameter is too small, it cannot be enlarged. The only solution is to order the required large impeller from the manufacturer.

Example 3-15: Reducing Impeller Diameter at Fixed RPM

If you have a non-cavitating (sufficient NPSH) operating 9-inch impeller producing 125 GPM at 85 feet total head pumping kerosene of SpGr = 0.8 at 1750 rpm using 6.2 BHP (not motor nameplate), what diameter impeller should be used to make a permanent change to 85 GPM at 60 feet head, at the same speed?

$$Q_2 = Q_1(d_2/d_1) \tag{3-23}$$

$$85 = 125(d_2/9)$$

$$d_2 = 6.1 \text{ in. diameter (new)}$$

The expected head would be

$$H_2 = H_1(d_2/d_1)^2 \tag{3-24}$$

$$H_2 = 85(6.1/9)^2$$

$= 39.0$ ft (must check system new total head to determine if it will satisfy this condition.)

The expected brake horsepower would be

$$BHP_2 = BHP_1(d_2/d_1)^3 \tag{3-25}$$

$$BHP_2 = 6.2(6.1/9)^3$$

$= 1.93$ BHP (use a 2- or 3-hp motor)

Effects of Viscosity

When viscous liquids are handled in centrifugal pumps, the brake horsepower is increased, the head is reduced, and the capacity is reduced as compared to the performance with water. The corrections may be negligible for viscosities in the same order of magnitude as water, but become significant above 10 centistokes (10 centipoise for SpGr = 1.0) for heavy materials. While the calculation methods are acceptably good, for *exact* performance charts test must be run using the pump in the service.

When the performance of a pump handling water is known, the following relations are used to determine the performance with viscous liquids [17]:

$$Q_{vis} = C_Q(Q_W) \tag{3-32}$$

$$H_{vis} = C_H(H_W) \tag{3-33}$$

$$E_{vis} = C_E(E_W) \tag{3-34}$$

$$BHP_{vis} = (Q_{vis})(H_{vis})(SpGr)/3960(E_{vis}) \tag{3-35}$$

Determine the correction factors from Figure 3-56 and Figure 3-57, which are based on water performance because this is the basis of most manufacturer's performance curves (except, note that the "standard" manufacturer's performance curves of *head vs GPM* reflect the head of any fluid, water, or other non-viscous). Do not extrapolate these curves!

Referring to Figure 3-56 [17]:

1. The values are averaged from tests of conventional single-stage pumps, 2-inch to 8-inch, with capacity at best efficiency point of less than 100 GPM on water performance.
2. Tests use petroleum oils.
3. The values are not exact for any specific pump.

Referring to Figure 3-57 [17]:

1. Tests were on smaller pumps, 1-inch and below.
2. The values are not exact for any specific pump.

The charts are to be used on Newtonian liquids, but not for gels, slurries, paperstock, or any other non-uniform liquids [17].

Figure 3-56 and 3-57 are used to correct the performance to a basis consistent with the conditions of the usual pump curves. In order to use the curves, the following conversions are handy:

Centistokes = centipoise/SpGr

SSU = Saybolt Seconds Universal
= (Centistokes)(4.620) at 100°F
= (Centistokes)(4.629) at 130°F
= (Centistokes)(4.652) at 210°F

Example 3-16: Pump Performance Correction For Viscous Liquid

When the required capacity and head are specified for a viscous liquid, the equivalent capacity when pumping

(text continued on page 206)

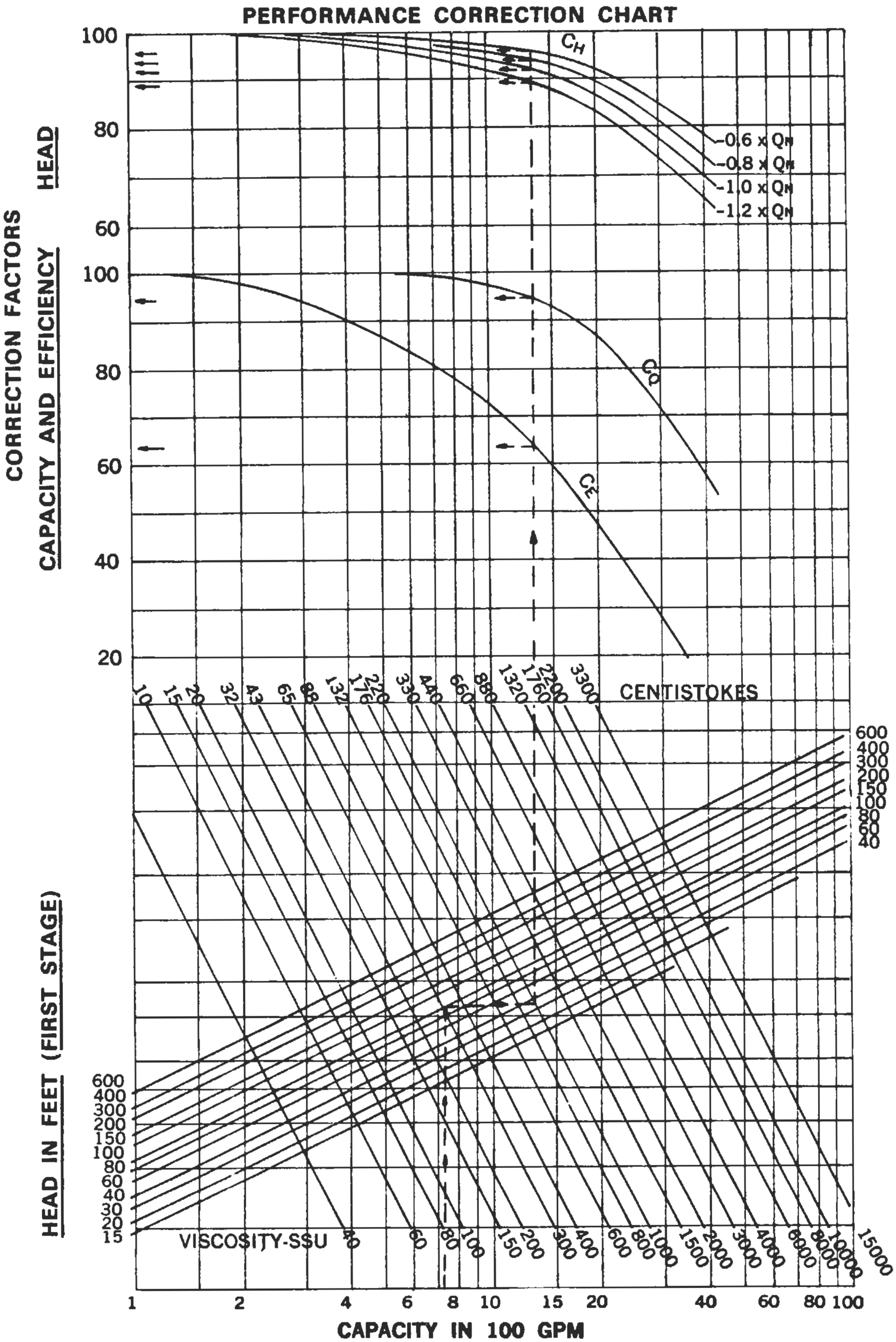


Figure 3-56. Viscosity performance correction chart for centrifugal pumps. Note: do not extrapolate. For centrifugal pumps only, not for axial or mixed flow. NPSH must be adequate. For Newtonian fluids only. For multistage pumps, use head per stage. (By permission, *Hydraulic Institute Standards for Centrifugal, Rotary, and Reciprocating Pumps,* 13th ed., Hydraulic Institute, 1975.)

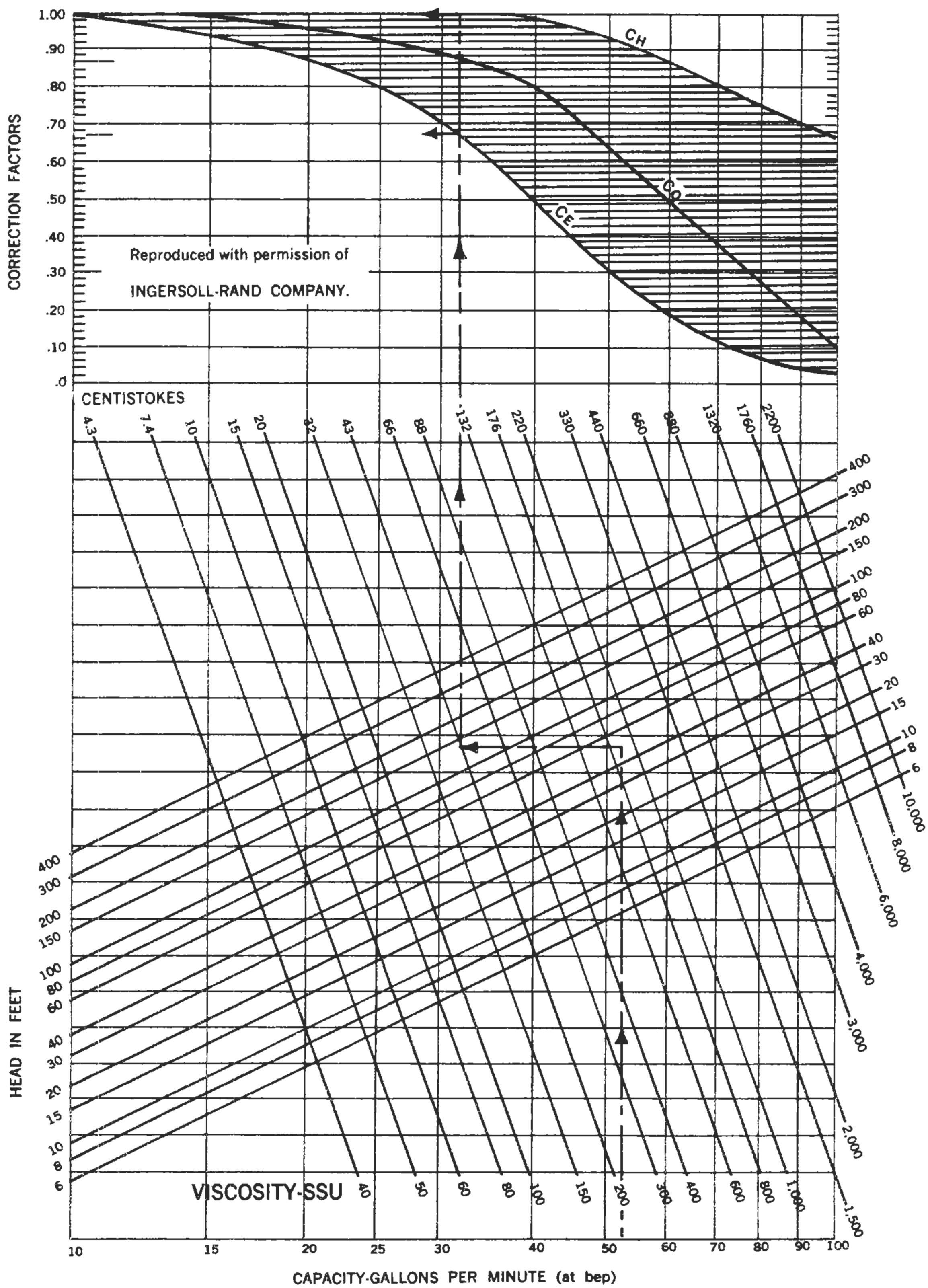


Figure 3-57. Viscosity performance correction chart for small centrifugal pumps with capacity at best efficiency point of less than 100 GPM (water performance). Note: Do not extrapolate. For small centrifugal pumps only, not for axial or mixed flow. NPSH must be adequate. For Newtonian fluids only. For multistage pumps, use head per stage. (By permission, *Hydraulic Institute Standards for Centrifugal, Rotary, and Reciprocating Pumps,* 13th ed., Hydraulic Institute, 1975.)

(text continued from page 203)
water needs to be determined using Figure 3-56 or 3-57 in order to rate pump selection from manufacturer's curves.

Determine proper pump selection and specifications when pumping oil with SpGr of 0.9 and viscosity of 25 centipoise at the pumping temperature, if the pump must deliver 125 GPM at 86 feet total head (calculated using the viscous liquid).

Viscosity conversion:

Centistokes = 25/0.9 = 27.8

Referring to Figure 3-56:

1. Enter capacity at 125 GPM, follow vertically to 86 feet of head, then to right to viscosity of 27.8 centistokes, and up to correction factors:

Efficiency, $C_E = 0.80$
Capacity, $C_Q = 0.99$
Head, $C_H = 0.96$ (for 1.0 Q_N),
Q_N = head at best efficiency point

Note this represents a flow rate using water under *maximum* efficiency conditions [17].

2. Calculate approximate *water* capacity:

$$Q_W = Q_{vis}/C_Q \quad (3\text{-}32)$$
$$Q_W = 125/0.99 = 126.3$$

3. Calculate approximate *water* head:

$$H_W = H_{vis}/C_H = 86/0.96 = 89.6 \text{ ft} \quad (3\text{-}33)$$

4. A pump may now be selected using water as the equivalent fluid with capacity of 126.3 GPM and head of 89.6 feet. The selection should be made at or very near to the point (or region) of peak performance as shown on the manufacturer's curves.
5. The pump described by the curves of Figure 3-36 fits these requirements. The peak efficiency is 71 percent using water.
6. Calculate the viscous fluid pumping efficiency:

$$e_{vis} = C_e(e_w) \quad (3\text{-}34)$$
$$= (0.80)(71) = 56.8\%$$

7. Calculate Brake Horsepower for viscous liquid

$$(BHP)_{vis} = \frac{Q_{vis} H_{vis} (SpGr)}{3960(e_{vis})} \quad (3\text{-}35)$$

$$= \frac{(125)\ (86)\ (0.9)}{(3960)\ (0.568)} = 4.3 \text{ B. horsepower}$$

Example 3-17: Corrected Performance Curves for Viscosity Effect

When a pump performance is defined for water, the corrected performance for a viscous fluid can be developed using Figure 3-56 or 3-57. In order to develop the curves for viscosity conditions of 100 SSU or 1,000 SSU as shown in Figure 3-58, the following general procedure is used [17].

1. Starting with performance curve based on pumping water:
   a. Read the water capacity and head at peak efficiency. This capacity is the value of (1.0 Q_{nW}).
   b. Using this value of GPM, calculate 0.6, 0.8 and 1.2 times this value, giving 0.6 Q_{nW}, 0.8 Q_{nW} and 1.2 Q_{nW} respectively, and read the corresponding heads and water efficiencies.
2. Using Figure 3-56 or 3-57 enter GPM at value corresponding to peak efficiency, 1.0 Q_{nW}, and follow up to the corresponding head value, H_W, then move to the viscosity value of the liquid, and up to the correction factors C_E, C_Q, C_H.
3. Repeat step 2 using GPM and head values of step (1b).
4. Correct head values:

$$H_{vis} = H_W C_H \quad (3\text{-}33)$$

5. Correct efficiency values:

$$e_{vis} = e_w C_E \quad (3\text{-}34)$$

6. Correct capacity values:

$$Q_{vis} = Q_W C_Q \quad (3\text{-}32)$$

7. Calculate the viscous BHP as indicated in the previous example.
8. Plot values as generally indicated on Figure 3-58 and obtain the performance curves corresponding to the viscous liquid conditions.

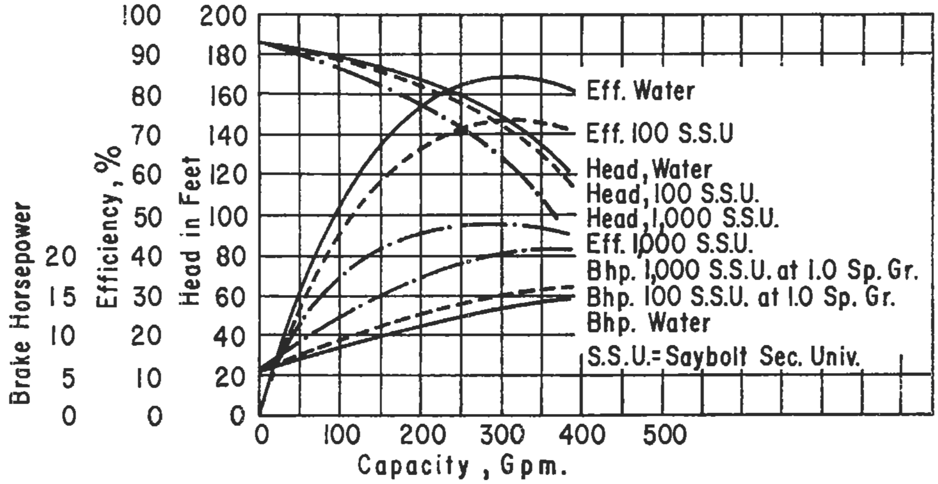


Figure 3-58. Typical curves showing the effect on a pump designed for water when pumping viscous fluids. (By permission, *Pic-a-Pump*, 1959, Allis-Chalmers Mfg. Co.)

Temperature Rise and Minimum Flow

When a pump operates near shut-off (low flow) capacity and head, or is handling a hot material at suction, it may become overheated and create serious suction as well as mechanical problems. To avoid overheating due to low flow, a minimum rate (GPM) should be recognized as necessary for proper heat dissipation. However, it is not necessarily impossible to operate at near shutoff conditions, provided (1) it does not operate long under these conditions, as temperature rises per minute vary from less than 1°F to 30–40°F, or (2) a by-pass is routed or recycled from the discharge *through* a cooling arrangement and back to suction to artificially keep a minimum safe flow through the pump while actually withdrawing a quantity below the minimum, yet keeping the flowing temperature down [31].

1. Temperature rise in average pump during operation [6].

$$\Delta T_r = \frac{42.4\, P_{so}}{W_1\, c_p}, {}^\circ F/\min\ [25] \qquad (3\text{-}36)$$

where [25]

ΔT_r = temperature rise, °F/min

P_{so} = brake horsepower at shutoff or no flow

W_1 = weight of liquid in pump, lbs

c_p = specific heat of liquid in pump

or, alternate procedure [33,6]. For low capacity:

$$\Delta T_r = \frac{H(1-e)}{778(c_p)(e)} \qquad (3\text{-}37)$$

where H_{so} = total head of pump at no flow or shutoff or at any flow rate with corresponding efficiency from pump curve, ft

e = pump efficiency at the flow capacity involved (low flow), decimal

Another alternate procedure [10]

$$\Delta T_r = (GPM)(H_{so})(SpGr)/3960 \qquad (3\text{-}38)$$

See Figure 3-59 and Figure 3-60 for a graphical solution to the equation above for temperature rise. Figure 3-59 illustrates the characteristics of a boiler feed water pump set to handle 500 GPM water at 220°F for a total of 2600 feet head. The temperature rise curve has been superimposed on the performance chart for the pump, and values of ΔT_r are calculated for each flow-head relationship. Note how rapidly the temperature rises at the lower flows. This heating of the fluid at low flow or no flow (discharge valve shut, no liquid flowing *through* the pump) can be quite rapid and can cause major mechanical problems in the pump's mechanical components. The maximum temperature rise recommended for any fluid is 15°F (can be a bit higher at times for the average process condition) except when handling cold fluids or using a special pump designed to handle hot fluid, such as a boiler feed water pump of several manufacturers.

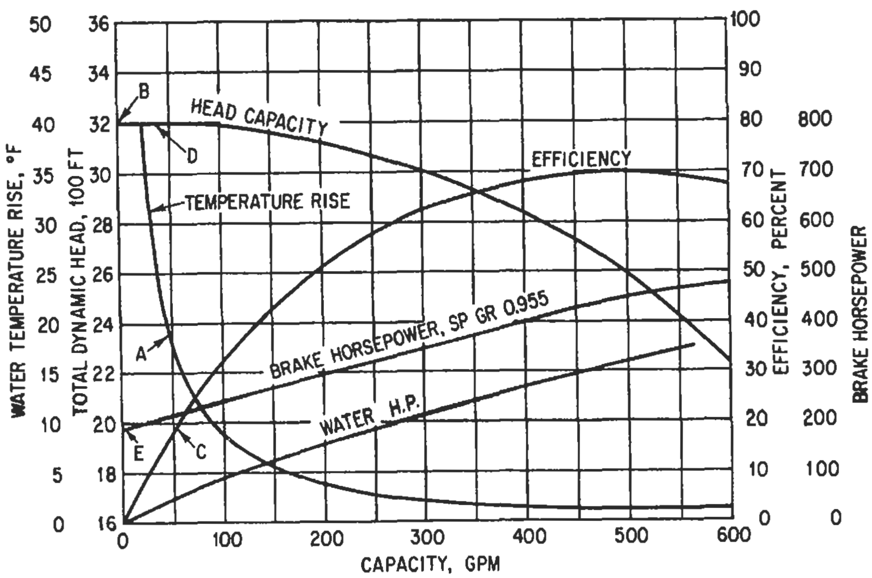


Figure 3-59. Typical temperature rise for boiler feed water pump. (By permission, *Transamerica Delaval Engineering Handbook,* 4th ed., H. J. Welch, ed., 1983, Transamerica Delaval, Inc., IMO Industries, Inc., Div.)

$$\text{Temp. rise, } ^\circ\text{F/min} = \frac{(\text{BHP at shutoff})\ (42.4)}{(\text{weight of liquid in pump})\ (c_p)} \quad (3\text{-}36)$$

or

$$\text{Temp. rise } ^\circ\text{F/min} = \frac{(\text{BHP} - \text{WHP})\ (2545)}{(\text{pump capacity})} \quad (3\text{-}39)$$

2. Minimum Flow (Estimate) [6]

The validity of the method has not been completely established, although it has been used rather widely in setting approximate values for proper operation [10]. For multistage pumps use only the head per stage in temperature limit by this method.

a. Determine $NPSH_A$ available at pump suction

b. Add the NPSH value to the vapor pressure of the liquid at suction conditions. This represents the vapor pressure corresponding to the temperature of the liquid at the flash point. Read temperature, t_2, value from vapor pressure chart of liquid.

c. Allowable temperature rise = t_2 − (actual pumping temperature). Boiler feed water practice uses 15°F rise for average conditions [10].

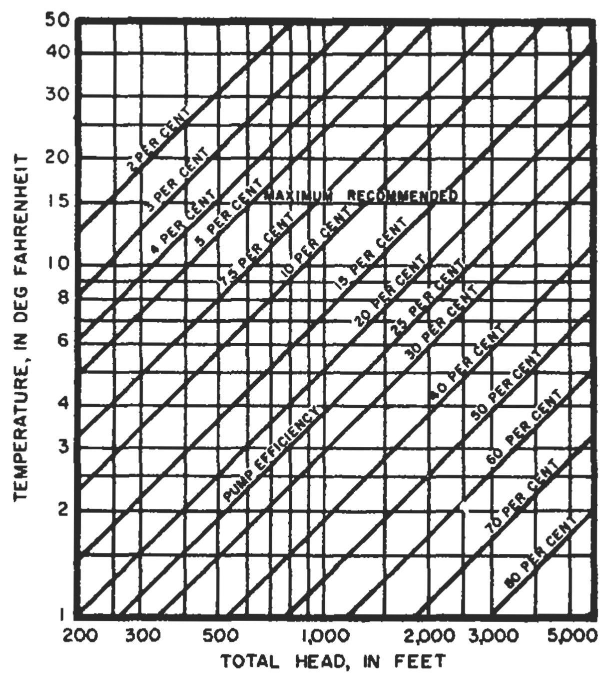


Figure 3-60. Temperature rise in centrifugal pumps in terms of total head and pump efficiency. (By permission, Karassik, I. and Carter, R., *Centrifugal Pumps,* McGraw-Hill Book Co. Inc., 1960, p. 438.)

d. Approximate minimum safe continuous flow efficiency:

$$e_M = \frac{H_{so}, \text{ at shutoff from curve}}{778\,(\Delta T_r)\, c_p + H_{so}, \text{ at shutoff}} \quad (3\text{-}40)$$

where e_M = minimum safe flowing efficiency, overall pump, fraction
H_{so} = head at no flow or shutoff, ft
c_p = specific heat of liquid, BTU/lb/°F
ΔT_R = temperature rise in liquid, °F

e. Read minimum safe flow in GPM from pump performance curve *at* value of minimum efficiency calculated in (d).

Example 3-18: Maximum Temperature Rise Using Boiler Feed Water

Using the example of Reference [6], assume a pump with characteristic curve and added temperature rise data as shown on Figure 3-59 is to handle boiler feed water at 220°F, with a system available $NPSH_A$ = 18.8 feet. The vapor pressure of water at 220°F is 17.19 psia from steam tables and the SpGr = 0.957. Correcting the 18.8 feet $NPSH_A$: psia = 18.8 (1/[2.31/0.957)] = 7.79 psia at 220°F.

The vapor pressure to which the water may rise before it flashes is 17.19 psia + 7.79 psia = 24.98 psia.

From steam tables (or fluid vapor pressure tables), read at 24.98 psia (for water of this example), temperature = 240°F.

Therefore, allowable temperature rise of the water (this example) = 240° − 220°F = 20°F.

A plotted curve as shown on Figure 3-59 [33] shows that at point A a rise of 20°F on the temperature rise curve corresponds to a flow of 47 GPM minimum safe for the pump handling 220°F, with $NPSH_A$ of 18.8 feet.

An alternate estimate for minimum flow [11]:

Minimum flow (for water) through pump,

$$Q_M = 0.3\ P_{so},\ \text{GPM} \tag{3-41}$$

where P_{so} = shutoff horsepower

For cold liquids, general service can often handle ΔT_r of up to 100°F, a rule with approximately 20% factor of safety:

$$Q_M = 6\ P_{so}/\Delta T_r,\ \text{GPM} \tag{3-42}$$

ΔT_r = permissible temperature rise, °F.

The $NPSH_R$ required at the higher temperature may become the controlling factor if cavitation is not to occur.

The minimum flow simply means that this flow must circulate through the pump casing (not recirculate with no cooling) back to at least the initial temperature of the feed, if excessive temperatures are not to develop. The best practice is to request the manufacturer to state this value for the fluid handled and the calculated $NPSH_A$ condition. For $NPSH_R$ refer to corrections discussed earlier.

Centrifugal Pump Specifications

Figure 3-61 presents specifications for a centrifugal pump. Although the process engineer cannot or should not specify each item indicated, he must give the pertinent data to allow the pump manufacturer to select a pump and then identify its features. Pumps are selected for performance from the specific characteristic curves covering the casing size and impeller style and diameter. Often the process fluids are not well known to the pump manufacturer, therefore the materials of construction, or at least any limitations as to composition, must be specified by the engineer.

Example 3-19: Pump Specifications, Figure 3-61

The pump specified identifies the design data, key portions of the construction materials and driver data as required information for the pump manufacturer. If the pump is to be inquiried to several manufacturers this is all that is necessary. The individual manufacturers will identify their particular pump selection and details of construction materials and driver data. From this information a pump can be selected with performance, materials of construction, and driver requirements specified.

In the example the manufacturer has been specified from available performance curves, and the details of construction must be obtained. The pump is selected to operate at 22 GPM and 196 to 200 feet head of fluid, and must also perform at good efficiency at 18 GPM and a head which has not been calculated, but which will be close to 196 to 200 feet, say about 185 feet. Ordinarily, the pump is rated as shown on the specification sheet. This insures adequate capacity and head at conditions somewhat in excess of normal. In this case the design GPM was determined by adding 10 percent to the capacity and allowing for operation at 90 percent of the rated efficiency. Often this latter condition is not considered, although factors of safety of 20 percent are not unusual. However, the efficiency must be noted and the increase in horsepower recognized as factors which are mounted onto normal operating conditions.

Sometimes the speed of the pump is specified by the purchaser. However, this should not be done unless there is experience to indicate the value of this, such as packing life, corrosion/erosion at high speeds, and suspended particles; as the limitation on speed may prevent the manufacturer from selecting a smaller pump. In some cases it must be recognized that high heads cannot be reached at low speeds in single stage pumps. Table 3-7 presents sug-

CENTRIFUGAL PUMP SPECIFICATIONS

Job No. ____
B/M No. ____

SPEC. DWG. NO. A-
Page 1 of 1 Pages
Unit Price 2(one as spare)
No. Units P-68 A & B
Item No.

DESIGN DATA

Service Forwarding to T-98 Liquid 73% Caustic
% Solids None Calc. GPM 18 Des. GPM 22
Vapor Press. - PSIA Temp. 265 °F Vis. 5.9 Cp Sp. Gr. 1.68

HEAD	SUCTION	DISCHARGE	DIFFERENTIAL
Static	- 8.0 Ft.	+ 4.1 Ft.	+ 12.1 Ft.
Pressure	+ 20.2 Ft. Abs.	+ 3.4 Ft. Abs.	- 16.8 Ft.
Friction	- 0.1 Ft.	+ 199.8 Ft.	+ 199.9 Ft.
Total	+ 12.1 Ft. Abs.	+ 207.3 Ft. Abs.	+ 195.2 Ft.

NPSH Available 9.5 Ft. Rating Head 196 to 200 Ft.

PUMP SELECTION

Mfgr. "A" Size & Type Centrifugal, 2 x 3 Model AAA Case Type Vert. split
Stages one Imp. Type Enclosed Design Imp. Dia. 13 1/8" Max. Imp. Dia. 14
Drive Motor RPM 1750 Design Eff. 73% Rotation* Counter Clockwise
Rating BHP 14.5 Non-Overload BHP 15 Shut-off Press. 143 psig NPSH Required 6 ft.
Suc. Location End Disch. Location Vertical Top Curve No. C - 1234

CONSTRUCTION MATERIALS

Case Cast 29 Cr-9 Ni, Misco "C" Impeller 29 Cr-9 Ni, Misco "C" Shaft Nickel

Packing ____ Shaft Seal: Internal ☐ External ☐ Plugged ☐
Stuffing Box ____ Seal Cage ____ Bushings ____ Gland Bolts ____
Mechanical Seal: Make "X" Type "Y"; Coolant Water; Inside ☐ Outside ☒ Single ☒ Double ☐
Baseplate Cast Iron Coupling: Mfg. "A" Type Flexible Guard Yes
Bearings: Type Bell, double Make ____ Coolant ____
Impeller Nut Nickel Thrust Bearings: Type Force Lubricated Make "Z"
Case Studs Nickel Case End Covers Nickel
Diffusers Nickel Diaphragm ____
Piping: Nickel
Suction Size 3" Rating 150# Std. Discharge Size 2" Rating 150# Std.

DRIVER

Type TEFC Mfgr. B H.P. 15
RPM 1750 Frame ____ Volts 440 Phase 3 Cycle 60
Elec. Class ____ Connection: Direct Coupling Belts ____ Gear ____
Inlet Steam ____ PSIG @ ____ °F Exhaust Steam ____ PSIG @ ____ °F.
Steam Rate ____ Lbs./Hr.

REMARKS

* Facing Pump from Driver End.

By	Chk'd.	App.	Rev.	Rev.	Rev.
Date					

P.O. To: ____

Figure 3-61. Centrifugal pump specifications.

gestions for materials of construction for pump parts in the services indicated. The effect of impurities, temperature, analysis variations and many other properties make it important to obtain specific corrosion service data in the specific fluid being pumped. Sometimes this is not possible, and generalized corrosion tables and experience of other users must be relied on as the best information for the materials selection.

Number of Pumping Units

A single pump is the cheapest first-cost installation. However, if downtime has any value such as in lost production, in hazards created in rest of process, etc., then a stand-by duplicate unit should be considered. A spare or stand-by can be installed adjacent to the operating unit, and switched into service on very short notice, provided it is properly maintained. Spare pumps which do not operate often should be placed in service on a regular schedule just to be certain they are in working order.

If solids are carried in the fluid, this can present a difficult problem if they are not properly flushed from the pump on shutdown. Some spare or second pumps are selected for 100 percent spare; others are selected so that each of two pumps operate in parallel on 50 percent of the flow, with each being capable of handling 67 to 75 percent of total load if one pump should fall off the line. This then only reduces production by about 25 percent for a short period, and is acceptable in many situations. These pumps are usually somewhat smaller than the full size spares.

When it is necessary to plan several pumps in parallel, the pump manufacturer must be advised, and care must be taken in arranging suction piping for the pumps, otherwise each may not carry its share of the flow.

There are many flow conditions, and pumps should be selected to operate as efficiently as possible over the widest range of capacity.

If the flow is expected to vary during the system operation, the high and low GPM (and corresponding heads) should be given to allow proper evaluation.

Fluid Conditions

The manufacturer must be told the conditions of the liquid, percent suspended solids, physical properties, corrosive nature and maximum and minimum temperature ranges. For extremely hot liquids, special hot pumps must be used, and temperature effects taken into account.

System Conditions

The manufacturer must know if the suction side of the pump is associated with vacuum equipment, or is to lift the liquid. This can make a difference as to the type of impeller suction opening he provides. If the system operates intermittently it should be noted. A piping diagram is often helpful in obtaining full benefit of the manufacturer's special knowledge.

Type of Pump

If there is a preference as to horizontal or vertical split casing, it should be stated. Also the suction and discharge connections should be stated as to top or end, or special, together with the preference as to flanged (rating) or screwed. Small pumps are commonly furnished with screwed connections unless otherwise specified.

Type of Driver

Pumps are usually driven by electric motors, steam or gas turbine or gas (or gasoline) engines either direct or

Table 3-7
Pump Materials of Construction

Table materials are for general use, specific service experience is preferred when available

Liquid	Casing & Wear Rings	Impeller & Wear Rings	Shaft	Shaft Sleeves	Type of Seal	Seal Cage	Gland	Remarks
Ammonia, Anhydrous & Aqua	Cast Iron	Cast Iron	Carbon Steel	Carbon Steel	Mechanical	....	Mall. Iron	NOTE: Materials of Construction shown will be revised for some jobs.
Benzene	Cast Iron	Cast Iron	Carbon Steel	Nickel Moly. Steel	Ring Packing	Cast Iron	Mall. Iron	
Brine (Sodium Chloride)	Ni-Resist*	Ni-Resist*	K Monel	K Monel	Ring Packing	Ni-Resist**	Ni-Resist**	*Cast Iron acceptable. **Malleable Iron acceptable.
Butadiene	Casing: C. Steel—Rings: C.I.	Impeller: C.I.—Rings: C. Steel	Carbon Steel	13% Chrome Steel	Mechanical	....	Carbon Steel	
Carbon Tetrachloride	Cast Iron	Cast Iron	Carbon Steel	Carbon Steel	Mechanical	....	Mall. Iron	
Caustic, 50% (Max. Temp. 200° F.)	Misco C	Misco C	18-8 Stainless Steel	Misco C	Ring Packing	Misco C	Carbon Steel	Misco C manufactured by Michigan Steel Casting Company. 29 Cr-9 Ni Stainless Steel, or equal.
Caustic, 50% (Over 200° F) & 73%	Nickel	Nickel	Nickel or 18-8 Stainless Steel	Nickel	Ring Packing	Nickel	Nickel	
Caustic, 10% (with some sodium chloride)	Cast Iron	23% Cr. 52% Ni Stainless Steel	23% Cr. 52% Ni Stainless Steel	23% Cr. 52% Ni Stainless Steel	Ring Packing	Cast Iron	....	Specifications for 50% Caustic (Maximum Temperature 200° F) also used.
Ethylene	Cast Steel	Carbon Steel	Carbon Steel	Carbon Steel	Mechanical	Cast Iron	Mall. Iron	
Ethylene Dichloride	Cast Iron	Cast Iron	Steel	K Monel	Mechanical	....	K Monel	
Ethylene Glycol	Bronze	Bronze	18-8 Stainless Steel	18-8 Stainless Steel	Ring Packing	....	Bronze	
Hydrochloric Acid, 32%	Impregnated Carbon	Impregnated Carbon	18-8 Stainless Steel	Impregnated Carbon	Mechanical	....	Impregnated Carbon	
Hydrochloric Acid, 32% (Alternate)	Rubber Lined C. Iron	Hard Rubber	Carbon Steel	Rubber or Plastic	Ring Packing	Rubber	Rubber	
Methyl Chloride	Cast Iron	Cast Iron	18-8 Stainless Steel	18-8 Stainless Steel	Mechanical	....	Mall. Iron	
Propylene	Casing: C. Steel—Rings: C.I.	Imp.: C.I.—Rings: C. Stl	Carbon Steel	Carbon Steel	Mechanical	Cast Iron	Mall. Iron	
Sulfuric Acid, Below 55%	Hard Rubber Lined C.I.	Special Rubber	Carbon Steel	Hastelloy C	Ring Packing	Special Rubber	Special Rubber	
Sulfuric Acid, 55 to 95%	Cast Si-Iron	Si-Iron	Type 316 Stn. Stl.	Si-Iron	Ring Packing	Teflon	Si-Iron	
Sulfuric Acid, Above 95%	Cast Iron	Cast Iron	Carbon Steel	13% Cr	Mechanical	....	Mall. Iron	
Styrene	Cast Iron	Cast Iron	Carbon Steel	13% Chrome Steel	Ring Packing	Cast Iron	Mall. Iron	
Water, River	Cast Iron	Bronze	18-8 Stainless Steel	Bronze	Mechanical	Cast Iron	Mall. Iron	
Water, Sea	Casing, 1-2% Ni, Cr 3-0.5% Cast Iron Rings: Ni-Resist, 2B	Impeller: Monel Rings: S-Monel	K Monel (Aged)	K Monel or Alloy 20 SS	Ring Packing	Monel or Alloy 20 SS	Monel or Alloy 20 SS	

through V-belts or gears. The pump manufacturer should know the preferred type of drive. If the manufacturer is to furnish the driver, the data on the specification sheet under Driver should be completed as far as applicable. If a gas or gasoline engine is to be used, the type of fuel and its condition must be stated. Engine cooling water (if air not used) must be specified.

Sump Design for Vertical Lift

The proper design of sumps for the use of vertical lift pumps or horizontal pumps taking suction from a sump is important to good suction conditions at the pump [2, 3, 14].

The arrangement and dimensions indicated in Figure 3-62 or Figure 3-63 are satisfactory for single or multiple pump installations. (For more details, refer to Reference [17]). A few key points in sump-pump relationships for good non-vortexing operation are:

1. Avoid sudden changes in direction or elevation of flow closer than five bell diameters to pump.
2. Avoid sump openings or projections in water path close to pump.

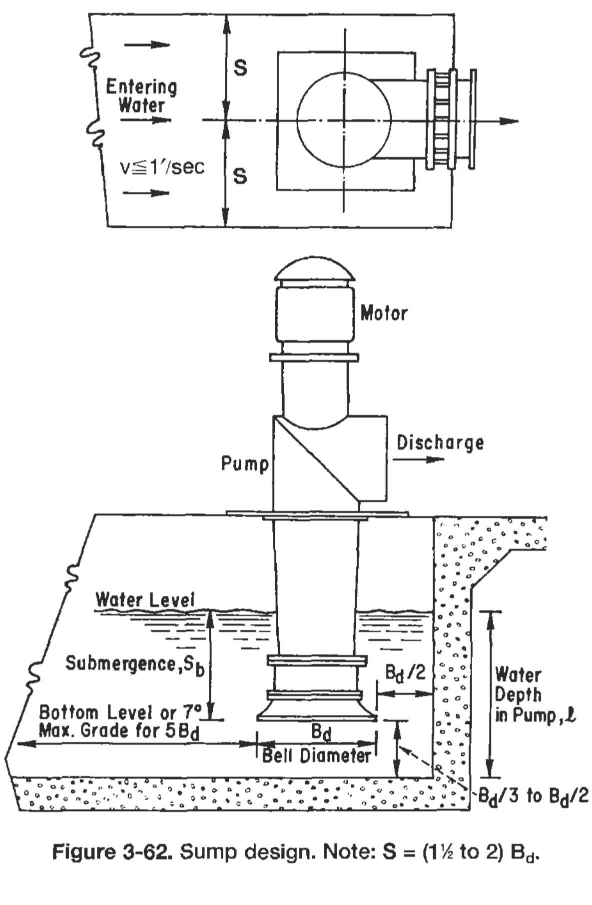


Figure 3-62. Sump design. Note: S = (1½ to 2) B_d.

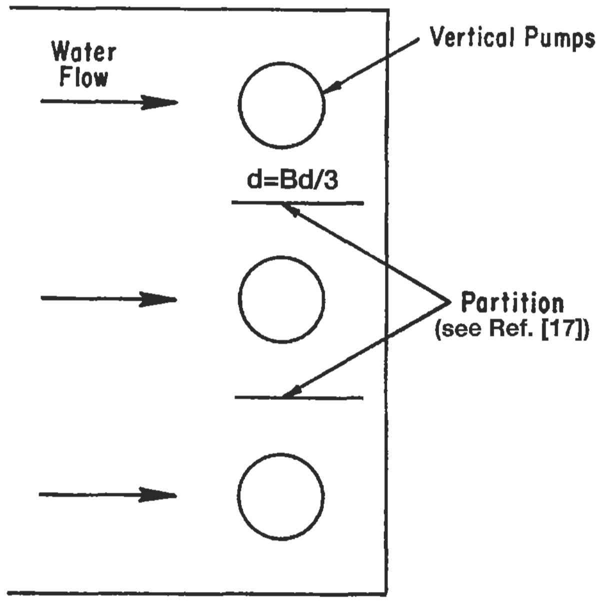


Figure 3-63. Acceptable sump arrangement for multiple pumps.

3. Have water flowing parallel to sump walls as it enters pump. Water should enter pump suction with as low a turbulence as possible.
4. Water velocity in sump must be low, 1½ feet/second is good practice.
5. Inlet channel width to each pump is considered optimum at 2 B_d to prevent secondary turbulence effects. [3]
6. Avoid placing several pumps in one open channel removing water in series fashion. If this must be done, velocity at each pump must be kept at same value as for single pump. The channel width at each pump would be taken from Reference [17].

A suction bell on the inlet of a vertical pump (or the inlet pipe of the suction side of a horizontal pump) is not necessary as far as pump or sump operation is concerned. If a bell is omitted, the entrance losses due to flow will be higher with only a straight pipe, and this must be considered in pump operation. An economic comparison will help decide the value of the bell. Strainers should not be placed on suction bells unless this is the only arrangement. Inlet water should be screened with trash racks, bars and screens to keep the sump free of debris.

Submergence of the inlet pipe column or bell inlet below the water level is necessary for good operation and to prevent vortexes and entrained air. The minimum submergence as recommended by the manufacturer must be maintained at all times. Generally, for 70°F water, each 1000 feet of elevation above sea level adds 14 inches to the required submergence. If the water is at 100°F at sea level, approximately 17 inches must be added to the 70°F submergence value [14].

Rotary Pumps

There are many different types of positive displacement rotary pumps [29] as illustrated in Figure 3-64 and Figures 3-65A, B, C.

The majority of this type are capable of handling only a clean solution essentially free of solids. The designs using rubber or plastic parts for the pressure device can handle some suspended particles. In general, these pumps handle materials of a wide range of viscosity (up to 500,000 SSU), and can develop quite high pressures (over 1000 psi). In addition, the units can handle some vapor or dissolved gases mixed with the liquid being pumped. The capacity is generally low per unit, and at times, they are used for metering. For specific performance characteristics of any type consult the appropriate manufacturer.

These pumps are low in cost, require small space, and are self priming.

Some can be rotated in either direction, have close clearances, require over-pressure relief protection on discharge due to positive displacement action, and have low volumetric efficiency [8].

Performance Characteristics of Rotary Pumps:

1. Flow proportional to speed and almost independent of pressure differential.
   (a) Internal slip reduces efficiency, and increases with pressure and decreasing viscosity.
   (b) Entrained gases reduce liquid capacity and cause pulsations.

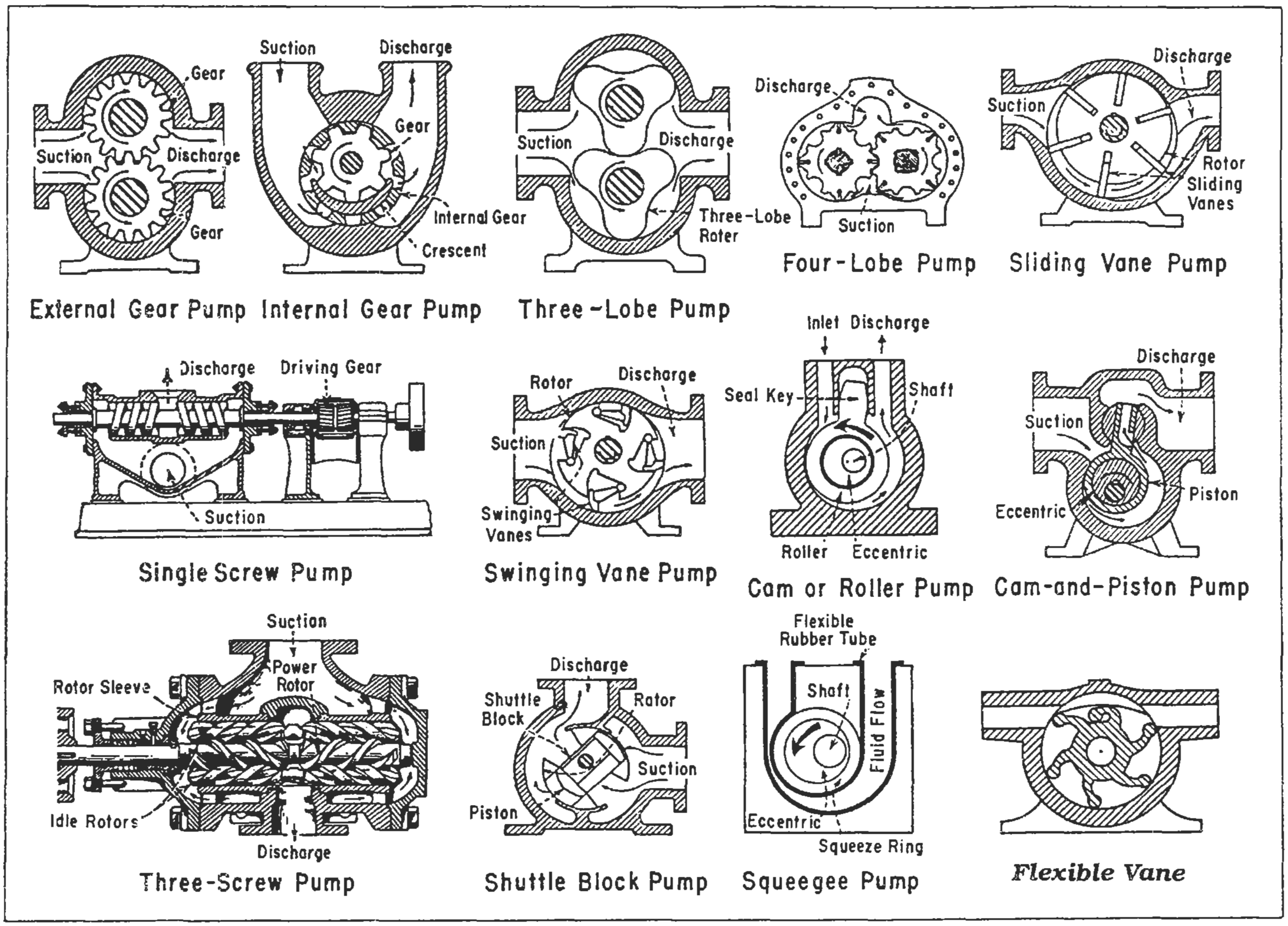


Figure 3-64. Rotary pumps. (By permission, Dolman, R. E., *Chemical Engineering,* Mar. 1952, p. 159.)

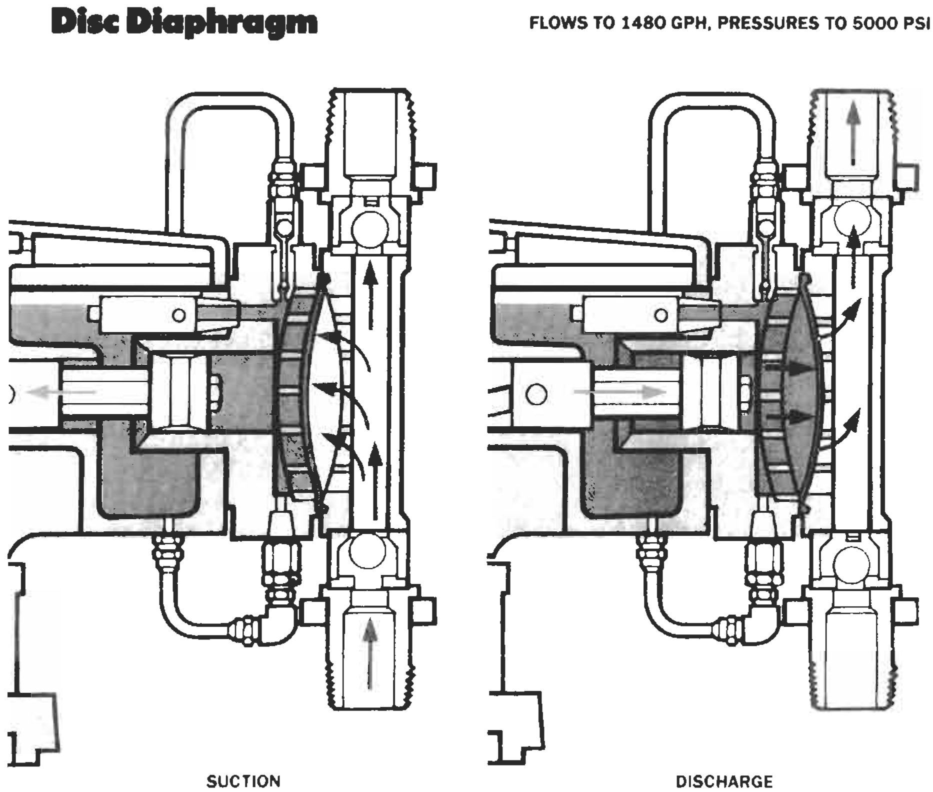


Figure 3-65A. Diaphragm metering pump, "Pulsa" series. One of several styles/types. (By permission, Pulsafeeder, Inc.)

(c) Liquid displacement [6]:

$$d' = \frac{d''(1 - E_n)}{(1 - E_n) + E_n\,(P/P_1)}, \text{ cu ft/min} \qquad (3\text{-}43)$$

where P is the atmospheric pressure, and P_1 is the inlet absolute pressure to the pump.

d'' = theoretical displacement, cu ft/min
d' = liquid displacement, cu ft/min
E_n = percent entrained gas by volume at atmospheric pressure

2. Volume displaced [17]

$$Q' = \frac{D''n}{231} - S'', \text{ GPM}$$
(for no vapor or gas present) (3-44)

where Q' = capacity of rotary pump, fluid plus dissolved gases/entrained gases, at operating conditions, GPM
D'' = displacement (theoretical) volume displaced per revolution(s) of driving rotor, cu in./revolution
n = speed, revolutions per minute of rotor(s), rpm
S'' = slip, quantity of fluid that leaks through internal clearances of pump per unit time, GPM

3. Pump power output (whp) [17]

$$whp_l = (Q'P_{td})/1714 \qquad (3\text{-}45)$$

where P_{td} = differential pressure between absolute pressures at the outlet and inlet to pump, psi
whp_l = power imparted by the pump to the fluid discharged (also liquid HP)
E_v = volumetric efficiency, ratio of actual pump capacity to the volume displaced/unit time

$$E_v = 231\,Q'(100)/(D''n) \qquad (3\text{-}46)$$

4. BHP varies directly with pressure and speed.
5. For speed and pressure constant, BHP varies directly with viscosity.

Selection

Suction and discharge heads are determined the same as for centrifugal pumps. Total head and capacity are used in selecting the proper rotary pump from a manufacturer's data or curves. Since viscosity is quite important in the

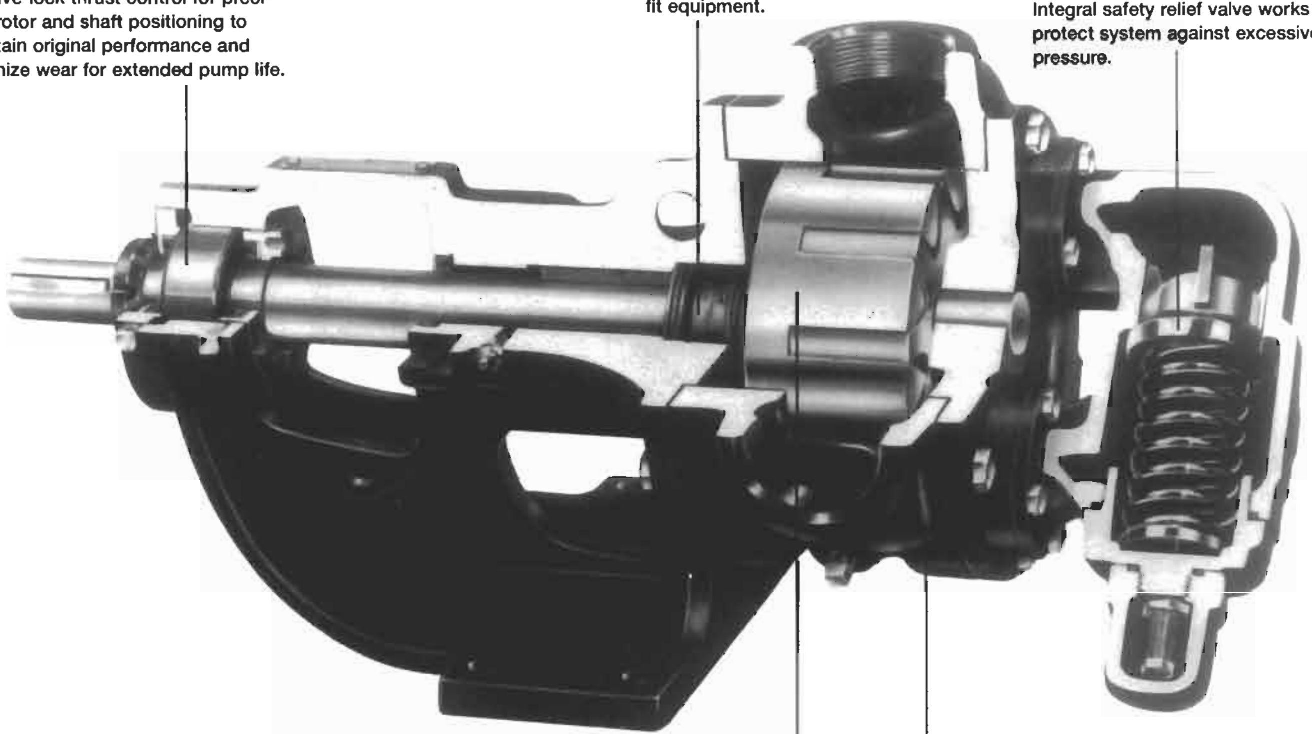


Figure 3-65B. Typical rotary gear pump. (By permission, Viking Pump, Inc., Unit of Idex Corporation.)

selection of these pumps, it is sometimes better to select a larger pump running at low speeds than a smaller pump at high speeds when dealing with viscous materials.

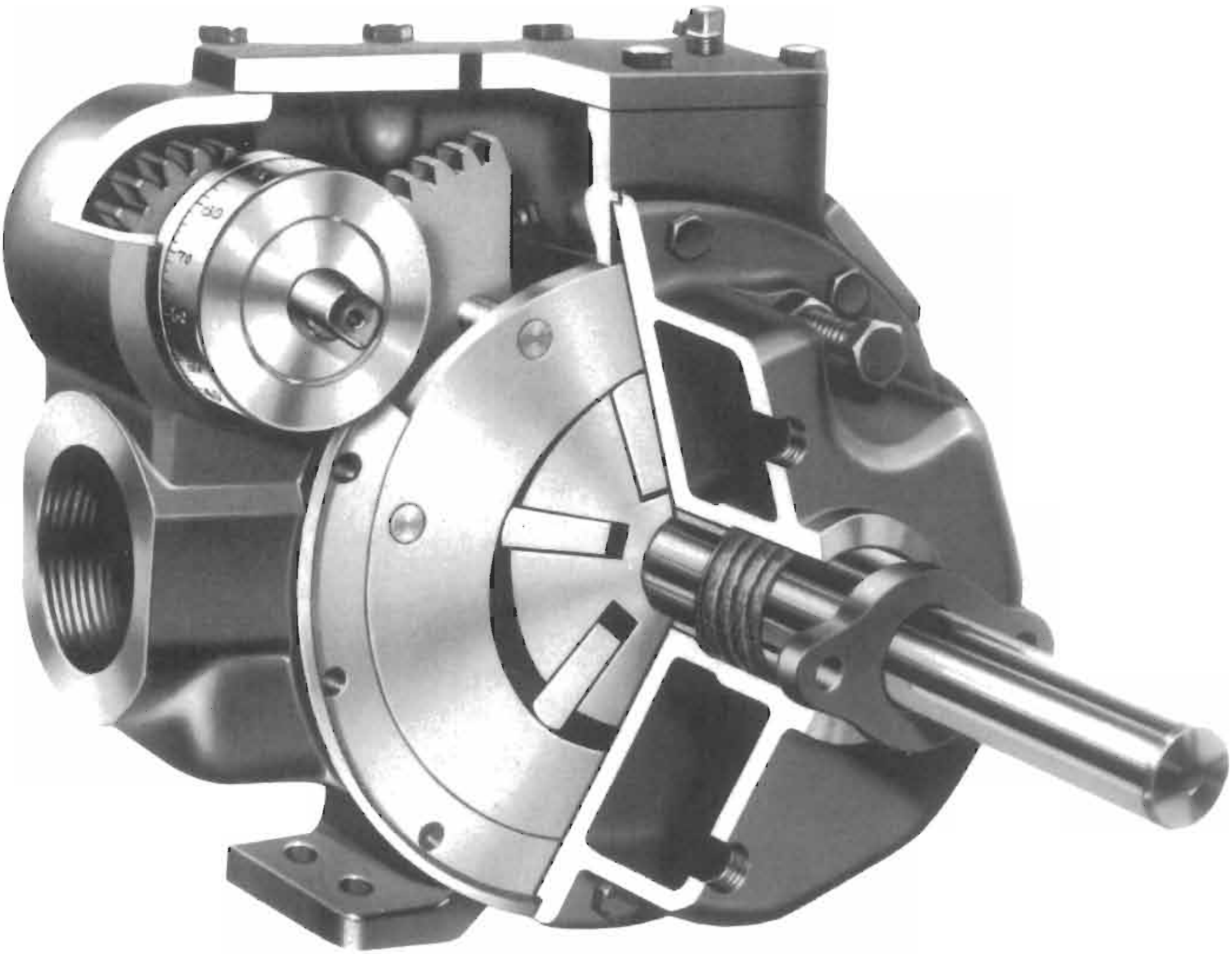
As a general guide, speed is reduced 25–35 percent below rating for each tenfold increase in viscosity above 1000 SSU. Also, generally, the mechanical efficiency of the pump is decreased 10 percent for each ten fold increase in viscosity above 1000 SSU, and referenced to a maximum efficiency of 55 percent at this point. [1]

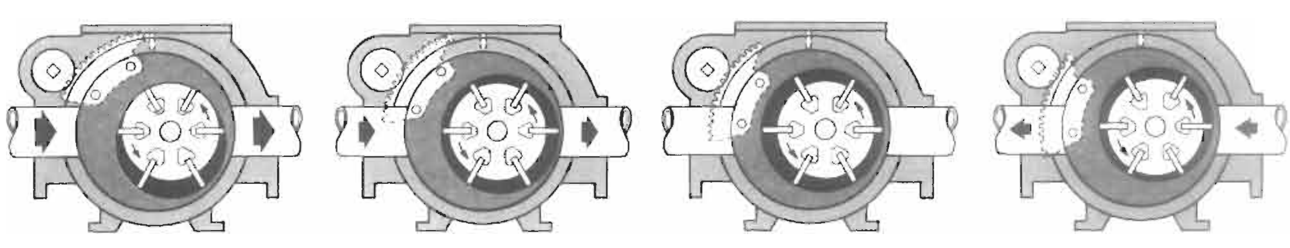
Reciprocating Pumps

Reciprocating pumps are positive displacement piston units driven by a direct connected steam cylinder or by an external power source connected to the crankshaft of the pump piston. Being positive displacement, these pumps can develop very high pressures (10,000 psi and higher) for very low or high capacities (up to 1000 GPM).

Significant Features in Reciprocating Pump Arrangements

I. Liquid Pump End
  A. Pump Pressure Component
    1. Piston
    2. Plunger
  B. Types
    1. Simplex, one piston
    2. Duplex, two piston (Figure 3-66)
    3. Triplex, three piston (not used as steam driven)
  C. Piston or Plunger Action
    1. Single acting, one stroke per rpm
    2. Double acting, two strokes per rpm, cylinder fills and discharges each stroke (Figure 3-67)
  D. Packing for Piston or Plunger
    1. Piston packed: packing mounted on piston and moves with piston; applied to comparatively low pressures





FULL FLOW
The liner (grey area) is in full-capacity position with the seal point at the top and the pumping chamber at the bottom. All liquid coming into the pump at the left is moving out of it at the right.

PARTIAL FLOW
Now the liner has been rotated counter-clockwise, which opens the seal point at the top. This allows part of the liquid to be recirculated, reducing the net flow by a proportionate amount.

NEUTRAL OR NO FLOW
Rotating the liner still further, a point is reached where displacement volumes above and below the rotor are equalized. This causes as much liquid to be returned over the top as is brought forward across the bottom, resulting in zero flow.

REVERSE FLOW
When the liner is rotated past the "no flow" point, the volume above the rotor exceeds that below, and net flow reverses direction even though pump speed and rotation have not changed. (Limited to approx. 30% of full forward flow on all models.)

Figure 3-65C. Sliding vane rotary pump. (By permission, Blackmer Pump, Dover Resources Co.)

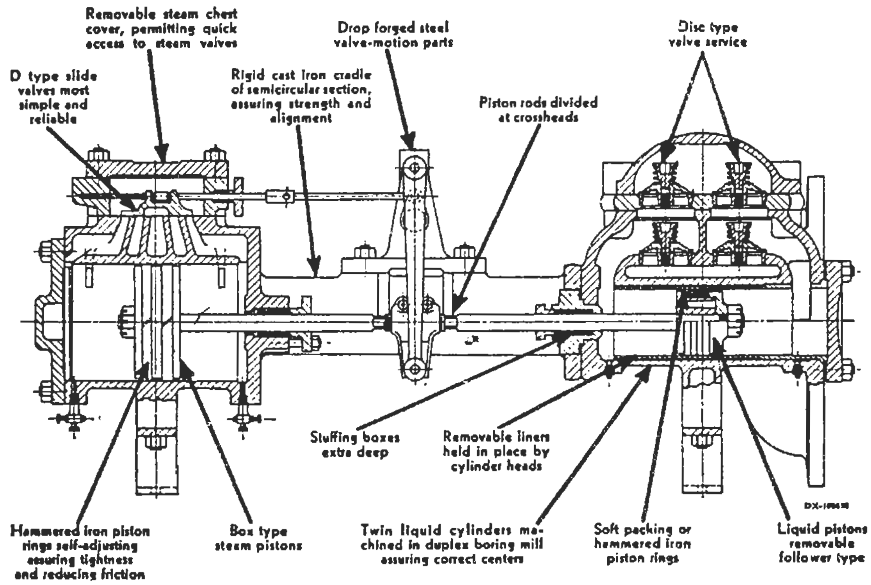


Figure 3-66. General service duplex steam driven piston pump. (Courtesy Worthington Corp.)

2. Cylinder packed: packing stationary; plunger moves; applied to high pressures; more expensive than piston packed.

II. Drive End: Steam

A. Steam Cylinders

1. Simple: single cylinder per cylinder of liquid pump; uses more steam than compound.
2. Tandem Compound; high and low pressure cylinder on same centerline; usually requires 80 psi or greater steam to be economical.
3. Cross Compound: high and low pressure cylinder arranged side-by-side with cranks 90° apart. Need for crank and flywheel arrangement only; usually requires 80 psi or greater steam to be economical. Percentage gain in compounding steam cylinders varies from 25–35 percent for non-condensing, and 25–40 percent for condensing [18].

B. Cylinder Action

1. Direct: steam piston direct connected to liquid piston or plunger through piston rod.
2. Crank and Flywheel: flywheel mounted on crank shaft driven by steam cylinder.

III. Drive End: Power

General features same as steam, except drive always through crankshaft; speed gear increasers or reducers; V-belts, or direct coupling connection to drive shaft.

IV. Designation

Units are identified as: (steam cylinder diameter, inches) (liquid cylinder diameter, inches) (length of stroke, inches).

Application

Piston Type: used for low pressure light duty or intermittent service. Less expensive than the plunger design, but cannot handle gritty liquids.

Plunger Type: used for high pressure heavy duty or continuous service. Suitable for gritty and foreign material service, and more expensive than the piston design.

Performance

The performance of reciprocating pumps provides for ease of operation and control. Depending upon the type of piston action, the fluid may be subject to pulsations unless accumulator or surge drums are provided.

The slip of a pump is fraction or percent loss of capacity relative to theoretical. Slip is $(1 - e_{vol})$, where e_{vol} is the volumetric efficiency. Volumetric efficiency is the actual liquid pumped (usually considered water) relative to that which should theoretically be pumped based on piston displacement.

The NPSH *required* is approximately 3–5 psi of *liquid above* the vapor pressure of the liquid.

The capacity of a pump is given in manufacturers tables as actual, after deducting for volume occupied by piston rod and slippage. Slip varies from 2–10 percent of displacement, with 3 percent being a fair average.

Capacity: actual, for single acting pumps, single cylinder

$$Q = \frac{(12\,a\,t)\,(e_{vol})}{(231)\,(2)} = 0.0204\, d_p^{\,2}\, t e_{vol},\ \text{GPM} \qquad (3\text{-}47)$$

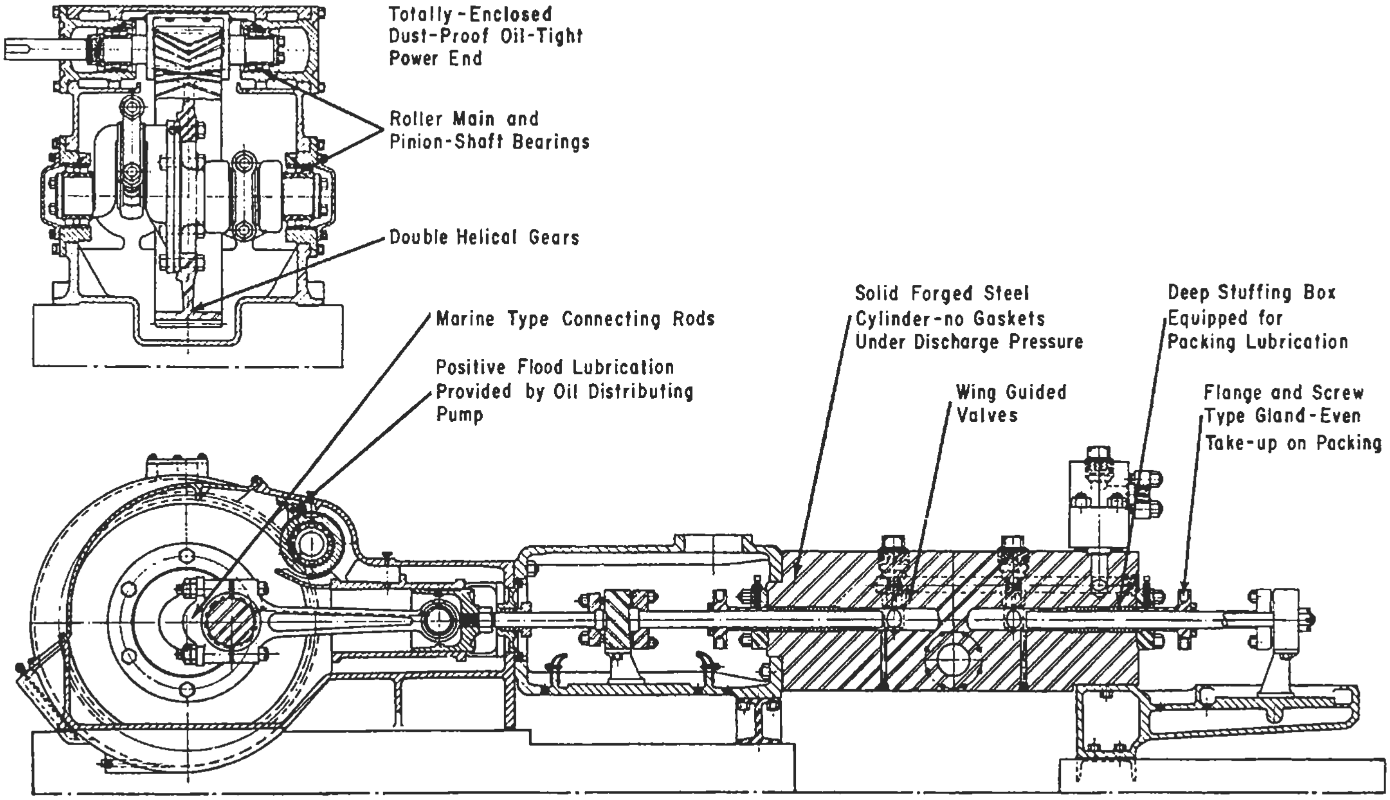


Figure 3-67. Duplex double-acting plunger pump, power driven. (Courtesy Worthington Corp.)

For double acting pumps, single cylinder

$$Q = \text{(two times value for single acting)} - 0.0204\, d^2_r\, t, \text{ GPM} \quad (3\text{-}48)$$

For multiple cylinders, multiply the capacities just obtained by the number of cylinders. If the piston rod does not replace pumping volume as in some arrangements, the last term of the double acting capacity equation is omitted.

Discharge Flow Patterns

Figure 3-68 shows the discharge flow patterns for several reciprocating power pump actions which are essentially the same for steam pumps. The variations above and below theoretical mean discharge indicate the magnitude of the pulsations to be expected. Although not shown, the simplex double acting discharge would follow the action of one piston on the duplex double acting curve from 0 to 360°. Its variation or pulsing is obvious by inspection, and accumulator bottles would be required to smooth the flow. The simplex single acting discharge would be one pumping stroke from 0 to 180°, then no pumping from 180° to 360°; and here again the pulse action is obvious.

Horsepower

Hydraulic

$$HHP = Q(\text{actual})H/3960 \quad (3\text{-}49)$$

Brake

$$BHP = HHP/e \quad (3\text{-}50)$$

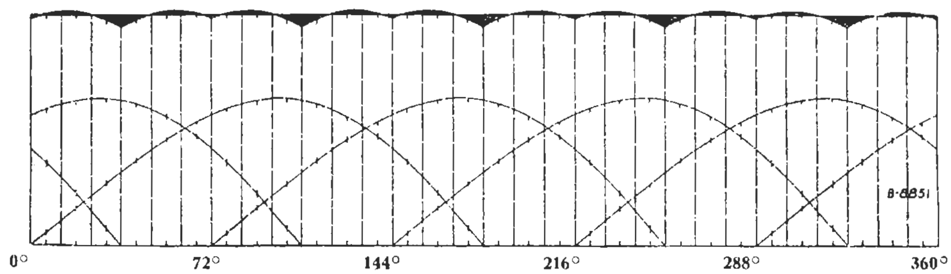
where e represents the total overall efficiency, and is $e = e_m(e_{vol})$, and e_m is the mechanical efficiency.

Mechanical efficiencies of steam pumps vary with the types of pump, stroke and the pressure differential. Some representative values are 55 to 80 percent for piston pumps with strokes of 3 inches and 24 inches respectively, and pressure differential up to 300 psi. For the same strokes a plunger design varies from 50 to 78 percent, and at over 300 psi differential the efficiencies are 41 to 67 percent [9]. Steam required is approximately 120 lbs/hour per BHP.

QUINTUPLEX SINGLE-ACTING PUMP

Variation Above Mean, 1.8%
Variation Below Mean, 5.2%

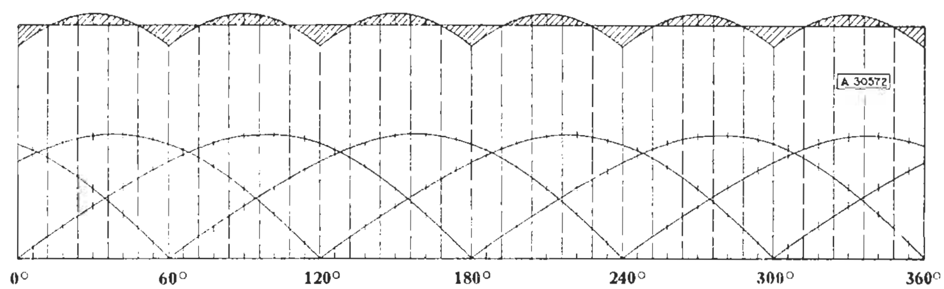
Total Variation, 7. %



SEXTUPLEX SINGLE-ACTING PUMP

Variation Above Mean, 4.82%
Variation Below Mean, 9.22%

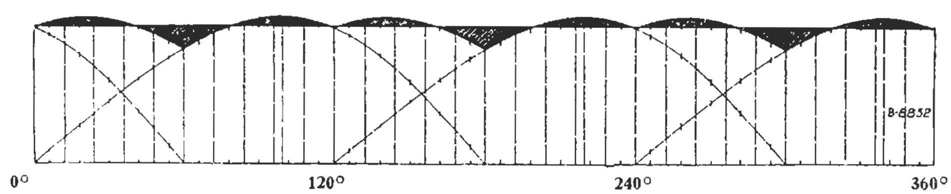
Total Variation, 14.04%



TRIPLEX SINGLE-ACTING PUMP

Variation Above Mean, 6.1%
Variation Below Mean, 16.9%

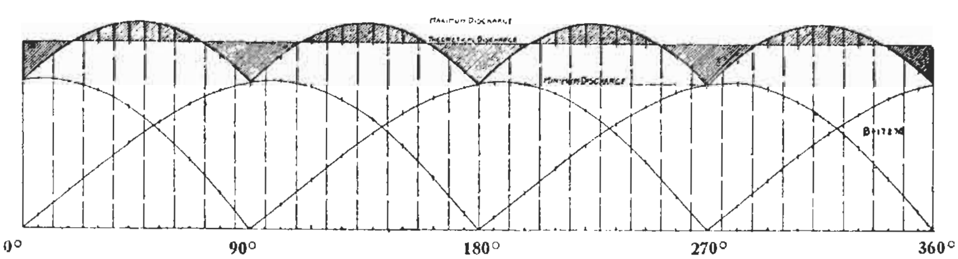
Total Variation, 23. %



QUADRUPLEX SINGLE-ACTING PUMP

Variation Above Mean, 11. %
Variation Below Mean, 21.5%

Total Variation, 32.5%



DUPLEX DOUBLE-ACTING PUMP

Variation Above Mean, 24.1%
Variation Below Mean, 21.5%

Total Variation, 45.6%

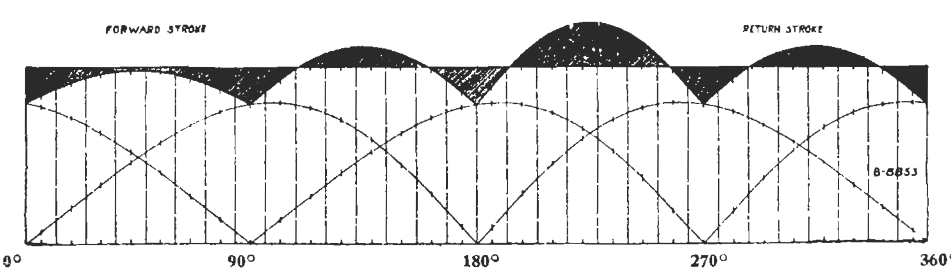


Figure 3-68. Reciprocating pump discharge flow patterns. (Courtesy the Aldrich Pump Co.)

HORIZONTAL DIRECT ACTING STEAM PUMP

OR POWER PUMP

ITEM ____________

____

APPARATUS ____________________________

OPERATE AT ________ # GA; MAX WP ________ # GA & MAX T ________ °F ________ DATE ________ BY ________

BASED ON STEAM AT ________ LBS./SQ. IN. GA AND ________ °F ________ DATED ________

ITEM NO. ____________________________

SIZE AND TYPE ____________________________

A. CONDITIONS OF SERVICE — **DATA**

- OPERATION ________
- PUMPED MATERIAL ________
- °API AT 60/60° F ________
- SPECIFIC GRAVITY AT 60° F ________
- SPECIFIC GRAVITY AT P. T. ________
- VISCOSITY AT P. T. ________
- PUMPING TEMPERATURE—°F ________
- U. S. G. P. M. AT 60° F ________
- U. S. G. P. M. AT P. T. ________
- DISCHARGE PRESSURE—LBS/SQ. IN. GA ________
- SUCTION PRESSURE—LBS/SQ. IN. GA ________
- APPROX. VAPOR PRESSURE OF LIQUID ________
- EXHAUST STEAM ________

B. PUMP SPECIFICATIONS

- SUCTION VALVES—NUMBER ________
- SUCTION VALVES—SIZE; INCHES ________
- SUCTION VALVES—AREA EACH, SQ. IN. ________
- DISCHARGE VALVES—NUMBER ________
- DISCHARGE VALVES—SIZE, INCHES ________
- DISCHARGE VALVES—AREA EACH, SQ. IN. ________
- SUCTION CONNECTION ________ SIZE: ________ SERIES: ________
- DISCHARGE CONNECTION ________ SIZE: ________ SERIES: ________
- TYPE OF RINGS—STEAM END: ________ PUMP END: ________

C. MATERIALS: — **STEAM END** — **LIQUID END**

- CYLINDERS ________
- LINERS ________
- PISTONS ________
- PISTON RODS ________
- VALVES ________
- VALVE SEATS ________
- VALVE SPRINGS ________
- PACKING ________

D. PERFORMANCE

- PISTON SPEED—FT/MIN. ________ REQ'D H. P. ________ INSTALLED H.P. ________
- R.P.M. ________ DRIVER RPM ________
- STALLING PRESSURE—LBS/SQ. IN. GA ________
- STEAM CONSUMPTION ________

COPY TO ________ DATE ________ CHECKED ________ DATE ________ APPROVED ________ DATE ________

REMARKS: ________________________

Figure 3-69. Horizontal direct-acting steam pump or power pump.

Pump Selection

Reciprocating pump selection follows the fundamentals of centrifugal pumps:

1. Evaluation suction side head loss.
2. Evaluate discharge side head loss.
3. Determine system static pressure.
4. Determine total differential head across pump.
5. Determine the $NPSH_A$ available on suction of pump.
6. From manufacturer's performance tables, select pump nearest to GPM and head requirements.
7. Contact manufacturer for final recommendations, give complete system requirements, and physical properties of liquid. Figure 3-69 is convenient for this purpose.

Nomenclature

a = Area of piston or plunger, sq in.
B_d = Bell diameter of vertical sump pump, ft
BHP = Brake horsepower
$(BHP)_{vis}$ = Brake horsepower when handling viscous material
C_E = Viscosity correction for efficiency to convert to water performance
C_H = Viscosity correction for head, to convert to water performance
C_Q = Viscosity correction for capacity, to convert to water performance
c_p = Specific heat of liquid, BTU/lb − °F
D = Height of liquid (static) above (+) or below (−) the centerline of the pump on discharge side, ft
D′ = Incremental height of liquid (static) above normal D level, to establish "worst case" condition, ft; Figure 3-38
D″ = Theoretical displacement volume displaced per revolution(s) of driving rotors, cu in./rev
d = Impeller diameter, in.
d_p = Diameter of piston or plunger, in.
d_r = Diameter of piston rod, in.
d′ = Liquid displacement, cu ft/min
d″ = Theoretical displacement, cu ft/min
eHP = Electrical horsepower
E = Efficiency, percent
E_n = Fraction entrained gas by volume at atmospheric pressure
E_v = Volumetric efficiency, ratio of actual pump capacity to volume displaced per unit of time
E_w = Pump efficiency with water, percent
E_{vis} = Pump efficiency with viscous fluid, percent
e = Pump efficiency, fraction
e_w = Pump efficiency with water, fraction
e_{vis} = Pump efficiency with viscous fluid, fraction
e_{vol} = Volumetric efficiency, fraction
e_M = Maximum safe flowing efficiency, overall pump, fraction
g = Acceleration of gravity, 32 ft/sec/sec
H = Total head developed by a pump, ft of fluid; or total head/stage, ft, or,
H = Static head discharge ft (Figure 3-38, -39, -40)
H_{so} = Head at no flow, or shutoff, ft
H_{vis} = Head of viscous fluid, ft
H_w = Water equivalent head, ft
h_d = Discharge head on a pump, ft of fluid
h_s = Suction head (or suction lift) on a pump, ft of fluid
h_{SL}, h_{DL} = Friction losses in pipe and fittings:, subscript SL for suction line; and DL for discharge line, ft of fluid
h_v = Velocity head, ft of fluid
L = S = Static head, suction side, ft (Figure 3-38)
l = Water depth in sump, ft (Figure 3-62)
N_s = Specific speed, dimensionless
N_p = Number of pumps
n = Rotative speed, revolutions per minute = RPM = rpm
P = Positive external pressure on surface of liquid (+) or partial vacuum on surface of liquid (−)
P_a = Atmospheric pressure or absolute pressure in vessel, psia
P_{so} = Brake horsepower at shutoff or no flow
P_{td} = Differential pressure between absolute pressures at outlet and inlet to pump, psi
P_{vp} = Vapor pressure of liquid at pumping temperature, psia
p′ = Absolute pressure, inches mercury abs
p'_a = Atmospheric pressure or absolute pressure in vessel expressed as ft of fluid
p'_{vp} = Vapor pressure of liquid at pumping temperature expressed as ft of fluid
Q = Flow rate, gal per minute
Q′ = Capacity of rotary pump, fluid plus dissolved gases/entrained gases at operating conditions, GPM
Q_M = Minimum flow, GPM
Q_N = Head at best efficiency point on pump curve, ft
Q_{vis} = Viscous liquid capacity, GPM
Q_w = Water capacity, GPM
S = Suction static head, ft, or height of liquid (static) above (+) or below (−) the center line of the pump on suction side, ft, or,
S = Suction lift, negative suction head, ft
S'_L = Worst case suction side static lift, ft (Figure 3-39)
S″ = Slip, quantity of fluid that leaks through internal clearances of rotary pump per unit time, GPM
SpGr = Specific gravity of liquid at pumping temperature referred to water = 1.0
s = Stroke, in.
Δt = Temperature rise, °F
ΔT_r = Temperature rise, °F/min
t = Piston speed or travel, ft/min
V = Liquid velocity, ft/sec
v = Average velocity, ft/sec
W = Width of channel with series pump, ft
W_1 = Weight of liquid in pump, lb
whp = Water or liquid horsepower
whp_1 = Power imparted by pump to fluid discharged (also liquid horsepower)

Subscripts

1,2 = Refer to first and second condition respectively
A = Available from pump system (NPSH)
L = Liquid
d = Discharge side of pump
d_1 = Friction losses for pipe fittings and related items on discharge side of pump

s_1 = Friction losses for pipe valves and other system losses, suction side of pump
R = Required by pump (NPSH)
s = Suction side of pump

Greek Symbols

ρ = Fluid mass gravity, lb/cu ft

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Chapter 4

Mechanical Separations

Practically every process operation requires the separation of entrained material or two immiscible phases in a process. This may be either as a step in the purification of one stream, or a principal process operation [64]. These separations may be:

1. liquid particles from vapor or gas
2. liquid particles from immiscible liquid
3. dust or solid particles from vapor or gas
4. solid particles from liquid
5. solid particles from other solids

These operations may sometimes be better known as mist entrainment, decantation, dust collection, filtration, centrifugation, sedimentation, screening, classification, scrubbing, etc. They often involve handling relatively large quantities of one phase in order to collect or separate the other. Therefore the size of the equipment may become very large. For the sake of space and cost it is important that the equipment be specified and rated to operate as efficiently as possible [9]. This subject will be limited here to the removal or separation of liquid or solid particles from a vapor or gas carrier stream (1. and 3. above) or separation of solid particles from a liquid (item 4). Reference [56] is a helpful review.

Other important separation techniques such as pressure-leaf filtration, centrifugation, rotary drum filtration and others all require technology very specific to the equipment and cannot be generalized in many instances.

Particle Size

The particle sizes of liquid and solid dispersoids will vary markedly depending upon the source and nature of the operation generating the particular particles. For design of equipment to reduce or eliminate particles from a fluid stream, it is important either to know from data the range and distribution of particle sizes, or be in a position to intelligently estimate the normal and extreme expectancies. Figures 4-1 and 4-1A give a good overall picture of dimensions as well as the descriptive terminology so important to a good understanding of the magnitude of a given problem. The significant laws governing particle performance in each range is also shown.

Particle sizes are measured in microns, μ. A micron is 1/1000 millimeter or 1/25,400 inch. A millimicron, $m\mu$, is 1/1000 of a micron, or 1/1,000,000 millimeter. Usually particle size is designated as the average diameter in microns, although some literature reports particle radius. Particle concentration is often expressed as grains/cubic feet of gas volume. One grain is 1/7000 of a pound.

The mechanism of formation has a controlling influence over the uniformity of particle size and the magnitude of the dimensions. Thus, sprays exhibit a wide particle size distribution, whereas condensed particles such as fumes, mists and fogs are particularly uniform in size. Table 4-1 gives the approximate average particle sizes for dusts and mists which might be generated around process plants. Figure 4-2 indicates the size ranges for some aerosols, dusts and fumes. Table 4-2 gives typical analysis of a few dusts, and Table 4-3 gives screen and particle size relationships. Table 4-4 gives approximate mean particle size for water spray from a nozzle.

Preliminary Separator Selection

The Sylvan Chart [2] of Figure 4-3 is useful in preliminary equipment selection, although arranged primarily for dust separations, it is applicable in the appropriate parts to liquid separations. Perry [23] presents a somewhat similar chart that is of different form but contains much of the same information as Figure 4-1 and 4-1A.

Table 4-1
Sizes of Common Dusts and Mists

Dust or Mist	Average Particle Diameter, Microns
Human Hair (for comparison)	50-200
Limit of visibility with naked eye	10-40
Dusts	
Atmospheric dust	0.5
Aluminum	2.2
Anthracite Coal Mining	
Breaker air	1.0
Mine Air	0.9
Coal Drilling	1.0
Coal loading	0.8
Rock drilling	1.0
Alkali fume	1-5
Ammonium Chloride fume	0.05-0.1-1
Catalyst (reformer)	0.5-50
Cement	0.5-40-55
Coal	5-10
Ferro-manganese, or silicon	0.1-1
Foundry air	1.2
Flour-mill	15
Fly Ash (Boiler Flue gas)	0.1-3
Iron (Gray Iron Cupola)	0.1-10
Iron oxide (steel open hearth)	0.5-2
Lime (Lime Kiln)	1-50
Marble cutting	1.5
Pigments	0.2-2
Sandblasting	1.4
Silica	1-10
Smelter	0.1-100
Taconite Iron ore	
(Crushing & Screening)	0.5-100
Talc	10
Talc Milling	1.5
Tobacco smoke	0.2
Zinc oxide fume	0.05
Zinc (sprayed)	15
Zinc (condensed)	2
Mists	
Atmospheric fog	2-15
Sulfuric acid	0.5-15

Compiled from References 1, 13, and 15.

Table 4-2
Typical Dust Size Analysis*

	DUST			
	Rock	Cement Kiln Exhaust	Foundry Sand	Limestone
Sp. Gravity	2.63	2.76	2.243	2.64
Apparent Wt., lbs./cu.ft.	61.3	52.0	45.9	72.0
Screen Analysis (Percent passed)				
100 Mesh	98.8	99.6	91.2	85.6
200 Mesh	92.8	92.2	78.4	76.4
325 Mesh	79.6	80.8	67.6	66.4
400 Mesh	70.8	73.2	64.4	63.2
Elutriation Analysis: Percent Under				
Terminal Velocity				
320 In./min.	75.8	78.0	64.2	70.5
80 In./min.	37.0	61.0	53.9	52.0
20 In./min.	17.5	40.8	42.0	33.0
5 In./min.	8.9	23.0	32.0	18.0

*Compiled from Bulletin No. 1128, American Blower Corp., Detroit, Michigan.

Table 4-3
Dry Particle Screen Sizes

W. S. Tyler Screen Scale	Micron (approximate)
80	174
100	146
115	123
150	104
170	89
200	74
250	61
270	53
325	43

Table 4-4
Approximate Particle Sizes From Liquid Full Cone Spray Nozzles*
Liquid: Water

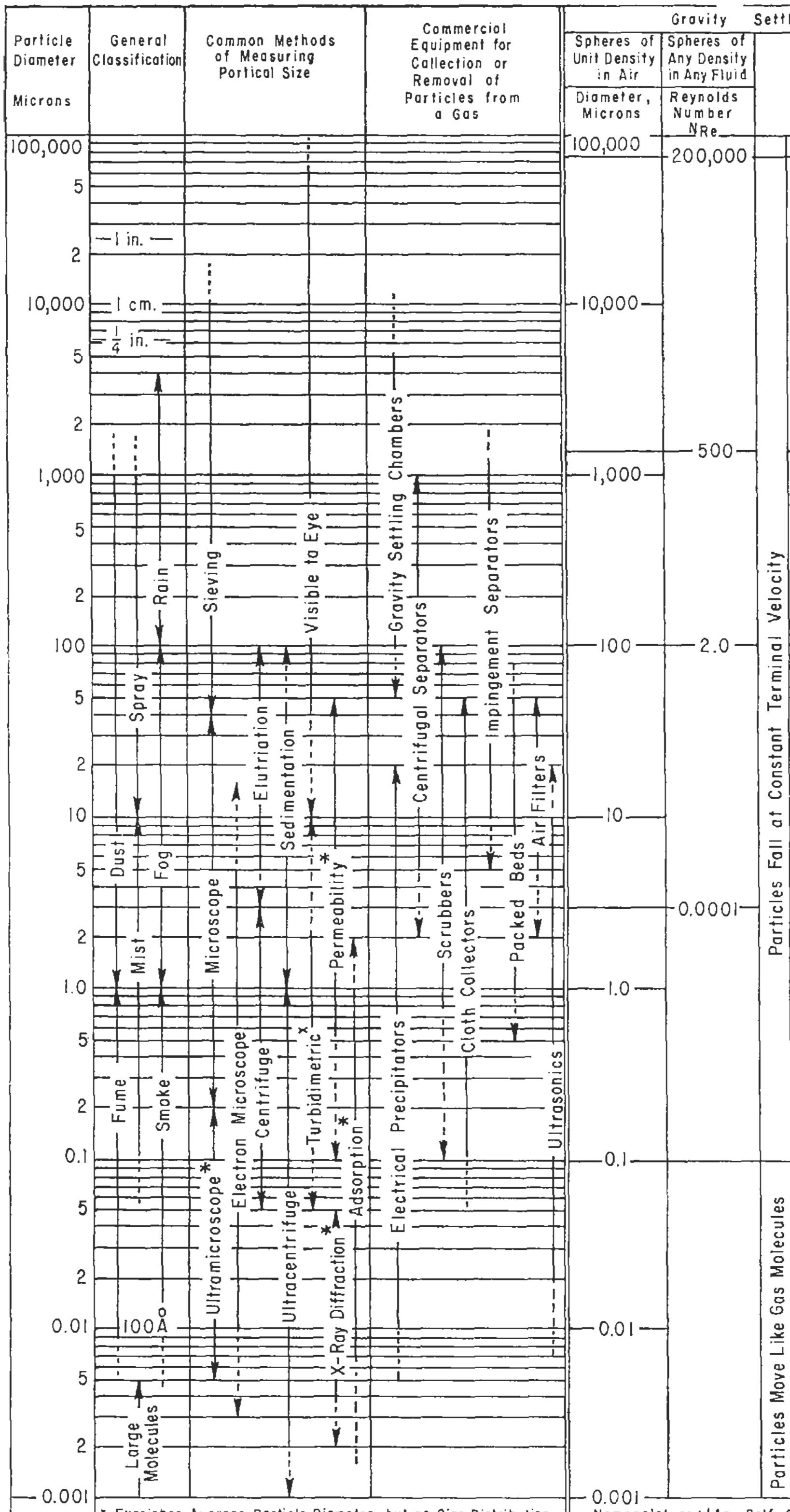
Nozzle Size, In.	Operating Pressure, Psig	Approx. Mean Particle Size, Microns
½	15	1200
	60	750
¾	10	1600
	40	1000
1	15	1750
	40	1250
1½	15	2300
	60	1800
3	10	5300
	30	4300

*Private communication, Spraying Systems Co., Bellwood, Ill.

Example 4-1: Basic Separator Type Selection [2,17]

A suitable collector will be selected for a lime kiln to illustrate the use of the Sylvan Chart (Figure 4-3). Referring to the chart, the concentration and mean particle size of the material leaving the kiln can vary between 3 and 10 grains/cu ft with 5 to 10 microns range of mass mean particle size. Assume an inlet concentration of 7.5 grains/cu ft and inlet mean particle size of 9 microns. Projection of this point vertically downward to the collection efficiency portion of the chart will indicate that a low resistance cyclone will be less than 50% efficient; a high efficiency centrifugal will be between 60 and 80% efficient, and a wet collector, fabric arrester and electro-static precipitator will be 97%-plus efficient. The last three collectors are often preceded by a precleaner so a high efficiency centrifugal will be selected. Using the average line of this group, the efficiency will be 70%. Therefore, the effluent from this collector will have a concentration of $7.5(1.00 - 0.70) = 2.25$ grains/cu ft.

Draw a line through the initial point with a slope parallel to the lines marked "industrial dust." Where deviation is not known, the average of this group of lines will normally be sufficiently accurate to predict the mean par-



Laws of Settling	Critical Particle Diameter Above Which Law Will Not Apply
Newton's Law $C = 0.44$ $u_t = 1.74\sqrt{\dfrac{g_L D_p(\rho_s-\rho)}{\rho}}$	$D_{p,crit} = K_{cr}\left[\dfrac{\mu^2}{g_L\rho(\rho_s-\rho)}\right]^{\frac{1}{3}}$ $K_{cr} = 2{,}360$ for Newton's Law
Intermediate Law $C = 18.5\, N_{Re}^{-0.6}$ $u_t = \dfrac{0.153\, g_L^{0.71} D_p^{1.14}(\rho_s\rho)^{0.71}}{\rho^{0.29}\mu^{0.43}}$	$K_{cr} = 43.5$ for Intermediate Law
Stokes' Law $C = 24\, N_{Re}^{-1}$ $u_t = \dfrac{g_L D_p^2(\rho_s-\rho)}{18\mu} = u_{ts}$	$K_{cr} = 33$ for Stokes' Law
Stokes-Cunningham Law $u_t = K_m\, u_{ts}$	The Cunningham Correction on Stokes' Law Becomes Important for Particles of Diameters Under 3 Microns for Settling in Gases and Under 0.01 Micron for Settling in Liquids.
$K_m = 1 + K_{me}\;(\lambda_m / D_p)$ The Value K_{me} has Experimentally been Shown to lie Between 1.3 and 2.3 for Different Gases, Particle Sizes, and Materials (Wasser, Physik. Z., 34, 257-278 [1933]). An Approximate Average Value Based on the Data of Millikan is Empirically Given by $K_{me} = 1.644 + 0.552\, e^{-(0.656\, D_p/\lambda_m)}$	
Brownian Movement $\Delta x = \sqrt{\dfrac{4 g_c\, RTK_m\, t}{3\pi^2\mu\, N D_p}}$	Brownian Movement is a Random Motion Superimposed upon the Gravitational Settling Velocity of the Particle it Becomes Appreciabe for Particles under 3 Microns Diameter and Becomes Entirely Predominant for Particles Under 0.1 Micron.

* Furnishes Average Particle Diameter, but no Size Distribution
x Size Distribution may be Obtained by Special Calibration

From Kinetic Theory of Gases:

$$\lambda_m = 3\mu/\rho\,\bar{v}$$

$$\bar{v} = \sqrt{8 g_c RT/\pi M}$$

Nomenclature: (Any Self-Consistent System of Units may be Employed; English units are given by way of Example.)

- C = Overall Drag Coefficient, Dimensionless
- D_p = Diameter of Spherical Particle, ft.
- $D_{p\,crit}$ = Critical Particle Diameter Above Which Law will Not Apply, ft.
- g_c = Conversion Factor, 32.17 (lb. mass/lb. Force)/(ft./sec.)
- g_L = Local Acceleration due to Gravity, (ft.)/(sec.)(sec.)
- K_{cr} = Proportionality Factor, Dimensionless
- K_m = Stokes-Cunningham Correction Factor, Dimensionless
- K_{me} = Proportionality Factor, Dimensionless
- M = Molecular Weight, lb./mole
- N = Number of Gas Molecules in a mole, 2.76×10^{26} Molecules/lb. mole
- N_{RC} = Reynolds Number, Dimensionless $= D_p\rho\mu\, E/\mu$
- R = Gas Constant, 1,546 (ft.-lb. Force)/(lb. mole)(°F)
- t = Time, sec.
- T = Absolute Gas Temperature, °F abs., or °R
- u_t = Terminal Settling Velocity of Particle Under Action of Gravity, ft./sec.
- u_{ts} = Terminal Settling Velocity of Particle as Calculated from Stokes' Law, ft./sec.
- $\bar{v}$ = Mean Molecular Speed, ft./sec.
- Δx = Average Linear Amplitude or Displacement of Particle in Time t, ft.
- ρ = Fluid Density, lb. mass/cu. ft.
- ρ_s = True Density of Particle, lb. mass/cu. ft.
- μ = Fluid Viscosity, (lb. mass)/(ft.)(sec.)
- λ_m = Mean Free Path of Gas Molecules, ft.

Figure 4-1. Characteristics of dispersed particles. By permission, Perry, J. H., Ed., *Chemical Engineers Handbook*, 3rd. Ed., 1950, McGraw-Hill Company, Inc.

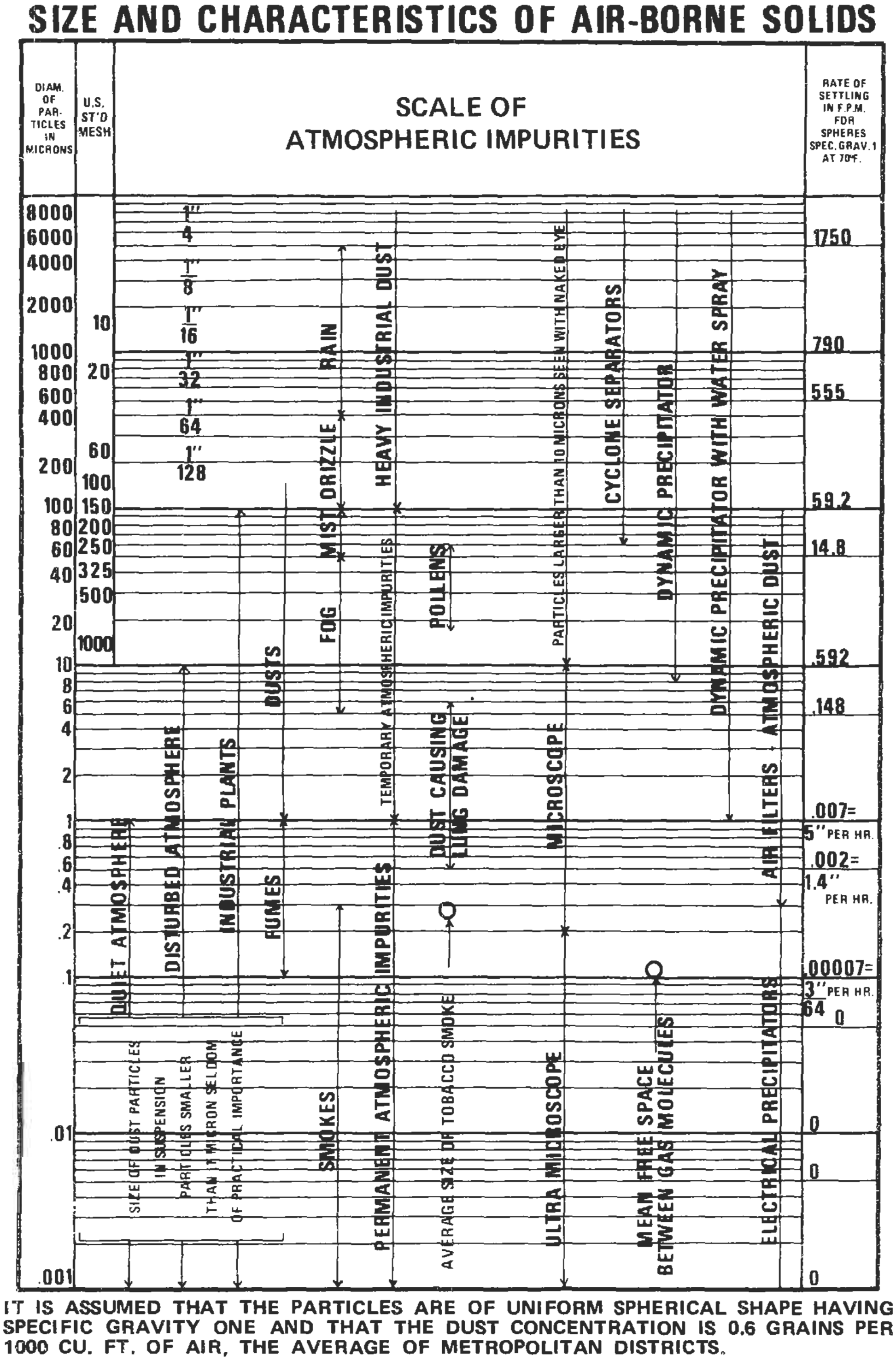


Figure 4-1A. Size and characteristics of air-borne solids. By permission, *Hoffman Handy Engineering Data,* Hoffman Air and Filtration Systems, Inc.

ticle size in the collector effluent. Where this line intersects the horizontal line marked 2.25 grains/cu ft, a vertical line through the point will indicate the effluent mean particle size of 6.0 microns.

A projection of this point of collector effluent vertically downward shows that a second high efficiency centrifugal will be less than 50% efficient. A wet collector, fabric arrester and electro-static precipitator will be not less than

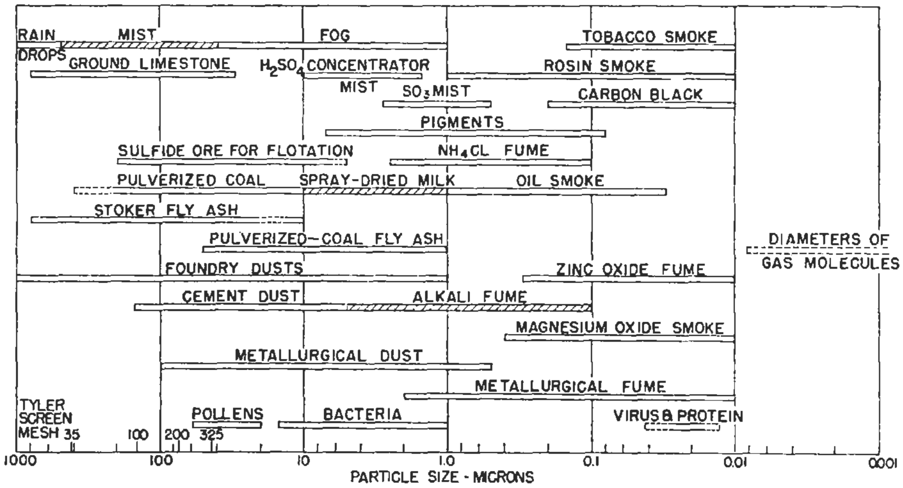


Figure 4-2. Particle-size ranges for aerosols, dusts, and fumes. Courtesy, H. P. Munger, Battelle Memorial Institute.

93% efficient. Selection of a good wet collector will show an efficiency of 98%. The effluent leaving this collector will have a concentration of 2.25(1.00 − 0.98) = .045 grains/cu ft. Using the line initially drawn, at the point where it intersects the line of 0.045 grains/cu ft will indicate a mean particle size in the effluent of 1.6 microns.

Guide to Dust Separator Applications

Table 4-5 [10] summarizes dry dust particle separators as to general application in industry, and Table 4-6 and Figures 4-4 and 4-5 [42] compare basic collector characteristics. Figure 4-5 presents a typical summary of dust collection equipment efficiencies which have not changed significantly for many years except for specialized equipment to specialized applications.

Guide to Liquid-Solid Particle Separators

Table 4-7 summarizes liquid particle separators as to the general process-type application.

Gravity Settlers

The use of these settlers is not usually practical for most situations. The diameters or cross-section areas become too large for the handling of anything but the very smallest of flowing vapor streams. In general, gravity settlers of open box or tank design are not economical for particles smaller than 325 mesh or 43µ [23].

They are much more practical for solids or dusts, although even for these situations the flow quantities must be small if the sizes are not to become excessive. With unusually heavy and/or large particles the gravity separator can be used to advantage.

The fundamentals of separation for a particle moving with respect to a fluid are given by the drag coefficient of Figure 4-6.

The motion of particle and fluid are considered relative, and the handling of the relations are affected only by conditions of turbulence, eddy currents, etc.

Terminal Velocity

When a particle falls under the influence of gravity it will accelerate until the frictional drag in the fluid balances the gravitational forces. At this point it will continue to fall at constant velocity. This is the terminal velocity or free-settling velocity. The general formulae for any shape particle are [13]:

$$u_t = \sqrt{\frac{2g_L m_p(\rho_s - \rho)}{\rho \rho_s A_p C}} \tag{4-1}$$

For spheres:

$$u_t = \sqrt{\frac{4g_L D_p(\rho_s - \rho)}{3\rho C}} \tag{4-2}$$

(a) Spherical particles between 1500 and 100,000 microns; Newton's Law:

$$u_t = 1.74\sqrt{\frac{g_L D_p(\rho_s - \rho)}{\rho}} \tag{4-3}$$

C = 0.445 average drag coefficient

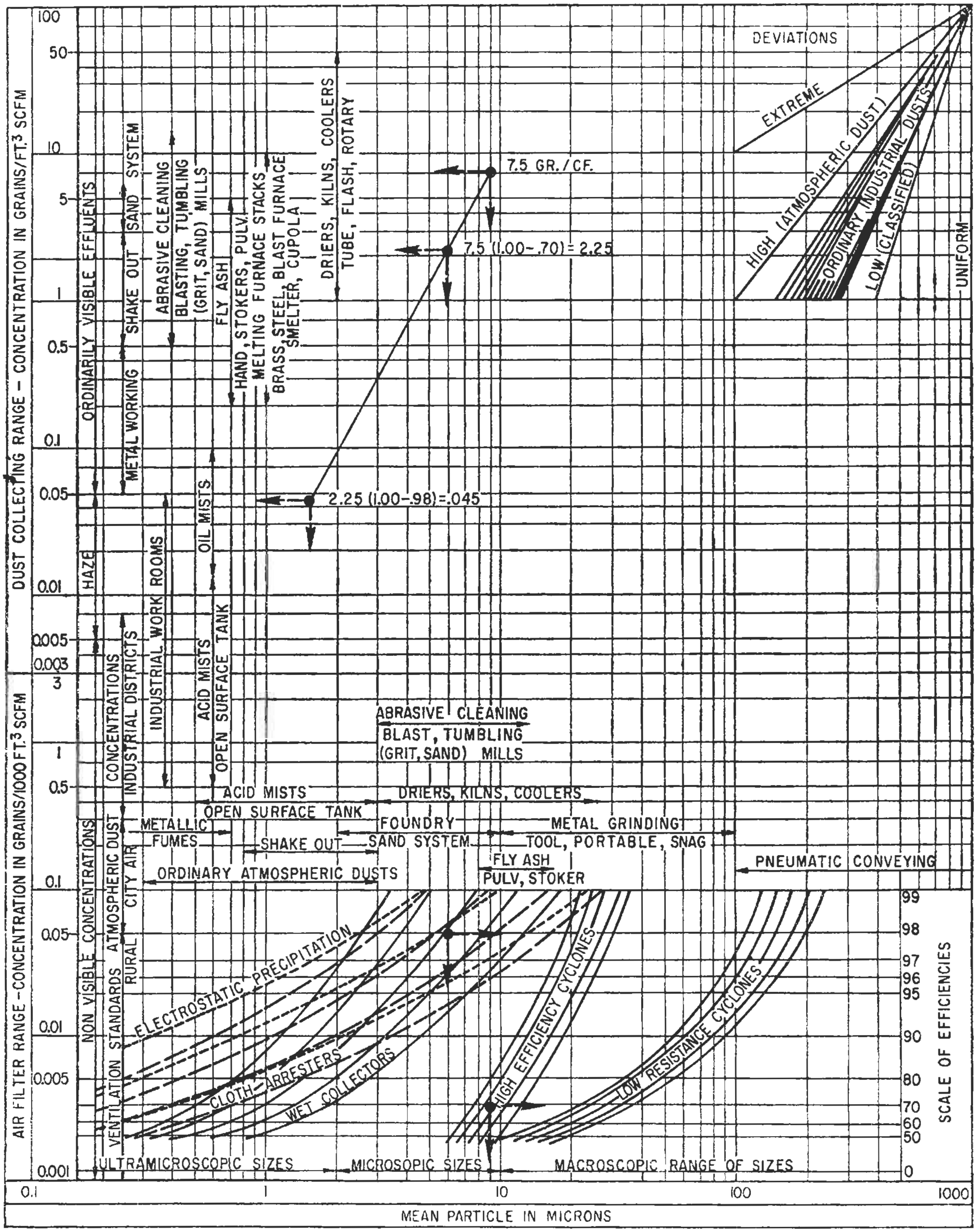


Figure 4-3. Range of particle sizes, concentration, and collector performance. Courtesy American Air Filter Co. Inc.

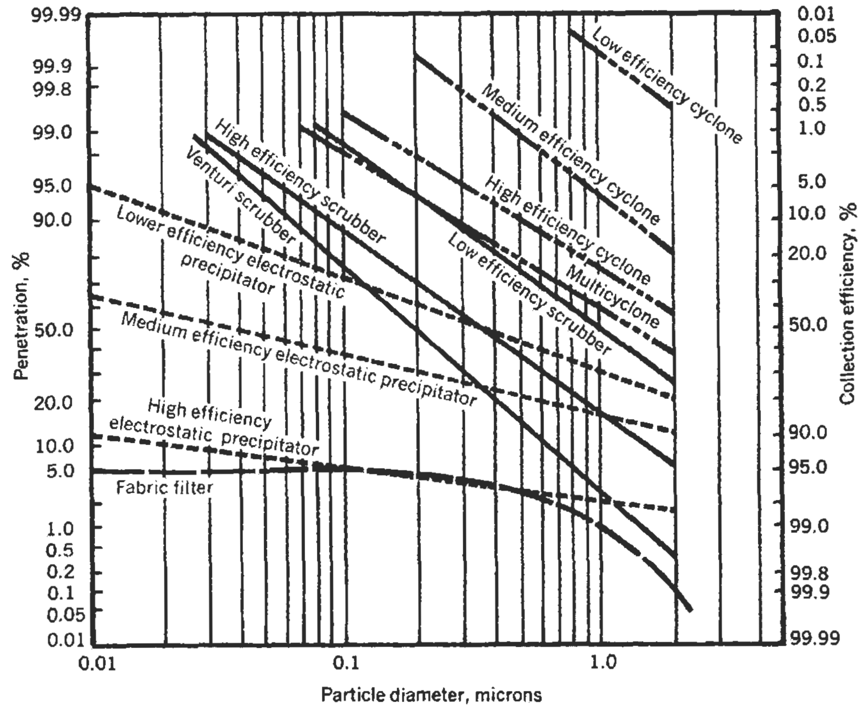


Figure 4-4. Comparison chart showing ranges of performance of several collection/control devices in air streams. By permission, Vandegrift, *et. al. Chemical Engineering,* Deskbook Issue, June 18, 1973, p. 109.

(b) Spherical particles between 100 and 1500 microns diameter [13]:

$$u_t = \frac{0.153 g_L^{0.71} D_p^{1.14} (\rho_s - \rho)^{0.7}}{\rho^{0.29} \mu^{0.43}} \quad (4\text{-}4)$$

$C = 18.5\, N_{Re}^{-0.6}$, (See Figure 4-1)

(c) For spherical particles between 3 and 100 microns and Reynolds numbers between 0.0001 and 2.0, Stokes Law:

$$CN_{Re} = 24$$

$$F_d = 3\pi\mu u D_p / g_L$$

and:

$$u_t = g_L D_p^2 \frac{(\rho_s - \rho)}{18\mu} \quad (4\text{-}5)$$

For particles smaller than 0.1μ the random Brownian motion is greater than the motion due to gravitational settling. Therefore the above relations based on Stokes Law will not hold.

(d) Spherical particles between 0.1 and 3 microns: Stokes-Cunningham Law [12]:

$$u_t = K_m u_{ts} \quad (4\text{-}6)$$

$$K_m = 1 + K_{mc}(\lambda_m / D_p) \quad (4\text{-}7)$$

$$K_{mc} = 1.64 + 0.552 e^{-(0.656 D_p/\lambda_m)} \quad (4\text{-}8)$$

This represents a correction on Stokes Law and is significant for 3 micron and smaller particles in gases and 0.01 micron and smaller particles in liquids. Table 4-8 gives values of K_m.

When two free settling particles of different dimensions, D'_{p1} and D'_{p2} and different densities, ρ_{p1} and ρ_{p2}, fall through a fluid of density, ρ_f, they will attain equal velocities when:

$$\frac{D'_{p1}}{D'_{p2}} = \left(\frac{\rho_{p2} - \rho_f}{\rho_{p1} - \rho_f}\right)^n \quad (4\text{-}9)$$

where n = 1 in eddy-resistance zone (more turbulent) and n = 0.5 in streamline fall.

Alternate Terminal Velocity Calculation

In contrast to individual particles settling in a very dilute solution/fluid, is the case of sedimentation where particles must settle in more concentrated environment,

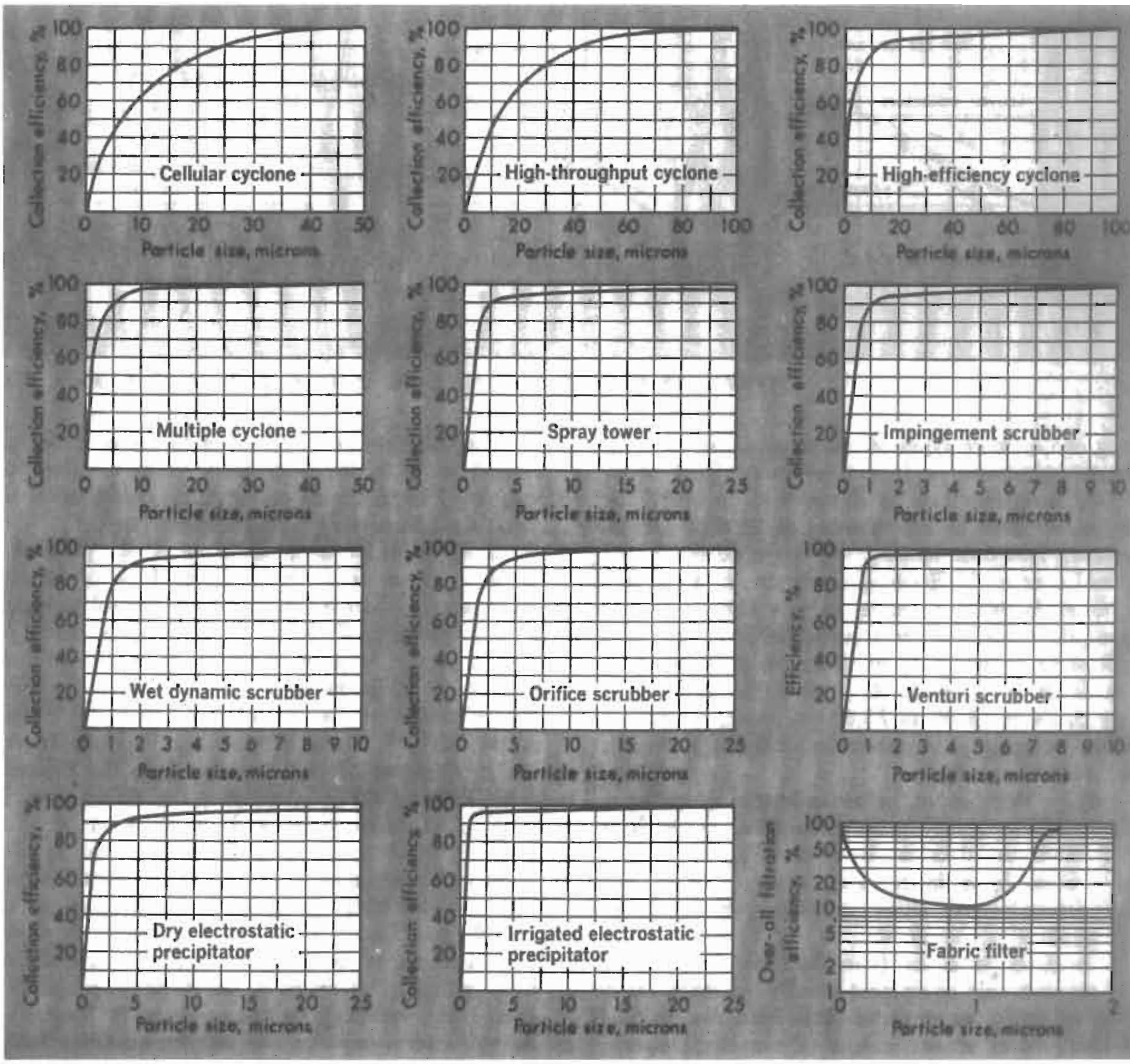


Figure 4-5. Efficiency curves for various types of dust collection equipment as of 1969. Only marginal improvements have been made since then. By permission, Sargent, G. D., *Chemical Engineering,* Jan. 27, 1969, p. 130.

and hence, particles influence adjacent particles. This is often termed *hindered settling* [23,46]. Depending upon the particles concentration, the hindered terminal settling velocity will generally be somewhat lower than for the terminal settling velocity of a single desired particle in the same medium.

From Reference [46]:

K is a dimensionless number that establishes the regime of settling class, reference to the settling laws:

$$K = 34.8\, D'_p \left[\frac{\rho_f(\rho_p - \rho_f)}{\mu^2}\right]^{1/3} \quad (4\text{-}10)$$

If, $K < 3.3$, then Stokes' law applies. If $3.3 \leqq K \leqq 43.6$, the intermediate law applies, and if $K > 43.6$, Newton's law applies. If $K > 2{,}360$, the equations should not be used.

Values of a and b_1

Law	Range	b_1	a
Stokes'	$K < 3.3$	24.0	1.0
Intermediate	$3.3 \leqq K \leqq 43.6$	18.5	0.6
Newton's	$K > 43.6$	0.44	0

(text continued on page 234)

Table 4-5
Applications of Dust Collectors in Industry

Operation	Concen-tration	Particle Sizes	Cyclone	High Eff. Centrif-ugal	Wet Collector	Fabric Arrester	Hi-Volt Electro-static	See Remark No.
			COLLECTOR TYPES USED IN INDUSTRY					
Ceramics								
a. Raw product handling	light	fine	rare	seldom	frequent	frequent	no	1
b. Fettling	light	fine to medium	rare	occasional	frequent	frequent	no	2
c. Refractory sizing	heavy	coarse	seldom	occasional	frequent	frequent	no	3
d. Glaze & vitr. enamel spray	moderate	medium	no	no	usual	occasional	no	
Chemicals								
a. Material handling	light to moderate	fine to medium	occasional	frequent	frequent	frequent	rare	4
b. Crushing grinding	moderate to heavy	fine to coarse	often	frequent	frequent	frequent	no	5
c. Pneumatic conveying	very heavy	fine to coarse	usual	occasional	rare	usual	no	6
d. Roasters, kilns, coolers	heavy	med-coarse	occasional	usual	usual	rare	often	7
Coal Mining and Power Plant								
a. Material handling	moderate	medium	rare	occasional	frequent	frequent	no	8
b. Bunker ventilation	moderate	fine	occasional	frequent	occasional	frequent	no	9
c. Dedusting, air cleaning	heavy	med-coarse	frequent	frequent	occasional	often	no	10
d. Drying	moderate	fine	rare	occasional	frequent	no	no	11
Fly Ash								
a. Coal burning—chain grate	light	fine	no	rare	no	no	no	12
b. Coal burning—stoker fired	moderate	fine to coarse	rare	usual	no	no	rare	
c. Coal burning—pulverized fuel	heavy	fine	rare	frequent	no	no	frequent	13
d. Wood burning	varies	coarse	occasional	occasional	no	no	no	14
Foundry								
a. Shakeout	light to moderate	fine	rare	rare	usual	rare	no	15
b. Sand handling	moderate	fine to medium	rare	rare	usual	rare	no	16
c. Tumbling mills	heavy	med-coarse	no	no	frequent	frequent	no	17
d. Abrasive cleaning	moderate to heavy	fine to medium	no	occasional	frequent	frequent	no	18
Grain Elevator, Flour and Feed Mills								
a. Grain handling	light	medium	usual	occasional	rare	frequent	no	19
b. Grain dryers	light	coarse	no	no	no	no	no	20
c. Flour dust	moderate	medium	usual	often	occasional	frequent	no	21
d. Feed mill	moderate	medium	usual	often	occasional	frequent	no	22
Metal Melting								
a. Steel blast furnace	heavy	varied	frequent	rare	frequent	no	frequent	23
b. Steel open hearth	moderate	fine to coarse	no	no	doubtful	possible	probable	24
c. Steel electric furnace	light	fine	no	no	considerable	frequent	rare	25
d. Ferrous cupola	moderate	varied	rare	rare	frequent	occasional	occasional	26
e. Non-ferrous reverberatory	varied	fine	no	no	rare	?	?	27
f. Non-ferrous crucible	light	fine	no	no	rare	occasional	?	28
Metal Mining and Rock Products								
a. Material handling	moderate	fine to medium	rare	occasional	usual	considerable	?	29
b. Dryers, kilns	moderate	med-coarse	frequent	frequent	frequent	rare	occasional	30
c. Cement rock dryer	moderate	fine to medium	rare	frequent	occasional	no	occasional	31
d. Cement kiln	heavy	fine to medium	rare	frequent	rare	no	considerable	32
e. Cement grinding	moderate	fine	rare	rare	no	frequent	rare	33
f. Cement clinker cooler	moderate	coarse	occasional	occasional	?	?	?	34
Metal Working								
a. Production grinding, scratch brushing, abrasive cut off	light	coarse	frequent	frequent	considerable	considerable	no	35
b. Portable and swing frame	light	medium	rare	frequent	frequent	considerable	no	
c. Buffing	light	varied	frequent	rare	frequent	rare	no	36
d. Tool room	light	fine	frequent	frequent	frequent	frequent	no	37
e. Cast iron machining	moderate	varied	rare	frequent	considerable	considerable	no	38

Table 4-5
Application for Dust Collectors in Industry (cont.)

Operation	Concen-tration	Particle Sizes	Cyclone	High Eff. Centrif-ugal	Wet Collector	Fabric Arrester	Hi-Volt Electro-static	See Remark No.
			COLLECTOR TYPES USED IN INDUSTRY					
Pharmaceutical and Food Products								
a. Mixers, grinders, weighing, blending, bagging, packaging	light	medium	rare	frequent	frequent	frequent	?	39
b. Coating pans	varied	fine to medium	rare	rare	frequent	frequent	no	40
Plastics								
b. Raw material processing	(See comments under Chemicals)							41
a. Plastic finishing	light to moderate	varied	frequent	frequent	frequent	frequent	no	42
Rubber Products								
a. Mixers	moderate	fine	no	no	frequent	usual	no	43
b. Batchout rolls	light	fine	no	no	usual	frequent	no	44
c. Talc dusting and dedusting	moderate	medium	no	no	frequent	usual	no	45
d. Grinding	moderate	coarse	often	often	frequent	often	no	46
Woodworking								
a. Woodworking machiues	moderate	varied	usual	occasional	rare	frequent	no	47
b. Sanding	moderate	fine	frequent	occasional	occasional	frequent	no	48
c. Waste conveying, hogs	heavy	varied	usual	rare	occasional	occasional	no	49

By Permission John M. Kane, *Plant Engineering*, Nov. (1954).

REMARKS REFERRED TO IN TABLE 4-5

1. Dust released from bin filling, conveying, weighing, mixing, pressing, forming. Refractory products, dry pan and screening operations more severe.
2. Operations found in vitreous enameling, wall and floor tile, pottery.
3. Grinding wheel or abrasive cut off operation. Dust abrasive.
4. Operations include conveying, elevating, mixing, screening, weighing, packaging. Category covers so many different materials that recommendation will vary widely.
5. Cyclone and high efficiency centrifugals often act as primary collectors followed by fabric or wet type.
6. Usual set up uses cyclone as product collector followed by fabric arrester for high over-all collection efficiency.
7. Dust concentration determines need for dry centrifugal; plant location, product value determines need for final collectors. High temperatures are usual and corrosive gases not unusual.
8. Conveying, screening, crushing, unloading.
9. Remote from other dust producing points. Separate collector usually.
10. Heavy loading suggests final high efficiency collector for all except very remote locations.
11. Difficult problem but collectors will be used more frequently with air pollution emphasis.
12. Public nuisance from boiler blow-down indicates collectors are needed.
13. Higher efficiency of electrostatic indicated for large installations especially in residential locations. Often used in conjunction with dry centrifugal.
14. Public nuisance from settled wood char indicates collectors are needed.
15. Hot gases and steam usually involved.
16. Steam from hot sand, adhesive clay bond involved.
17. Concentration very heavy at start of cycle.
18. Heaviest load from airless blasting due to higher cleaning speed. Abrasive shattering greater with sand than with grit or shot. Amounts removed greater with sand castings, less with forging scale removal, least when welding scale is removed.
19. Operations such as car unloading, conveying, weighing, storing.
20. Collection equipment expensive but public nuisance complaints becoming more frequent.
21. In addition to grain handling, cleaning rolls, sifters, purifiers, conveyors, as well as storing, packaging operations are involved.
22. In addition to grain handling, bins, hammer mills, mixers, feeders, conveyors, bagging operations need control.
23. Primary dry trap and wet scrubbing usual. Electrostatic is added where maximum cleaning required.
24. Cleaning equipment seldom installed in past. Air pollution emphasis indicates collector use will be more frequent in future.
25. Where visible plume objectionable from air pollution standards, use of fabric arresters with greater frequency seems probable.
26. Most cupolas still have no collectors but air pollution and public nuisance emphasis is creating greater interest in control equipment.
27. Zinc oxide loading heavy during zinc additions. Stack temperatures high.
28. Zinc oxide plume can be troublesome in certain plant locations.
29. Crushing, screening, conveying, storing involved. Wet ores often introduce water vapor in exhaust air stream.
30. Dry centrifugals used as primary collectors, followed by final cleaner.
31. Collection equipment installed primarily to prevent public nuisance.
32. Collectors usually permit salvage of material and also reduce nuisance from settled dust in plant area.
33. Salvage value of collected material high. Same equipment used on raw grinding before calcining.
34. Coarse abrasive particles readily removed in primary collector types.
35. Roof discoloration, deposition on autos can occur with cyclones and less frequently with dry centrifugal. Heavy duty air filters sometimes used as final cleaners.
36. Linty particles and sticky buffing compounds can cause trouble in high efficiency centrifugals and fabric arresters. Fire hazard is also often present.
37. Unit collectors extensively used, especially for isolated machine tools.

(Remarks cont. on next page)

Remarks from Table 4-5 (Cont.)

38. Dust ranges from chips to fine floats including graphitic carbon.
39. Materials involved vary widely. Collector selection may depend on salvage value, toxicity, sanitation yardsticks.
40. Controlled temperature and humidity of supply air to coating pans makes recirculation from coating pans desirable.
41. Manufacture of plastic compounds involve operations allied to many in chemical field and vary with the basic process employed.
42. Operations are similar to woodworking and collector selection involves similar considerations. See Item 13.
43. Concentration is heavy during feed operation. Carbon black and other fine additions make collection and dust free disposal difficult.
44. Often no collection equipment is used where dispersion from exhaust stack is good and stack location favorable.
45. Salvage of collected material often dictates type of high efficiency collector.
46. Fire hazard from some operations must be considered.
47. Bulky material. Storage for collected material is considerable, bridging from splinters and chips can be a problem.
48. Production sanding produces heavy concentration of particles too fine to be effectively caught by cyclones or dry centrifugals.
49. Primary collector invariably indicated with concentration and partial size range involved, wet or fabric collectors when used are employed as final collectors.

Table 4-6

Comparison of Some Important Dust Collector Characteristics*

Type	Higher Efficiency Range on Particles Greater than Mean Size in Microns	Pressure Loss, Inches Water	Water, Gal. per 1,000 CFM	Space	SENSIVITY TO CFM CHANGE		Humid Air Influence	Max. Temp., F, Standard Construction
					Pressure	Efficiency		
Electro-Static	0.25	½	......	Large	Negligible	Yes	Improves Efficiency	500
Fabric:								
Conventional	0.4	3 —6	......	Large	As cfm	Negligible	May make reconditioning difficult	180
Reverse Jet	0.25	3 —8	......	Moderate	As cfm	Negligible		200
Wet:								
Packed Tower	1 —5	1½—3½	5 —10	Large	As cfm	Yes		
Wet Centrifugal	1 —5	2½—6	3 —5	Moderate	As $(cfm)^2$	Yes		
Wet Dynamic	1 —2	Note 1	½ to 1	Small	Note 1	No	None	Unlimited
Orifice Types	1 —5	2½—6	10—40	Small	As cfm or less	Varies with design		
Higher Efficiency:								
Nozzle	0.5—5	2 —4	5 —10	Moderate	As $(cfm)^2$	Slightly to Moderately	None	Note 2
Venturi	0.5—2	12 —20	......	Small				Unlimited
Dry Centrifugal:								
Low Pressure Cycle	20—40	¾—1½	......	Large	As $(cfm)^2$	Yes	May cause condensation and plugging	750
High Eff. Centrif.	10—30	3 —6	......	Moderate	As $(cfm)^2$	Yes		750
Dry Dynamic	10—20	Note 1	......	Small	Note 1	No		750
Louver	15—60	1 —3	......	Small	As $(cfm)^2$	Moderately		750

Note 1: A function of the mechanical efficiency of these combined exhauster and dust collectors.
Note 2: Precooling of high temperature gases will be necessary to prevent rapid evaporation of fine droplets.
* By permission, John M. Kane, "Operation, Application and Effectiveness of Dust Collection Equipment," *Heating and Ventilating*, Aug. 1952, Ref. (10)

(text continued from page 231)

The terminal settling velocity for single spheres can be determined using the contrasts for the flow regime.

$$V_t = \left[\frac{4a_e D_p'^{(1n)} (\rho_p - \rho_f)}{3 b_1 \mu^n \rho_f^{(-n)}}\right]^{1/(2-n)}, \text{ ft/sec} \qquad (4\text{-}11)$$

For hindered particle settling in a "more crowded" environment, using spherical particles of uniform size:

$$V_{ts} = V_t (1 - c)^m, \text{ ft/sec} \qquad (4\text{-}12)$$

Referring the above to other than uniform spherical particles does not create a significant loss in accuracy for industrial applications. For higher concentration, the values of V_{ts} are lower than V_t. In large particles in small ves-

Table 4-7
General Applications of Liquid Particle Separators

Operation	Concentration	Particle Sizes	Gravity	COLLECTOR TYPES Impinge-ment	Cyclone	Scrubbers	Electrical
Pipeline entrained liquid	light	fine to coarse	No	Frequent	Yes	Occasional	Few
Compressor discharge liquid	light	fine	No	Frequent	Occasional	Occasional	Rare
Compressor oil haze	very light	very fine	No	Frequent	Frequent	Frequent	Occasional
Flashing liquid	light to mod.	fine to medium	No	Frequent	Frequent	Occasional	Rare
Boiling or bubbling	light to heavy	fine to coarse	Occasional	Frequent	Frequent	Occasional	Rare
Spraying	light to heavy	fine to coarse	No	Frequent	Frequent	Rare	Rare
Corrosive liquid particles	light to heavy	fine to coarse	Occasional	Frequent	Occasional	Frequent	Rare
Liquid plus solid particles	light to heavy	medium	Occasional	Occasional	Frequent	Frequent	Occasional

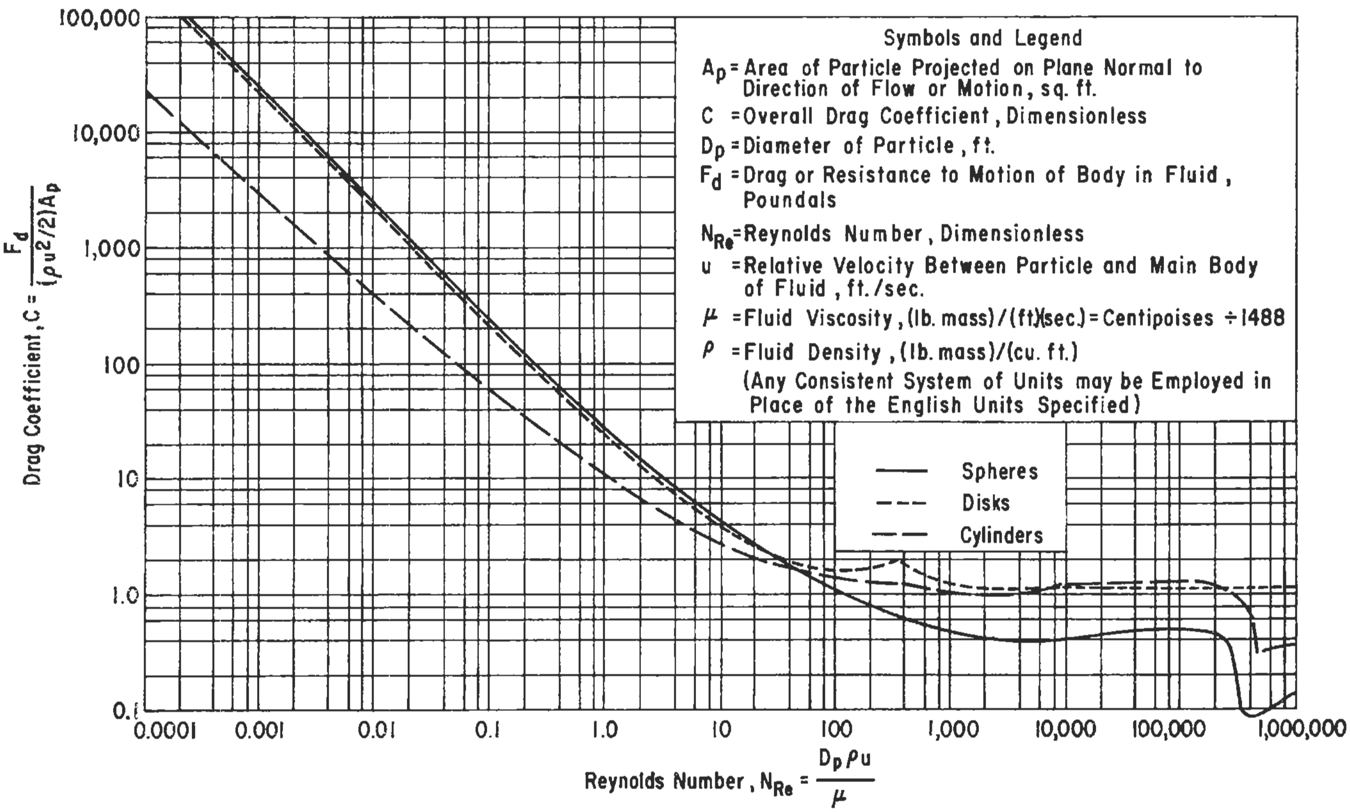


Figure 4-6. Drag coefficients for spheres, disks, and cylinders in any fluid. By permission, Perry, J. H., *Chemical Engineers Handbook*, 3rd Ed., McGraw-Hill Company, 1950.

sels, the wall effect can become significant (see Reference [23]).

For a single particle, D_p can be taken as 2 (hydraulic radius), and the Sauter mean diameter for hindered particles.

Where D'_p = diameter of particle, in. or mm

a_c = acceleration due to gravity, 32.2 ft/s^2 or 9.8 m/s^2

ρ_p = density of particles, lb/ft^3 or kg/m^3

ρ_f = density of fluid, lb/ft^3 or kg/m^3

μ = viscosity of fluid, cp

b_1 = constant given above

n = constant given in text.

Table 4-8

Values of K_m for Air at Atmospheric Pressure[12]

Particle Diameter, Microns	70° F.	212° F.	500° F.
0.1	2.8	3.61	5.14
0.25	1.682	1.952	2.528
0.5	1.325	1.446	1.711
1.0	1.160	1.217	1.338
2.5	1.064	1.087	1.133
5.0	1.032	1.043	1.067
10.0	1.016	1.022	1.033

m = exponent given by equations in Reynolds number table below

V_t = settling velocity for single spherical particle, ft/s and m/s (terminal)

V_{ts} = settling velocity for hindered uniform spherical particle, ft/s or m/s (terminal)

c = volume fraction solids

K = constant given by equation above

N_{Re} = Reynolds number, $D_p V_t \rho_f/\mu$

Values of m	N_{Re}
4.65	< 0.5
$4.375(N_{Re})^{-0.0875}$	$0.5 \leqq N_{Re} \leqq 1{,}300$
2.33	$N_{Re} > 1{,}300$

$$N_{Re} = D_p V_t \rho_f/\mu, \text{ dimensionless} \quad (4\text{-}13)$$

Example 4-2: Hindered Settling Velocities

Using the example of Carpenter [46]:

ρ_f = fluid density = 0.08 lb/cu ft
μ = viscosity = 0.02 cp
ρ_p = 500 lb/cu ft
D'_p = particle diameter, in. = 0.01
c = volume fraction solids, 0.1.

Solving equation for K, for unhindered particle:

$$K = 34.81\ (0.01)\left[\frac{0.08\ (500 - 0.08)}{(0.02)^2}\right]^{1/3}$$

$$K = 16.28$$

Then, for K = 16.28 (intermediate range), b = 18.5; n = 0.6.

Solve for settling velocity, V_t:

$$V_t = \left[\frac{4(32.2)\ (0.01)^{(1 + 0.6)}\ (500 - 0.08)}{3(18.5)\ (0.02)^{0.6}\ (0.08)^{(1 - 0.06)}}\right]^{1/(2-0.6)}$$

$$V_t = 9.77 \text{ ft/sec}$$

Reynolds number, $N_{Re} = D_p V_t \rho_s/\mu$

$$= \left(\left[\frac{0.01}{(12)}\right]\frac{(9.77)\ (0.08)}{(0.02)\ (6.72 \times 10^{-4})}\right)$$

$\mu = (\text{cp})(6.72 \times 10^{-4})$, lb/ft sec

$N_{Re} = 48.46$

Then, $m = 4.375(N_{Re})^{-0.0875} = 4.375(48.46)^{-0.0865} = 3.1179$

For 0.1 volume fraction solids for hindered settling velocity:

$$V_{ts} = V_t\ (1 - c)^m$$
$$= 9.77(1 - 0.1)^{3.1179}$$
$$= 7.03 \text{ ft/sec}$$

(e) Particles under 0.1 micron:

Brownian movement becomes appreciable for particles under 3 microns and predominates when the particle size reaches 0.1 micron [13]. This motion usually has little effect in the average industrial process settling system except for the very fine fogs and dusts. However, this does not mean that problems are not present in special applications.

Figure 4-1 gives the limiting or critical diameter above which the particular settling law is not applicable. Figure 4-7 gives terminal velocities for solid particles falling in standard air (70°F and 14.7 psia), and Figure 4-8 gives particles falling through water. If a particle (liquid or solid) is falling under the influence of gravity through a vapor stream, the particle will continue to fall until, or unless the vapor flow rate is increased up to or beyond the terminal velocity value of the particle. If the vapor velocity exceeds this, then the particle will be carried along with the vapor (entrained).

Pressure Drop

Pressure drop through gravity settlers is usually extremely low due to the very nature of the method of handling the problem.

Figure 4-9 is convenient for quick checks of terminal settling velocities of solid particles in air and in water [23].

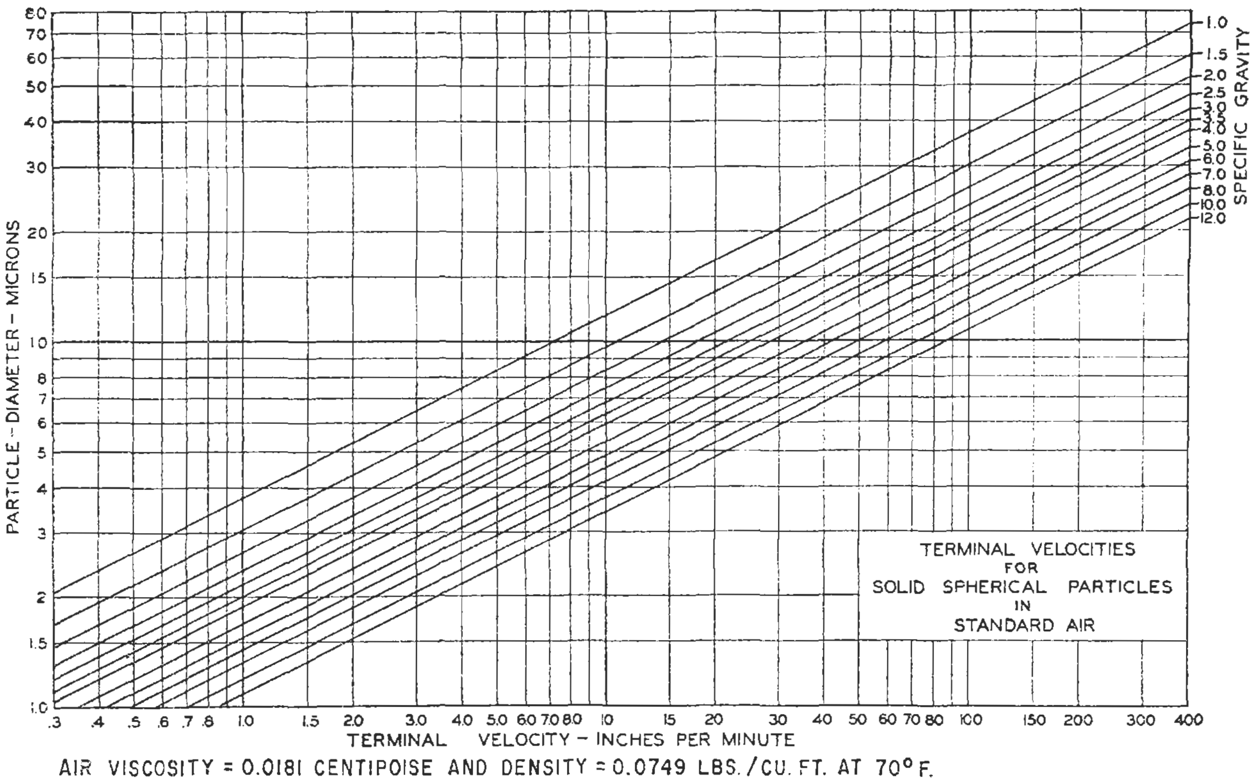


Figure 4-7. Terminal velocities for solid spherical particles in standard air. Courtesy of American Blower Div. American Radiator and Standard Sanitary Corporation.

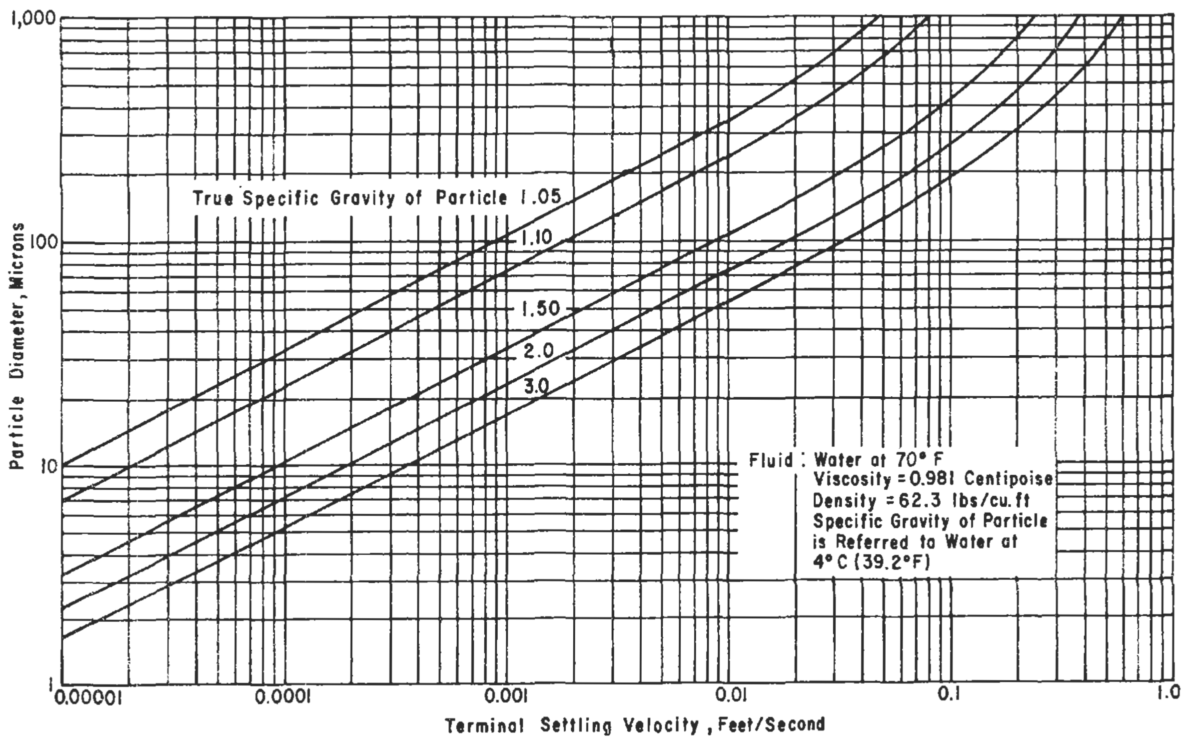


Figure 4-8. Terminal settling velocity of particles in water. By permission, Lapple, C. E., *Fluid and Particle Mechanics,* 1st Ed., University of Delaware, Newark, 1954.

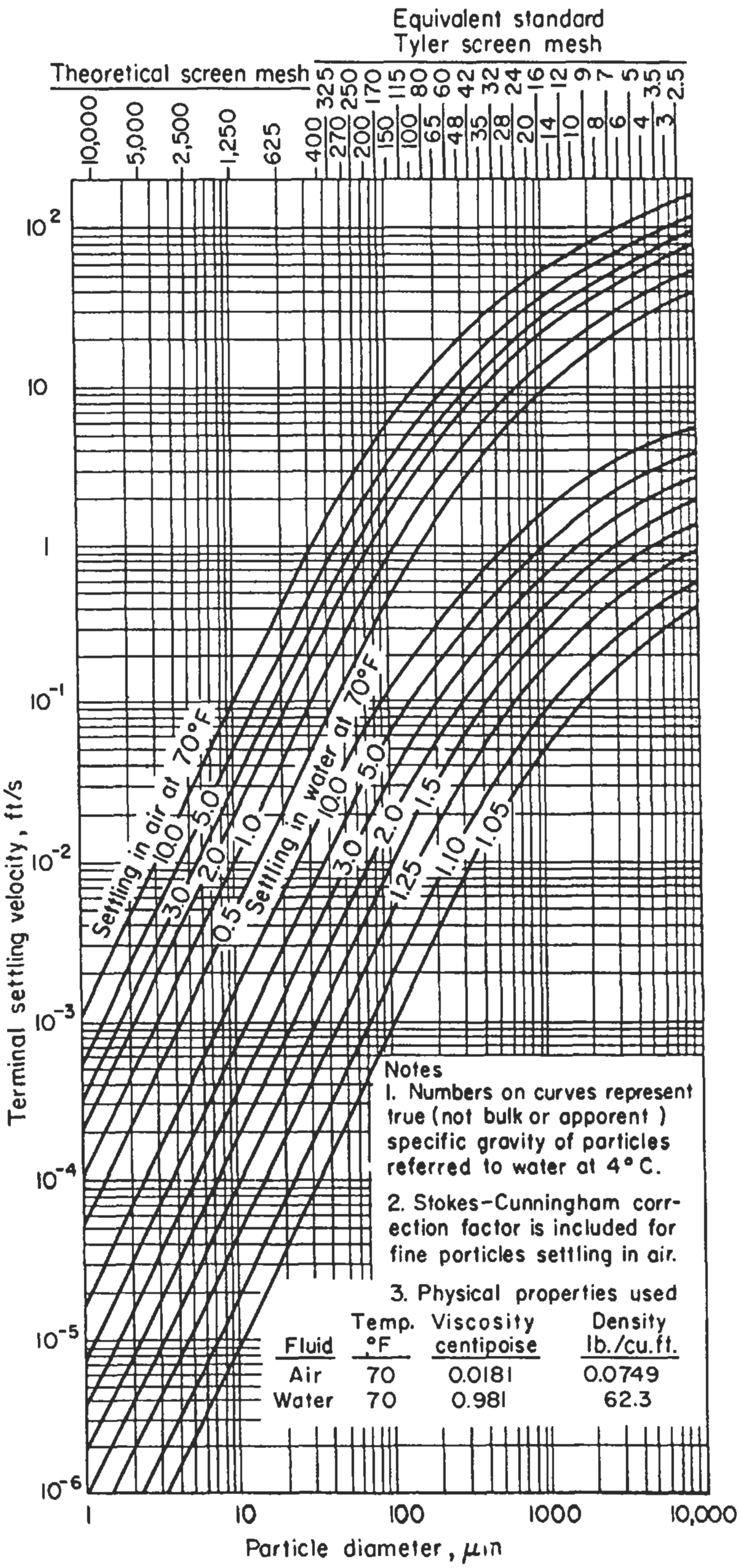


Figure 4-9. Terminal velocities of spherical particles of different densities settling in air and water at 70°F under the action of gravity. To convert feet per second to meters per second, multiply by 0.3048. (From Lapple, *et. al., Fluid and Particle Mechanics,* University of Delaware, 1951, p. 292. By permission, Perry, J. H. *Chemical Engineers Handbook,* 6th Ed., McGraw-Hill Book Co., Inc. 1984, p. 5–67.

API-Oil Field Separators

The American Petroleum Institute *Manual on Disposal of Refinery Wastes* provides specific design and construction standards for API-separators that are often used in oilfield waste disposal [24].

Liquid/Liquid, Liquid/Solid Gravity Separations, Decanters, and Sedimentation Equipment

Lamella Plate Clarifier, [see Figure 4-10]

This angle plate gravity separator removes suspensions of solids from a dilute liquid. The unit is more compact than a box-type settler due to the increased capacity achieved by the multiple parallel plates. The concept is fairly standard (U.S. Patent 1,458.805—year 1923) but there are variations in some details. For effective operation, the unit must receive the mixture with definite particles having a settling velocity. The units are not totally effective for flocculants or coagulated masses that may have a tendency to be buoyant.

Flow through the plates must be laminar, and the critical internal areas are [25,54,61]:

- Distribution of the inlet flow
- Flow between each parallel plate surface
- Collection of clarified water

Because these units are essentially open, the "standard" design does not fit into a closed process system without adding some enclosures, and certainly is not suitable for a pressurized condition.

Although the main suspended flow is through a top mixing chamber, the mixture flows around the angled (avg. 55°) parallel plate enclosure and begins its settling path from the bottom of the plates, flowing upward while depositing solids that slide countercurrent into the bottom outlet. The purified liquid flows overhead and out the top collecting trough.

Manufacturers should be contacted for size/rating information, because the efficiency of a design requires proper physical property information as well as system capacity and corrosion characteristics.

Thickeners and Settlers

Generally, large volume units for dewatering, settling of suspensions, and thickening of solids, and concentration of solids and clarification must be designed by the specific manufacturer for the process conditions and physical properties. Some typical processes involving this class of equipment are lime slurrying, ore slurrying, ore

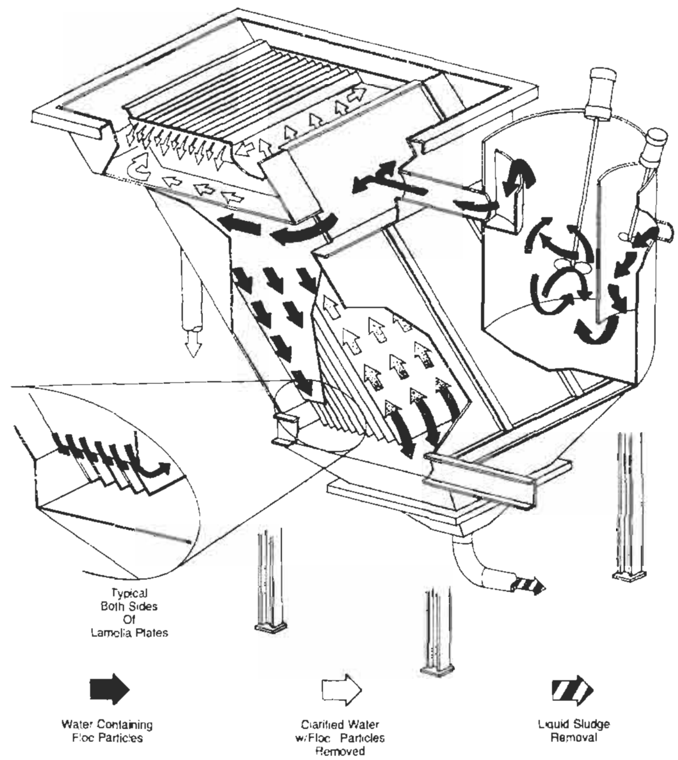


Figure 4-10. Lamella Clarifier. By permission, Graver Water Div. of the Graver Co.

clarifying, wastewater clarification and thickening. A good summary discussion is given in Reference [27].

Horizontal Gravity Settlers or Decanters, Liquid/Liquid

Many processes require the separation of immiscible liquid/liquid streams; that is, water/hydrocarbon. The settling unit must be of sufficient height (diameter) and length to prevent entrainment of the aqueous phase into the hydrocarbon and vice versa. Horizontal units are usually best for settling and possibly vented units for decantation (but not always).

Residence time of the mixture in the vessel is a function of the separation or settling rate of the heavier phase droplets through the lighter phase. Most systems work satisfactorily with a 30 minute to 1 hour residence time, but this can be calculated [26]. After calculation, give a reasonable margin of extra capacity to allow for variations in process feedrate and in the mixture phase composition.

From Stokes' Law, the terminal settling velocity:

$$V_t = gD_p^2 (\rho_p - \rho)/18\mu, \text{ ft sec} \quad (4\text{-}14)$$

μ = viscosity of surrounding fluid, lb/ft sec
D_p = diameter of particle, ft

for assumed spherical particles in a surfactant-free system [26]. The minimum particle diameter for many fine dispersions is 100 microns; however, Reference [28] has reviewed a wide variety of liquid drop data and suggests that a good choice is 150 micron or 0.15 cm or 0.0005 ft. This is also the particle size used in the API Design Manual [24]. Using too large an assumed particle diameter will cause the settler unit to become unreasonably small.

Assuming a horizontal unit, as illustrated in Figure 4-11, has a segment of a circle equal to 25% to 75% [27,26] of the circular area (the highest of this segment will be about 30% to 70% of the diameter), then height of the interface will be [26]:

$$H/D = 0.8A/(\pi D^2/4) + 0.1, \text{ ft} \quad (4\text{-}15)$$

$$\text{or, } h_c = 38.4A/(\pi D) + 1.2D, \text{ in} \quad (4\text{-}15A)$$

where H = height of segment of a circle, ft
h_c = height of segment of a circle, in.
D = diameter of vessel, ft
A = area of segment of circle, sq ft

The average volumetric residence time in the settler is:

$$t_{avg} = 7.48\ (V_{set}/F), \text{ min} \quad (4\text{-}16)$$

V_{set} = active volume of settler occupied by one of the phases, cu ft
F = flow rate of one phase GPM
t_{avg} = average residence time based on liquid flow rate and vessel volume, min

The minimum residence time as determined by Stokes' Law terminal settling velocity is:

$$t_{min} = h/v_t, \text{ min} \quad (4\text{-}17)$$

h_c = height of segment of circle, in.

v_t = terminal settling velocity, in./min

Average residence time related to minimum residence time is:

$$t_{ave} = (f)(t_{min}) \quad (4\text{-}18)$$

f = factor relating average velocity to maximum velocity

This relationship is related to the viscosities of the hydrocarbon and aqueous phases at the interface. Based on data from different systems:

f = 2.0 (use for design)

The active volume occupied by either phase is:

$$V = AL, \text{ cu ft} \quad (4\text{-}19)$$

L = length of vessel, ft, inlet to outlet

$$h_c = 7.48\ ALv_t/(fF) \quad (4\text{-}20)$$

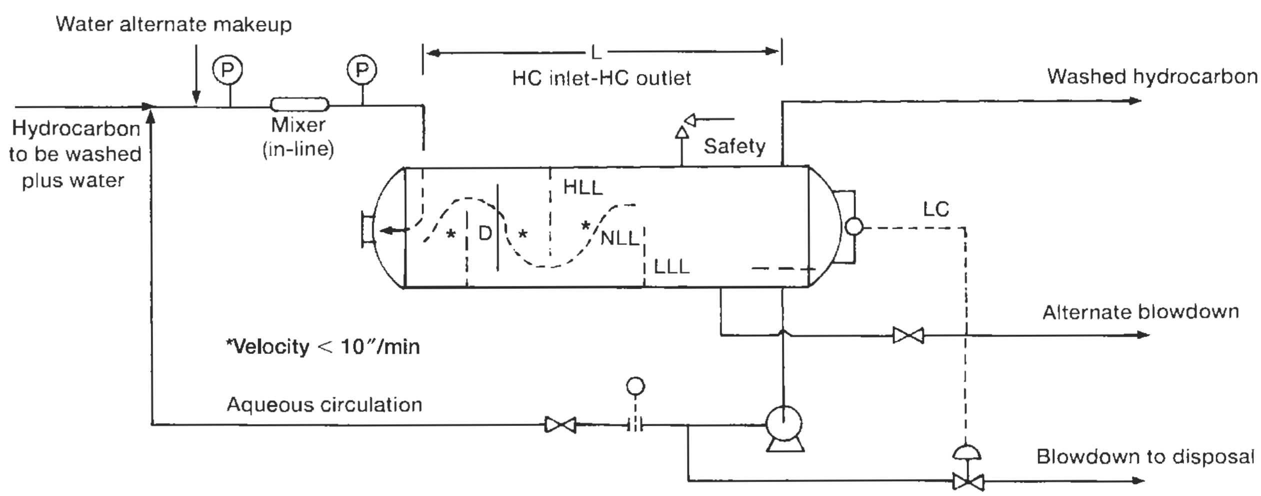


Figure 4-11. Settler vessel; runs full. Adapted by permission, Abernathy, M. W., Vol. 25, *Encyclopedia of Chemical Processing and Design*, J. J. McKetta, Ed., Marcel Dekker, 1987, p. 77 and *Hydrocarbon Processing*, Sept. 1977, p. 199 [25] and private communication.

For an aqueous-hydrocarbon or organic solvent mixture:

The top layer will be hydrocarbon, with the aqueous layer droplets settling through the hydrocarbon. The terminal velocity is:

$$v_{hc} = 12.86\ (\Delta SpGr)/\mu_{hc},\ \text{in./min} \quad (4\text{-}21)$$

v_{hc} = terminal settling velocity of aqueous droplets in hydrocarbon phase in top of vessel, in./min
$\Delta SpGr$ = differences in specific gravity of the particle and surrounding fluid
μ_{hc} = viscosity of surrounding fluid, cp

Height of hydrocarbon layer to the interface:

$$h_t = (7.48)A_tLv_{ha}/f_{hc}F_{hc} \quad (4\text{-}21A)$$

$$h_t = 38.4\ A_t/(\pi D) + 1.2D \quad (4\text{-}15A)$$

ht = height of continuous hydrocarbon phase in the top of vessel, in.

Then, $A_t = 1.2D[(7.48)Lv_{hc}/(f_{hc}F_{hc}) - 38.4/(\pi D)]^{-1}$ (4-22)

A_t = cross-sectional area at top of vessel occupied by the continuous hydrocarbon phase, sq ft
A_b = cross-sectional area at bottom of vessel occupied by continuous aqueous phase, sq ft

For the bottom aqueous phase:

hydrocarbon droplets settle out of the continuous aqueous phase. The terminal velocity is for hydrocarbon droplets:

$$v_{aq} = 12.86\ (\Delta SpGr)/\mu_{aq},\ \text{in./min} \quad (4\text{-}23)$$

v_{aq} = terminal settling velocity of hydrocarbon droplets in aqueous phase in bottom of vessel, in./min
μ_{aq} = viscosity of aqueous phase, cp

Height of aqueous layer to the interface:

$$h_b = (7.48)(A_bLv_{aq})/(f_{aq}F_{aq}) \quad (4\text{-}21A)$$

$$h_b = 38.4\ A_b/\pi D + 1.2D \quad (4\text{-}15A)$$

h_b = height of continuous aqueous phase in bottom of vessel, in.
A_b = cross-sectional area at bottom of vessel occupied by continuous aqueous phase, sq ft

$$A_b = 1.2D[(7.48)Lv_{aq}(f_{ag}F_{ag}) - 38.4/(\pi D)]^{-1} \quad (4\text{-}22)$$

Optimum vessel diameter:

Assume 20% cross-sectional area is occupied by an emulsion and is recognized as a "dead volume." This is actually the height over which the interface level will vary during normal operations [26].

$$A_t + A_b = 0.8\ \pi D^2/4 \quad (4\text{-}24)$$

$$D = \pm[a/2 \pm (a^2 - 4b)^{1/2}/2]^{1/2},\ \text{ft} \quad (4\text{-}25)$$

where $a = (1.889)(v_{hc}f_{aq}F_{aq} + v_{aq}f_{hc}F_{hc})/(rv_{hc}v_{aq})$ (4-26)

$$b = (3.505)(f_{hc}F_{hc}f_{aq}F_{aq})/(r^2v_{hc}v_{aq}) \quad (4\text{-}27)$$

The economical vessel ratio is L/D = r

Modified Method of Happel and Jordan [29]

This method is a modification of the earlier method [30] by Reference [26], as follows, and can be less conservative [26] than the original method [30]. A basic assumption is that particles must rise/fall through one-half of the drum vertical cross-sectional area [26].

$$t = h/v$$

$$t = (1/2)(7.48)[0.8\ \pi D^2L/4]F_t \quad (4\text{-}28)$$

F_t = flow rate of both phases
v_t = v = terminal settling velocity, in./min

This assumes 20% of the cross-sectional even as "dead volume." The height from the interface can be determined by combining the above equations:

$$h = (0.748)\pi D^2Lv/F_t \quad (4\text{-}29)$$

The height for each interface is:

$$h_t = (0.748)\pi D^2Lh_{hc}/F_t \quad (4\text{-}30)$$

$$h_b = (0.748)\ \pi D^2Lv_{aq}/F_t \quad (4\text{-}31)$$

$$A_t = [(0.748)\ \pi D^2Lv_{hc}/F_t - 1.2D]\ \pi D/38.4 \quad (4\text{-}32)$$

$$A_b = [(0.748)\ \pi D^2Lv_{aq}/F_t - 1.2D]\ \pi D/38.4 \quad (4\text{-}33)$$

Example 4-3: Horizontal Gravity Settlers

Using the data from Sigales [31] and following the design of [26]:

Data for propane/caustic wash:

F_{hc} = 95 GPM
F_{aq} = 39 GPM
v_{aq} = 5 in./min
v_{hc} = 120 in./min
r = 3.4

The terminal (highest calculated) settling velocity of the aqueous droplet in/through the hydrocarbon phase is:

v_{hc} = (1.2)(5 in./min)(95/39 GPM) = 14.6 in./min

Because this is more than the 10 in./min recommended earlier, then use:

v_{hc} = 10 in./min

Assume for design: $f_{hc} = f_{ag} = 2$ (from earlier discussion).

Then, a = (1.889[(10)(2)(39) + (5)(2)(95)]/[(3.4)(10)(5)]

a = 19.22

b = (3.505)(2)(95)(2)(39)/[(3.4)^2(10)(5)]

b = 89.87

Solving for D:

$$D = [19.22/2 \pm [(19.22)^2 - 4(89.87)]^{1/2}/2]^{1/2}$$

D = 3.34 ft or −2.83 ft (latter is an unreal negative number, so use 3.34 ft)

Area of segment at top of vessel = A_t, substituting into Equation 4-22:

$$A_t = 1.2\,D\,[(7.48)(3.4)D(10)]/[(2)(95)] - 38.4/(\pi D)]^{-1}$$

Using: L/D = 3.4:

For the bottom segment of the vessel, aqueous layer:

$$A_b = 1.2(3.34)\ [(7.48)(3.34)(3.4)(5)]/[(2)(39)] - (38)/\pi(3.34)]^{-1}$$

A_b = 2.2448 sq ft

Then, using Equation 4-21A:

h_t = 7.48(4.942)(3.4)(10)/(2.0)(95) = 22.1 in.

h_b = 7.48(2.2448)[(3.34)(3.4)](5)/(2)(39)] = 12.2 in.

Then, h_t/D = (22.1)/(12)(3.34) × 100 = 55%

h_b/D = 12.2/(12)(3.34) × 100 = 30%

Since h_t and h_b are between 30% and 70% of the diameter, the solution is acceptable.

In summary:

Design Calculation	Practical Design Use
Diameter 3.34 ft (40.08 in.)	3.5 ft. (42 in.) or 3.83 ft (46 in.)
Length HC inlet/outlet: 11 ft	12 or 14 ft

Abernathy [26] has compared several design methods as follows:

	Sigales	This Method	Modified Happel	Happel	Rule-of-Thumb
Diameter	2.67 ft	3.34 ft	3.36 ft	4.01 ft	4.1 ft
h_t	10 in.	22 in.	22.6 in.	24 in.	32.5 in.
h_b	8 in.	12 in.	11.3 in.	24 in.	16.7 in.
Interface	14 in.	6 in.	6.4 in.	0 in.	0 in.
HC residence time	1.1 min	4.4 min	4.6 min.	6.8 min.	10 min.

Decanter [32]

In most general applications, a decanter is a continuous gravity separation vessel that does not run full, as contrasted to a settler that usually runs full, with one stream exiting at or near the top of a horizontal vessel. For most decanters, one phase of a two-plane mixture overflows out of the vessel (see Figure 4-12). The concept of the decanter involves the balancing of liquid heights due to differences in density of the two phases, as well as settling velocity of the heavier phase falling through the lighter, or the lighter rising through the heavier.

Settling Velocity: Terminal [32]

$$v_d = gd^2\frac{(\rho_d - \rho_c)}{18\mu_c},\ \text{ft/sec} \qquad (4\text{-}34)$$

where v_d = terminal settling velocity of a droplet, ft/sec
g = acceleration due to gravity, 32.17 ft/sec-sec
d = droplet diameter, ft(1 ft = 304, 800μm, or 1μm = 0.001mm)
ρ_d = density of fluid in the droplet, lb/cu ft
ρ_c = density of fluid continuous phase, lb/cu ft
μ_c = viscosity of the continuous phase, lb/(ft)(sec)
Note: 1 cp = 6.72 × 10^{-4} lb/(ft)(sec)
μm = millimicron

For a decanter that operates under gravity flow with no instrumentation flow control, the height of the heavy phase liquid leg above the interface is balanced against the height of one light phase above the interface [23]. Figures 4-12 and 4-13 illustrate the density relationships and the key mechanical details of one style of decanter.

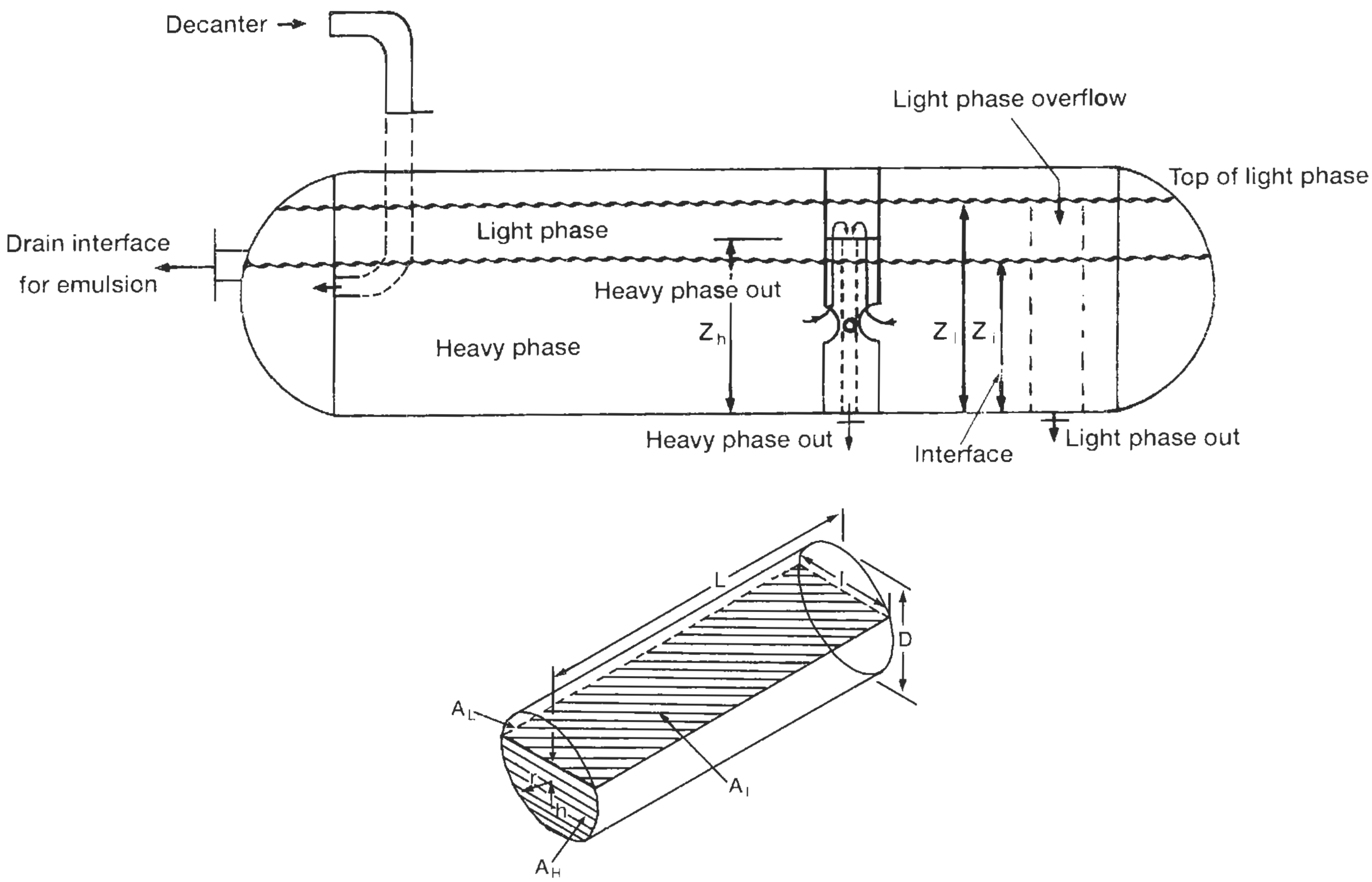


Figure 4-12. Gravity decanter basic dimensions. Adapted by permission, Schweitzer, P.A., McGraw-Hill Book Co. (1979) [32].

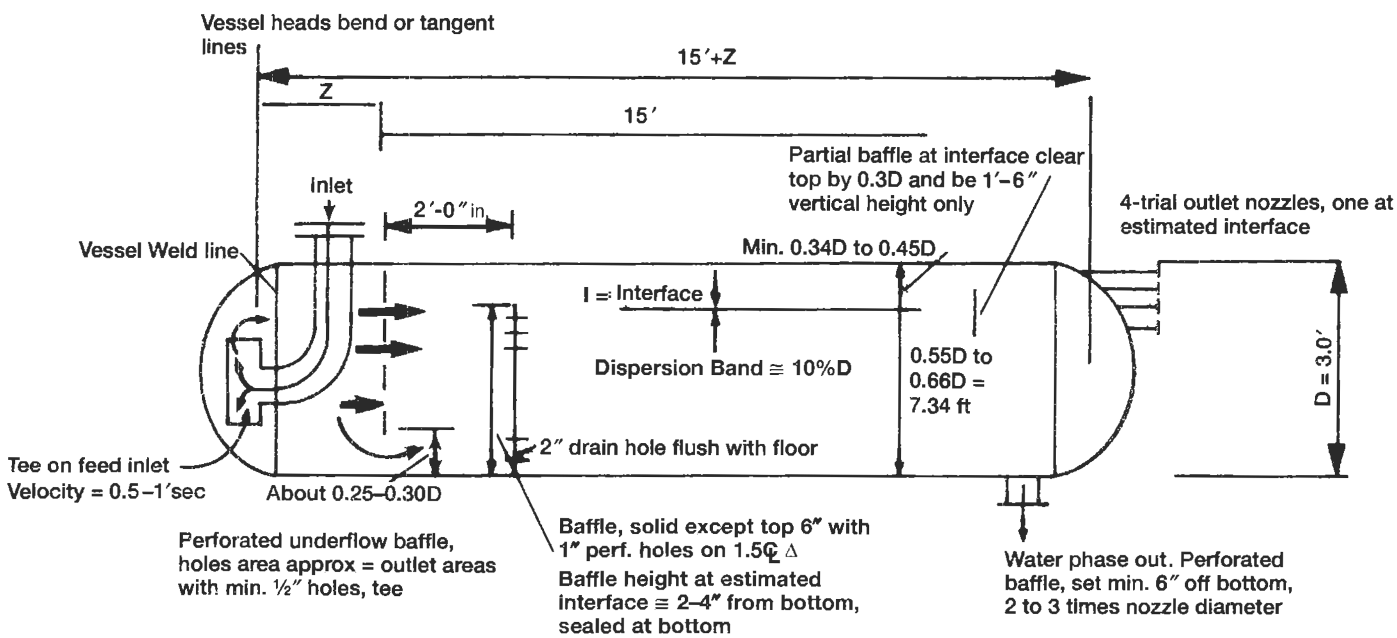


Figure 4-13. Decanter for Example 4-1.

The same results can be achieved with internal flat plate baffles and outlet nozzles.

$$(z_h - z_i)\rho_h = (z_l - z_i)\rho_L \tag{4-35}$$

z_h = heavy phase outlet dimension from bottom of horizontal decanter
z_i = interface measured from bottom
z_l = light phase outlet measured from bottom of decanter

Droplet diameter, when other data is not available:

= 150μm (d = 0.0005 ft)

Reference [32] recognizes that this is generally on the safe side, because droplets generated by agitation range 500 to 5000 μm, turbulent droplet range 200 to 10,000 μm. Due to limitations of design methods, decanters sized for droplets larger than 300 μm often result in being too small to work properly [32].

The continuous phase moves through the vessel on a uniform flow equal to the overflow rate. To identify which is the continuous phase (from [65] by [32]):

$$\theta = \frac{Q_L}{Q_H}\left(\frac{\rho_L \mu_H}{\rho_H \mu_L}\right)^{0.3} \tag{4-36}$$

θ	*Result*
< 0.3	light phase always dispersed
0.3–0.5	light phase probably dispersed
0.5–2.0	phase inversion probable, design for worst case
2.0–3.3	heavy phase probably dispersed
> 3.3	heavy phase always dispersed

where Q_d = dispersed volumetric flow rate, cu ft/sec
Q_L = volumetric flow rate, cu ft/sec, light phase
Q_H = volumetric flow rate, cu ft/sec, heavy phase
ρ_L = density of light phase fluid, lb/cu ft
ρ_H = density of heavy phase fluid, lb/cu ft
μ_H = viscosity of heavy phase, lb/(ft)(sec)
μ_L = viscosity of light phase, lb/(ft)(sec)

To begin, there is a dispersion band through which the phases must separate. Good practice [32] normally keeps the vertical height of the dispersed phase, H_D < 10% of decanter height (normally a horizontal vessel), and:

$1/2 H_D A_I / Q_D > 2$ to 5 min

where A_I = area of interface assuming flat interface, sq ft
A_L = cross-sectional area allotted to light phase, sq ft
A_H = cross-sectional area allotted to heavy phase, sq ft
H_D = height of the dispersion band, ft
Q_D = volumetric flow, dispersed phase, cu ft/sec

h = distance from center to given chord of a vessel, ft
I = width of interface, ft
D = decanter diameter, ft
L = decanter length, ft
r = vessel radius, ft

Horizontal vessels as cylinders are generally more suitable for diameters up to about 8 feet than other shapes, or vertical, due in part to the increased interfacial area for interface formation. For a horizontal drum (See Figure 4-12):

$$I = 2(r^2 - h^2)^{1/2} \tag{4-37}$$

$$A_I = IL \tag{4-38}$$

$$A_L = 1/2\,\pi r^2 - h(r^2 - h^2)^{1/2} - r^2 \arcsin(h/r) \tag{4-39}$$

or use the methods from the Appendix to calculate area of a sector of a circle. The arc is in radians:
Radians = (degrees) (π/180)

$$A_H = \pi r^2 - A_L \tag{4-40}$$

$$D_L = 4\,A_L/(I + P) \tag{4-41}$$

$$D_H = 4\,A_H/(I + 2\,\pi r - P) \tag{4-42}$$

where $P = 2r \arccos(h/r)$

Degree of turbulence [32]:

$$N_{Re} = \frac{v_c D_H \rho_c}{\mu_c} \tag{4-43}$$

c = continuous phase
D_H = hydraulic diameter, ft = 4 (flow area for the phase in question/wetted perimeter of the flow channel)
v_c = velocity down the flow channel

Guidelines for successful decanters [32]:

R_e	Results
< 5000	little problem
5000–20,000	some hindrance
20,000–50,000	major problem may exist
Above 50,000	expect poor separation

Velocities of both phases should be about the same through the unit. By adjusting mechanical internals, a ratio of < 2:1 is suggested (internals do not need to be equal) [32]. Velocities for entrance and exit at the vessel nozzle should be low, in the range of 0.5 to 1.5 ft/sec. The feed must not "jet" into the vessel, and should be baffled to prevent impingement in the main liquid body, keeping turbulence to an absolute minimum to none. Baffles can/should be placed in the front half of the unit to provide *slow* flow of the fluids either across the unit or up/down paths followed by the larger stilling chamber, before fluid exits. (See Figure 4-13)

Example 4-4: Decanter, using the method of Reference [32]

A plant process needs a decanter to separate oil from water. The conditions are:

Oil flow = 8500 lb/hr
ρ = 56 lb/cu ft
μ = 9.5 centipoise
Water flow = 42,000 lb/hr
ρ = 62.3 lb/cu ft
μ = 0.71 centipoise

Units conversion:

Q_{oil} = (8500)(56)(3600) = 0.0421 cu ft/sec
μ_{oil} = (9.5)(6.72 × 10^{-4}) = 63.8 × 10^{-4} lb/ft-sec
Q_{water} = 42,000/(62.3)(3600) = 0.187 cu ft/sec
μ_w = (0.71)(6.72 × 10^{-4}) = 4.77 × 10^{-4} lb/ft-sec

Checking dispersed phase, Equation 4-36:

$$\theta = \frac{0.0421}{0.187}\left[\frac{(56)\ (4.77 \times 10^{-4})}{(62.3)\ (63.8 \times 10^{-4})}\right]^{0.3}$$
$$= 0.010009$$

Therefore, light phase is always dispersed since θ is less than 0.3.

Settling rate for droplets of oil through water:

Assume droplet size is d = 0.0005 ft (150 μm), as earlier discussed.

$$V_{oil} = (32.17)(0.0005)^2(56 - 62.3)/[(18)(4.77 \times 10^{-4})]$$
$$= -0.005 \text{ ft/sec}$$

The (−) sign means the oil rises instead of settles.

Overflow rate:

Assume I (Figure 4-12) is 80% diameter, D, of vessel and that L/D = 5.

Then, $Q_c/A_I < v_d$
$A_I = IL = (0.8D)(5D) = 4D^2$, then,
$Q_c/4D^2 < v_d$

and:

$$D \geqq 1/2(Q_c/v_d)^{1/2} \geqq 1/2(0.187/0.005)^{1/2}$$
$$D = 3.057 \text{ ft}$$

Length, L = 5D = 5(3.0) = 15 ft

Interface Level: Assume: Hold interface one foot below top of vessel to prevent interface from reaching the top oil outlet.

Then, h = 0.5 ft
r = 3.0/2 = 1.5 ft
$I = 2(r^2 - h^2)^{1/2} = 2[(1.5)^2 - (0.5)^2]^{1/2} = 2.828$ ft
$A_{oil} = (1/2)(\pi)(1.5)^2 - 0.5[(1.5)^2 - (0.5)^2]^{1/2} - (1.5)^2$ arc sin (0.5/1.5)
= 3.534 − 0.707 − 0.765
= 2.062 sq ft

Note: In radians: Arc sin (0.5/1.5) = (19.47/180)π = 0.3398

$A_{water} = \pi(1.5)^2 - A_{oil} = \pi(2.25) - 2.06 = 5.01$ sq ft
P = 2(1.5)[arc cos (0.5/1.5)] = 3.69 sq ft
Area interface, A_I = (2.828)(15) = 42.42 sq ft

Secondary settling: Continuous phase water droplets to resist the oil overflow rate if it gets on wrong side of interface.

$v_{water} \leqq Q_{oil}/A_I = 0.0421/42.42 = 0.0009924$ ft/sec

Then, from settling-velocity equation:

$d = [(18)(6.38 \times 10^{-3})(0.0009924)/(32.17)(62.3 - 56)]^{1/2}$
d = 0.0007498 ft, (216 μm)

Checking coalescence time:

Assume H_D = height of dispersion band = 10% of D = 0.3 ft

Time available to cross the dispersed band

= $1/2(H_D A_I/Q_D)$ should be > 2 to 5 min
= 1/2[(0.3)(42.42)/(0.0421)]
= 150 sec, which is 2.5 min

Should be acceptable, but is somewhat low.

Then D_{oil} = 4(2.062)/(2.828 + 3.69) = 1.265 ft
V_{oil} = 0.0421/(2.062) = 0.0204 ft/sec

$$N_{Re\ oil} = \frac{(0.0204)\ (1.265)\ (56)}{6.38 \times 10^{-3}} = 226.5$$

$$D_{water} = 4(5.01)/[2.828 + 2\pi\,(1.5) - 3.69] = 2.34 \text{ ft}$$

$$v_{water} = 0.187/5.01 = 0.0373 \text{ ft/sec}$$

$$N_{Re\,(water)} = \frac{(0.0373)\ (2.34)\ (62.3)}{4.77 \times 10^{-4}} = 11,399$$

d = droplet diameter, ft

The degree of turbulence would be classified as acceptable, but the unit must not be increased in capacity for fear of creating more water phase turbulence.

B. Impingement Separators

As the descriptive name suggests, the impingement separator allows the particles to be removed to strike some type of surface. This action is better accomplished in pressure systems where pressure drop can be taken as a result of the turbulence which necessarily accompanies the removal action.

Particle removal in streamline flow is less efficient than for turbulent flow, and may not be effective if the path of travel is not well baffled.

The "target" efficiency for impingement units expresses the fraction of the particles in the entraining fluid, moving past an object in the fluid, which impinge on the object.

The target efficiencies for cylinders, spheres, and ribbonlike particles are given for conditions of Stokes Law in an infinite fluid by Figure 4-14.

If the particles are close enough together in the fluid to affect the path of each other, then Figure 4-14 gives conservative efficiencies. For particles differing considerably from those given in the curves, actual test data should be obtained.

There are basically three construction types for impingement separators:

1. Wire mesh
2. Plates (curved, flat or special shaped)
3. Packed Impingement Beds

Knitted Wire Mesh

A stationary separator element of knitted small diameter wire or plastic material is formed of wire 0.003 in. to 0.016 in. (or larger) diameter into a pad of 4 inches, 6 inches or 12 inches thick and serves as the impingement surface for liquid particle separation. Solid particles can be separated, but they must be flushed from the mesh to prevent plugging. Although several trade name units are available they basically perform on the same principle, and have very close physical characteristics. Carpenter [4] presented basic performance data for mesh units. Figure 4-15 shows a typical eliminator pad.

Figure 4-16 pictorially depicts the action of the wire mesh when placed in a vertical vessel.

Referring to Figure 4-16, the typical situation represents a vapor disengaging from a liquid by bursting bubbles and creating a spray of liquid particles of various sizes. Many of these particles are entrained in the moving vapor stream. The largest and heaviest particles will settle by gravity downward through the stream and back to the bottom of the vessel or to the liquid surface. The smaller particles move upward, and if not removed will carry along in the process stream. With wire mesh in the moving stream, the small particles will impinge on the wire surfaces; coalesce into fluid films and then droplets, run to a low point in their local system; and fall downward through the up-flowing gas stream when sufficiently large. The gas leaving is essentially free from entrained liquid unless the unit reaches a flooding condition.

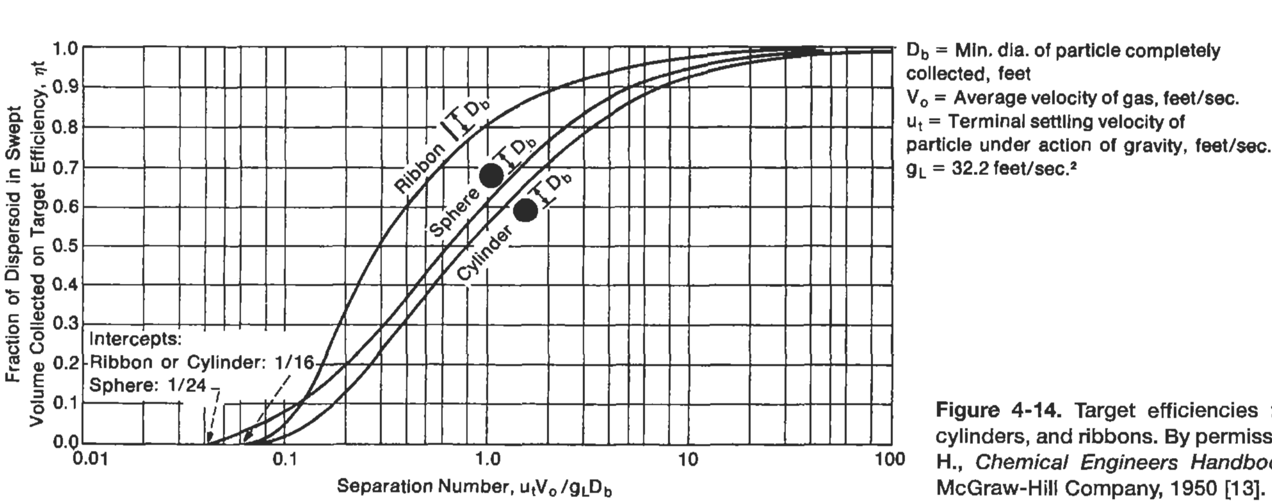


Figure 4-14. Target efficiencies for spheres, cylinders, and ribbons. By permission, Perry, J. H., *Chemical Engineers Handbook,* 3rd Ed., McGraw-Hill Company, 1950 [13].

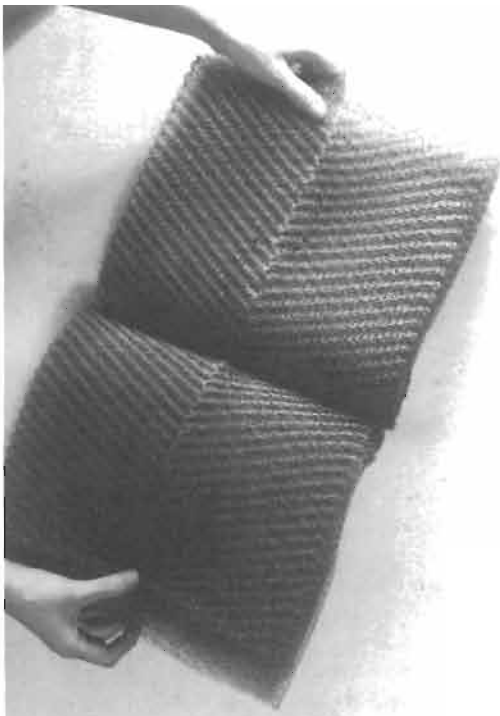
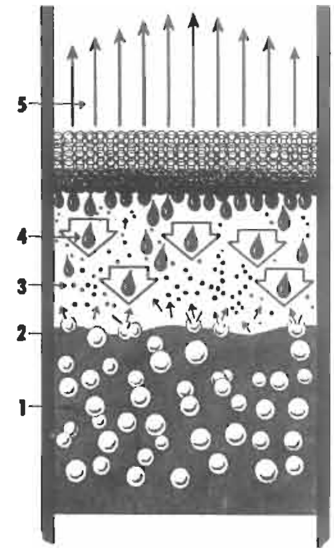


Figure 4-15. Details of wire mesh construction. Courtesy of Otto H. York Co.



When a gas is generated in, or passes through, a liquid (1), the gas, on bursting from the liquid surface (2) carries with it a fine spray of droplets—liquid entrainment—which are carried upward in the rising gas stream (3). As the gas passes through the mist eliminator, these droplets impinge on the extensive surface of the wire, where they are retained until they coalesce into large drops. When these liquid drops reach sufficient size, they break away from the wire mesh (4) and fall back against the rising gas stream. In this way, the entrained droplets are literally "wiped out" of the gas which, freed from liquid entrainment, (5) passes on unhindered through the mesh.

Figure 4-16. Diagram of action of wire mesh in liquid-vapor separation. Courtesy of Metal Textile Corp., Bulletin ME 9-58.

For special applications the design of a mist eliminator unit may actually be an assembly in one casing of wire mesh and fiber packs/pads or in combination with Chevron style mist elements (see Figure 4-17A and 17B and −17C.) This can result in greater recovery efficiencies for small particles and for higher flow rates through the combined unit. Refer to the manufacturers for application of these designs.

Mesh Patterns

There are several types of mesh available, and these are identified by mesh thickness, density, wire diameter and weave pattern. Table 4-9 identifies most of the commercial material now available. The knitted pads are available in any material that can be formed into the necessary weaves, this includes: stainless steels, monel, nickel, copper, aluminum, carbon steel, tantalum, Hastelloy, Saran, polyethylene, fluoropolymer, and glass multi-filament.

Capacity Determination

The usual practice in selecting a particular mesh for a given service is to determine the maximum allowable velocity and from this select a vessel diameter. In the case of existing vessels where mesh is to be installed, the reverse procedure is used, i.e., determine the velocity conditions which will prevail and select a mesh to fit as close to the conditions as possible. The procedure is outlined below:

Allowable vapor velocity (mesh in horizontal position)

$$Va = k\sqrt{\frac{\rho_L - \rho_v}{\rho_v}}$$

Va = maximum allowable superficial vapor velocity across inlet face of mesh, ft/sec

k = constant based on application, Table 4-10, average for free flowing system = 0.35 for 9–12 lb/cu ft mesh

ρ_L = liquid density, lb/cu ft

ρ_v = vapor density, lb/cu ft

For other mesh densities, use k(52) of 0.4 for 5 lb/cu ft mesh (high capacity), and 0.3 for plastic mesh such as Teflon® and polypropylene.

Table 4-9
Identification of Wire Mesh Types

General Type	Density, Lbs./cu. ft.*	Surface Area Sq. ft./cu. ft.	Thickness, In.**	Min. Eff. Wt. %	Application
High Efficiency	12	115	4+	99.9+	Relatively clean, moderate velocity.
Standard Eff.	9	85	4+	99.5+	General purpose
Optimum Eff. or VH Efficiency, and Wound type	13—14	120	4+	99.9+	For very high efficency
Herringbone, High through-put or Low Density	5—7	65±	4—6+	99.0+	For services containing solids, or "dirty" materials

*If the mesh is made of nickel, monel or copper, multiply the density values by 1.13, referenced to stainless steel.
** 4" is minimum recommended thickness; 6" is very popular thickness; 10" and 12" recommended for special applications such as fine mists, oil vapor mist.
Compiled from references (3) and (21).

Reference [52] suggests "dry" mesh pressure drop of:

$$\Delta p_D = [f_c l a \rho_v V_s / g_c \varepsilon^3](27.7/144) \quad (4\text{-}45)$$

$$\Delta p_T = \Delta p_D + \Delta p_L \quad (4\text{-}46)$$

For Δp_L see manufacturer's curves.

A rough approximation of operating mesh pressure drop is 1 inch water or less. The calculated pressure drop at the maximum allowable velocity is close to 1.5 inches of water. Therefore:

$$p_T = 1.5\,(V_{act}/V_{max})^2 \quad (4\text{-}47)$$

How FLEXICHEVRON® Mist Eliminators Work

Gases with entrained liquid droplets flow between the zig-zag baffles. The gas can easily make the turns while the liquid droplets impinge upon the walls of the baffles and coalesce to a size such that they drop downward, being too heavy to be carried in the gas.

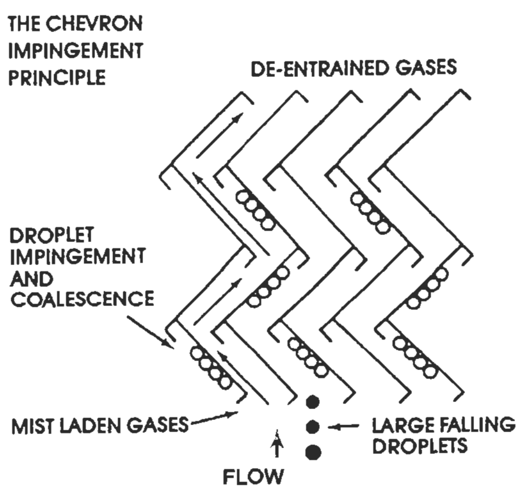


Figure 4-17A. Separation/Impingement action of Chevron-style mist eliminators. Flow is up the V-shaped plates assembly. Courtesy of Bulletin KME - 12, Koch Engineering Co.

where
- a = specific surface area, sq ft/cu ft
- f_c = friction factor, dimensionless
- g_c = gravitational constant, 32.2 lb-ft/lb-sec-sec
- l = wire mesh thickness, ft
- Δp_D = pressure drop, no entrainment, in. of water
- $\Delta p_L'$ = pressure drop, due to liquid load, in. of water
- Δp_T = pressure drop, total across wet pad, in. of water
- V_s = superficial gas velocity, ft/sec
- ε = void fraction of wire mesh, dimensionless
- ρ_L = liquid density, lb/cu ft
- ρ_v = vapor density, lb/cu ft
- f = generally ranges 0.2 to 2 for dry mesh

Subscript:

Act = actual
Max = maximum

The correlation factor, k, is a function of the liquid drop size, liquid viscosity, liquid load, disengaging space, type of mesh weave, etc. k varies somewhat with system pressure; as pressure increases the k value decreases. The manufacturers should be consulted for final design k valves for a sys-

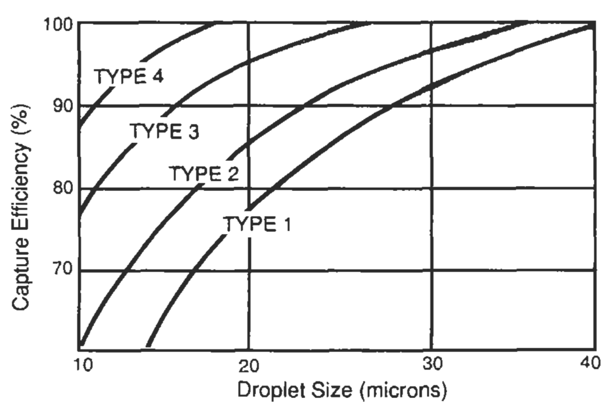


Figure 4-17B. Capture efficiency vs particle size for four standard York-Vane mist eliminators. By permission, Otto H. York Co. Inc.

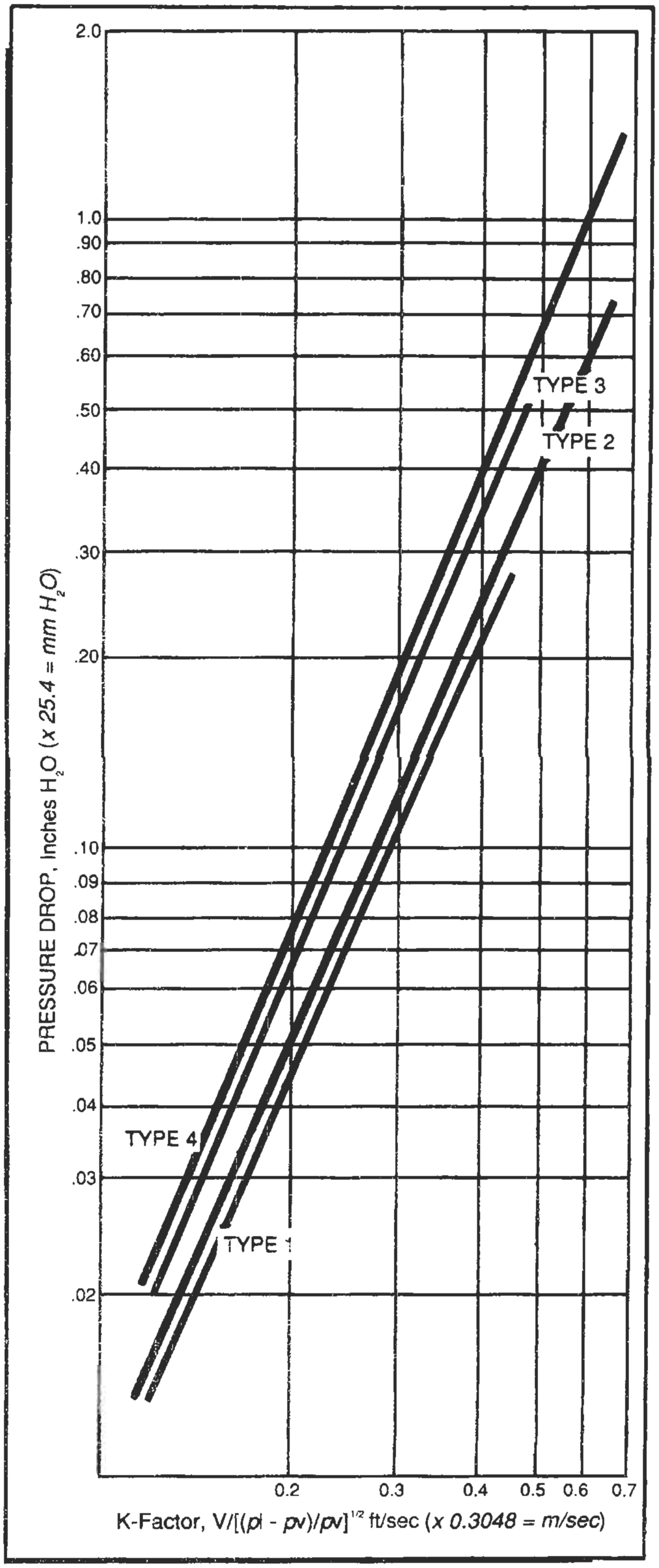


Figure 4-17C. Pressure drop vs K-factor for standard York-Vane mist eliminators, air-water system. By permission, Otto H. York Co., Inc.

tem, because the wire style, size, and material also affect the value. For pressures below 30 psig, k = 0.35 avg., then above 30 psig, k value decreases with pressure with an approximate value of 0.30 at 250 psig and 0.275 at 800 psig. Certain values have been found satisfactory for estimating systems described in Table 4-10 and Table 4-11.

For conditions of high liquid loading, use caution in design. Use the high velocities for very fine mist to remove the small particles, and use two mesh pads in series with the second mesh operating at a lower velocity to remove the larger drops re-entrained from the first mesh. Systems involving high viscosity fluids should be checked with the various manufacturers for their case history experience. Lower k values are used for systems with high vacuum, high viscosity liquids, low surface tension

Table 4-10
"k" Values for Knitted Mesh

Bottom of mesh at least 12 inches above liquid surface

Service Conditions	"k"	General Type Mesh
Clean fluids, moderate liquid load, fits 90% of process situations	0.35 to 0.36 0.35 0.25	Standard High Efficiency Very High Efficiency
High viscosity, dirty suspended solids	0.40	Low density or Herringbone, high through-put
Vacuum operations:		
2" Hg. abs.	0.20	Standard or
16" Hg. abs.	0.27	High Efficiency
Corrosive Chemical	0.21	Plastic coated wire, or plastic strand

Compiled from various manufacturer's published data. Note: k values for estimating purposes, not final design unless verified by manufacturer. Unless stated, all values based on stainless steel wire.

Table 4-11
Variation of k with Disengaging Height*

Disengaging Height Above Mesh, Inches	Allowable k Value
3	0.12
4	0.15
5	0.19
6	0.22
7	0.25
8	0.29
9	0.32
10	0.35
11	0.38
12	0.40
13	0.42
14	0.43

*By permission, O. H. York, Reference (21).

Note: Values based on 12 lb/cu ft wire mesh. Design practice normally does not exceed k of 0.4 even for higher disengaging height.

liquids and systems with very bad fouling conditions. Table 4-11 indicates the effect of disengaging height on the allowable k value. Similar relations should hold for other mesh densities.

Velocity Limitations

Very low velocities will allow particles to drift through the mesh and be carried out with the leaving vapor. Also, very high velocities will carry liquid to the top of the mesh, establish a "flooding" condition, and then re-entrain the liquid from the surface of the mesh. For most situations very good performance can be expected for all velocities from 30% to 100% of the optimum allowable design velocity. The minimum allowable safe design velocity is 10 percent of the value calculated by the equation. The flooding velocity of the mesh is usually about 120 percent to 140 percent of the maximum allowable velocity.

Generally the maximum allowable velocities are lower under conditions of pressure, and higher under conditions of vacuum. The limits and ranges of each area being determined by the relative operating densities of the vapor and liquid, the nature of the entrainment, and the degree of separation required.

When the mesh is installed with the pad vertical or inclined, the *maximum* allowable velocity is generally used at 0.67 times the allowable value for the horizontal position.

Design Velocity

To allow for surges, variations in liquid load and peculiarities in liquid particle size and physical properties, use:

$$V_D = 0.75\ V_a \qquad (4\text{-}48)$$

for the design of new separators. When checking existing vessels to accept wire mesh, some variation may have to be accepted to accommodate the fixed diameter condition, but this is no great problem since the range of good operation is so broad.

Efficiency

For most applications the efficiency will be 98–99 percent plus as long as the range of operating velocity is observed. The typical performance curves for this type of material are given in Figures 4-17B, 4-18, and 4-19. For hydrocarbon liquid-natural gas system, guarantees are made that not more than 0.1 gallon of liquid will remain in the gas stream per million cubic feet of gas. Special designs using a 3-foot thick pad reduce radioactive entrainment to one part per billion [21].

For the average liquid process entrainment the mesh will remove particles down to 4 to 6 microns at 95%+ recovery efficiencies [see Figure 4-19]. Particles smaller than this usually require two mesh pads or the fiber pack style discussed later. Carpenter [4,5] shows the calculated effect of decreasing particle size on percent entrainment removed at various linear velocities. For water particles in air at atmospheric pressure, the 8μ particles are 99 percent removed at 3.5 ft/sec, the 7μ at 5 ft/sec, and the 6μ at 6.8 ft/sec. Excellent performance may be obtained in most systems for velocities of 30% to 110% of calculated values [35].

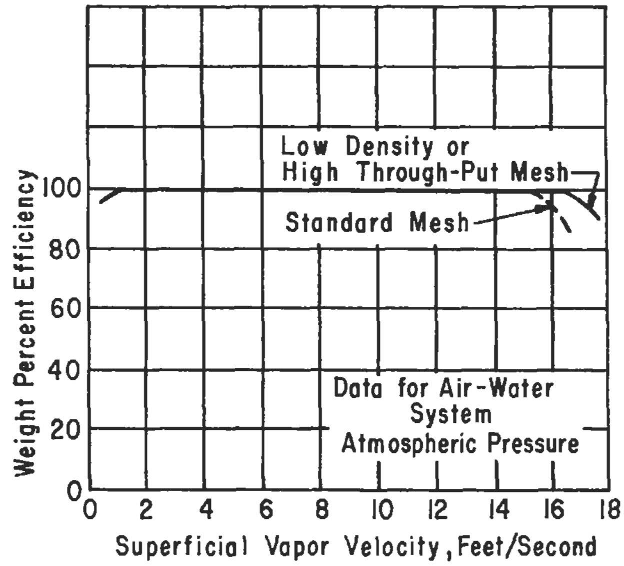


Figure 4-18. Typical wire mesh efficiency.

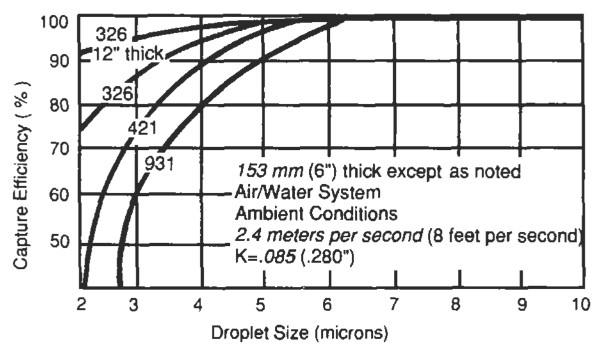


Figure 4-19. Capture efficiency vs particle size for four types of DEMISTER® knitted mesh mist eliminators. By permission, Otto H. York Co., Inc.

Pressure Drop

Pressure drop through wire mesh units is usually very low, in the order of 1-inch water gauge for a 4-inch or 6-inch thick pad. For most pressure applications this is negligible. If solids are present in the particle stream, then solids build-up can become appreciable, and is usually the guide or indicator for cleaning of the mesh. A 12-inch pad may require a 3-inch water drop. Figures 4-20 and 4-21 present the range of expected pressure drops for a spread of 3 to 1600 lb/hr-ft^2 for liquid rates. Although this is for air-water system at atmospheric pressure it will not vary much unless the physical properties of the vapor and liquid deviate appreciably from this system, in which case the general Fanning equation can be used to approximate the pressure drop under the new conditions. Approximate values based upon air-water tests suggest these relations [3]:

For the standard weave, 4 inches thick:

$$\Delta p = 0.2\ V_D^2 \rho_v, \text{ in. water} \tag{4-49}$$

For the low density weave (high through-put), 6 inches thick:

$$\Delta p = 0.12\ V_D^2 \rho_v \tag{4-50}$$

Installation

The knitted mesh separator unit may be placed in a pipe in which case a round flat rolled unit is usually used, or it may be placed in a conventional vessel. Although the vessel may be horizontal or vertical, the mesh must always be in a horizontal plane for best drainage. Some units in special situations have been placed at an angle to the horizontal, but these usually accumulate liquid in the lower portion of the mesh. Since the material is not self-supporting in sizes much over 12 inches in diameter, it requires support bars at the point of location in the vessel. In most instances it is wise to also install hold-down bars across the top of the mesh in accordance with manufacturers' instructions as the material will tend to blow upward with a sudden surge or pulsation of vapor in the system. Many early installations made without the bars on top were soon found ineffective due to blowout holes, and wire particles were found in pipe and equipment downstream of the installation. Figures 4-22 and 4-23 show a typical installation arrangement in a vertical vessel. The mesh is wired to the bottom support bars and the hold-down on top.

A few typical arrangements of mesh in vessels of various configurations are shown in Figure 4-24.

Note that in some units of Figure 4-24 the mesh diameter is smaller than the vessel. This is necessary for best

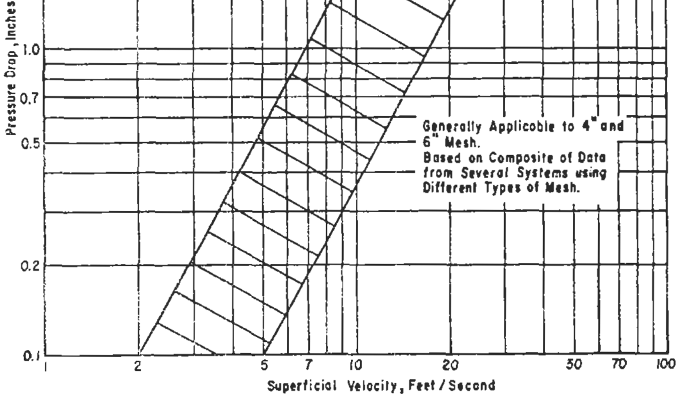


Figure 4-20. Typical pressure drop range for most wire mesh separators.

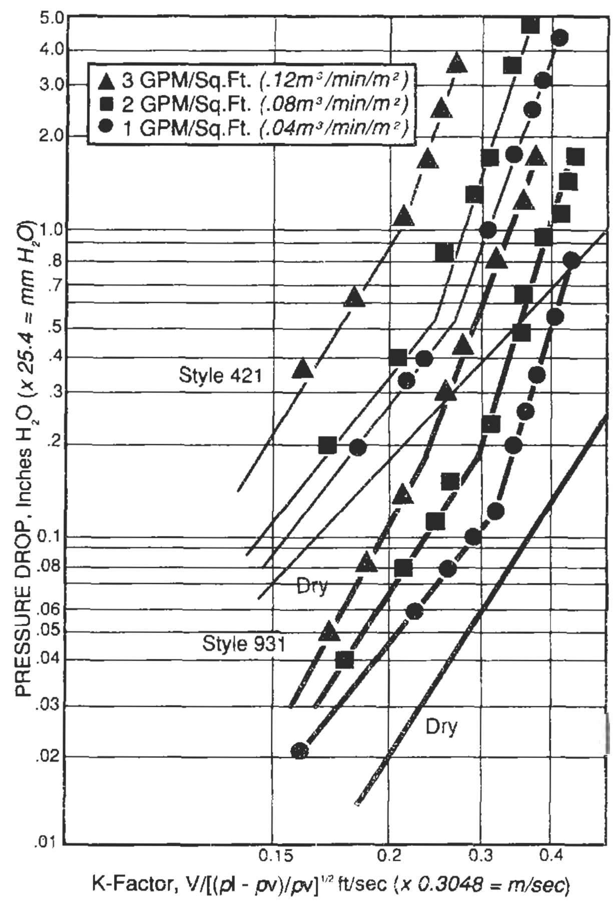


Figure 4-21. Typical wire mesh mist eliminator pressure drop curves for one style of mesh at three different liquid loadings. Others follow similar pressure drop patterns. By permission, Otto H. York Co., Inc.

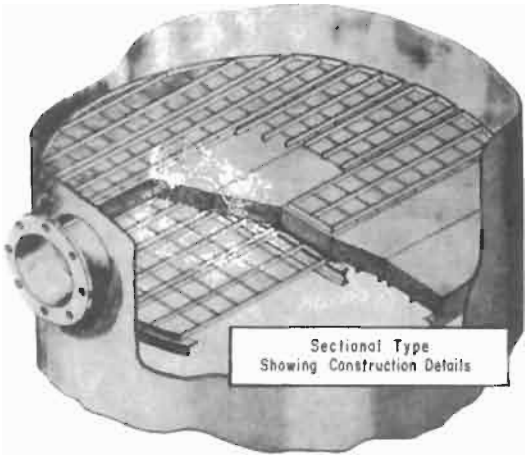


Figure 4-22. Typical installation of mesh strips in vertical vessel. Courtesy of Otto H. York Co., Inc.

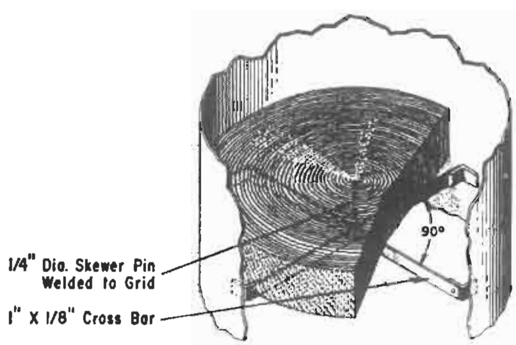


Figure 4-23. Typical installation of wound mesh pads in vertical vessel. Courtesy of Metal Textile Corp., Bulletin ME 9-58.

operating efficiency under the system conditions, and applies particularly when using an existing vessel.

When placing mesh in small diameter vessels it is important to discount the area taken up by the support ring before determining the operating velocity of the unit. For small 6-, 8-, and 12-inch vessels (such as in-line, pipe-with-mesh units) it is usual practice to use 6- or 8-inch thickness of mesh for peak performance.

Provide at least 6- to 12-inch minimum (preferably 18-inch min.) disengaging space ahead of the inlet face of the mesh, i.e., above any inlet nozzle bringing the liquid-carrying vapors to the vessel, or above any liquid surface held in the vessel. Leave 12-inch minimum of disengaging space above the mesh before the vapors enter the vessel vapor exit connection. The mesh may be installed in horizontal, vertical or slanting positions in circular, rectangular or spherical vessels. For locations where the liquid drains vertically through the mesh pad perpendicular or angular to its thickness dimension, care must be taken to keep velocities low and not to force or carry the liquid through to the downstream side of the mesh.

Example 4-5: Wire Mesh Entrainment Separator

Design a flash drum to separate liquid ethylene entrainment for the following conditions:

Volume of vapor = 465 CFM @—110°F and 35 psig
Density of vapor = 0.30 lb/cu ft
Density of liquid ethylene = 33 lb/cu ft
Allowable velocity for wire mesh:

$$V_a = k\sqrt{\frac{\rho_L - \rho_v}{\rho_v}}$$

Use, k = 0.35 for clean service, moderate liquid loading

$$V_a = 0.35\sqrt{(33 - 0.3)/0.3}$$
$$= 3.66 \text{ ft/sec, allowable loading velocity}$$

Use, $V_D = 0.75\ V_a$

Design velocity:

$$V_D = 0.75(3.66) = 2.74 \text{ ft/sec}$$

Required vessel cross-section area:

$$A = 465/(60)(2.74) = 2.83 \text{ sq ft}$$

Vessel diameter:

$$D = \sqrt{\frac{2.83\ (4)}{\pi}} = 1.898 = 1' - 11''$$

Try: 2'-0" I.D. vessel
Deduct 4 inches from effective diameter for 2-inch support ring inside.

$$24'' - 4'' = 20''$$

Net area:

$$A = \frac{\pi\,(20)^2}{4\,(144)} = 2.18 \text{ sq ft}$$

$$\text{Actual velocity at ring: } 2.74\left(\frac{2.83}{2.18}\right) = 3.56 \text{ ft / sec}$$

This is 97% of maximum allowable design, too high.
Second Try:
Increase diameter to next standard dimension, 2 ft, 6-in. Although intermediate diameters could have been

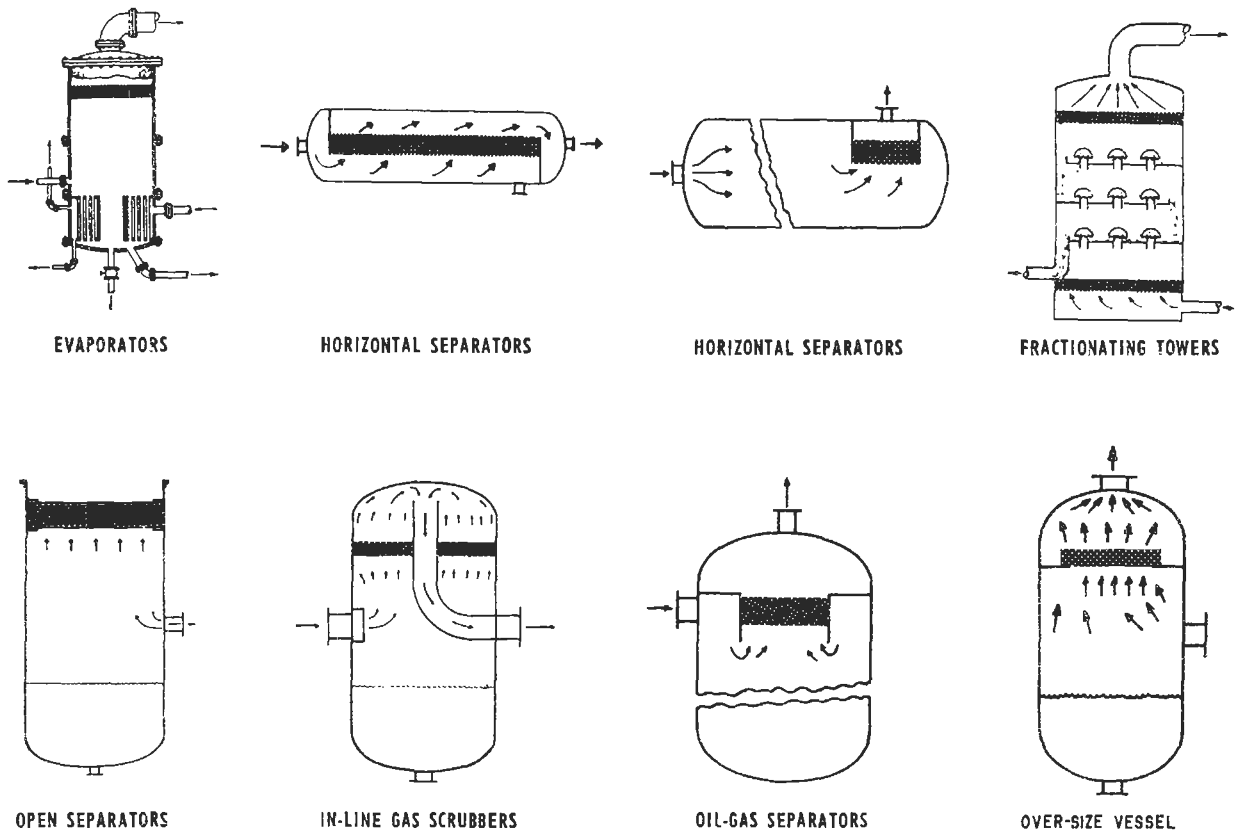


Figure 4-24. Typical mesh installations in process equipment. Courtesy of Metal Textile Corp., Bulletin ME-7.

selected, the heads normally available for such vessels run in 6-inch increments (either O.D. or I.D.).

Net inside diameter at support ring:

30″ O.D. − 4″ − 3/4″ = 25 1/4″

Note that vessel wall assumed 3/8-inch thick.

Net area = 3.46 sq ft

$$\text{Actual velocity at ring: } \frac{2.83}{3.46}(2.74) = 2.24 \text{ ft/sec}$$

Percent design velocity: 2.24(100)/3.66 = 61.3%. This is acceptable operating point.

Note that if 28-inch O.D. × 3/8-inch wall pipe is available this could be used with weld cap ends, or dished heads. The percent design velocity would be = 71.8%.

This is also an acceptable design.

Pressure drop is in the order of 0.1 inch to 0.5 inches water.

Notes: Since this vessel will operate as a flash drum with a liquid level at approximately 1/4 of its height up from bottom, place the inlet at about center of vessel. See Figure 4-25.

Use stainless 304 mesh due to low temperature operation. Carbon steel is too brittle in wire form at this temperature.

The check or specification form of Figure 4-26 is necessary and helpful when inquiring wire mesh entrainment units, either as the mesh alone, or as a complete turnkey unit including vessel.

For services where solids are present or evaporation of droplets on the mesh might leave a solid crust, it is usual practice to install sprays above or below the mesh to cover the unit with water (or suitable solvent) on scheduled (or necessary) operating times, as the plugging builds up. This is checked by a manometer or other differential pressure meter placed with taps on the top and bottom side of the mesh installation.

A few case examples for guidance include:

- 2–3% caustic solution with 10% sodium carbonate. This condition might plug the wire mesh. Sprays would be recommended.
- Raw river water. This presents no plugging problem
- Light hydrocarbon mist. This presents no plugging problem

Job No. ______

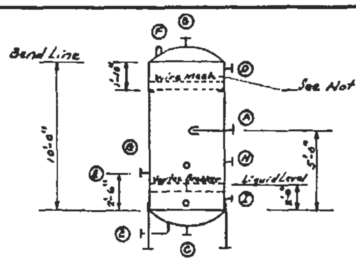
B/M No. ______

SPEC. DWG. NO.
A-

Page 1 of 1 Pages
Unit Price
No. Units 1
Item No. D-3

DRUM OR TANK SPECIFICATIONS

Service Flash Drum and Separator
Size 30" O.D. x 10'-0" Bend Line Type ______



DESIGN DATA

Operating Pressure 35 PSI G   Operating Temp. -110 °F.
Design Pressure 100 PSI G   Design Temp. -130 °F.
Code ASME Stamp Yes Lethal Const. No Density of Contents 33 Lbs/cu. ft
Materials: Shell Low temperature steel Heads Low temp. steel Supports Carbon steel
Lining: Metal No Rubber or Plastic No
Brick No Cement No
Internal Corrosion Allowance 1/64 * Self Supporting Yes Insulation: ☒ Yes ☐ No, Class -110°F

NOZZLES

Service	No. Req'd.	Size	Press Class	Facing	Mark No.
Inlet	1	6"	150	RTJ	A
Vapor Out	1	6"	150	RTJ	B
Liquid Out	1	3"	150	RTJ	C
Drain					
Safety Valve	1	4"	150	RTJ	D
Level Control	2	2"	150	RTJ	E
Pressure Tap	1	1"	6000	Coupling	F
Vent					
Gage Glass	2	2"	150	RTJ	G
Manhole *	1				
High Level Alarm	1	2"	150	RTJ	H
Low Level Alarm	1	2"	150	RTJ	I

REMARKS

*Follow Code, Provide convient to wire mesh installation
Wire Mesh to be Standard Weave, 6" thick; Material Type 304 stainless steel.
Provide support and top hold-down for mesh.

By	Chk'd	Rev.	Rev.	Rev.	Rev.
Date					

P.O. To: ______

Figure 4-25. Specification design sheet for separator using wire mesh.

- Heavy oil with suspended matter. This might plug. A light oil or solvent spray would be recommended for flushing the mesh.

Fiber Beds/Pads Impingement Eliminators

The use of fiber packing held between wire mesh containing screens is best applied in the very low micron range, generally 0.1 to > 3 microns with recoveries of entrained liquid of up to 99.97% (by weight). Figures 4-27A, B, and C illustrate the design concept and its corresponding data table indicates expected performance. The fibers used mean the bed packing can be fabricated from fine glass, polypropylene fibers, or can be selected to be the most resistant to the liquid mist entering the unit from corrosive plant operations such as sulfuric acid, chlorine, nitric acid, ammonia scrubbing for sulfur oxides control, and many others. The entrained particles are removed by direct interception, inertial impaction, and Brownian capture.

The design rating for this equipment is best selected by the manufacturers for each application.

The concept of removal of entrained liquid particle is essentially the same as for wire mesh designs, except the

Wire Entrainment Mesh Specifications

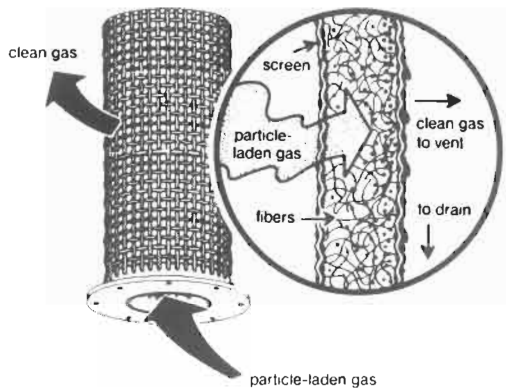
A. Application Service
   1. Source of Entrainment: ______
   2. Operating Conditions: Give (1) Normal (2) Maximum (3) Minimum, where possible
      Temperature ______
      Pressure ______
      Vapor Phase
         Flow Rate ______
         *Velocity ______
         Density ______ at operating conditions
         Molecular Weight ______
         Composition or Nature of Phase ______
      Liquid Entrainment Phase
         Quantity (if known): ______
         Density: ______
         Viscosity: ______
         Surface Tension: ______
         Composition or Nature of Entrainment: ______
         Droplet Sizes or distribution (if known): ______
         Solids Content (Composition and Quantity): ______
            Dissolved: ______
            Suspended: ______
   3. Performance
      Allowable Total Separator Pressure Drop: ______
      Allowable Mesh Pressure Drop: ______
      Allowable Entrainment: ______
      Mesh Thickness Recommended: ______

B. Construction and Installation
   1. Vessel
      *Diameter. I.D: ______ Length ______
      Construction Material: ______
      *Position (Horizontal, Vertical, Inclined): ______
      *Shape (Circular, Square, etc.): ______
      Type (Evaporator, Still, Drum, etc.): ______
      Existing or Proposed: ______
   2. Entrainment Mesh
      Construction Material
         Separator Mesh ______
         Support Grid ______
      Installation Method (Dimensions)
         Vessel Open End: ______
         Manhole (size): ______

C. Special Conditions: ______

*Assumes vessel size fixed prior to mesh inquiry

Figure 4-26. Wire entrainment mesh specifications.



Materials of construction
Packing of York-Fiberbed high efficiency mist eliminators consists of ceramic, glass, polypropylene, fluoropolymer fibers. Cages and frames are fabricated from all stainless steels and other weldable alloys as well as FRP.

Figure 4-27A. Details of a cylindrical York-Fiberbed® mist eliminator. Courtesy of Otto H. York Co., Inc., Bullet in 55B.

particle size removed may be much smaller. Just as for other types of mist eliminators, the performance is affected by the properties of the liquid particles, entraining gas, temperature, pressure, liquid viscosity, particle size distribution of entrained material and the quantity of total entrainment, and the desired process removal requirement. Some designs of these units provide excellent performance removal efficiencies at a wide range of rates (turndown), even at low gas rates.

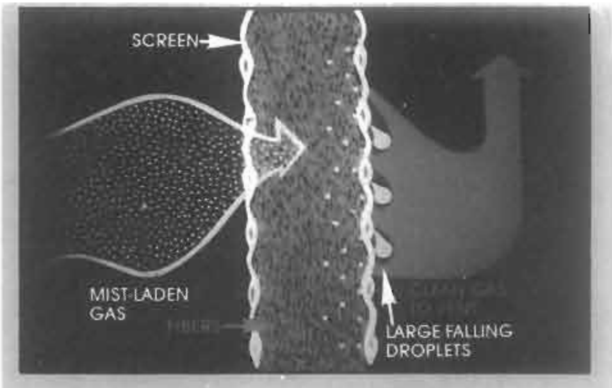
Pressure drop is usually low depending on many factors, but can be expected in the range of 2 to 20 inches of water [33].

Baffle Type Impingement

There are many baffle type impingement separators. The efficiency of operation for entrainment is entirely a function of the contacting action inside the particular unit. There are no general performance equations which will predict performance for this type of unit; therefore manufacturers' performance data and recommendations should be used. A few of the many available units are shown in Figures 4-28 to 4-31. Many use the Chevron-style vertical plates as shown in Figures 4-17A and 4-30.

Baffles (Chevrons/Vanes)

One of the common impingement plate assemblies is of the Chevron "zig-zag" style of Figures 4-17A and 4-30. This style of impact separation device will tolerate higher gas velocities, high liquid loading, viscous liquids, reasonable solids, relatively low pressure drops. The collected



Available FLEXIFIBER® Mist Eliminator Styles

KOCH Type	Primary Collection Mechanism	Collection Efficiency: Particle Size (Microns)	Collection Efficiency: Efficiency (%)	Element Pressure Drop (Inches WG)	Bed Velocity (Ft/Min)
BD	Brownian Diffusion	>3	Essentially 100	2–20	5–40
		<3	Up to 99.95		
IC	Impaction	>3	Essentially 100	7–9	250–450
		1-3	95–99 +		
IP	Impaction	>3	Essentially 100	5–7	400–500
		1-3	85–97		
		0.5–1	50–85		
IS	Impaction	>3	Essentially 100	1–2	400–500
		<3	15–30		

NOTE: Since Types IC, IP and IS operate primarily by impaction, the above collection efficiencies drop off at gas flows below about 75% of design rates and depend on the specific gravity of the collected liquid.
Flexifiber® Type BD: Normally cylindrical in shape and available in a wide variety of materials and sizes. Bulk packed elements are Mark I series.
Also available as wound beds (Mark II or III) for lower initial cost and ease of repacking, and with equivalent collection efficiency at equal or lower pressure drop.
Flexifiber® Type IC: Normally cylindrical in shape and available in a variety of materials and sizes.
Flexifiber® Types IP and IS: Normally rectangular in shape and available in various metals.

Figure 4-27B. Fiber-pack® mist eliminator pack separators. By permission, Koch Engineering Co., Inc. Note that other manufacturers have basically the same concept; however, the identification of types are peculiar to each.

liquid droplets run down on the plate surfaces countercurrent to the up-flowing gas stream. See Reference [59] for performance study.

Spacing of the plates and their angles is a part of the design using the manufacturers' data. Multiple pass designs can result in higher recovery efficiencies. The units can be designed/installed for vertical or horizontal flow.

Some of the same physical properties of the liquid and gas phases as well as temperature and pressure and the amount of entrained liquids (or solids if present) and the expected particle size and its distribution control the design and performance of these units also.

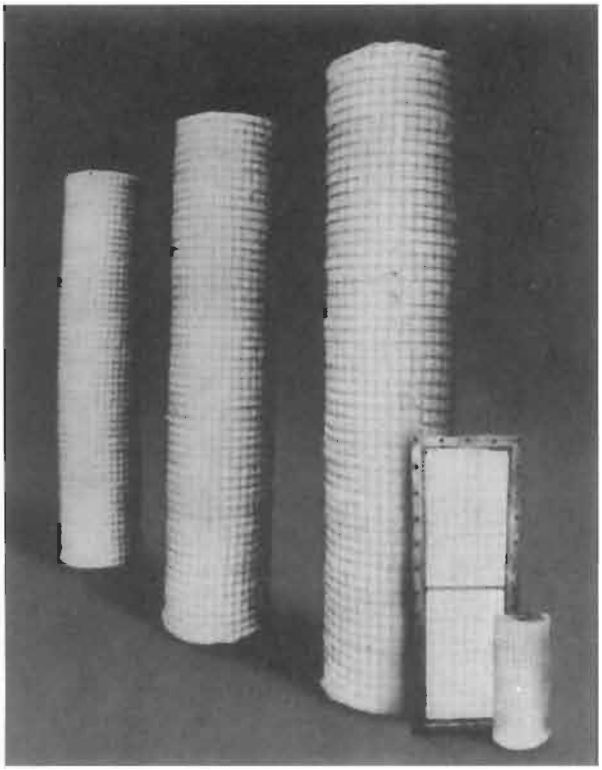


Figure 4-27C. Typical fiberbed mist eliminators are available in both candle and panel configurations. By permission, Otto H. York Co., Inc.

For preliminary selection:

$$V_D = k[(\rho_l - \rho_v)/\rho_v]^{1/2} \tag{4-51}$$

ρ_v = vapor density, lb/cu ft at actual conditions
ρ_L = liquid density, lb/cu ft at actual conditions
k = 0.40 for up-flow at 0.65 for horizontal flow, for estimating

Required flow area estimate only,

A = (ACFS)/V_D, sq ft
A = area sq ft
ACFS = actual flow, cu ft/sec
V_D = design velocity, ft/sec

Generally, this style of unit will remove particles of 12 to 15 microns efficiently. The typical droplet separator is shown for an air-water system in Figure 4-17A. This will vary for other systems with other physical properties. The variations in capacity (turndown) handled by these units is in the range of 3 to 6 times the low to maximum flow, based on k values [33].

A liquid-liquid separator used for removing small, usually 2% or less, quantities of one immiscible liquid from another is termed a coalescer. These units are not gravity settlers, but agglomerate the smaller liquid by passing through a surface contact medium such as excelsior, hay, cotton or wool bats, or cartridges of fibers similar in nature and weave to those of Table 4-12A and -12B. Figure 4-32 illustrates some of these types.

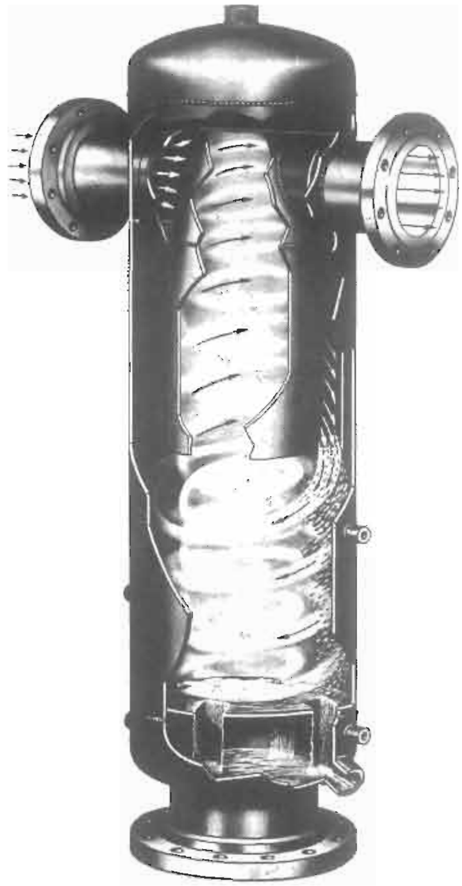


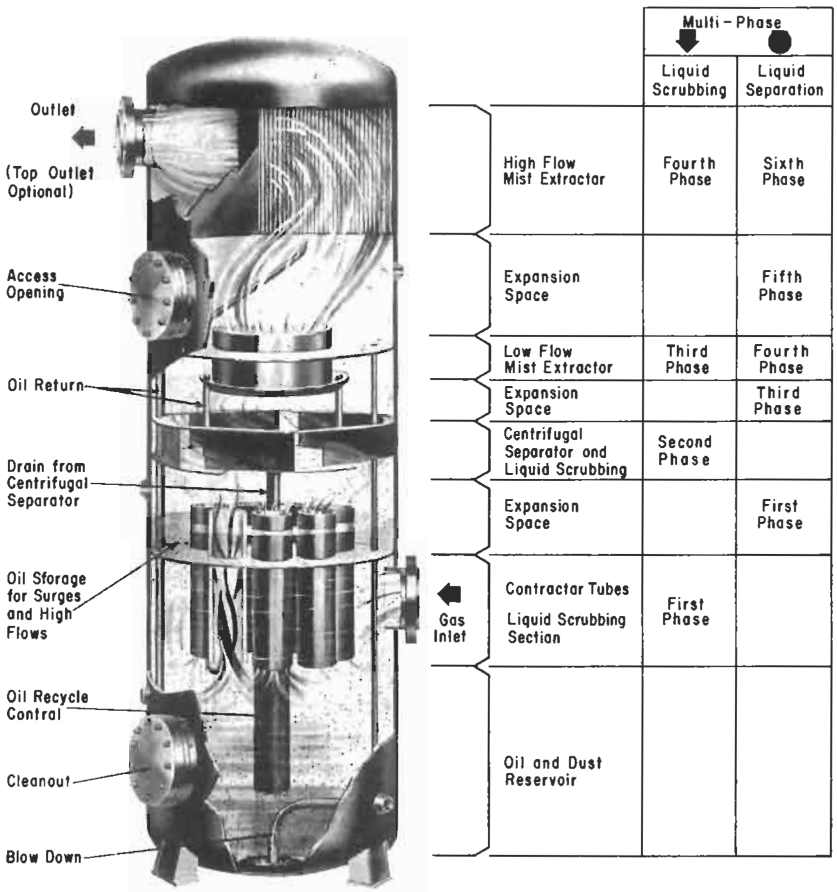
Figure 4-28. Wall-wiping centrifugal type separator. Courtesy of Wright-Austin Co.

Efficiency

The efficiency of this type of unit varies, and is a function of the effectiveness of the impingement baffling arrangement. About 70% of separator applications can use the line-type unit; the other 30% require the vessel construction. The preference of the designer and problems of the plant operator are important in the final selection of a unit to fit a separation application.

The efficiency for removal of liquid and solid suspended particles is 97–99%+ when handling 15-micron particles and larger. For steam service, a typical case would be 90% quality entering steam with 99.9 percent quality leaving.

Some units will maintain a reasonable efficiency of separation over a range of 60%–120% of normal performance rating while other types will not. This flexibility is



	Multi-Phase: Liquid Scrubbing	Multi-Phase: Liquid Separation
High Flow Mist Extractor	Fourth Phase	Sixth Phase
Expansion Space		Fifth Phase
Low Flow Mist Extractor	Third Phase	Fourth Phase
Expansion Space		Third Phase
Centrifugal Separator and Liquid Scrubbing	Second Phase	
Expansion Space		First Phase
Contractor Tubes Liquid Scrubbing Section	First Phase	
Oil and Dust Reservoir		

Figure 4-29. Multiphase gas cleaner. Courtesy of Blaw-Knox Co.

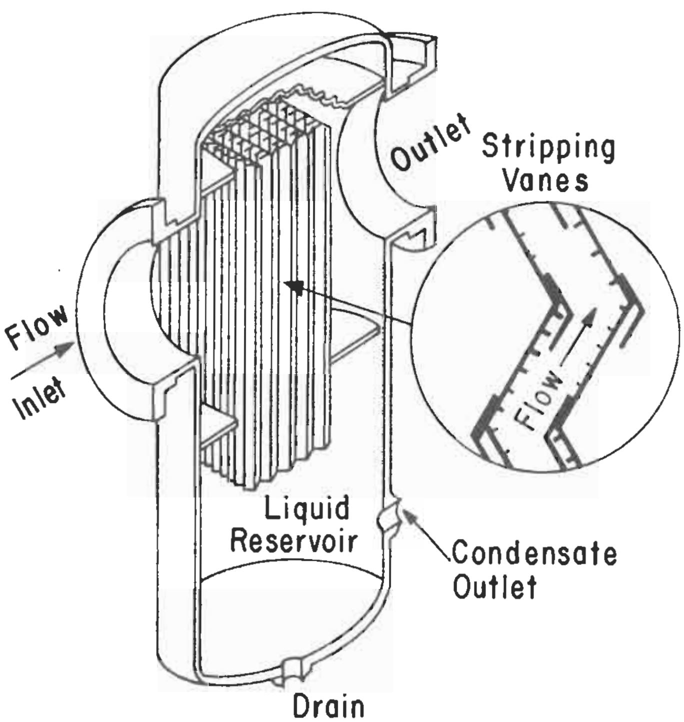


Figure 4-30. Impingement separator. Courtesy of Peerless Manufacturing Co.

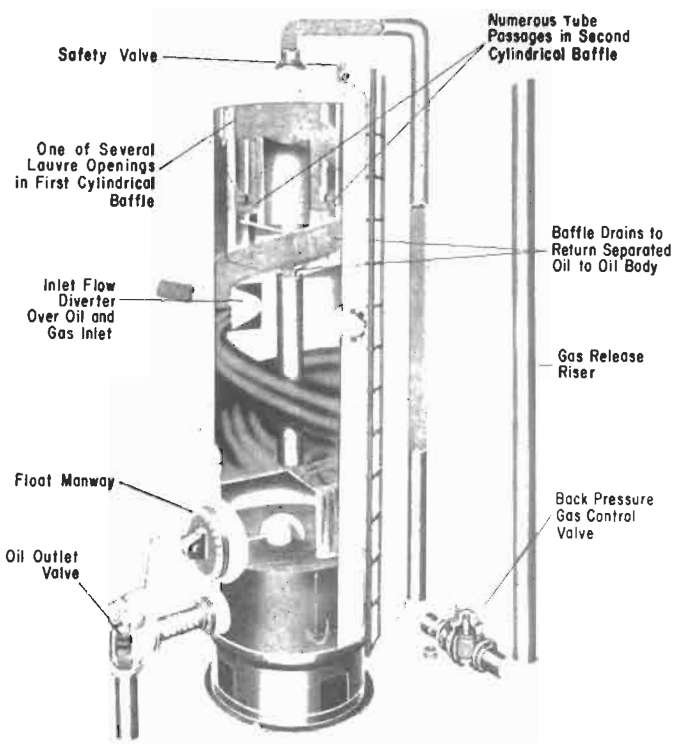


Figure 4-31. Combination separator. Courtesy of National Tank Co.

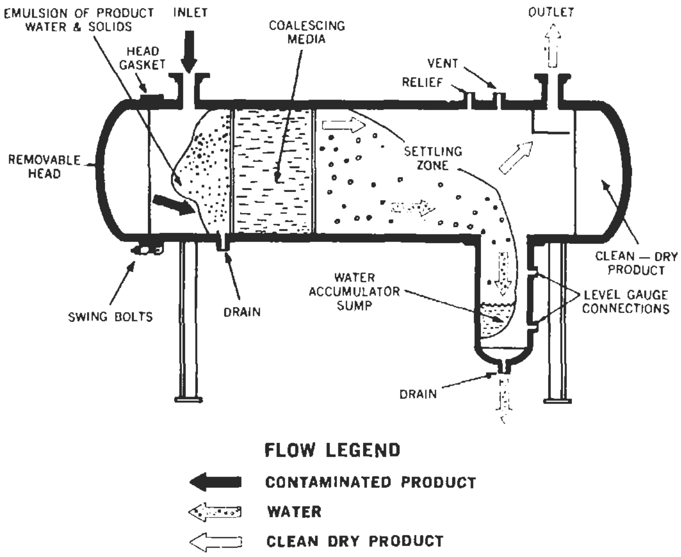


Figure 4-32. Typical coalescer unit. By permission, Facet Enterprises, Inc., Industrial Div.

very peculiar to the internal design of the unit. Some units are guaranteed to reduce mechanical entrainment loss to less than 0.1 gallon per million standard cubic feet of entraining gas.

Pressure Drop

Pressure drop in most units of this general design is very low, being in the order of 0.1 to 3 psi.

How to Specify

Manufacturers' catalogs are usually available and complete with capacity tables for the selection of a unit size. However, it is good practice to have this selection checked by the manufacturer whenever conditions will allow. This avoids misunderstandings and misinterpretations of the catalog, thereby assuring a better selection for the separation operation, and at the same time the experience of the manufacturer can be used to advantage.

With a manufacturer's catalog available:

1. Establish the normal, maximum, and minimum gas flow for the system where the unit will operate. This is usually in standard cubic feet per minute, per hour or per day. Note the catalog units carefully, and also that the reference standard temperature is usually 60°F for gas or vapor flow.
2. Use the rating selection charts or tables as per catalog instructions. For specific gravity or molecular weight different than the charts or tables, a correction factor is usually designated and should be used.
3. Note that some units use pipe line size for the separator size designation, others do not.
4. From the system operating pressure, establish the pressure rating designation for the separator selection.
5. Note that most separators for pressure system operations are fabricated according to the ASME code.
6. Specify special features and materials of construction, such as alloy or nonferrous impingement parts, or entire vessel if affected by process vapor and liquid. Specify special liquid reservoir at base of unit if necessary for system operations. Line units normally have dump traps or liquid outlet of separator, while vessel type often use some type of liquid level control.
7. Specification sheet: see Figure 4-33.

Baffle Type Separator Specifications

Separator Application: (Give Service) ____________

Design Operating Conditions:

Main Stream Flow Rate _____ Sp. Gr. or Mol. Wt. _____

Entrained Material rate(if known)Source of entrainment _____

Min. Pressure _____ psi (g) or (a), Max. Temp. _____°F.

Max. Pressure _____ psi (g) or (a)

Entrained Particle size ________(mesh)(Microns)

Vessel Specifications:

Design Pressure _____ PSI _____ Design Temp. _____ °F.

Code: API-ASME _____ ASME 1949 Ed. _____ASME 1950 Ed. _____

State Code _____; Non-Code _____, Customer's Spec. _____

X-Ray _____ Stress Relief _____

Corrosion allowance ________

Dimensions: _____ "O.D. x _____ " long bend line to bend line

Base Support ________

Mist Extractor: ________, Mat'l. of Construction ________

Connections

1. Gas inlet and outlet:(Size, ASA pressure rating, type flange)
2. Liquid Outlet ________
3. Liquid Level Gage ________
4. Liquid Level Control ________
5. Pressure Gauge ________
6. Relief Valve ________
7. Bursting Disc ________
8. High Level Alarm ________
9. Low Level Alarm ________
10. Thermometer ________
11. Equalizer ________
12. Drain ________
13. Others: (specify)

Special Features: ________

Figure 4-33. Baffle-type separator specifications.

Note that these units should not be connected in lines larger than their pipe inlet, since inlet velocity conditions are very important, the swaging down or reduction tends to produce a jet effect by the gas upon the mist eliminator unit. This may erode the unit and cause other erratic performance

Dry-Packed Impingement Beds

Although this type of unit is not used as frequently as most of the others discussed, it does have some specific applications in sulfuric acid mist removal and similar very difficult applications. The unit consists of a bed of granular particles or ceramic packing, sometimes graduated in size, operating dry as far as external liquid application to aid in the separation. The superficial velocities of 0.5 to 8 feet per second through the unit are rather low for most separators therefore the vessels become large. Due to the packed heights of 2 feet (min.) and higher, the pressure drop can be appreciable. Particle removal may be 0.5 to 5 microns at 99% efficiency for a good design. These units will plug on dust service and must be back washed to regain operability at reasonable pressure drops.

Centrifugal Separators

Centrifugal separators utilize centrifugal action for the separation of materials of different densities and phases. They are built in (a) stationary and (b) rotary types. Various modifications of stationary units are used more than any other kind for separation problems. The cost is moderate; it is simple in construction, and is reasonably flexible in service, being useful for gas-liquid or gas-solid systems. In addition to serving as finishing separators centrifugal units are also used to take a "rough cut" into a separation problem. They may be followed by some additional unit of special cyclone action or filtration through woven cloth pads, etc., to completely remove last traces of entrained particles.

Stationary Vane

The stationary vane type is quite popular and adapts to many applications. It is used in vessels or pipe lines as illustrated in Figures 4-34 to 4-40. They are usually of high efficiency for both liquid and solid particles such as rust, scale, dirt, etc. When the system is dry with dust a special design is used.

Efficiency

The efficiency of centrifugal units is:

Type	Efficiency Range
High Velocity Stationary Vanes	99% or higher, of entering liquid. Residual entrainment 1 ppm or less
Cyclone	70–85% for 10 micron, 99% for 40 micron and larger. For high entrainment, efficiency increases with concentration.
Rotary	98% for agglomerating particles

Cyclone Separators

The cyclone type unit is well recognized and accepted in a wide variety of applications from steam condensate to dusts from kilns. In this unit the carrier gas and suspended particles enter tangentially or volutely into a cylindrical or conical body section of the unit, then spiral downward forcing the heavier suspended matter against the walls. Solids tend to slide down the wall while liquid particles wet the wall, form a running film and are removed at the bottom. Figure 4-41 gives a good typical cyclone arrangement, but this is by no means the highest efficiency or best design. References [43,45,51] provide good design and performance analysis.

Some commercial units are shown in Figures 4-42 to 4-45.

The zone of most efficient separation is in the conical region designated by dimension Z_c in Figure 4-41. The larger particles have already been thrown against the wall before the outlet was reached. The finer particles are thrown out in the inner vortex as the direction of motion is reversed. Here, the relative velocity difference between the particle and the carrier is the greatest for any point in the separator. Although the tangential velocity component predominates throughout the cyclone, the axial velocity prevails in the turbulent center. Van Dongen and ter Linden [20] measured pressure patterns in a typical cyclone and found the lowest total pressure at the extreme bottom point of the cone, even lower than at the gas exhaust. Their pressure profiles indicate considerable eddy or secondary gas movement in the unit near the vertical axis.

Particle Size Separation

The theoretical minimum diameter particle to be separated in a cyclone of the basic type given by Figure 4-41 is given by the relation of Rosin [13].

$$\underset{(\text{Min.})}{D_p} = \sqrt{\frac{9\mu W_i}{\pi N_t V_c (\rho_s - \rho)}} \qquad (4\text{-}52)$$

N_t has been found to be about 5 turns of the gas stream in the unit, and is considered somewhat conservative. When re-entrainment takes place, N_t may drop to 1.0 or 2.0. The API study presents an excellent survey of cyclone dust collectors [7].

Solid Particle Cyclone Design

Following the general dimensional relations of the typical cyclone as shown in Figure 4-41, the following general guides apply. This cyclone is better suited to solid particles removal than liquid droplets. To avoid re-entrainment it is important to keep the separated material from entering the center vortex of the unit. Solid particles generally slide down the walls with sufficient vertical velocity to avoid re-entrainment.

a. Select outlet diameter to give gas velocity out not exceeding 600 ft per min. Bear in mind that higher velocities can be used in special designs.
b. Due to the usual conditions of limiting pressure drop, entrance velocities range from 1000 to 4000 ft per min. 3000 fpm is good average, although velocities to 6000 fpm are used in some applications.

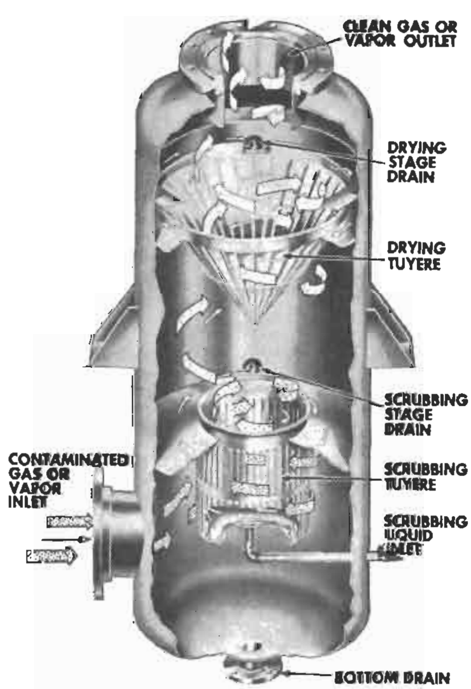


Figure 4-34. Scrubber with internal liquid feed. Courtesy of Centrifix Corp.

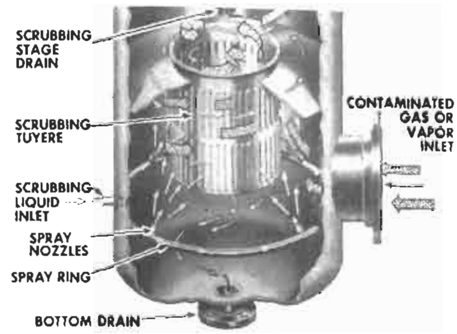


Figure 4-35. Scrubber with spray ring as alternate arrangement for Figure 4-34. Courtesy of Centrifix Corp.

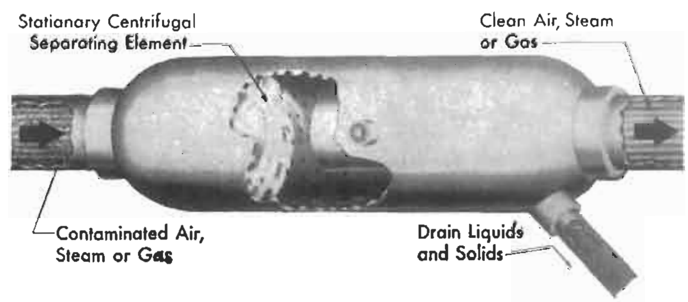


Figure 4-36. Line-type centrifugal separator. Courtesy of V. D. Anderson Co.

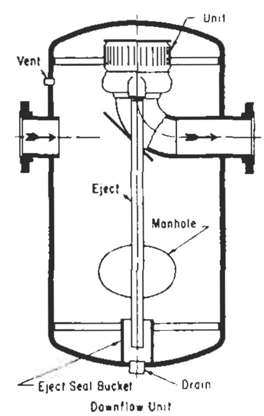


Figure 4-37. Centrifugal separator applications. By permission, Centrifix Corp.

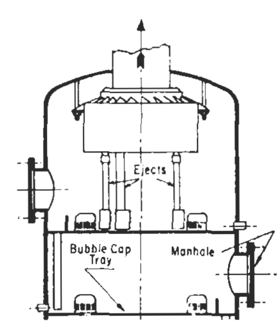


Figure 4-38. Centrifugal separator applications. By permission, Centrifix Corp.

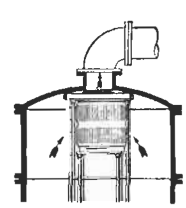


Figure 4-39. Centrifugal separator applications. By permission, Centrifix Corp.

c. Select cylindrical shell diameter, D_c, with two considerations in mind:

- Large diameter reduces pressure drop
- Small diameter has higher collection efficiency for the same entrance conditions and pressure drop.

d. The length of the inverted cone section, Z_c, is critical, although there is no uniformity in actual practice. The dimensions suggested in Figure 4-41 are average.

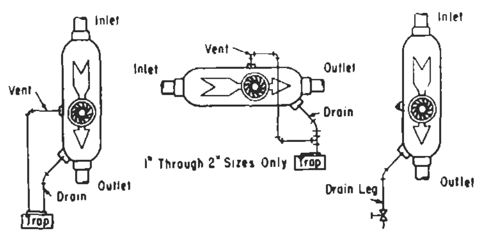


Figure 4-40. Centrifugal separator applications. By permission, Centrifix Corp.

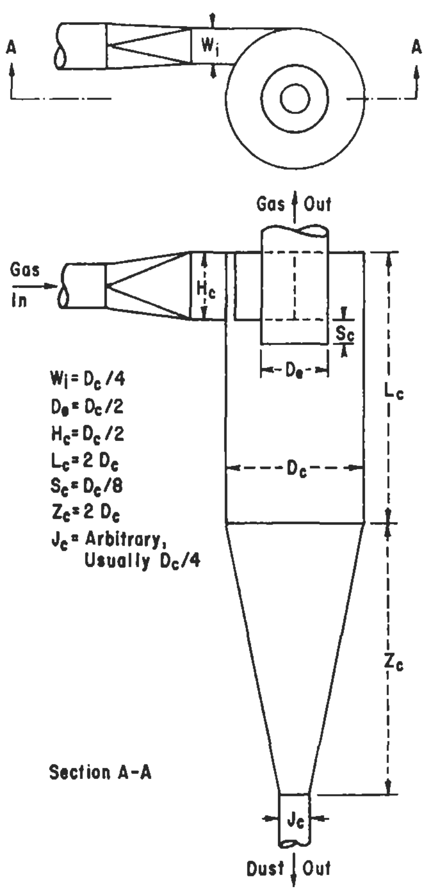


Figure 4-41. Cyclone separator proportions—dust systems. By permission, Perry, J. H., *Chemical Engineers Handbook*, 3rd Ed., McGraw-Hill Company, 1950.

Efficiency

Typical estimating efficiencies are given in Figures 4-46 and 4-47. Note the curves indicate how much dust of each particle size will be collected. The efficiency increases as the pressure drop increases; that is, a smaller separator might have a higher efficiency due to the higher gas velocities and increased resistance than a larger unit for the same gas flow. For example, there are several curves of the typical shape of Figure 4-46, with each curve for a definite resistance to flow through the unit.

The pressure drop in a typical cyclone is usually between 0.5 and 8 inches of water. It can be larger, but rarely exceeds 10 inches water for single units. The API study [7] summarizes the various factors. Lapple [13,16] gives calculation equations, but in general the most reliable pressure drop information is obtained from the manufacturer.

Here is how the pressure drop may be estimated.

For the typical cyclone of Figure 4-41 [13]

(a) Inlet velocity head based on inlet area:

$$h_{ui} = 0.003\rho\ V_c^2 \qquad (4\text{-}53)$$

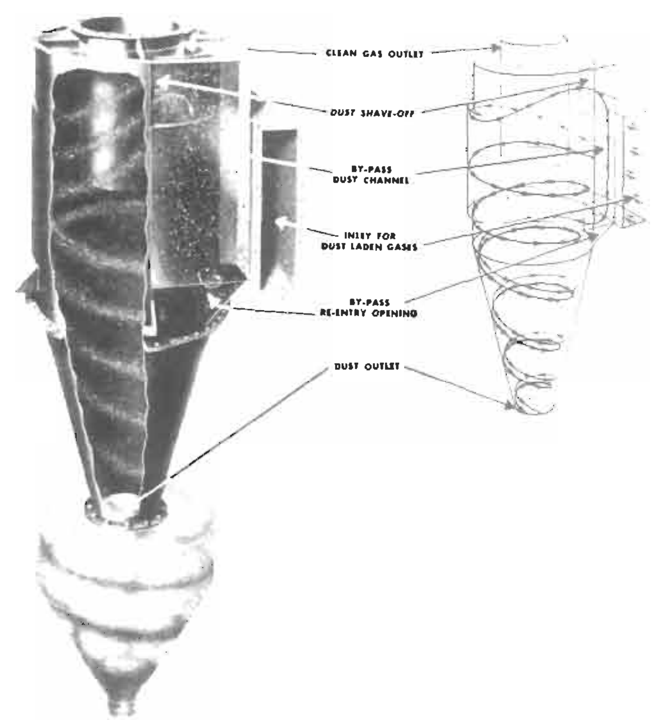


Figure 4-42. Van Tongeran dust shave-off design. Courtesy of Buell Engineering Co.

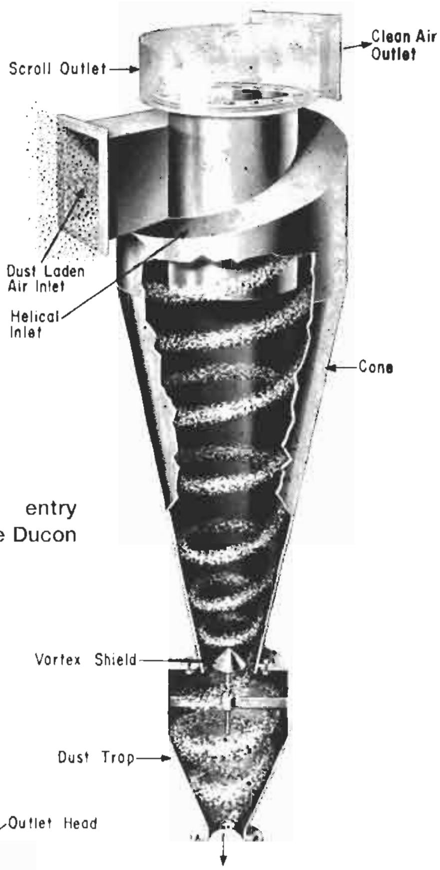


Figure 4-43. Helical entry cyclone. Courtesy of The Ducon Co., Inc.

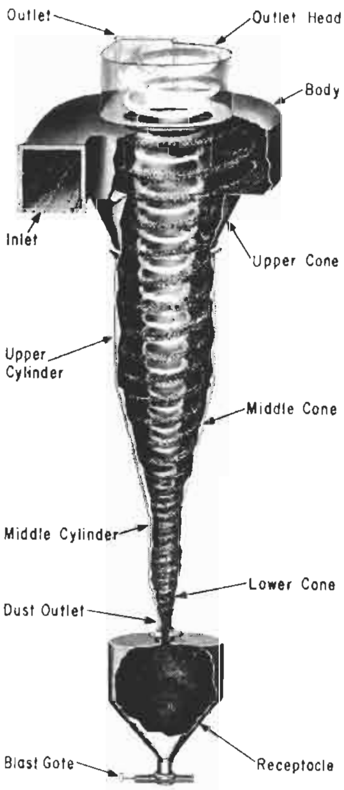


Figure 4-44. Involute entry cyclone. By permission, American Blower Div. American Radiator and Standard Sanitary Corp.

(b) Internal cyclone friction loss:

$$F_{cv} = \Delta P_{cv} + 1 - \left(\frac{4A_c}{\pi D_e^2}\right)^2 \qquad (4\text{-}54)$$

$$\Delta P_{cv} = K\left(\frac{D_c}{D_e}\right)^2 \qquad (4\text{-}55)$$

K has been found to be constant at 3.2 for cyclones with an involute entrance

$W_i/D_c = 1/8$ to $3/8$

$H_c/D_c = 1.0$

$D_e/D_c = 1/4$ to $3/4$

For the typical cyclone of Figure 4-41 with tangential inlet:

$$F_{cv} = \frac{KW_iH_c}{D_e^2} \qquad (4\text{-}56)$$

$K = 16.0$

$F_{cv} = 8.0$

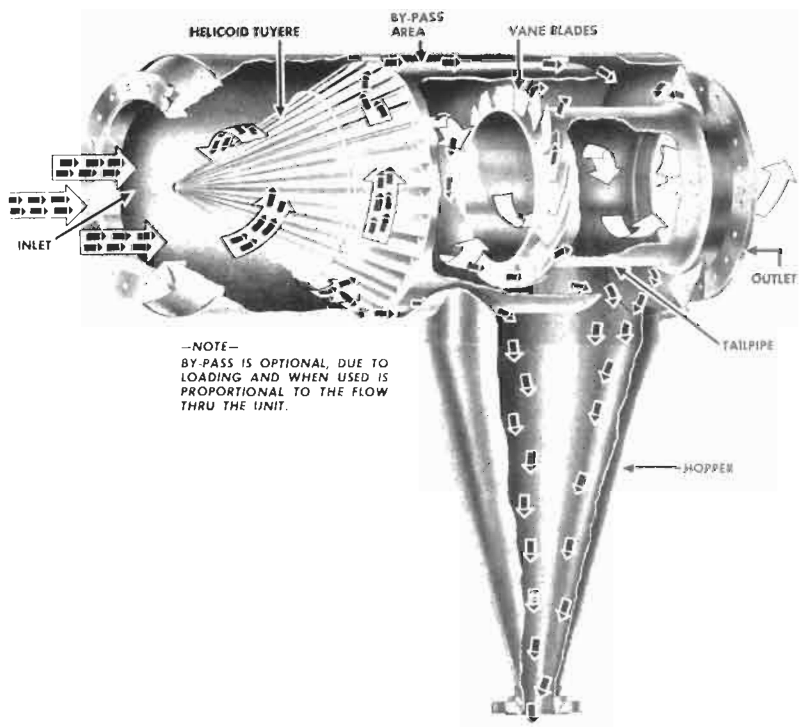


Figure 4-45. Stationary vane centrifugal separator. Courtesy of Centrifix Corp.

If inlet vane is formed with inlet connection:

K = 7.5

Example 4-6: Cyclone System Pressure Drop

A cyclone system is to be installed as a part of a bagging operation. The unit is shown in Figure 4-48. Determine the head required for purchase of the fan. The conditions are:

Air volume: 4000 cu ft per min of air at 70°F

Air density: 0.075 lb/cu ft

Areas:		Gas Velocity	Velocity Head*
Inlet duct:	1.398 sq ft	47.7 ft/sec	0.50 in. water
Cyclone inlet:	2.0 sq ft	33.33 ft/sec	0.25 in. water
Cyclone exit duct:	3.14 sq ft	21.2 ft/sec	0.10 in. water

*Velocity head, inches water = $V_D^2/(16)(10^6)$, V_D = ft/min

Friction Loss ① to ② :

$N_{Re} = 398{,}000 \qquad f = 0.0038$

$$\text{No. Vel. Hd.} = \frac{4fL}{D} = \frac{4\ (.0038)\ (18)}{(16/12)} = 0.204\ \text{vel. head}$$

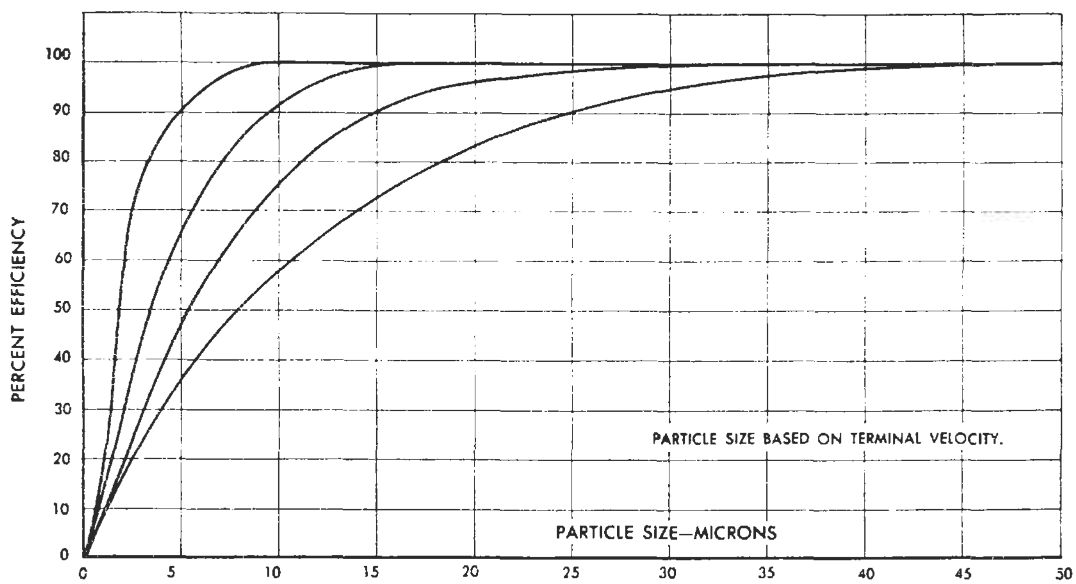


Figure 4-46. General efficiency curves, applies specifically to helical entry cyclone dust separators. Courtesy of The Ducon Co.

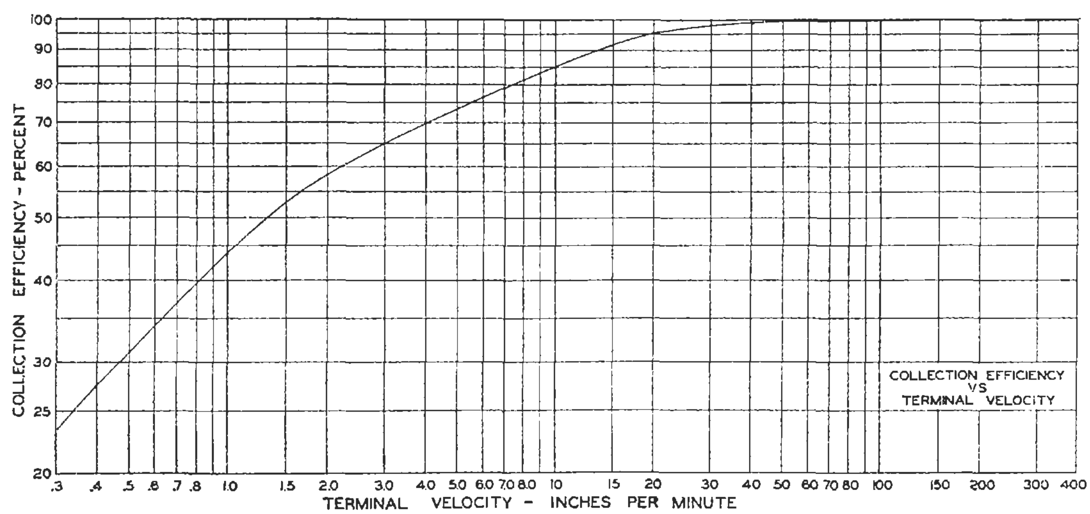


Figure 4-47. General efficiency curve applies specifically to involute entry cyclone dust separators. Courtesy of American Blower Div., American Radiator and Standard Sanitary Corp.

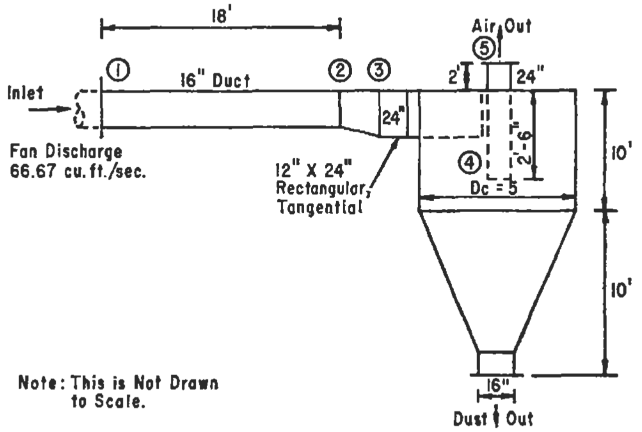


Figure 4-48. Pressure drop for cyclone separator system. Adapted by permission, Lapple, C. E., *Fluid and Particle Dynamics,* 1st Ed., University of Delaware, 1954.

Loss = (0.204)(.5) = 0.102 in. water

Friction Loss ② to ③ :
Assume as 1 vel. head (conservative)

Friction loss = (1)(.50) = 0.50 in. water

Friction Loss ③ to ④ (thru cyclone):

$$F_{cv} = \frac{KW_iH_c}{D_e^2} = \frac{(16)\ (1)\ (2)}{(2)^2} = 8.0 \text{ vel. heads}$$

Friction loss = 8.0(0.25) = 2.0 in. water

Friction Loss ④ to ⑤ :

$$N_{Re} = 280{,}000,\ f = 0.004$$

$$\text{No. vel. heads} = \frac{4fL}{D} = \frac{(4)\ (.004)\ (4.5)}{2} = .036 \text{ vel. head}$$

Loss = 0.036(.10) = 0.0036 in. water

Since the unit exhausts to atmosphere with no additional restrictions, the *total pressure drop* is:

ΔP (total) =
Friction loss + downstream vel. head at ⑤ − upstream vel. head
= (0.102 + 0.50 + 2.0 + 0.0036) + 0* − 0.50
ΔP (total) = 2.6056 + 0 − 0.50 = 2.10 in. water

*Note that point (5) is at atmospheric pressure and the velocity head is zero; however, if there had been a back pressure or resistance at this point before discharging it would have to be added in.

Liquid Cyclone-Type Separator

The unit shown in Figure 4-49 has been used in many process applications with a variety of modifications [18,19,20]. It is effective in liquid entrainment separation, but is not recommended for solid particles due to the arrangement of the bottom and outlet. The flat bottom plate serves as a protection to the developing liquid surface below. This prevents re-entrainment. In place of the plate a vortex breaker type using vertical cross plates of 4-inch to 12-inch depth also is used, (Also see Reference [58].) The inlet gas connection is placed above the outlet dip pipe by maintaining dimension of only a few inches at point 4. In this type unit some liquid will creep up the walls as the inlet velocity increases.

In order to handle higher loads, the liquid baffle is placed at the top to collect liquid and cause it to drop back down through the gas body. If the baffle is omitted, the liquid will run down the outlet pipe and be swept into the outlet nozzle by the outgoing gas as shown in Figure 4-50B. Figure 4-50 and 4-51 show several alternate entrance and exit details. The unit with a tangential entry is 30%–60% more efficient than one with only a turned-down 90° elbow in the center.

If the design of Figure 4-41 is used for liquid-vapor separation at moderately high liquid loads, the liquid sliding down the walls in sheets and ripples has somewhat of a tendency to be torn off from the rotating liquid and become re-entrained in the upward gas movement.

Liquid Cyclone Design (Based on air-water at atmospheric pressure) Figure 4-49

For maximum liquid in outlet vapor of 4 weight percent based on incoming liquid to separator: Figure 4-49.

a. Select inlet pipe size to give vapor velocity at inlet of 100 to 400 ft per second for tangential pipe inlet.
b. Select separator diameter to give velocity of 0.02 to 0.2 (max.) times the inlet velocity. At 400 feet/second pipe velocity the separator velocity should be 0.018 to 0.03 times the pipe velocity. At 130 feet/second pipe velocity the separator velocity should be 0.15 to 0.2 times the pipe velocity.
c. Establish dimensions from typical unit of Figure 4-49. Always evaluate the expected performance in terms of the final design, adjusting vertical dimensions to avoid gas whipping on liquid films or droplets. Do not direct inlet gas toward an outlet. Place manway on same side of vessel as tangential inlet.
d. Pressure drop is essentially negligible for the average conditions of use. Some estimate of entrance and exit losses can be made by fluid flow techniques, and

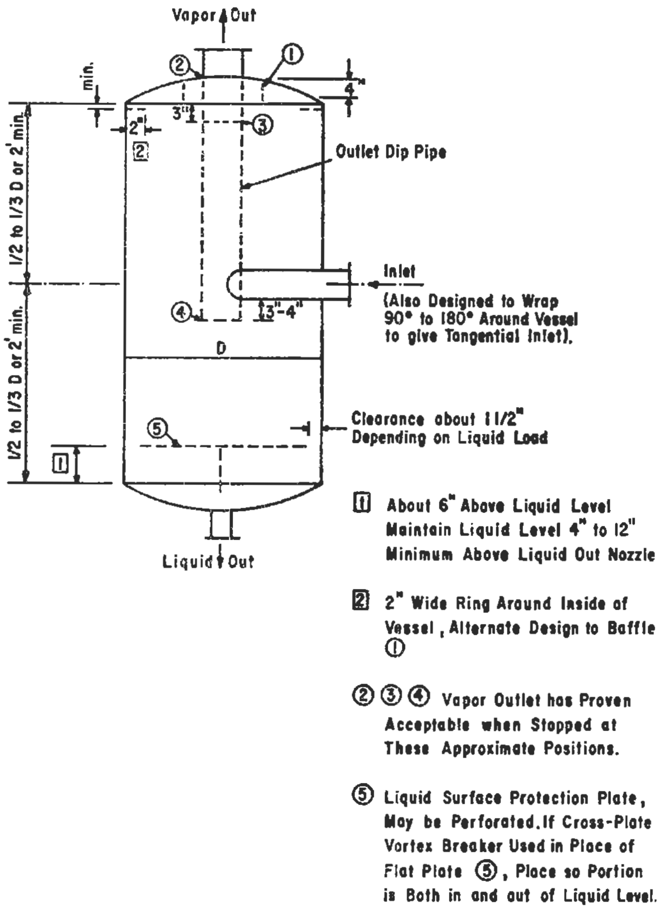


Figure 4-49. Centrifugal liquid separator.

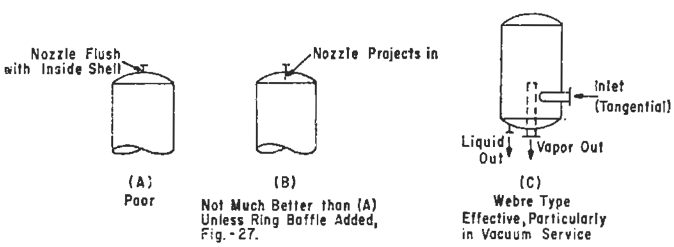


Figure 4-50. Separator outlets for liquid-vapor service.

an internal loss of 0.25 to 2.0 psi assumed, depending upon system pressure and general unit dimensions.

e. For liquids and vapors other than air-water:

$$V\ (\text{separator}) = 0.1885\, V_{sa} \left(\frac{\rho_L - \rho_v}{\rho_v}\right)^{0.25} \qquad (4\text{-}57)$$

where V_{sa} is the selected separator velocity when using an air-water system, feet/sec.

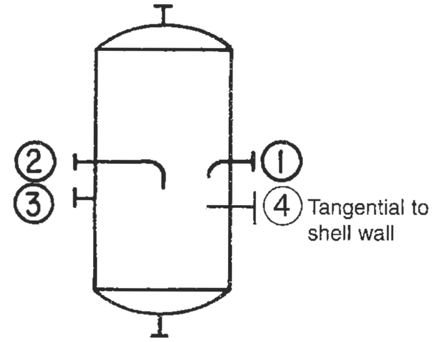


Figure 4-51. Separator inlets for liquid-vapor service.

The Webre design as tested by Pollock and Work [14] showed (Figure 4-50C) that internal action in the separator was responsible for some of the entrainment, particularly liquid creep up the vessel walls.

The performance of this unit correlated for several different types of particle distribution by [14]:

$$\text{Log}\ [L_v(V'/L_1)^a] = b + cV' \qquad (4\text{-}58)$$

a = 2 for Webre unit
L_v = entrainment, lb liquid/min/ft^2 of inlet
V' = vapor vel. entering, lb/min/ft^2 of inlet
L_1 = liquid entering, lb/min/ft^2 of inlet

a, b, c, constants associated with the type and physical conditions of the system. For the unit of Reference [14]:

b = 1.85 and c = 0.00643

Liquid-Solids Cyclone (Hydrocyclones) Separators

This type of solids removal device, Figures 4-52A, and B, is a relatively low cost approach to remove/separate solids from solid/liquid suspensions. The incoming feed to such a unit is injected along the inner wall where the centrifugal force causes rotation at high angular velocity. The kinetic energy of this feed is converted to centrifugal force. The coarse/heavier particles will be concentrated at the bottom as underflow. Most of the feed liquid and part of the fine solids will discharge through the vortex and overflow.

This unit is good to pre-thicken feed to centrifugal filters and similar applications. One cyclone may satisfy a requirement, or the units can be arranged in parallel for large capacities or in series for removal of extreme fines. See Figure 4-53 for a counter-current wash system. Solids as small as 10 microns can be separated.

These units are made of abrasion resistant metals, solid plastics, or with corrosion/wear resistant plastic liners,

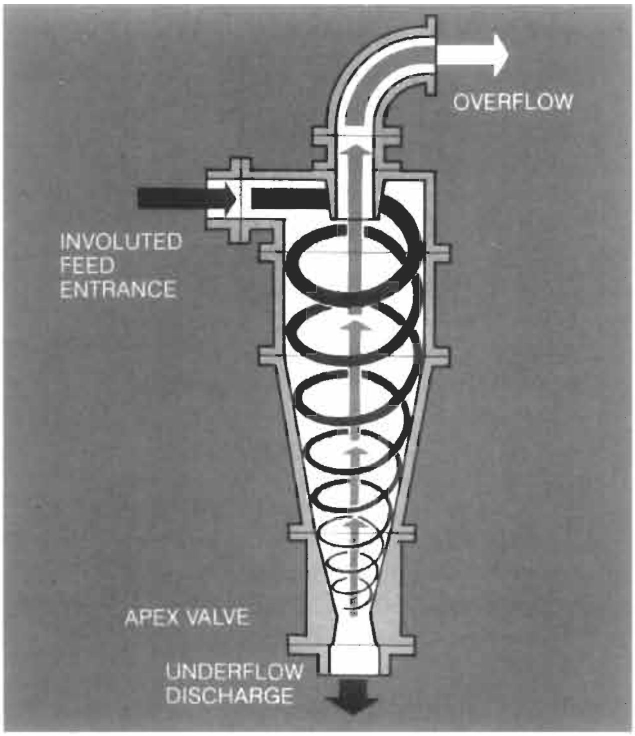
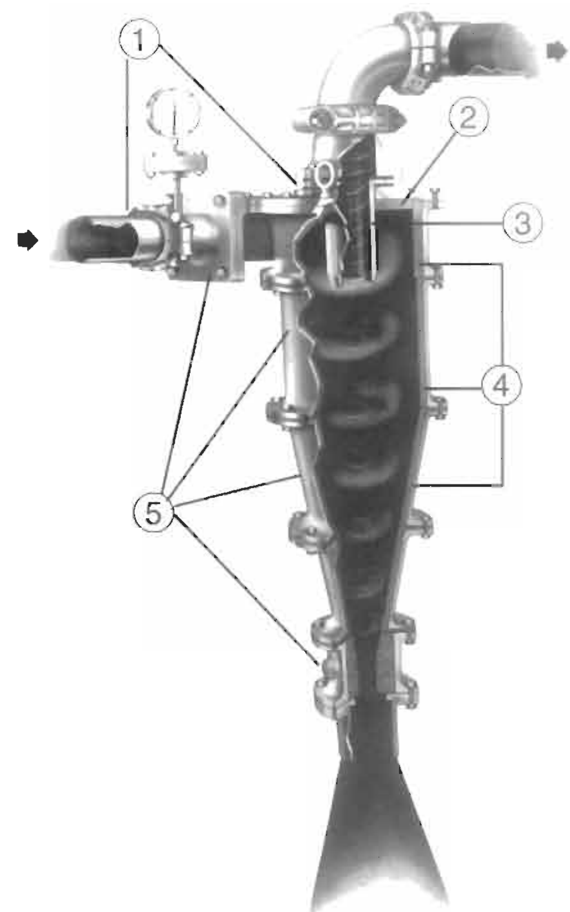


Figure 4-52A. Liquid-solids removal cyclones. Feed enters tangentially along sidewall. By permission, Krebs Engineers.



1. Feed inlet and overflow connections are elastomer lined spool-piece adapters.
2. The top cover plate has all wetted surfaces lined, including the area mating with the overflow adapter.
3. The vortex finder is completely elastomer covered.
4. The molded liners for the inlet head, cylinder section, and conical sections have integral molded gaskets for sealing at the flanged joints. Molded liners and vulcanized linings are offered in gum rubber, polyurethane, nitrile rubber, butyl, Neoprene®, Viton®, Hypalon®, and other liner materials can be supplied.

Many of the molded elastomer liners are interchangeable with ceramic liners of silicon carbide or high purity alumina.

5. All metal housings are of cast or fabricated mild steel. Standard housings are for system pressures up to 25 psi, and special designs are available for higher system pressures.

Figure 4-52B. Liquid-solids cyclone fabricated to resist corrosion and abrasion. By permission, Krebs Engineers.

such as molded rubber and elastomers; for example butyl, Hycar®, Hypalon®, urethane, and metal alloys, silicon carbide, alumina ceramics. These units require little or no maintenance.

The manufacturer requires complete solids and/or liquids data, feed size analysis, and requirements for separation. In some instances, it may be best to have a sample tested by the manufacturer in their laboratory.

References [44,62] give good performance analysis of these designs.

Solid Particles in Gas/Vapor or Liquid Streams

The removal of solid particles from gas/vapor or liquid streams can be accomplished by several techniques, some handling the flow "dry," others wetting the stream to settle/agglomerate the solids (or even dissolve) and remove the liquid phase from the system with the solid particles. Some techniques are more adaptable to certain industries than others. Figure 4-54 illustrates typical ranges of particle size removal of various types of common equipment or technique. All of these will not be covered in this chapter. Attention will be directed to the usual equipment associated with the chemical/petrochemical industries.

Inertial Centrifugal Separators

Specification Sheet, Figure 4-55, can be used as a guide in summarizing and specifying conditions for this type of equipment.

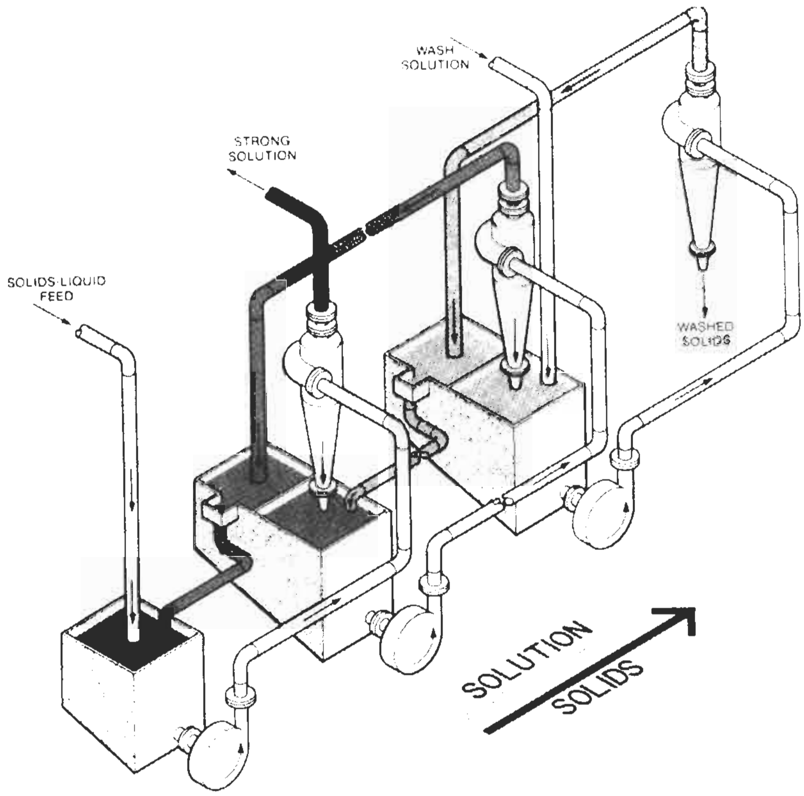


Figure 4-53. Cyclones used for countercurrent washing system. By permission, Krebs Engineers.

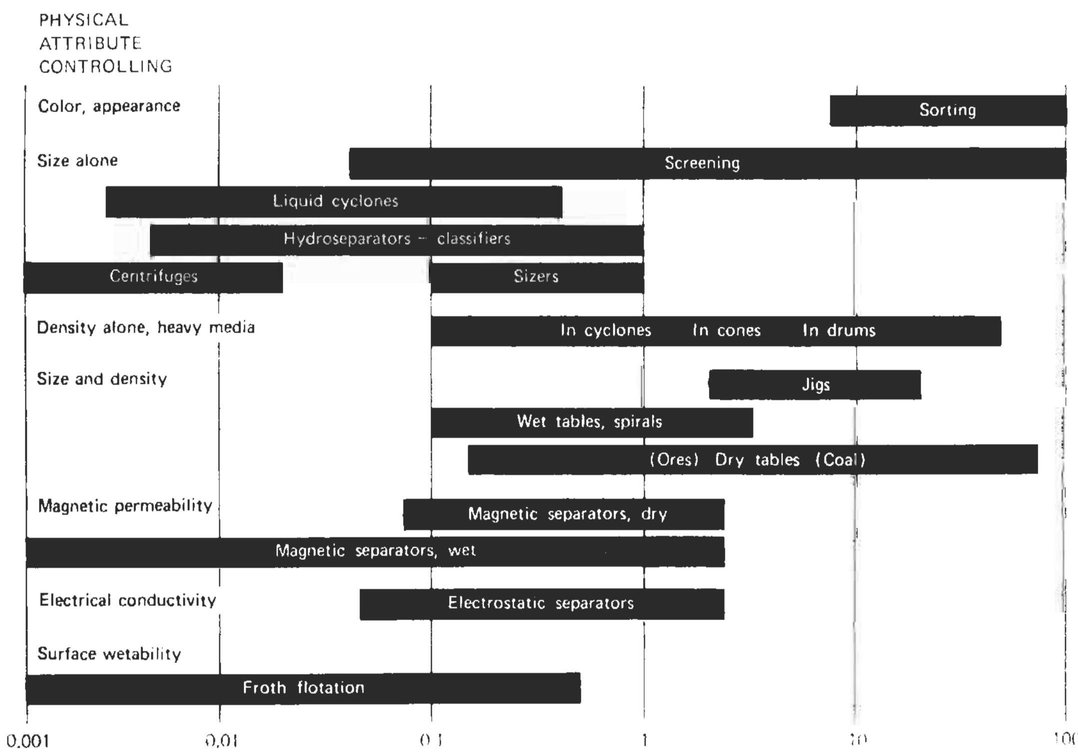


Figure 4-54. Size ranges where particular solid-solid/solid-liquid separation techniques can be applied. By permission, Roberts, E. J. *et. al.*, *Chemical Engineering*, June 29, 1970 [35].

Specification Sheet

Gas Phase Centrifugal Entrainment Separator

(Liquid or Solid Particles)

1. Application: (Describe service application of unit when possible) ______

2. Fluid Stream: ______ Composition: (Vol. %) ______

3. Entrained Particles: (Liquid or solid) ______

   Composition

   a. Size range ______ microns (or Mesh) ______

   b. Size percentage distribution: ______

   c. True Specific Gravity ______ (of particle), referred to water = 1.0

   d. Bulk density ______ of particle

   e. Source of entrainment: (Boiling liquid, kiln dust, etc.) ______

4. Operating Conditions:

	Maximum	Minimum	Normal
Gas Flow rate	______	______	______
Entrained Flow Rate	______	______	______
Temperature, °F.	______	______	______
Pressure, PSI	______	______	______
Moisture Content	______	______	______
Dew Point, °F.	______	______	______

5. Installation Altitude: ______

   a. Normal barometer ______ mm Hg.

6. Nature of entrained material: ______

   Solids: (a) Describe (dry, moist, sticky, at operating conditions) ______

   (b) Hygroscopic: ______

   (c) Angle of repose ______

   Liquid: (a) Describe: (Corrosive, oily) ______

   (b) Surface tension at operating conditions: ______

   (c) Viscosity at operating conditions: ______

7. Insulation required: ______ Reason ______

8. Construction Features:

   (a) Describe separator location in system ______

   (b) Indoors, outdoors, inside another vessel (Provide sketch if possible) ______

   (c) Storage required for collected dust or liquid ______ (hours)

   (d) Preliminary size inlet connection: ______ inches, (diam., rect., sq.) ______

   (e) Type of dust removal required ______

   (f) Suggested Materials of Construction shell: ______ internals ______

9. Special conditions: ______

______

Figure 4-55. Specification Sheet, gas phase centrifugal entrainment separator (liquid or solid particles).

There are a few mechanical arrangements that use external power to exert centrifugal force on the gas particle stream. The fan type blades direct the separating particles to the collection outlet. Figures 4-56 and 4-57 show such a unit. These units are compact and have been used in various dust applications. However, caution should be used to avoid installations involving sticky or tacky materials which might adhere to the walls and blades of the unit. The efficiency of these is about 90%–99%, similar to a small, high pressure drop cyclone. The air handling performance can be predicted using the fan laws.

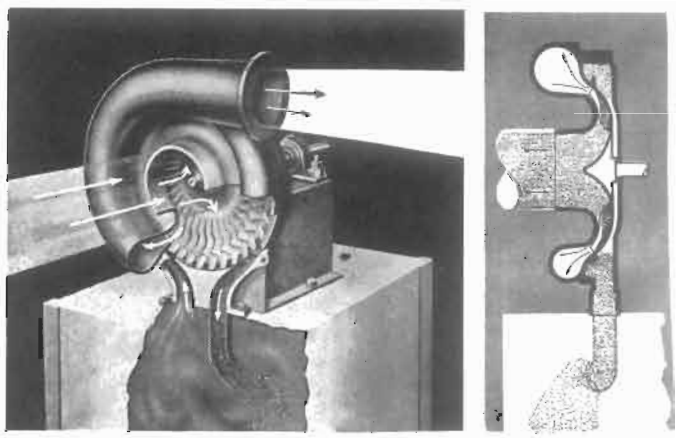


Figure 4-56. Inertial centrifugal dust separator. Courtesy of American Air Filter Co.

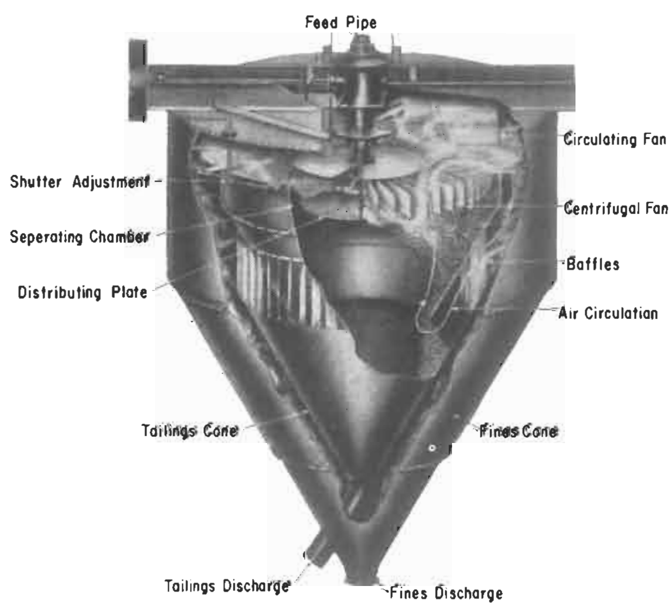


Figure 4-57. Inertial centrifugal dust separator. Courtesy of Universal Road Machinery Co.

Scrubbers

Scrubber separators use a liquid to form some type of liquid surface (spray droplets, film, etc.) to assist the internal arrangements of the separator in the separating action. Essentially the incoming dust or liquid particles are wet by the action of the liquid (usually water or oil) and are made larger and/or heavier and thus can be separated from the moving stream. There are many types and styles of units falling in this classification (see Figures 4-58 to 4-64. Reference [36] provides a good summary of manufacturers and their products for wet scrubbing.

One or more of the following mechanisms are employed in the separating action of the wet scrubbers.

1. Impingement—on internal parts.
2. Wetting—of particle to help agglomerate and prevent re-entrainment.
3. Diffusion—dust particles deposited on the liquid droplets. Predominant for the submicron and particles up to about 5μ.

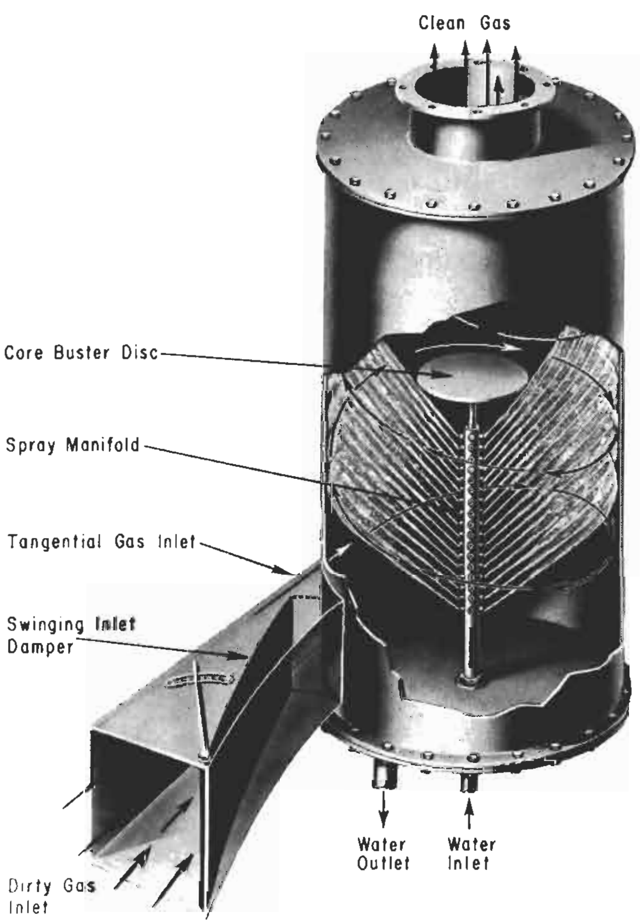


Figure 4-58. Cyclonic scrubber. Courtesy of Chemical Construction Corp.

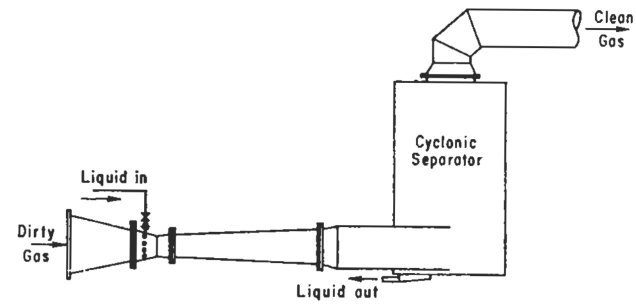


Figure 4-59. Venturi scrubber. Courtesy of Chemical Construction Corp.

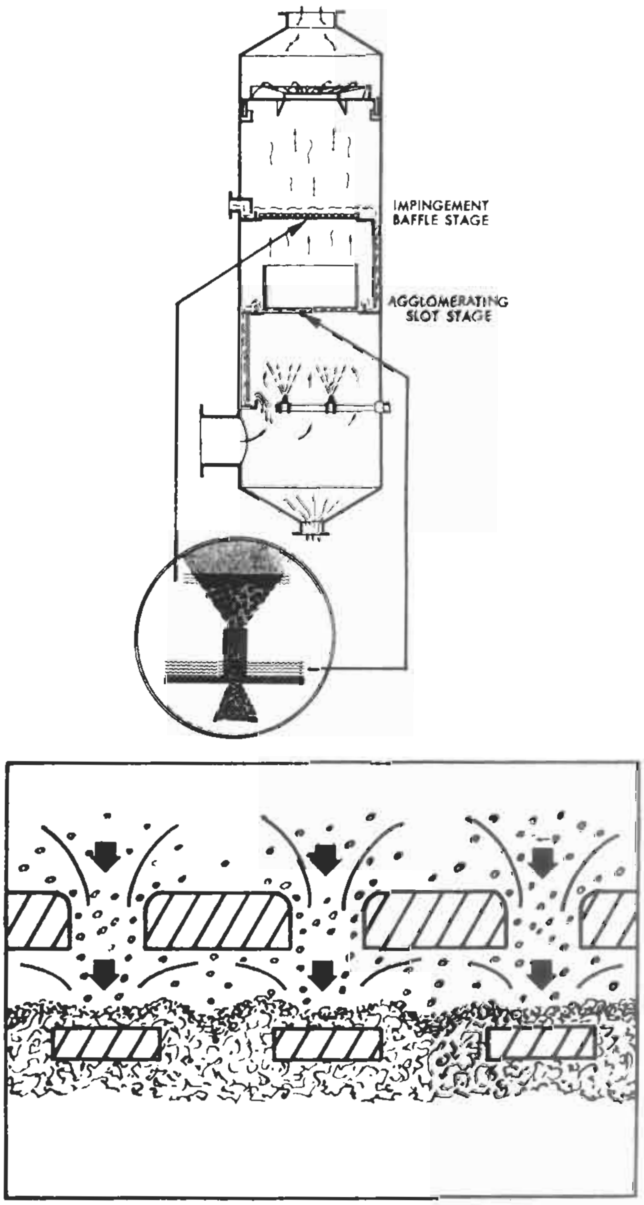


Figure 4-60. Impingement scrubber. Courtesy of Peabody Engineering Corp.

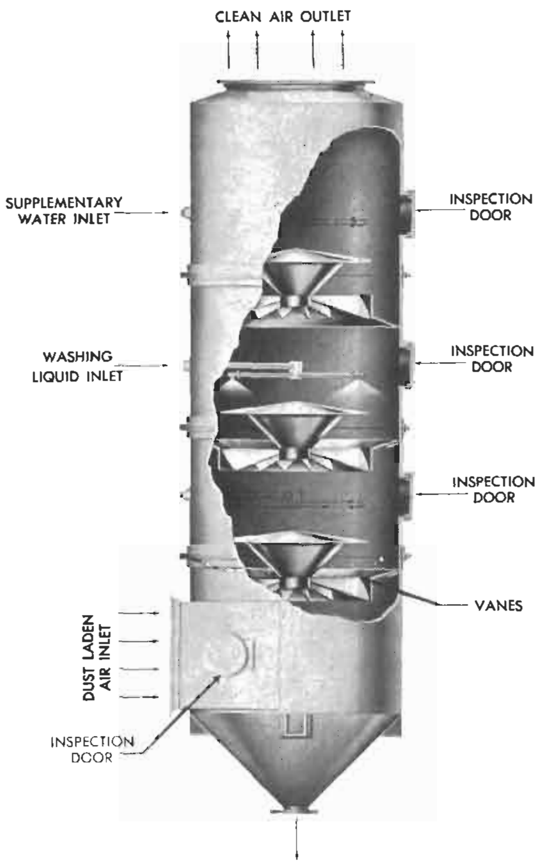


Figure 4-61. Spray scrubber. Courtesy of The Ducon Co., Inc.

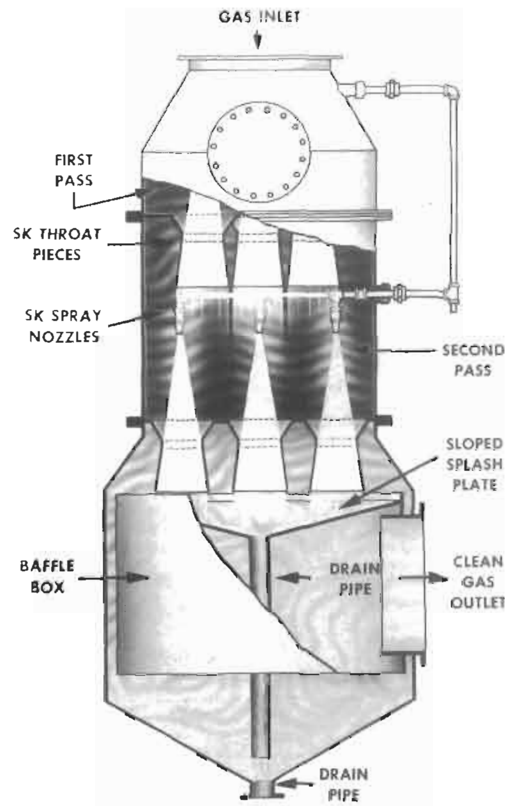


Figure 4-62. Spray scrubber—fume scrubber arranged for vertical downflow. Courtesy of Schutte & Koerting Co.

4. Humidification—aids in flocculation and agglomeration of particles.
5. Condensation—will cause particle size to grow if gas cooled below its dew-point.
6. Dust Disposal—running film action of liquid washes dust and collected liquid out of scrubber.
7. Gas Partition—segregates gas into small streams and segments when flowing through a liquid or foam.
8. Electrostatic Precipitation—the electrical charging of the liquid droplets may come about by the interaction of the gas and liquid streams. Not much known of this action.

The separating ability of most units is limited to 5-micron particles. However, some will take out 1 to 5μ particles at a sacrifice in collection efficiency. Due to the peculiarities of each system as well as the equipment available to perform the separation, it is well to consult manufacturers regarding expected performance. Quite often they will want to run test units, particularly on difficult separations. References [12,13] give good descriptions of the various types of equipment illustrated in Figures 4-58 to 4-64.

Figures 4-64 and -64A use a floating valve variable orifice opening as used in distillation contacting on the one or more trays included in the manufacturer's design. This provides for good contact to wet down the solid particles as well as scrub many water soluble gas/vapors in the incoming stream (such as chloride, sulfur, and nitrogen compounds). Heat and mass transfer can take place under these conditions. The pressure drop through this type unit typically ranges from 1 inch water to 2 inches of water for a five-fold increase in gas flow rates. Particle removal can go as low as 0.5 micron to greater than 30 microns. Usually a wire mesh entrainment pad is mounted in the outgoing "clean" vapors to knock out liquid entrained particles, not solids.

Cloth or Fabric Bag Separators or Filters

Reference [55] provides additional details beyond the bag filter applications, and Reference [60] provides a technical and analytical review of flowing gas-solids suspensions.

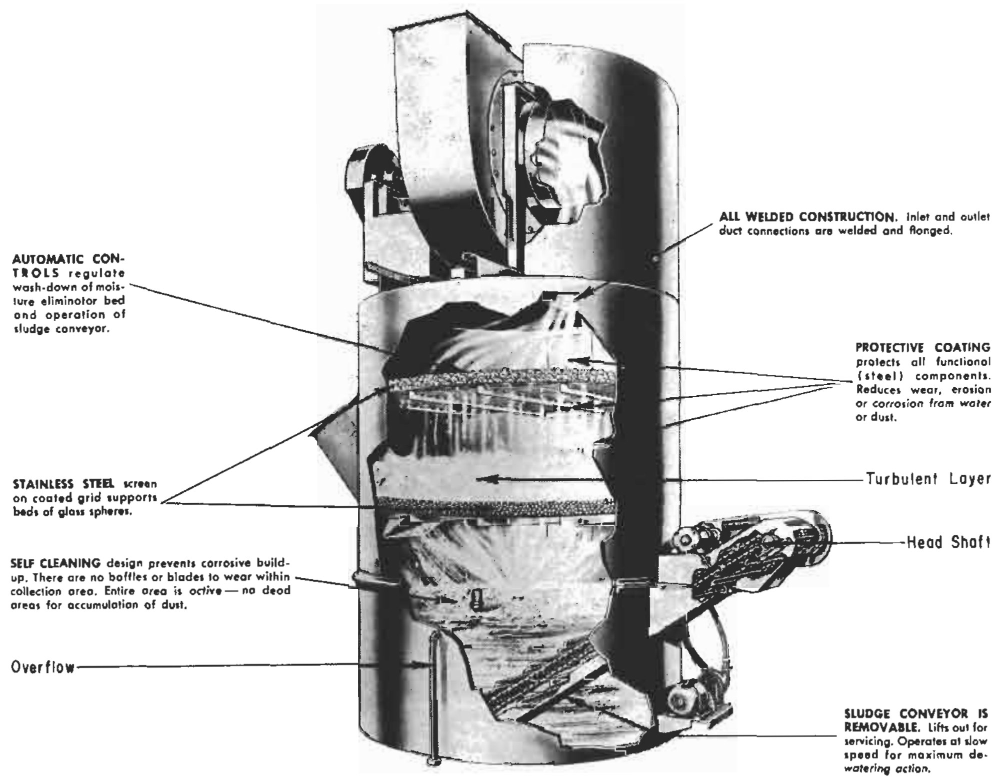


Figure 4-63. Tray-type scrubber with continuous sludge removal. Courtesy of National Dust Collector Corp.

Filters of this type or class may be of the large bag filter type for large volumes of low pressure dust laden gases or vapor, or of the generally smaller cartridge or pack types for gas/vapors or liquids containing suspended solid materials.

Figures 4-65, 4-66, and 4-67 show several units of the bag. The bags may be of cotton, wool, synthetic fiber, and glass or asbestos with temperature limits on such use as 180°F, 200°F, 275°F, 650°F respectively, except for unusual materials. (See Table 4-12A and B.) These units are used exclusively on dry solid particles in a gas stream, not being suitable for wet or moist applications. The gases pass through the woven filter cloth, depositing the dust on the surface. At intervals the unit is subject to a de-dusting action such as mechanical scraping, shaking or backflow of clean air or gas to remove the dust from the cloth. The dust settles to the lower section of the unit and is removed. The separation efficiency may be 99%+, but is dependent upon the system and nature of the particles. For extremely fine particles a precoat of dry dust similar to that used in some wet filtrations may be required before re-establishing the process gas-dust flow.

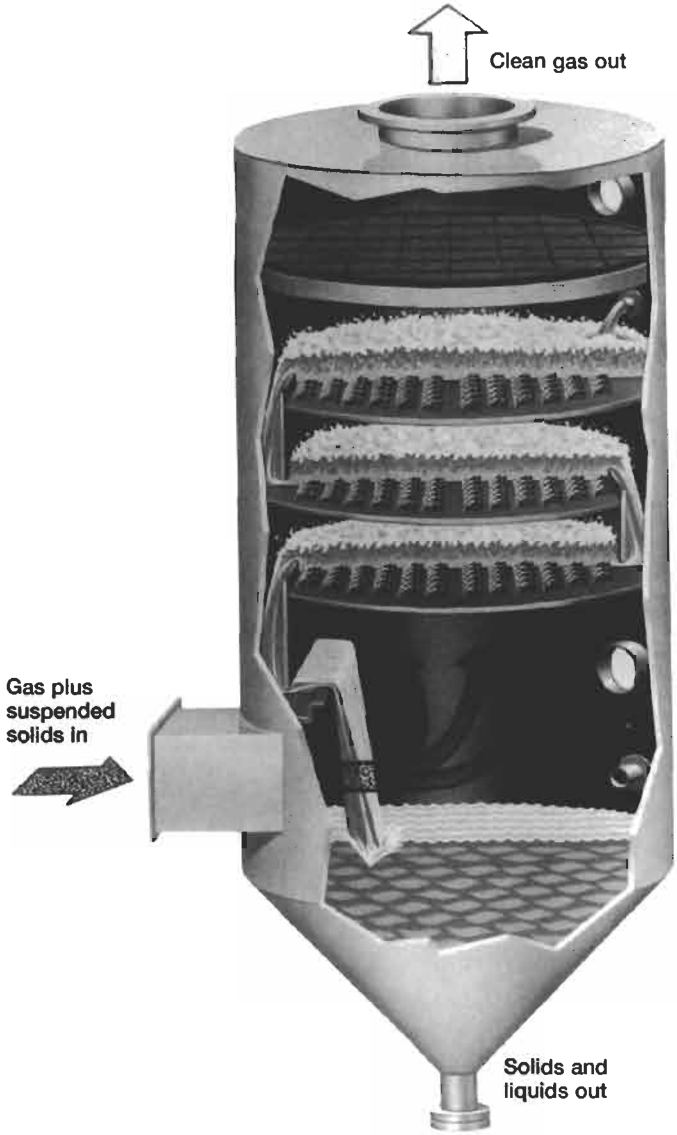
For heavy dust loads these units are often preceded by a dry cyclone or other separator to reduce the total load on the bags.

Suggested air-to-cloth ratios are given in Table 4-13.

Specifications

The details of specifications for bag filter dust collectors are important to a proper and operable design selection. There are many variables which must be furnished by the manufacturer so that the user can understand how the unit operates mechanically and the unit's dust loading capabilities. The larger the air/cloth ratio for the unit, the smaller will be its physical dimensions and generally, cost; however, the higher will be the frequency of cleaning. This can be quite troublesome, therefore low values of this ratio are preferable, consistent with the analysis of overall performance.

The removal or filtration of the entrained dust from the gas stream is accomplished by passing the mixture through a sufficiently porous fabric filter bag(s) (Table 4-14). These bags allow some air to flow through and are either cylindrical tubes or oblong tubes/bags. The dust is retained on the outside or inside (depending on unit design) of the bag surface and the small spaces between the fibers of the cloth (or felt). This dry cake builds up and acts as a pre-coat and then as the actual filtering medium as the dust particles build up. After a period of time, unique to the filter system of dust laden air plus bag type, the pressure drop will build up. (These are low pressure and low pressure drop systems.) Therefore, the dust or "cake" is removed (cleaned) from the outside of the



The scrubber is comprised of one or more trays. Each tray contains numerous venturi openings. Each of the MultiVenturi openings is surmounted by a spider cage holding a floating Flexicap (see insert). In addition, each tray is equipped with one or more "downcomers" and weir flow baffles that control the scrubbing liquid as it flows across the tray and then to the tray below.

Figure 4-64. Variable orifice MultiVenturi Flexitray® scrubber at essentially constant pressure drop maintains good efficiency over wide flow rates. By permission, Koch Engineering Co., Inc.

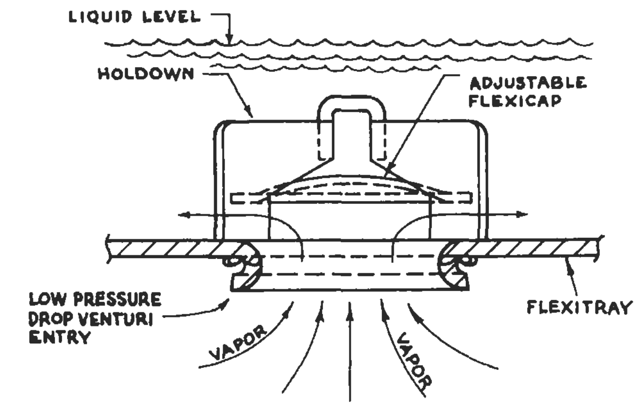


Figure 4-64A. Adjustable "floating" caps for vapor flow. By permission, Koch Engineering Co., Inc.

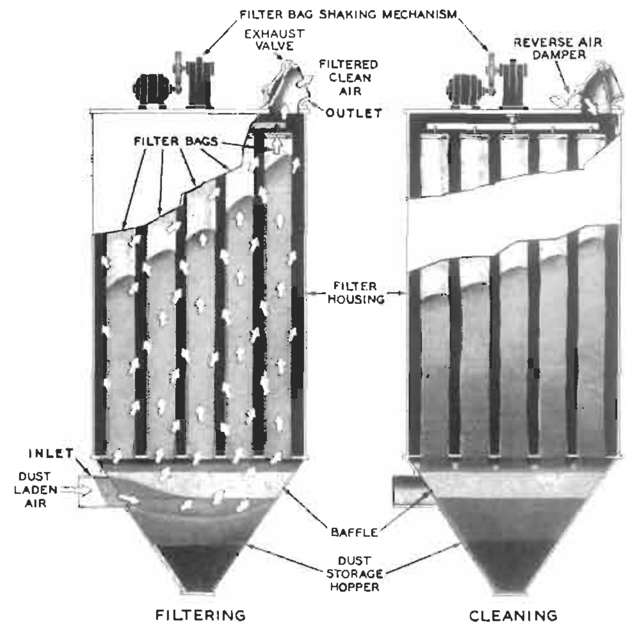


Figure 4-65. Bag filtration with mechanical shaking for bag cleaning. Courtesy of Dracco Div. Fuller Co.

bag by internal arrangements in the "bag house" or housing by such techniques as (1) shaking or vibrating the bag or bag assembly to drop the dust into an integral hopper while there is no-flow of air-dust feed into the unit or compartment, or (2) back pulse with jets of air into each bag (Figure 4-67). The criterion should be a constant pressure drop across the fabric for a fixed air flow and a specified dust loading [47]. The air-to-cloth ratio so often used is only useful when comparing a particular manufacturer's equipment for handling different materials, and not for comparing manufacturers. Reference [49] is an excellent summary of many details associated with specifying and selecting bag filters.

The following are suggested filter specifications:

1. Performance: define air/gas and dust rates, particle size distribution, and percent of particle sizes.
   a. Temperature at inlet to baghouse
   b. Moisture concentration, dewpoint

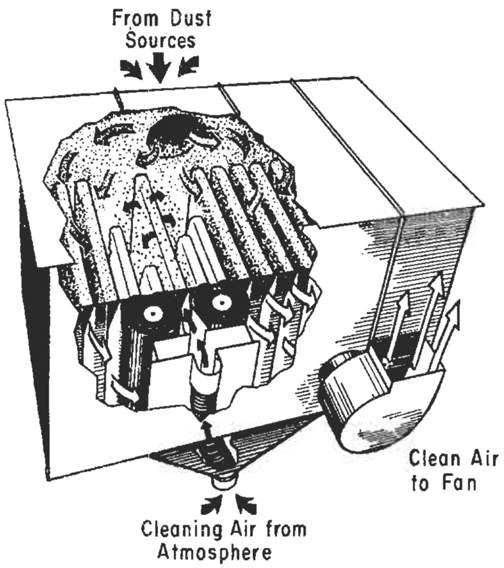


Figure 4-66. Bag filtration with continuous reverse air cleaning. Courtesy of W. W. Sly Mfg. Co.

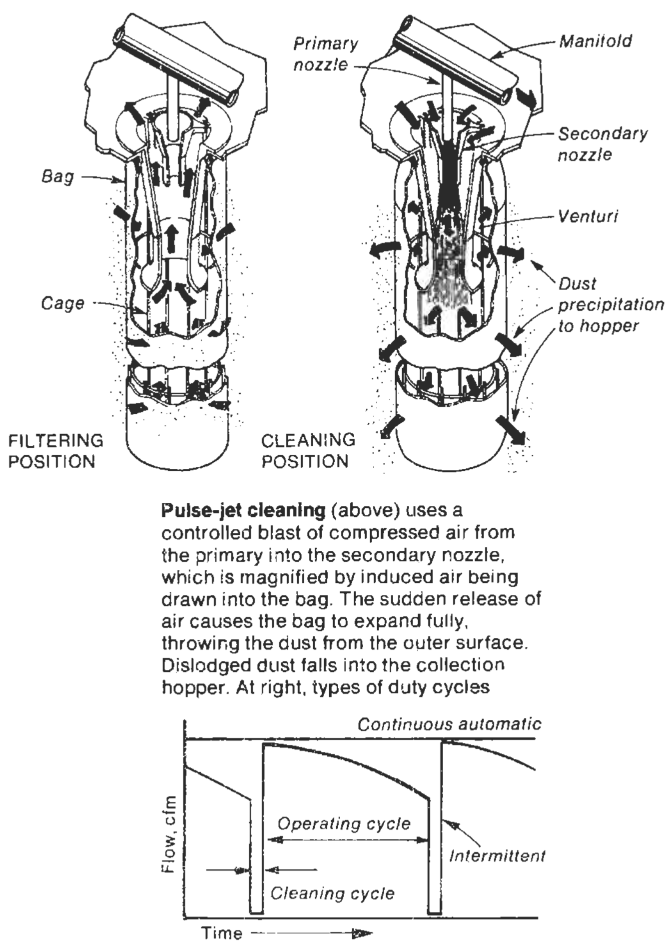


Figure 4-67. Pulse-jet air cleaning of fabric bags. By permission, *Power,* November 1975, McGraw-Hill Co., Inc., New York, p. 41.

c. Chemical composition of vapor and of dust, including any abrasive, hydroscopic or other characteristics.

2. Define dust recovery, as percentage below a certain particle size.

3. Indicate, if known, preferred bag material that will withstand environment, e.g., fibers of glass, polyester, Teflon®, Nomex®, polypropylene, polyethylene, cotton, wool, nylon, Orlon®, Dacron®, and Dynel®. The type of weave of fiber should be recommended by the manufacturer. The fabrics may be felted or woven [47,48] in weaves of plain, satin, or twill, and should be resistant to any corrosive material in the solid particles or the gas stream.

4. Manufacturer should recommend
   a. Bag size, (diameter, length).
   b. Bag holding hardware: anti-collapse spreader rings, snap rings, etc.
   c. Number of baghouse compartments, number of bags per compartment.
   d. Air/gas flow cycle to compartments.
   e. Complete description with mechanical details of bag cleaning system (shaking, air-jet, etc.)
   f. Dust removal arrangement.

The cleaning system set up for a particular bag house will determine whether the filtering system operates continuously or batch/intermittently. Some systems operate as a continuous batch, with sections of the entering chambers being isolated by valving to automatically switch from one section of one bag house to another. Thereby, one or more bag groups/sections filter while another one or more are not operating, but are in the dust removal cycle. The permeability of the fabric is generally stated as the clean airflow in (cu ft/min)/square foot of fabric at a pressure differential of 0.5 inches water as determined by the ASTM standard D-737 (Frazier Test) [47]. Whereas this test is useful, several fabrics may have the same permeability yet have different fiber surfaces, and thereby do not perform the same for a specific application.

The felted fabrics are generally used for maximum recovery of product and are used at high face velocity for airflow-to-cloth-area ratio. The felt promotes the greatest dust collection surface.

Monofilament fibers require special attention to ensure a uniform open space between the filaments.

Table 4-12A
Partial List of Commercial Crossflow Microfilter Media-materials and Geometries

Material	Geometries					
	Pleated sheet	Tubular	Spiral wound	Tubular MC*	Hollow fiber	Flat sheet
1. *Polymers*						
Cellulosics	■		■		■	■
Polysulfone		■	■		■	■
Polyvinylidene fluoride		■	■			
Acrylic					■	
Polytetrafluoroethylene		■				■
Polybenzimadazole			■			■
Polypropylene					■	■
Nylon	■				■	■
2. *Ceramics*						
Alumina				■		■
Zirconia/alumina				■		
Zirconia/sintered metal		■				
Zirconia/carbon		■				
Silica		■				
Silicon carbide				■		
3. *Sintered metal*						
Type 316 stainless steel		■				
Other alloys		■				

*MC=Multi-channel monolithic elements

By permission, Michaels, S. L. [38].

The woven fabrics have various yarn patterns for different spacings between the yarn fabrics (Table 4-14). There is a wide variety of choices for not only the materials of construction but the tightness of weave and the size of the yarn. All of these factors along with the others noted earlier, make the selection of bag fabric an art that requires manufacturer's and plant's actual field tests. Woven fabrics have a low ratio of weave openings for yarn area and generally have a limited face velocity for air flow of about 1.5 to 3.0 cu ft/min/sq ft [47].

Newer fabrics, not in common use but in development, test, and field trials, are described for higher temperature applications by Reference [50]. Application to 400°F—2100°F are potentially available using ceramic fibers Nextel 312®, laminated membrane of expanded PTFE on a substrate, polyimid fiber P-84, Ryton® polyphenylene sulfide, and woven fiberglass. The heat and acid resistance of these new materials varies, but one promise seems to be that higher temperatures will be handled.

New cartridge designs for bag houses will allow improved servicing and cleaning techniques.

It is important to keep bolts, nuts and other potentially loose items to a minimum inside the unit, as vibration from air/gas flow and bag cleaning can loosen nuts, break small welds, and ultimately tear holes or rip bags. The bag construction is likewise extremely important, since loose edges and "unlocked" seams will fray and tear, allowing fibers into the product dust. The bag construction must have straight seams in order for them to bend and flex properly on cleaning and/or loading.

Cartridge filters may be single units or clusters in a single container or canister. Figures 4-68 to 4-74 illustrate typical units. These may be designed to filter suspended

Table 4-12B
Partial List of Crossflow Microfilter Media in Chemical Service Applications

Chemicals TABLE II	Ceramics	Acrylics	Carbon	Poly—benzimadazole	Polysulfone	Polytetra-fluoroethylene	Polypropylene	Polyvinylidene fluoride	Other polymers	Sintered halide-resistant alloys	Sintered 316 stainless steel	Other sintered metals	Sintered chloride-resistant alloys
Alkanes, alkenes and aromatic hydrocarbons, below 100°C	■		■			■				■	■	■	■
Oxygen-containing organics, below 100°C	■		■			■	■	■		■	■	■	■
Chlorinated organics, below 100°C	■		■			■	■			■	■	■	■
Esters, below 100°C	■		■			■	■	■		■	■	■	■
Organics at 100 - 200°C	■		■			■				■	■	■	■
Organics at 200 - 600°C	■		■							■	■	■	■
Organics at 600 - 900°C	■		■										
Gases:													
Inert, or low reactivity	■		■	■	■	■	■	■		■	■	■	■
Oxygen	■												
Chlorine	■		■			■							■
Other reactive gases	■					■							
Aqueous solutions:													
pH = 3 - 7, no chlorides	■		■		■	■	■	■	■		■		
pH = 7 - 10, no chlorides	■		■		■	■	■	■	■		■		
pH = 0 - 3 (except HF)	■		■	■	■	■	■	■				■	
pH = 3 - 10, chlorides present	■		■		■	■	■	■	■				■
HF, with pH < 3			■			■	■	■		■			
pH = 10 - 13	■	■	■	■	■	■	■	■		■	■	■	■
pH > 13			■			■	■	■		■	■	■	■
Concentrated acids	■		■									■	
Steam (> 100°C)	■		■			■		■		■	■	■	■
Oxidants (e.g., bleach)	■		■		■	■	■	■		■	■	■	■

By permission, Michaels, S. L. [38].

Table 4-13
Suggested Air-to-Cloth Ratios for Dust Removal from Air*

Type of Dust	Ratio
Abrasives	2-2.5
Asbestos	2.5-3
Blast cleaning	3-3.5
Carbon	2-2.5
Cement (mills)	1.5-2
Cement (conveying and packing)	2-2.5
Clay	2-2.5
Coal	2-2.5
Feed	2.5-3
Graphite	1.5-2
Grinders	3-3.5
Gypsum	2-2.5
Lampblack	1.5-2
Limestone	2-2.5
Rubber	2-2.5
Salt	2.5-3
Sand	3-3.5
Silica Flour	2-2 5
Soap	2-2.5
Soapstone	2-2.5
Talc	2-2.5
Wood Flour	2-2.5

Ratio is the volume in cubic feet per minute of dust-laden air to each square foot of active cloth area. If grain loading is above normal, ratios must be reduced.

*By permission, Bulletin 104 The W. W. Sly Mfg. Co., Cleveland, Ohio

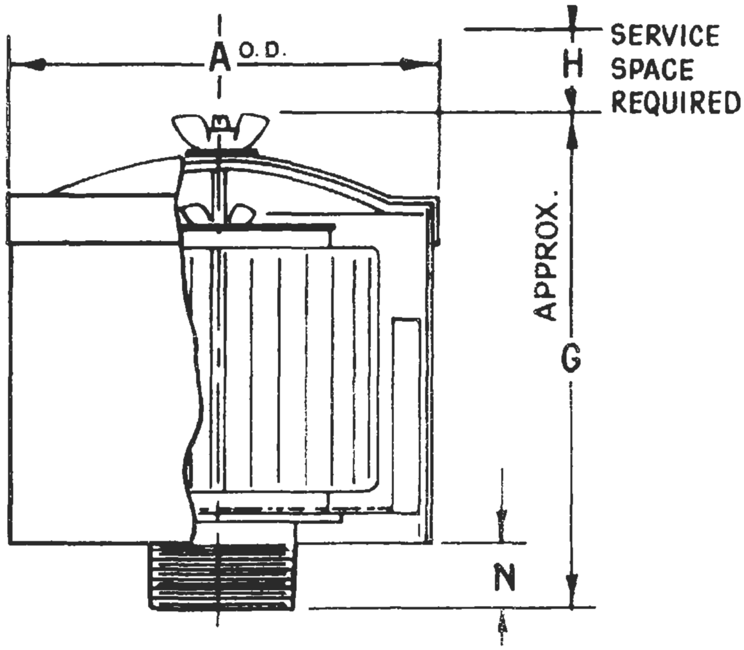


Figure 4-68. Typical blower intake filter-silencer. Air to blower leaves through pipe connection, which may be screwed or flanged. Courtesy of Dollinger Corp.

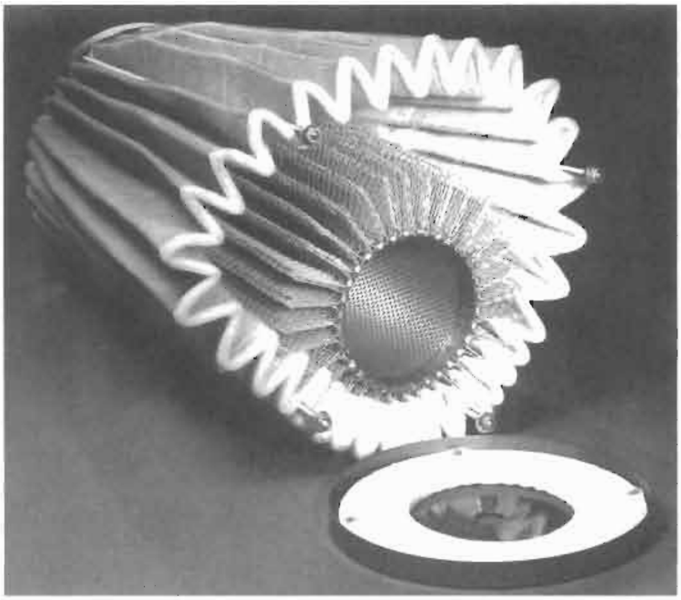


Figure 4-69. Pleated radial-fan filter cartridge. Filtration is from outside to inside. Courtesy of Dollinger Corp.

Figure 4-70. Wound filter tube on stainless steel core. Courtesy of Filterite Corp.

solids from gases or liquids. Table 4-14 presents representative physical property and application data for the more commonly used filter media. These media may be in filament, fiber, or "felt" form and arranged by weaving techniques to control the pore or free spaces to specific size for removal of various sizes of particles. The particle size retention listed in the table ranges from 0.006 micron to over 100 micron. A micron is often termed "micrometer" or a millionth of a meter, using symbol μm.

Filter cartridges as illustrated are considered "throwaway" and are removed from service when the pressure drop builds up to a predetermined value, or when the effluent changes color or becomes opaque with suspend-

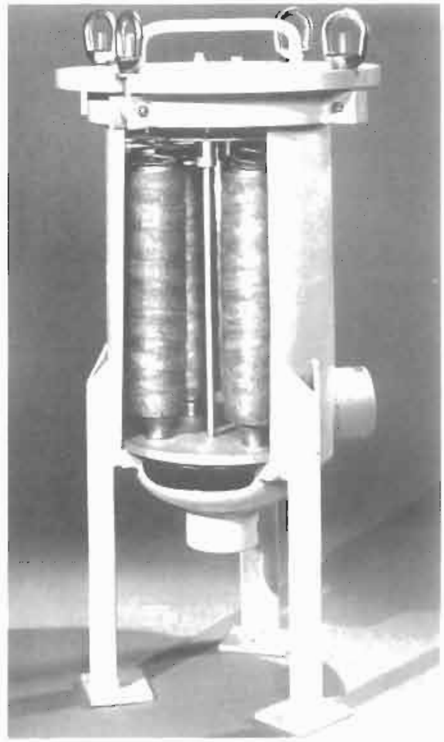


Figure 4-71. Cluster of filter cartridges in a single chamber. Courtesy of Filterite Corp.

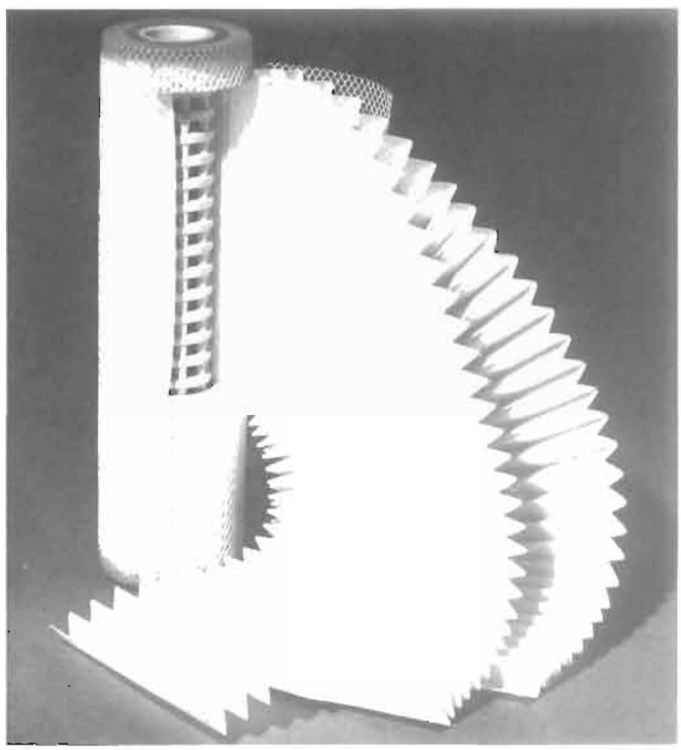
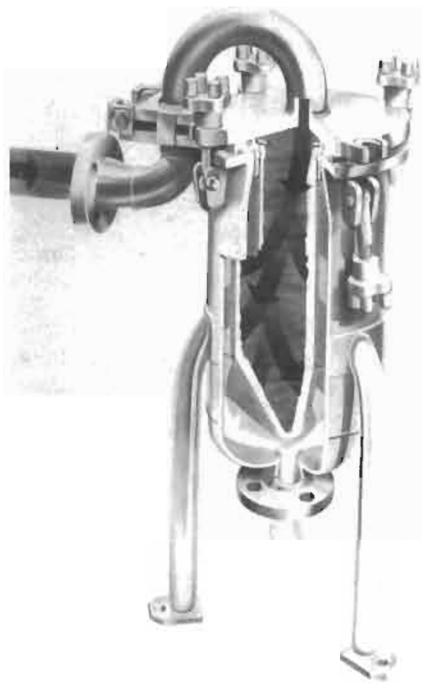


Figure 4-72. Cartridge-type filter-pleated membrane. Courtesy of Gelman Instrument Co.



This three-dimensional cutaway drawing illustrates the filtering operation of the GAF® filter-bag pressure filter system, showing the flow patterns of unfiltered liquid through a preselected micronrated felt filter bag which renders the desired quality of filtered product.

Figure 4-73. Flow scheme for GAF filter-bag pressure filter system for liquids. Courtesy of GAF Corporation, Chemical Group, Greenwich, Conn.

ed material breaking through. The flow in most applications is from outside cartridge to inside and into the hollow metal or plastic collection tubes. It then flows into the outlet pipe to the process. Materials for these cartridges are most commonly selected from cellulose, glass fibers, polypropylene (woven and non-woven) fibers, or monofilaments, molded resins, ceramics, or resin-impregnated fiberglass. The last three are termed "depth" filters, as they can hold a large amount of solids before the pressure drop builds up excessively. "Surface" filters are usually made of paper, non-woven fabrics, or cast membranes and are usually pleated to provide more working surface area. This type is fabricated from sheets of porous non-woven fabric often used for the absolute capture of sub-micron particles and has a sharp cutoff in particle size retention [37]. Yarn wound filters often have a graded-density or decreasing pore size structure.

To aid in selection of the most probable successful filter media for the service, the summaries of Table 4-12A and Table 4-12B can be a useful guide [38]; however, for some applications, actual testing in the plant using plant fluid streams can be the most conclusive. This plant testing is not necessary for every situation because the manufacturer has large data files to often aid in a good selection. Generally the ability to collect solids at low flow rates is greater for the wound filter.

Because the suspended particles are "captured" by different physical mechanisms depending on the particle size, shape, density, and concentration, all cartridges do not perform the same. The "capture" may be by (1) direct interception, (2) sieving, and/or (3) bridging [39]. (See Figure 4-75.) The cartridges from one manufacturer are generally consistent in performance; however, all cartridges from just any manufacturer may not be interchangeable in performance.

The micron ratings of a cartridge are intended to indicate the smallest particle that will be retained by the pores of the filter element. Often a "rough-cut" pre-filter is installed ahead of a final or "polishing" filter in order to increase the life of the final unit. Unfortunately, the method for determining the micron rating is not a universal standard between manufacturers. Thus, one manufacturer's "50 micron" filter may not perform the same as another manufacturer's with the same rating number. The only reliable approach is to send the manufacturer an actual sample of the fluid and let him test it to select the filter to do your job, or actually test the unit in your plant's field application [37].

An important feature of these cartridge units is the mechanism for assembling one or more in the housing. The top/bottom sealing mechanisms determine what style of cartridge is required (open both ends, open one end) and the method of pressure loading/sealing each cartridge into its bracket in the housing. The housing may hold one or 40 cartridges, and the assembly inside to prevent leakage and cross-contamination is essential to good performance as a filtering device. The housings can be made of various metals (carbon steel, stainless, alloy) or plastic-lined steel using corrosion resistant polymers, or elastomers, or solid plastic.

The cartridges can be selected to be useful over the range of low to high viscosities, that is, 100,000 cp with temperature ranges to 750°F at higher pressure of up to 3,000 psi [38]. Usually for the average application, the concentration of the suspended solids is not over 100 ppm, but can be higher. These units do not perform well with pressure pulsations or surges in the system. Note the differences in expected performance of Figure 4-76 between a pleated cartridge. This does not necessarily mean that *all* cartridges perform in this manner, but these are typical of expected performance curves. When examining particle retention ratings, examine Reference [39].

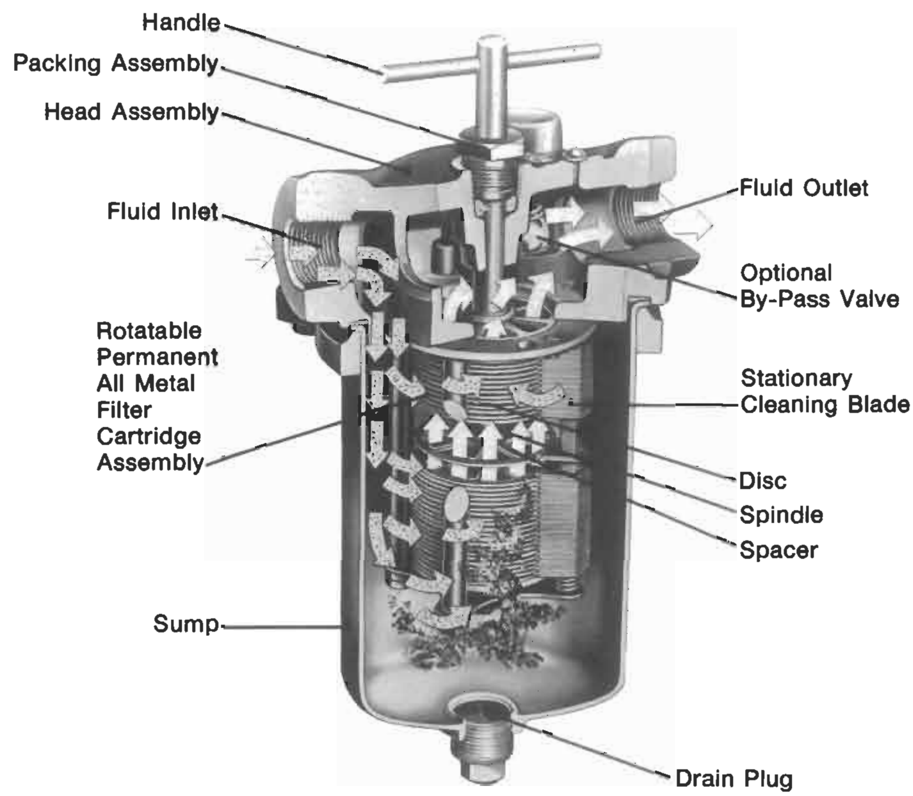


Figure 4-74. Edge-type filter with the external on-line cleaning. Courtesy of AMF Corp., Cuno Div.

Table 4-14
Physical Properties of Filter Media

	Cotton	Wool	Glass	Polyester	Polypropylene	Nylon	Nomex (high temp. Nylon)	Rayon	Dynel	Teflon	Paper	Sintered Metal	Woven Wire Cloth	Porous Ceramics
Particle Retention	2μ to >100μ	2μ to >100μ	3μ to >100μ	2μ to >100μ	2μ to >100μ	2μ to >100μ	2μ to >100μ	2μ to >100μ	2μ to >100μ	2μ to >100μ	3μ to 100μ	1μ to 40μ	2μ to >500μ	1μ to >40μ
Contaminant Holding Ability	G to E	E	F to G	G to E	F to E	F to E	F to E	F to E	F to E	G to E	G	F to G	F to G	F to G
Permeability	G	G	F	G	G	G	G	G	G	G	G	F	G	F
Chemical Compatibility	F	F	G	G	E	G	G	F	E	E	P	E	E	E
Temperature Limits (°F)	200	200	700	300	200	250	450	200	200	450	200	1200	1200	2000
Strength	G	F	G	G	G	G	G	F	G	G	P	E	E	G
Abrasion Resistance	G	G	P	E	E	E	E	G	G	E	F	E	E	G
Machineability (Workability)	E	E	F	E	E	E	E	G	E	E	F	F	F	F
Cleanability	G	G	P	G	E	G	G	G	E	E	P	P	G	P
Cost	L	L	M	M	L	M	M	L	L	H	L	H	M	H

Legend
P-poor F-fair G-good E-excellent L-low M-medium H-high

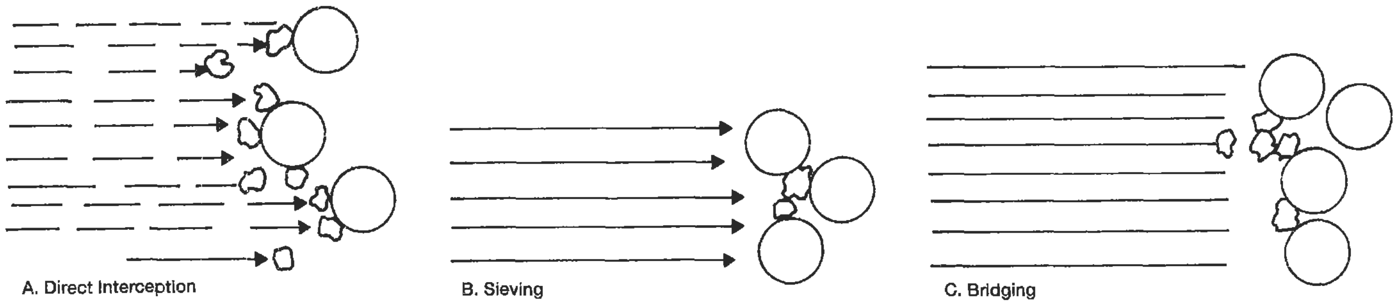


Figure 4-75. "Capture" mechanism for cartridge filters. Adapted by permission after Shucosky, A. C., *Chemical Engineering,* V. 95, No. 1, 1988, p. 72.

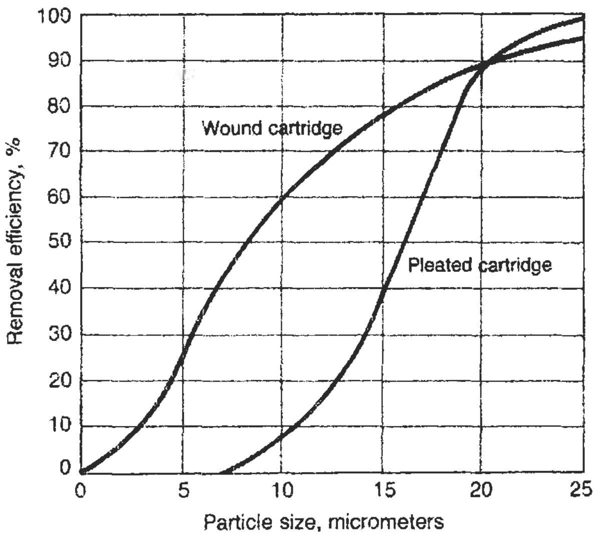


Figure 4-76. Pleated and wound cartridges differ in removal-efficiency profile. By permission, Shucosky, A. C., *Chemical Engineering,* V. 95, No. 1, 1988, p. 72.

Note: (a) Designations for both nominal and absolute ratings are based on the measure of a particle size, *not* a pore size. (b) Ratings are based on arbitrary laboratory tests by the filter manufacturer and can vary in actual plant conditions as previously discussed.

For some critical applications (such as polymer melt, beverage, or pharmaceutical filtration), it may be important to avoid cartridges that have a "nap" or "fuzz" on the fiber used, because these extremely fine fibers tend to break off and drift through the cartridge and go out with the finished product, thereby creating a visual acceptance problem, if not outright contamination.

In actual practice some companies have cartridges that will remove to 0.25 micrometer. Of course, the smaller the particle size that is specified to be removed from the vapor or liquid, the higher will normally be the ultimate pressure drop or lower the holding capacity. In normal operation, the pressure drop initially is quite low, perhaps 1 to 3 psig depending on flow rate, but as the solids build up, the pressure drop will rise to 10 to 35 psig, in which range most companies recommend replacement.

These replaceable cartridges or packs are the most commonly used; however, there are cartridges of wire mesh, sintered or porous metal which can be removed, cleaned, and replaced. Usually, the fine pores of the metal become progressively plugged and the cartridges lose capacity. They are often used for filtering hot fluids, or polymers with suspended particles, pharmaceuticals, and foods (liquids). In the case of polymers and other applications a special solvent and blow-back cleaning system may be employed.

The small cartridge units can be conveniently placed ahead of instruments, close-clearanced pumps, or a process to remove last indications of impurities in suspension.

Other useful cartridges are:

1. woven stainless steel (or other wire) wire screen mesh, Figure 4-77A and Figure 4-78
2. wire wound, Figure 4-77A
3. sintered metal, Figure 4-77B

The woven wire mesh type are formed to control the open space between the wires, thereby limiting the maximum size particle that can pass through. The cartridge is installed in cases or small vessels to facilitate quick replacement, or they can be arranged for backwash by use of proper piping connections. The wire wound units have consistent spaces for uniform particle size filtering.

The sintered metal units have uniform permeability with void spaces approximately 50% by volume for some metals and manufacturing techniques. The pore sizes can be graded to remove particles from 1 micron to 20 microns for liquids and smaller sizes when used in gaseous systems. (See Figure 4-77B.)

Nominal Rating Microns	Equivalent Standard Absolute Rating Microns
2	20
5	25
10	40
20	55
40	90

Figure 4-77A. Woven wire mesh filter cartridges. By permission, AMF Corp., Cuno Div., Catalog MP-20.1.

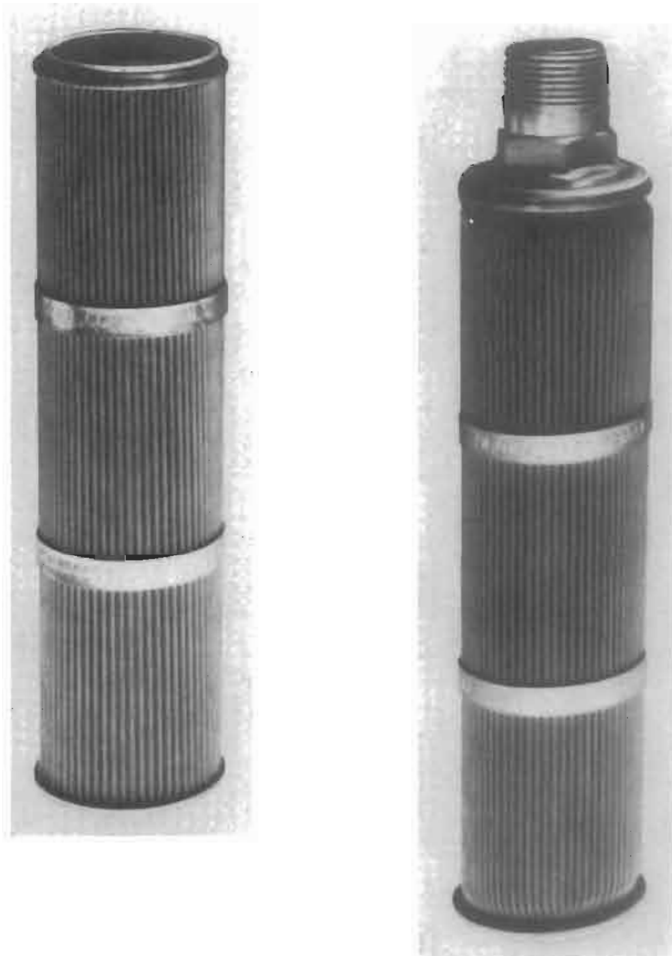


Figure 4-77B. Porous sintered metal filter elements. By permission, Pall Process Filtration Co.

Metals usually used are stainless steel, nickel, monel, inconel, high nickel alloys, and special designs for unique services.

The pressure drops for these units are typically low, ranging from 0.2 to 10–15 psi. The woven wire mesh runs even lower in pressure drops for the same or larger flow rates. Consult the manufacturers for specific application data.

With some types of particles the porous metal tends to plug, but they can usually be backwashed or washed with a solvent or acid/alkali to remove the particles from within the metal pores. This is one reason why manufacturer's testing or plant testing can be important to the proper selection. Once the internal pluggage has reached a point of reduction in flow-through capacity, it must be discarded. The actual cost of this type of cartridge is several times that of the non-metallic unit; therefore, the economics involving the life span of each unit should be examined.

Electrical Precipitators

The electrical precipitator is a dry dust or liquid mist removal unit which utilizes the ionization of the process gas (usually air) to impart electrical charges on the suspended entrained particles and effect particle collection by attraction to an oppositely charged plate or pipe. This type of unit is in use in services which are difficult for other types of entrainment removal equipment. Figures 4-79, 4-80, and 4-81 illustrate the usual fundamental action of these units.

For these units the usual particle size for removal is greater than 2 microns with a loading rate of greater than 0.1 grains/cu ft, with a collection efficiency of 99%±. The pressure drop is very low for a range of gas velocity through the unit of 100–600 ft/min [40].

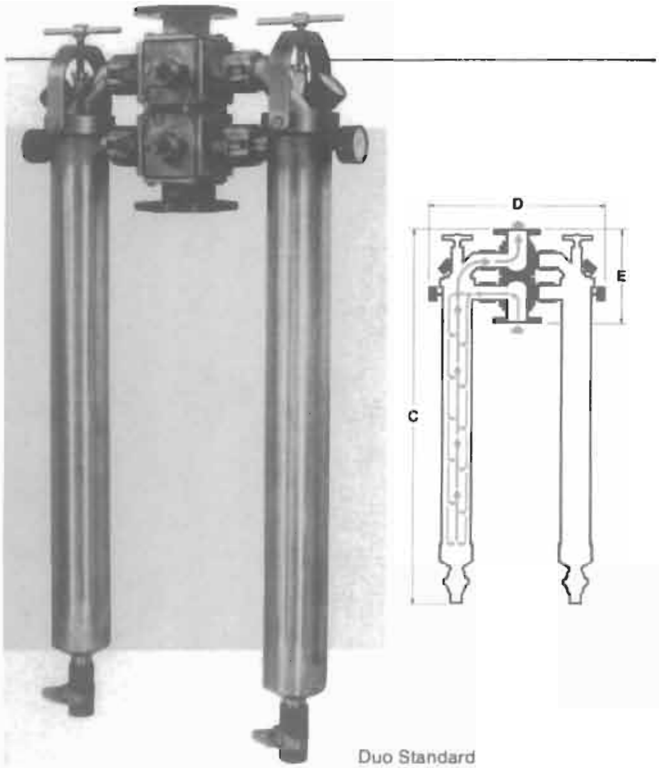


Figure 4-78. Tubular in-line pressure filter with reusable elements. The flow: unfiltered liquid enters the inlet port, flows upward, around, and through the media, which is a stainless steel or fabric screen reinforced by a perforated stainless steel backing. Filtered liquid discharges through the outlet (top) port. Because of outside-to-inside flow path, solids collect on the outside of the element so screens are easy to clean. By permission, Ronningen-Petter® Engineered Filter Systems, Bulletin RP-2.

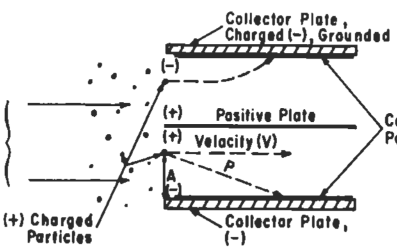


Figure 4-79. Charging particles in electrostatic precipitator. By permission, adapted after A. Nutting, American Air Filter Co.

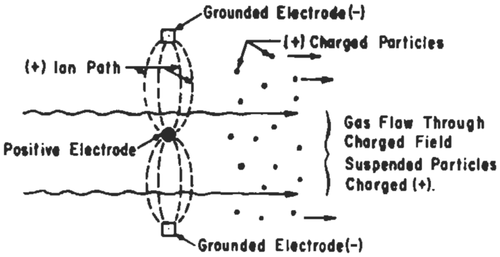


Figure 4-80. Particle collection. By permission, Nutting, A., American Air Filter Co.

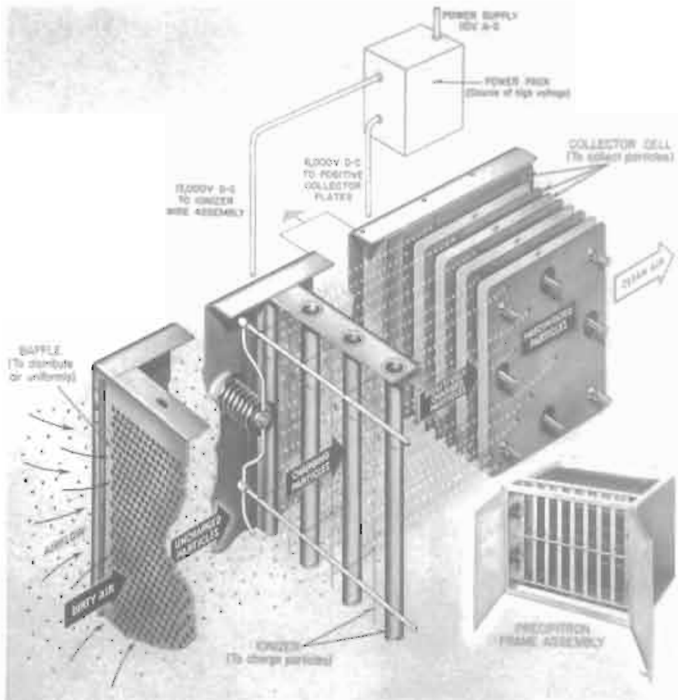


Figure 4-81. Electrical precipitator principle of operation. Courtesy of Sturtevant Div. Westinghouse Electric Corp.

Operating temperatures can be as high as 1000°F and above [41].

To improve the efficiency of collection, several units can be installed in series. The plate type unit is the most common design for dry dust removal, while pipe design is mainly for removal of liquid or sludge particles and volatilized fumes. The plates/pipes are the collecting electrodes, with the discharge electrodes suspended between the plates or suspended in the pipes [41,53,57].

In operation, the voltage difference between the discharge and collecting electrodes sets up a strong electrical field between them [63]. The "dirty" gas with particles passes through this field, and the gas ions from the discharge electrode attach to the suspended "dirty" particles, giving them a negative charge. The charged particles are then attracted to the positively charged collecting electrode, discharging their charge on contact, becoming electrically inert.

Collected liquids flow down the pipes and drain to a collection sump. Collected solids are washed off the plates with water or other liquid. Sometimes the dust/solids can be removed by mechanically vibrating or knocking on the plates while the particles are dry. The electrical power of the precipitator is applied only

(text continued on page 284)

Table 4-15
Precipitator Operating Data for Common CPI Applications

Precipitator operating data for common CPI applications	Ventilating				Grinding		Drying					
	Asphalt converter saturator	Glass melting	Pot line	Carbon plant	Cement finish grind	Jet pulverized catalyst	Cement dryers	Bauxite dryers	Gypsum dryers	Copper concentrator	Other drying uses	Dry process cement
Gas flow, 1,000 cfm.	5.0 to 24.8	7.0 to 10.0	94 to 120	15.5 to 27.0	10.0 to 13.5	10.0 to 13.5	40 to 80	100 to 150	11 to 43	39 to 45	20 to 187	62 to 305
Gas temperature, F	100 to 150	525 to 550	190 to 220	53 to 80	75 to 200	75 to 200	100 to 280	150 to 250	130 to 200	350 to 400	105 to 400	600 to 750
Gas pressure, in. water	−4 to −2	−6 to +6	Negative	Negative	−10 to +10	−10 to +10	−10 to +10	−10 to +10	Negative	—	−10 to +10	−4 to +4
Gas moisture, volume %	5	Ambient	2 to 3	Ambient	1 to 10	1 to 15	5 to 12	30 to 35	5 to 15	10 to 20	15 to 100	5.7 to 15.0
Inlet dust concentration, grs/cu ft	0.1 to 0.5	0.66 to 0.80	0.07 to 0.16	7.0 to 17.2	29 to 70	29 to 70	20 to 40	20 to 40	6.5 to 150	50 to 52	0.05 to 40	3.7 to 43.0
Outlet dust concentration, grs/cu ft	0.02 to 0.05	0.13 to 0.03	0.003 to 0.007	0.03 to 0.13	0.021 to 0.044	0.021 to 0.044	0.05 to 0.5	0.02 to 0.07	0.10 to 0.30	0.025 to 0.04	0.02 to 0.4	0.001 to 0.938
Power input, kw	3 to 7	7 to 12	10 to 14	3 to 7	4 to 6	4 to 6	4 to 12	15 to 20	7 to 12	21	7 to 24	2.4 to 51.0
Collection efficiency, %	90 to 98	98 to 98.5	86 to 97.6	98.5 to 99.8	95 to 99.96	95 to 99.96	90 to 99	99+	97 to 99.7	99.0 to 99.5	90 to 99	80 to 99.97

Calcining					Metals	Acid recovery		Miscellaneous							
Wet process cement	Gypsum kettles	Gypsum calciners	Alumina calciners	Mg $(OH)_2$ calciners	Nonferrous metals production	Sulfuric acid	Phosphoric acid	Shale oil	Phosphorus production	Acetylene	Waste incinerator	Titanium dioxide	Coal pyrolysis	Pickle liquor	Sulfur cleaning
157 to 346	3.0 to 42.9	7.6 to 82	25 to 135	27 to 115	3.6 to 600	15 to 102	13 to 43	12 to 15	10 to 37	42 to 46	32 to 270	37 to 45	83 to 90	32 to 36	15 to 20
352 to 630	200 to 300	250 to 375	250 to 720	400 to 750	120 to 800	95 to 170	68 to 170	140	520 to 750	100	500 to 700	650	220	750	1292
−4 to +4	−6 to −2	−6 to −2	−10 to +10	Negative	−10 to +10	−36 to 0	−5 to +5	10	-0.3 to +0.3	50	−10 to +10	Negative	Negative	2 psig	—
25 to 35	20 to 40	20 to 40	40 to 50	25 to 50	0.5 to 15	5 to 25	3 to 25	—	None	6	Various	27	—	26	21
9.9 to 53.0	5.0 to 48.0	5.0 to 48.0	100 to 150	2 to 17	0.6 to 30	20 to 200	40 to 100	40	1.5 to 15.0	1.0	1.0 to 4.0	0.9 to 1.2	2.5	13	—
0.006 to 0.08	0.011 to 0.92	0.011 to 0.92	0.04 to 0.10	0.02 to 0.3	0.015 to 0.8	0.5 to 1.0	0.8 to 5.0	0.2	0.02 to 0.04	0.001	0.02 to 0.08	0.009 to 0.012	0.12	0.08	—
20 to 94	5 to 10	5 to 10	14 to 36	13 to 30	7 to 30	8 to 40	8 to 16	14	15	14	14 to 50	14	20	12	14
98.8 to 99.93	95 to 99.94	95 to 99.94	99 to 99.96	98 to 99.5	90 to 99.6	95 to 99	90 to 98	99.5	99	99.9	90 to 99	99	95	99	99

By permission, Sickels, R. W. [41].

(text continued from page 281)

to the particles collected, thereby allowing for large volumes of gas to be handled with very low pressure drop.

For corrosive gases/liquid particles, corrosion resistant metals can be used for construction.

The performance of the unit involves the gas characteristics, analysis, velocity, flow rate, dust or liquid particle size and analysis, resistivity and required final particle efficiency of removal. Some particle materials of high electrical resistivity prevent proper electrical operation.

Table 4-15 illustrates some industrial application of electrostatic precipitators; however, it is not intended to be all inclusive.

Nomenclature

a = Specific surface area, sq ft/cu ft
a_e = Acceleration due to gravity, 32.2 ft/s^2 or 9.8 m/s^2
A = Area of segment of a circle, sq ft
or, A = Cross-sectional flow area, sq ft
A_b = Cross-sectional area at bottom of vessel occupied by continuous aqueous phase, sq ft
A_c = Cyclone inlet area = W_iH_c for cyclone with rectangular inlet, sq ft
A_I = Area of interface, assumes flat horizontal, sq ft
A_H = Cross-sectional area allocated to heavy phase, sq ft
A_L = Cross-sectional area allocated to light phase, sq ft
A_p = Area of particle projected on plane normal to direction of flow or motion, sq ft
A_t = Cross-sectional area at top of vessel occupied by continuous hydrocarbon phase, sq ft
ACFS = Actual flow at conditions, cu ft/sec
b_1 = Constant given in table
c = Volume fraction solids
C = Overall drag coefficient, dimensionless
D = Diameter of vessel, ft
D_b = See D_p, min
D_c = Cyclone diameter, ft
D_e = Cyclone gas exit duct diameter, ft
D_H = Hydraulic diameter, ft = 4 (flow area for phase in question/wetted perimeter); also, D_H in decanter design represents diameter for heavy phase, ft
D_L = Diameter for light phase, ft
D_p = Diameter of particle, ft or equivalent diameter of spherical particle, ft
D_{p-min} = Minimum diameter of particle that is completely collected, ft
D'_p = Diameter of particle, in. or mm
d = Droplet diameter, ft
f = Factor relating average velocity to maximum velocity
f_c = Friction factor, dimensionless
F = Flow rate of one phase, GPM
F_{aq} = Aqueous phase flow rate, GPM
F_{cv} = Cyclone friction loss, expressed as number of cyclone inlet velocity heads, based on A_c
F_d = Drag or resistance to motion of body in fluid, poundals
F_{hc} = Hydrocarbon phase flow rate, GPM
F_t = Total flow rate of both phases, GPM
g = g_c = g_L = Acceleration due to gravity, 32.2 ft/(sec)(sec)
h = Distance from center to given chord of a vessel, ft
h_b = Height of continuous aqueous phase in the bottom of the vessel, in.
h_c = Height of a segment of a circle, in.
h_t = Height of continuous hydrocarbon phase in the top of the vessel, in.
h_{vi} = Cyclone inlet velocity head, in. water
H = Height of a segment of a circle, ft
H_c = Height of rectangular cyclone inlet duct, ft
H_D = Height of dispersion band, ft
I = Width of interface, ft
k = K = Empirical proportionally constant for cyclone pressure drop or friction loss, dimensionless
K′ = Constant for stationary vane separators, based on design
K_m = Stokes-Cunningham correction factor, dimensionless
K_{me} = Proportionality factor in Stokes-Cunningham correction factor, dimensionless
k = Constant for wire mesh separators
l = Wire mesh thickness, ft
L = Length of vessel from hydrocarbon inlet to hydrocarbon outlet, or length of decanter, ft
L_l = Liquid entering Webre separator, lbs per minute per square foot of inlet pipe cross-section
L_v = Entrainment from Webre unit, lb liquid per minute per square foot of inlet pipe cross section
m = Exponent given by equations
m_p = Mass of particle, lb mass
n = Constant given in table
N_{Re} or (R_e) = Reynolds number, dimensionless (use consistent units)
N_t = Number of turns made by gas stream in a cyclone separator
ΔP = Pressure drop, lbs/sq in.
Δp = Pressure drop, in. water
Δp_D = Pressure drop, no entrainment, in. water
Δp_L = Pressure drop due to liquid load, in. water
Δp_T = Pressure drop, total across wet pad, in. water
Q_D = Dispensed phase volumetric flow rate, cu ft/sec
Q_H = Volumetric flow rate, heavy phase, cu ft/sec
Q_L = Volumetric flow rate, light phase, cu ft/sec
r = Vessel radius, ft
SpGr = Specific gravity of continuous phase at flow conditions
$SpGr_p$ = Specific gravity of settling particle at flow conditions
$\Delta SpGr$ = Difference in specific gravity of the particle and the surrounding fluid
t_{avg} = Average residence time based on liquid flow rate and vessel volume, min
t_{min} = Minimum residence time to allow particles to settle based on Stokes Law, min
u = Relative velocity between particle and main body of fluid, ft/sec
u_t = Terminal settling velocity determined by Stokes Law, of particle under action of gravity, ft/sec

u_{ts} = Terminal settling velocity as calculated from Stokes Law, ft/sec
$v = V_t$ = Terminal settling velocity, in./min
v_a = Average velocity of gas, ft/sec
v_{ag} = Terminal settling velocity of hydrocarbon droplets in aqueous phase in bottom of vessel, in./min
v_c = Velocity down flow channel for continuous phase, ft/sec
v_d = Terminal settling velocity of a droplet, ft/sec
v_{hc} = Terminal settling velocity of aqueous droplets in hydrocarbon phase in top of vessel, in./min
v_t = Terminal settling velocity of particle under action of gravity, ft/sec
v_{ts} = Terminal settling velocity of particle as calculated from Stokes Law, ft/sec
V = Velocity of gas or vapor entering, ft/min
V (separator) = Separator vapor velocity evaluated for the gas or vapor at flowing conditions, ft/sec
V′ = Vapor velocity entering unit, lbs, per minute per square foot of inlet pipe cross section
V_a = Maximum allowable vapor velocity across inlet face of mesh calculated by relation, ft/sec
V_{act} = Actual operating superficial gas velocity, ft/sec or ft/min, for wire mesh pad
V_D = Design vapor velocity (or selected design value), ft/sec
V_c = Cyclone inlet velocity, average, based on area A_c′ ft/sec
V_{max} = Calculated maximum allowable superficial gas velocity, ft/sec, or ft/min wire mesh pad
V_s = Superficial gas velocity, ft/sec
V_{sa} = Separator vapor velocity evaluated for air-water system, ft/sec
V_{set} = Active volume of settler occupied by one of the phases, cu ft
V_t = Settling velocity for single spherical particle, ft/s or m/s
V_{ts} = Settling velocity for hindered uniform spherical particle, ft/s or m/s
W_i = Width of rectangular cone inlet duct, ft
z_h = Heavy phase outlet dimensions of decanter measured from horizontal bottom, shown on Figure 4-12
z_i = Interface of decanter liquids measured from bottom, Figure 4-12
z_l = Light phase outlet measured from bottom of decanter, Figure 4-12

Subscripts

L, or l = Light phase
H, or h = Heavy phase
C, or c = Continuous phase
D, or d = Dispersed phase
l = Liquid
v = vapor or gas

Greek Symbols

ε = Void fraction of wire mesh, dimensionless
η = Fraction of dispersoid in swept volume collected on target
θ = Factor for establishing type of flow for decanters, Reference [32]
μ = Viscosity of surrounding fluid, cp, except where it is lb/(ft-sec)
μ_c = Viscosity of continuous phase, lb/(ft)(sec)
μ_H = Viscosity of heavy phase, lb/(ft)(sec)
μ_v = Viscosity of fluid, cp
μ_L = Viscosity of light phase, lb/ft sec
μ = Fluid viscosity, (lb mass)/(ft)(sec) = centipoise/1488
μm = Milli-micron = 0.001 millimeter
π = 3.1416
$\rho = \rho_d$ = Fluid density, or density of fluid in droplet, lb mass/cu ft
ρ_c = Density of fluid continuous phase, lb/cu ft
ρ_f = Density of fluid, lb/ft^3 or kg/m^3
ρ_L = Liquid density, lb/cu ft
ρ_d = Density of fluid continuous phase, lb/cu ft
ρ_L = Density of light phase fluid, lb/cu ft
ρ_p = Density of particle, lb/cu ft
$\rho_S = \rho_s$ = True density of particle, lb mass/cu ft
ρ_v = Vapor density, lb/cu ft

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Chapter 5

Mixing of Liquids

Mixing of fluids is necessary in many chemical processes. It may include mixing of liquid with liquid, gas with liquid, or solids with liquid. Agitation of these fluid masses does not necessarily imply any significant amount of actual intimate and homogeneous distribution of the fluids or particles, and for this reason mixing requires a definition of degree and/or purpose to properly define the desired state of the system.

In order for the mixing operation to accomplish the overall process requirement of this step in the system, it is necessary to establish which factors are significant for a mixing device that provides the required end result for the industrial application. Because the "art" of mixing is still not an exact science, it is really not practical for the design engineer to expect to totally design a mixer, that is, define its type, diameter, operating speed, and shape/type of impeller. Rather it is reasonable for the engineer to understand the mechanical and processing essentials and anticipated performance when dealing technically with a mixing equipment representative. For standard nomenclature See references [47, 48]. The technical performance and economics of various designs often need to be examined in order to make a good, cost-effective selection of the device that will be the "heart" of this step in a process. In some situations, particularly chemical reaction and/or mass transfer, it may be necessary to conduct test work to develop a sound basis for a larger scale industrial unit. In other cases, the needed data may be drawn from the public technical literature or a manufacturer's application files (see References [1, 4, 10, 11, 19, 20, 24, 25, 26, 27, 28, 29, 31, 33, 42, 43, 44, 45, 46, 47, 48]).

Mixer performance is often related in terms of the fluid velocity during agitation, total pumping capacity (flow of the fluid in the system) generated by one impeller, and the total flow in the tank (or sometimes as blending time or a solids-suspension criterion) [25].

Mixing applications often include one or more of the following [26]:

- bulk mixing
- chemical reaction
- heat transfer
- mass transfer
- phase interaction (suspending/dispersing)

Mixing is accomplished by the rotating action of an impeller in the continuous fluid. This action shears the fluid, setting up eddies which move through the body of the system. In general the fluid motion involves (a) the mass of the fluid over large distances and (b) the small scale eddy motion or turbulence which moves the fluid over short distances [21, 15].

The size and shape of the vessel to be used for the mixing operation is important in achieving the desired mixing results; therefore, this aspect of the design must accompany the actual mechanical mixer design/size selection.

The performance of mixers involves high volume or flow operations, or high head or shear operations. Many mixing processes utilize a combination of these two, although, surprisingly enough there are many which can have only high volume or only high head. Some operations listed in decreasing order of high volume requirements include: blending, heat transfer, solids suspension, solids dissolving, gas dispersion, liquid-liquid dispersion (immiscible), solid dispersion (high viscosity).

Impeller types usually used with mixing and listed in decreasing order of high volume ability (hence in increasing order of high head ability or requirement) are: paddle, turbine, propeller, sawtooth impeller or propeller, cut-out impeller disc (no blades), colloid mill.

Figures 5-1 and 5-2 are useful as guides in the *general* selection of mixing impellers and associated vessels. Note

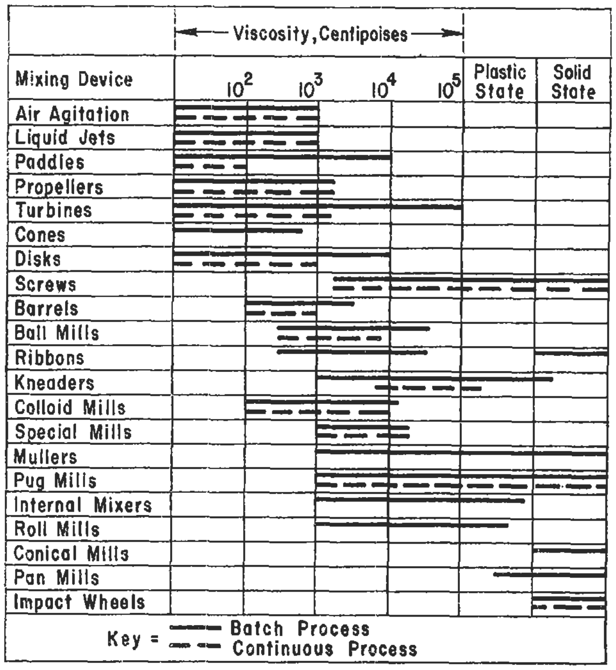


Figure 5-1. Range of operation of mixers. By permission, Quillen, C. S., *Chem. Engr.*, June 1954, p. 177 [15].

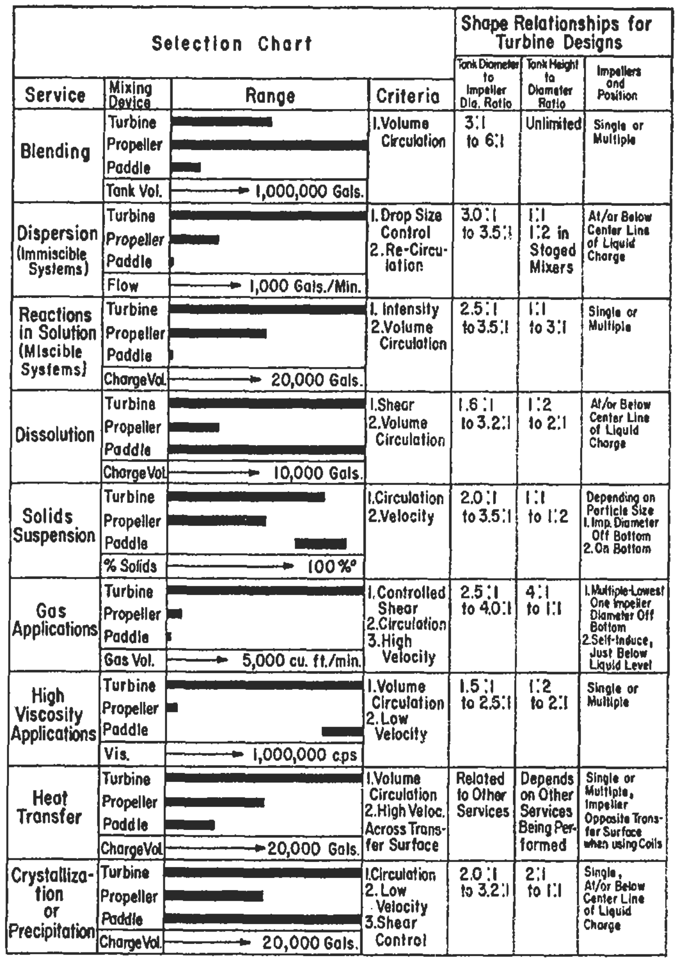


Figure 5-2. General selection chart for mixing. By permission, Lyons, E. J. and Parker, N. H., *Chem. Engr. Prog.*, V. 50, 1954, p. 629 [12].

that the shape relationships of Figure 5-2 are applicable to turbine type impellers only.

An example of the use of the chart occurs in the leaching of a 50% water slurry of a 20-mesh, 3.8-gravity ore by a dilute acid of equal volume, the heat of solution to be removed by cooling coils. The controlling factor is suspension of the solids to promote the reaction in which heat is developed. The criteria for solid suspension are circulation and liquid velocity sufficient to overcome the settling rate of the solids. The same criteria are also pertinent to good heat transfer and reaction. The large particle size and gravity difference between solids and solution suggest fast settlement. Best impeller position is therefore on the vessel bottom so that its radial discharge will sweep all solids up into the tank. In order to maintain maximum distribution of solids yet allow sufficient depth of liquid for the cooling coils, the maximum tank-height ratio of 1:1 from the chart would be used. Impeller ratio is regulated by reaction and suspension, with the latter controlling because of particle size. Tank depth and particle size in this case suggest a large impeller diameter, or a ratio of about 2.5:1. As the circulation pattern now established is radially across the bottom and up the sides, the slurry will flow up across and through a helical coil for good transfer rate. This pattern will be assured by four full vertical baffles mounted inside the cooling coil [12]. With this size information and reference to the horsepower charts, the preliminary design is complete.

All styles and designs of mixing impellers produce either an axial-flow or a radial-flow of the fluid during the impeller rotation. There are, of course, degrees of variation of each of these patterns, which then become a part of the selection and specifying process to achieve the mixing objective.

Axial flow impellers in an unbaffled tank will produce vortex swirling about the vertical shaft. This will be discussed later in more detail.

Mechanical Components

Figure 5-3 highlights the most commonly used radial and axial flow impeller styles for process applications. Other styles/designs are used for special specific applications to accomplish the mixing objectives (Figures 5-4 and

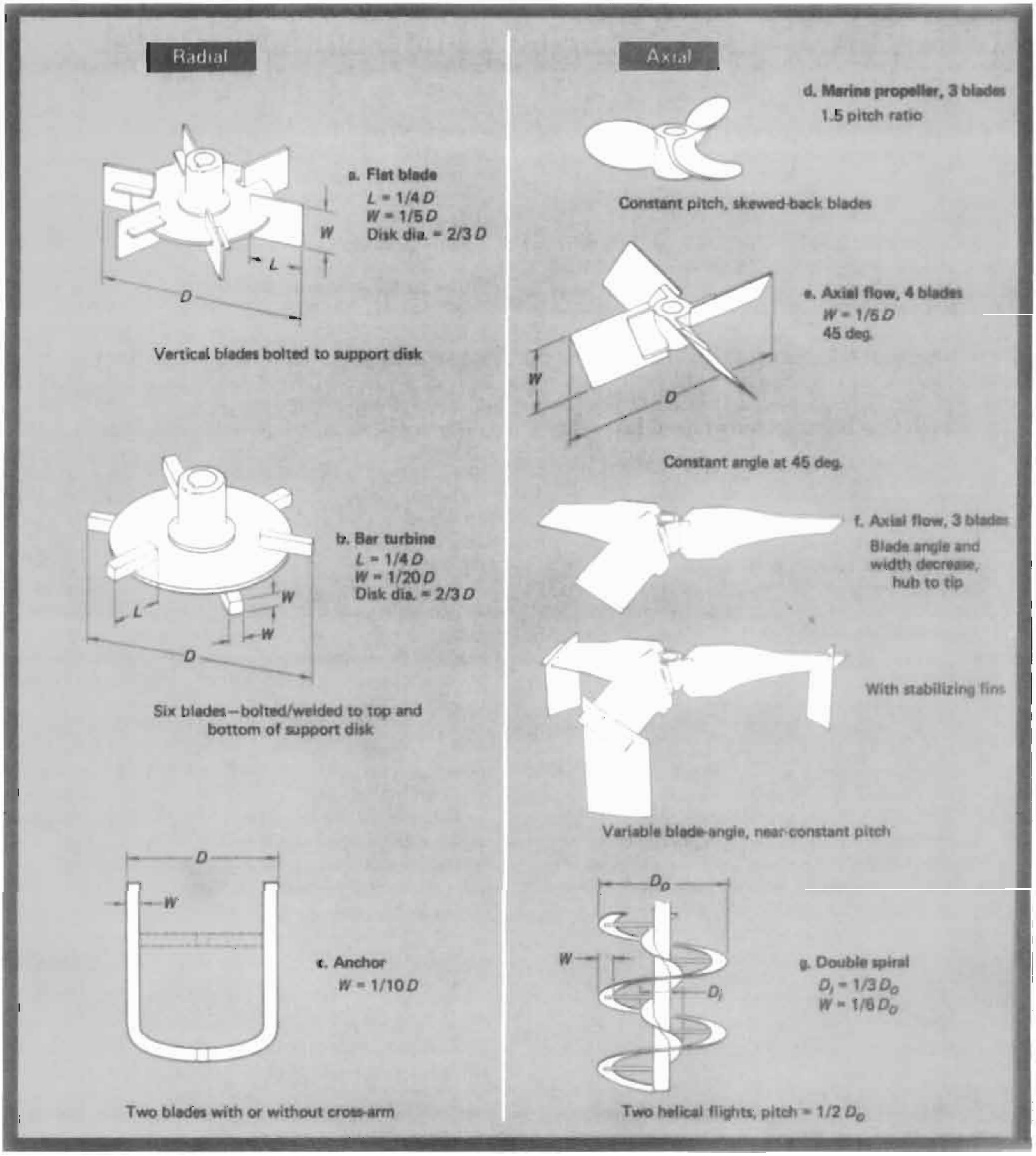


Figure 5-3. Impeller styles and general sizes commonly in use in process industry plant. By permission, Oldshue, J. Y., "Fluid Mixing Technology and Practice," *Chem. Engr.,* June 13, 1983, p. 84 [25].

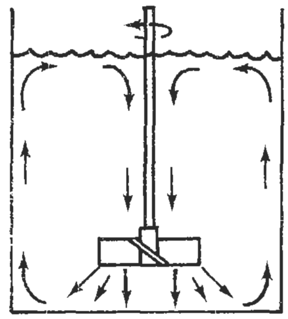


Figure 5-4A. Axial-flow pattern produced by a pitched-blade turbine. By permission, Oldshue, J. Y. [25].

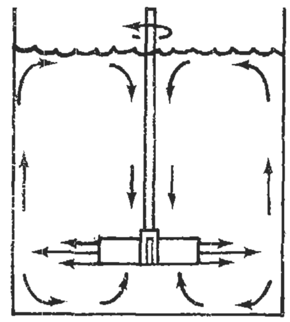


Figure 5-4B. Radial-flow pattern produced by a flat-blade turbine. By permission, Oldshue, J. Y. [25].

5-5), but Figure 5-3 has been found to cover most of the common applications. Fluid flow patterns for the axial and radial flows are shown in Figures 5-4A and B.

Impellers

General Types, Figure 5-5.

A. Propeller, Marine type
1. Circulates by axial flow parallel to the shaft and its flow pattern is modified by baffles, normally a downward flow.
2. Operates over wide speed range.
3. Can be pitched at various angles, most common is three blades on square pitch (pitch equal to diameter).
4. Shearing action very good at high speed, but not generally used for this purpose.
5. At low speed it is not easily destroyed.
6. Economical on power.
7. Generally self cleaning.
8. Relatively difficult to locate in vessels to obtain optimum performance.
9. Not effective in viscous liquids, unless special design.
10. Cost: moderate

B. Open Turbine: Radial
1. Circulates by radially directed centrifugal force using turbine blades. Circulation good for tank extremes; less danger of fluid short circuiting in tank.
2. Generally limited to a maximum speed, range may be narrow for some services.
3. Used for fairly high shear and turbulence.
4. Better than axial unit for tanks with cone bottom of greater than 15° angle, to lift material from bottom of cone and mix with bulk of liquid.
5. Effective in high viscosity systems.
6. Generally requires slower speeds and hence greater gear reduction than propeller, higher power per unit volume.
7. Cost: moderate.

C. Open Turbine: Axial Four Blades [25], Most Common Applications
1. Four-bladed 45° pitched blade, blade width is function of diameter.
2. Made in wide range of sizes for top entering mixers from 1 to 500 motor hp, and in diameters of 18 in. to 120 in.
3. Primarily for flow controlled requirements, such as solids suspensions, heat transfer, and other high pumping efficiency applications.
4. Preferred for shallow tanks of low Z/T with low liquid cover over turbine of 1.20% of turbine diameter.
5. Cost: moderate.

D. Open Turbine: Bar Turbine, Radial Flow
1. Produces highest shear rate of any basic impeller. Runs at high speed, uses lower torque.
2. Blade width and height normally 1/20 of impeller diameter.

E. Open Turbine: Axial, Three Blades [25]
1. Provides more flow and less shear than the four bladed design.
2. Produces nearly constant, uniform velocity across entire discharge area; has nearly constant pitch ratio.
3. Close to hub the blade angle is steeper, and blade is wider than at tip.
4. Size ranges 20 to 120 in. diameter for motors of 1 to 500 hp. Impeller speeds range from 56 to 125 rpm.
5. Large flow-directing stabilizer fins improve pumping capacity for viscosity ranges 500 to 1500 cen-

(text continued on page 294)

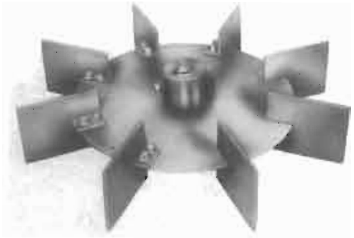


Figure 5-5A. A flat blade turbine can handle the majority of all fluid mixing applications when correctly applied. Its high pumping capacity makes it preferable for general mixing operations. It is well adapted to the application of protective coverings, such as lead, rubber and plastics. Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

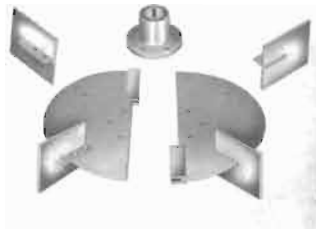


Figure 5-5B. Small openings are no problem with turbines. With blades removed, a turbine can pass through openings about ⅔ as large as the assembled impeller, or impeller can be split, as above, to pass through openings 1/3 of turbine diameter. Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

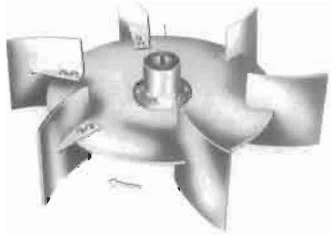


Figure 5-5C. Curved-blade turbine creates a dual suction flow pattern the same as the flat blade. This design is used when relatively low shear is a requirement, when abrasion must be considered, and when many other variables are of prime importance. Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

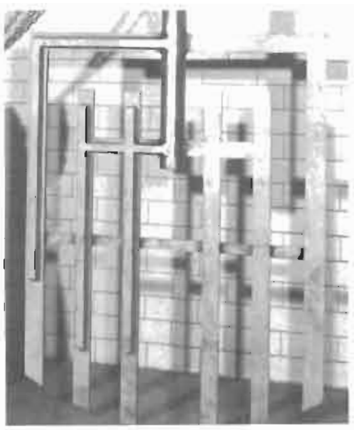


Figure 5-5D. Gate paddle impeller is designed for materials of high viscosity and operates at low shaft speeds. It is most desirable for shallow, wide tanks and wherever low shear is a requirement. Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

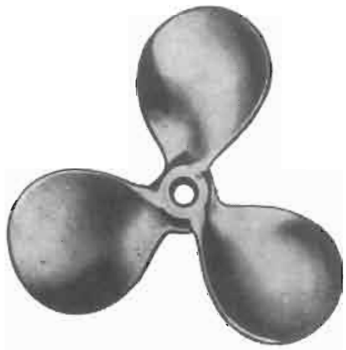


Figure 5-5E. Marine propeller is designed with extra section thickness to give longest life in corrosive or abrasive materials. It is polished to a high finish and accurately balanced. Many special propeller types and alloys available. Satisfactory in 95% of applications. It drives liquid ahead in a helical cone while doing considerable "work" on material passing through it. Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

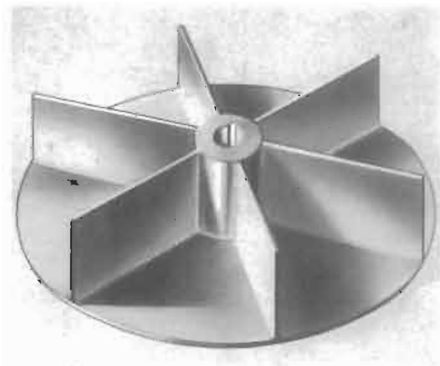


Figure 5-5F. Lifter turbine is efficient for pumping large volumes against static heads of less than 36 inches. As shown, it is used below a draft tube. Inverted, it is used above an orifice plane in tank bottom. Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

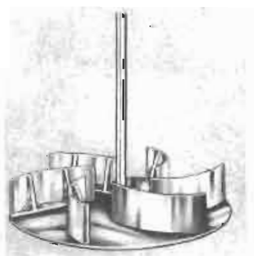


Figure 5-5G. Curved-blade turbine, developed especially for agitating fibrous materials such as paper stock. Also used on oil well drilling muds. This impeller gives fast, thorough turnover without need for the usual tank baffling or mid-feather construction. Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

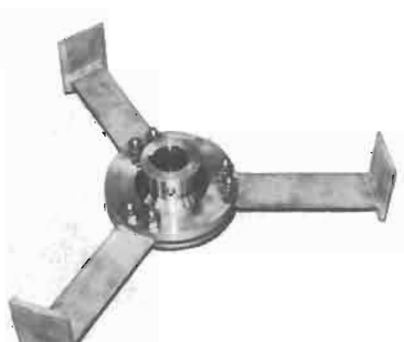


Figure 5-5H. A typical radial impeller agitator. Operates as agitating turbine or a conventional propeller, wide range of applications. Courtesy of Struthers-Wells Corp., Warren, Pa.

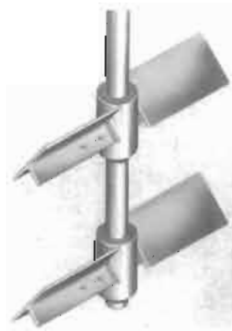


Figure 5-5I. Flat blade pitched paddle. A simple, low cost design that handles a wide variety of jobs. Operating at low speeds, it gives maximum pumping capacity with a minimum of turbulence. Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

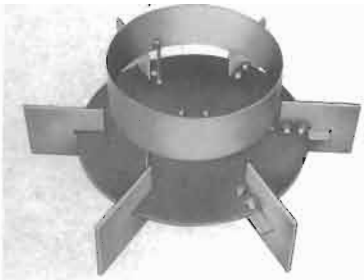


Figure 5-5J. Bottom of flat-blade turbine with stabilizing ring to prevent shaft whip. Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

Figure 5-5K. Plain cage beater. Imparts a cutting and beating action. It is usually combined with a standard propeller, which supplies movement in the mix. Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

Figure 5-5L. Studded cage beater. Enormous contact area gives extremely violent cutting and shredding action to certain emulsions, pulps, etc. Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

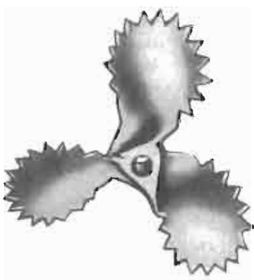


Figure 5-5M. Saw-toothed propeller. Displaces a large amount of liquid and combines a cutting and tearing action. Suitable for fibrous materials. Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

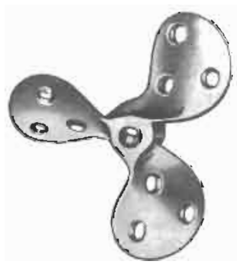


Figure 5-5N. Perforated propeller. Occasionally recommended for wetting dry powders, especially those that tend to form into lumps. Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

Figure 5-5O. Folding propeller. May be passed through a very small opening. Blades assume working position through centrifugal force only while rotating. Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

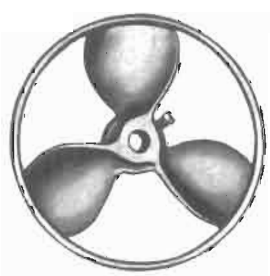


Figure 5-5P. Propeller with ring guard. For extra safety where sounding rods are used or where samples are taken by hand dipping. Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

Figure 5-5Q. Weedless propeller. Handles long fibrous materials that would become entangled in the ordinary propeller. Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

Figure 5-5R. Cut-out propeller. Displaces a small amount of liquid combined with a high rate of shear for shredding, breaking up pulps, etc. Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

(Figure 5-5 continued on next page)

(Figure 5-5 continued from previous page)

Figure 5-5S. Four-blade, vertical flat blade turbine impeller. Very versatile, one of the most used in wide application range. Courtesy of Philadelphia Gear Corp.

Figure 5-5T. Standard six-blade vertical curved blade turbine impeller. Gives good efficiency per unit of horsepower for suspensions, mixing fibrous materials. Gives high pumping capacity. Courtesy of Philadelphia Gear Corp.

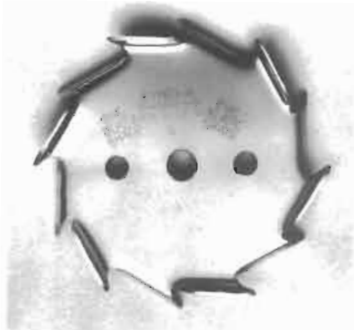


Figure 5-5X. Type R-500. Very high shear radial flow impeller for particle size reduction and uniform dispersion in liquids. By permission, Lightnin, (Formerly Mixing Equipment Co.) a unit of General Signal.

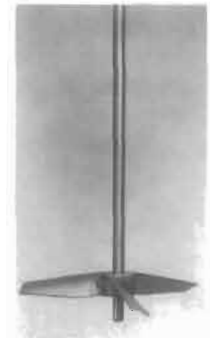


Figure 5-5Y. A-310 Impeller. Develops 50% more action than ordinary propellers and is geometrically similar for accurate scale-up. By permission, Lightnin, a unit of General Signal. (Formerly Mixing Equipment Co.)

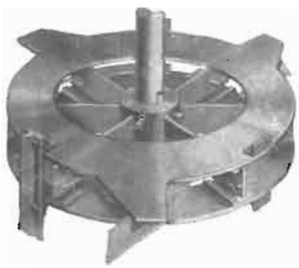


Figure 5-5U. Shrouded turbine for high pumping capacity. Usually used with low static heads, creates minimum of direct shear. Courtesy of International Process Equipment Co., Div. of Patterson Foundry and Machine Co.

Figure 5-5V. Turbine with three, four, or six radial blades. Handles wide range of applications. Courtesy of International Process Equipment Co., Div. of Patterson Foundry and Machine Co.

Figure 5-5Z. A-410 Composite Impeller. Strong axial flow at very high flow efficiency. Operates through a wide range of viscosities. By permission, Lightnin, (Formerly Mixing Equipment Co.) a unit of General Signal.

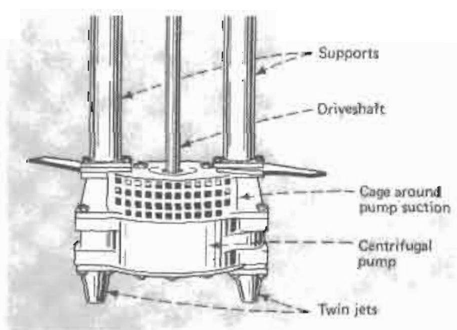


Figure 5-5W. Jet-flow mixer. Twin flow jets from submerged centrifugal pump allow for a maximum hydraulic shear per unit of power input, high velocities useful for thick slurries. By permission, Penny, W. R., *Chem. Eng.*, Mar. 22, 1971, p. 97 [31].

(text continued from page 291)

tipoise. Performs much like a marine propeller, but does not have some of its disadvantages.

6. Cost: moderate, less than marine propeller.

F. Double-Spiral Impeller [25]

1. Used for viscous materials; has inner and outer flights.
2. The inner flights pump down, and the outer flights pump upward.
3. Diameter of inner flight is one-third the impeller diameter.
4. Width of outer ribbon is one-sixth diameter of impeller.
5. Impeller height is equal to its diameter.
6. Available in 20 in. to 120 in. diameter for one to 250 motor hp, and speeds of 5.5 to 45 mph.
7. Cost: moderate

G. Shrouded Turbine
1. Circulates by radially directed centrifugal force using enclosed impeller stators. Circulation very good.
2. Speed range may be limited.
3. At reasonable speeds not easily destroyed.
4. Not self cleaning, fouls and plugs relatively easily.
5. Flow capacity limited, relatively low.
6. Effective in high viscosity systems.
7. Cost: relatively high.

H. Paddle
1. Circulates radially, but has no vertical circulation unless baffles used.
2. Covers wide viscosity range, blending.
3. Not easily destroyed in operation.
4. Not easily fouled.
5. Flow capacity can be high for multiple blades.
6. Cost: relatively low.

I. Anchor, Two blades, Contoured [25]
1. For higher viscosity applications: 40,000 to 50,000 centipoise.
2. Nominal blade width is impeller/10, with little power change from D/8 to D/12 (D = impeller diameter).
3. Power requirements vary directly with the impeller height-to-diameter ratio.
4. Used for blending and heat transfer for viscosities between 5,000 to 50,000 cp. Pumping capacity falls off above 50,000 cp, as it "bores a hole" in the fluid. Speed range 5.5 to 45 mph, for motor of 1 to 150 hp and impeller diameter 24 to 120 in.

J. Lifter Turbine, Figure 5-6A and Figure 5-6B

This type unit [29] is used for a combination of pumping and mixing purposes. The unit has a closed disk on the top side. The feed flow into the unit comes from directly below the rotating impeller. The performance is dependent on the size of the unit and the physical location with respect to the distance up from the bottom of the vessel. As this clearance increases, the head decreases for constant flow and increases the power requirement.

Figures 5-3 and 5-5 illustrate a few of the types of impellers used for mixing. They may be basically classified as axial, radial and mixed. In general the most generally applicable are the 3-bladed propeller, the flat-blade turbine, the curved blade turbine, and the paddle. The many other designs are either modifications of these or specially designed for a very special purpose with respect to a fluid system and/or its performance.

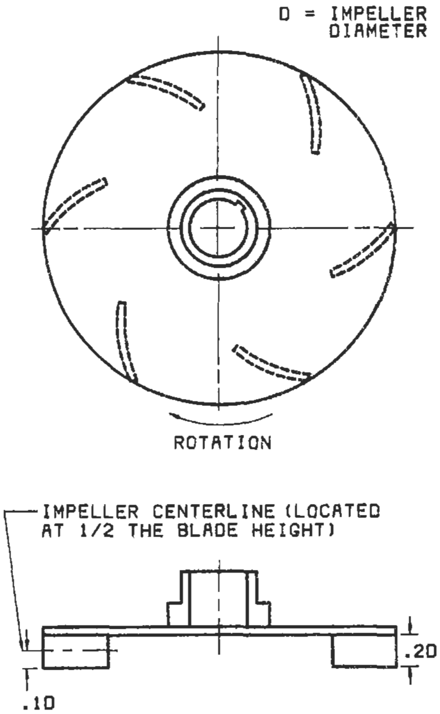


Figure 5-6A. Drawing of typical lifter turbine. By permission, Oldshue, J. Y., *Fluid Mixing Technology,* 1983, Chemical Engineering McGraw-Hill Publications Co. [29].

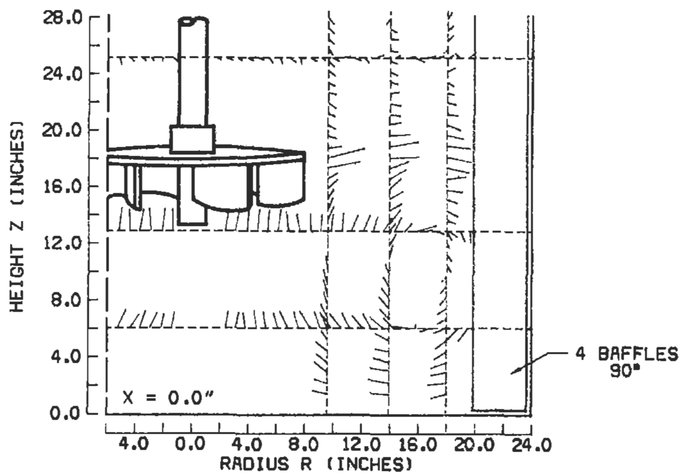


Figure 5-6B. Velocity vectors in R–Z plane (Lifter Turbine). By permission, Oldshue, J. Y., *Fluid Mixing Technology,* 1983, Chemical Engineering McGraw-Hill Publications Co. [29].

Figure 5-7 is an analysis flow chart for examining types of turbine impeller performance requirements.

For some services there may be more than one impeller on the shaft, attached part-way up the shaft from the lower one (Figures 5-8A and 5-8B).

The use of dual impellers on a shaft should be determined by the physical properties and characteristics of

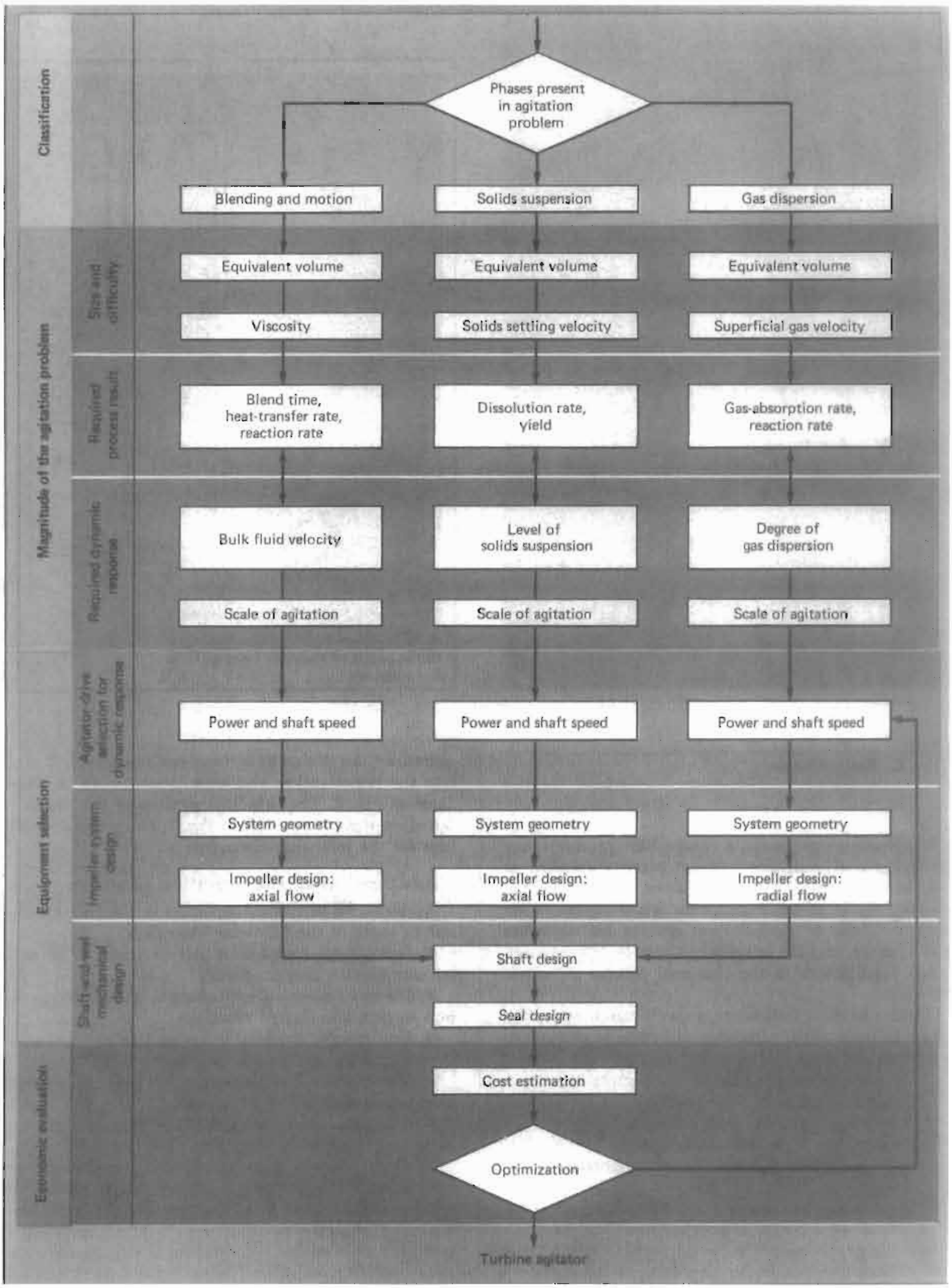


Figure 5-7. Analysis flow chart for examining types of turbine impeller applications. By permission, Gates, L. E., *et al., Chem. Eng.*, Dec. 8, 1975, p. 110 [26].

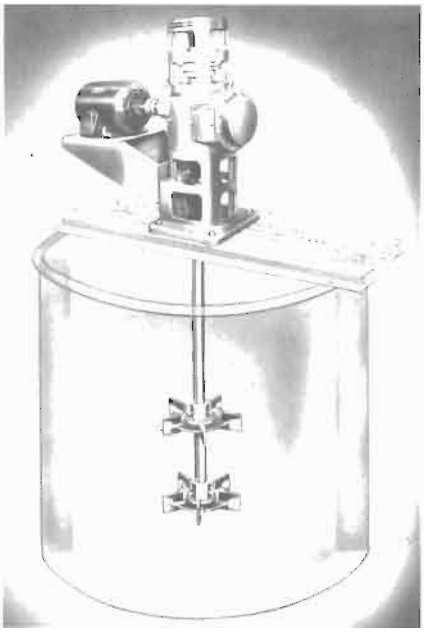


Figure 5-8A. Dual impeller mixer and drive. Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

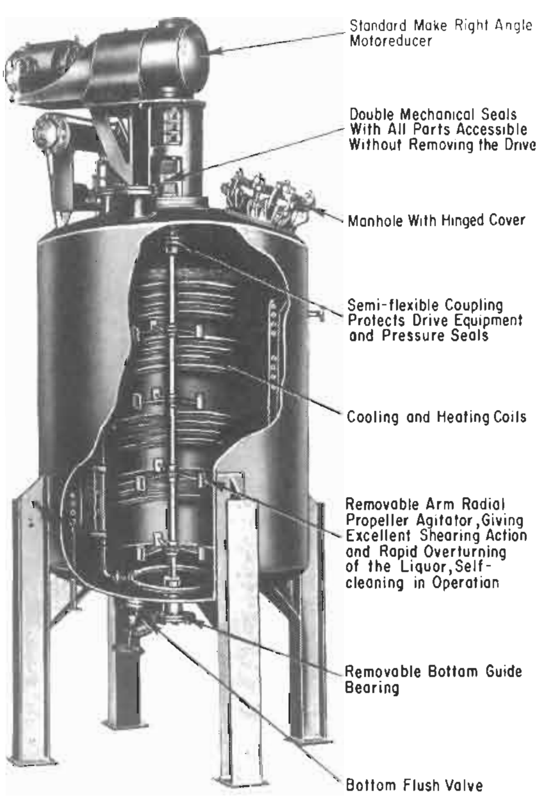


Figure 5-8B. Multiple impellers. Courtesy of Struthers-Wells Corp.

the system, in general being a function of viscosity, impeller diameter and liquid depth in the tank. In general, dual impellers may be indicated for fluids of 45 centipoise and greater and where the fluid travels more than four feet before being deflected.

The circulating capacity of 3-blade square pitch propellers is theoretically of the magnitude given in Figure 5-9. The speed ranges indicated may be grouped as [2]:

high speed, 1750 rpm:	for low viscosity fluids, such as water
medium speed, 1150 rpm:	for medium viscosity fluids, such as light syrups and varnishes
low speed, 420 rpm:	for high viscosity fluids such as oils, paints, or for tender crystals or fibers: or if foaming is a problem.

The mixing efficiency is generally higher (40%–60%) for the slow 400 rpm speed and lower (25%–45%) for the 1750 and 1150 rpm speeds. This is given in Figure 5-10 for general estimating use. Note that the turnover of tank capacity is involved through the selected impeller diameter and speed.

Mixing Concepts, Theory, Fundamentals

A mixer unit or impeller, regardless of its physical design features, is a pump of varying efficiency for pumping of fluids. Generally speaking, all designs are low heads compared to a conventional centrifugal pump, because there is no defined confining casing for the mixing element.

The action of the impeller design produces flow of the fluid, head on the fluid, or shear in the fluid, all to varying degrees depending on the specific design. A general identification of these characteristics for several types of impellers is given by [27]. (Note: Use consistent dimensions).

Type Impeller	Flow Decreases from Top	Head (Shear) Increases from Top
Rakes, Gates	↓	↓
Spirals, Anchors, Paddles		
Propellers		
Axial Flow Turbines	Decreases	Increases
Flat Blade Turbine		
Bar Turbine		
Bladeless Impeller		
Close Clearance Impeller and Stator		
Colloid Mill, Homogenizer	↓	↓

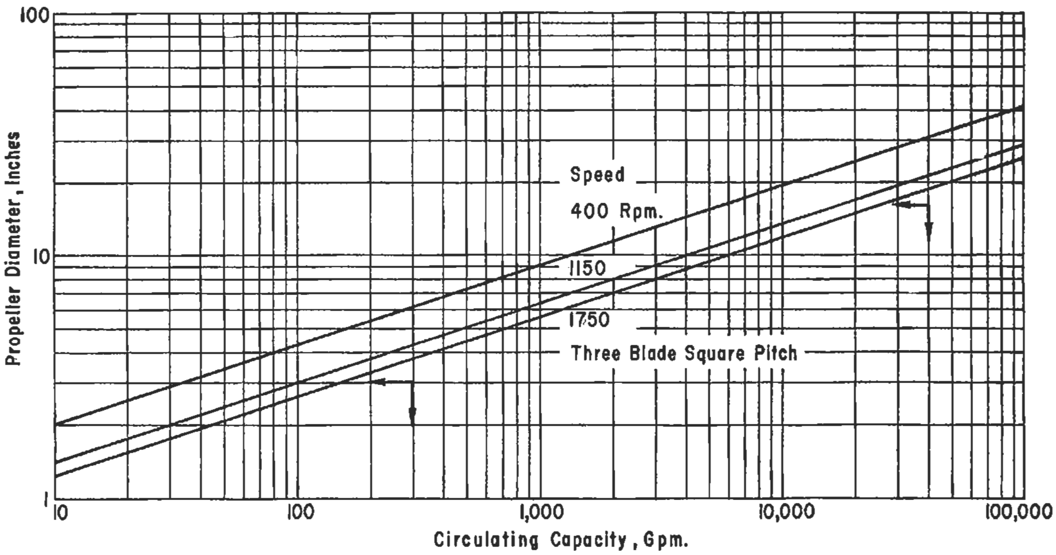


Figure 5-9. Theoretical circulating capacity of single propeller mixers. By permission, *Fluid Agitation Handbook,* Chemineer, Inc.

The horsepower required for any impeller is partly used for pumping flow and partly for shear requirements. To accomplish a given mixing performance for a process operation, the objective usually becomes a matching of the quantity of flow from an impeller with the shear characteristics at a specific power input. The flow/shear input ratio to a fluid system can be shifted or changed by changing the type/physical characteristics of the impeller, not the dimensions of a specific impeller design. For particular dimensional features (angles of blades, height/depth of blades, number of blades, etc.), the performance will remain the same as long as the dimensions are in the same relative relationship as the impeller, that is, in the same performance family.

Flow

The quantity of flow is defined as the amount of fluid that moves axially or radially away from the impeller at the surface or periphery of rotation. This flow quantity is never actually measured, but its relative relation to head characterizes the particular system. The flow rate, Q, is usually available from the manufacturer for a given impeller [21].

$$Q = K_1 N\, D^3 \cong N\, D^3 \qquad (5\text{-}1)$$

where Q = flow rate from impeller, cu ft/sec
N = speed of rotation, revolutions per sec
D = impeller diameter, ft
K_1 = proportionality constant, a function of the impeller shape, = 0.40 for 3-blade propeller in water

Figure 5-9 indicates the theoretical circulation from a propeller, and Figure 5-10 gives its efficiency for estimating purposes. Efficiency must be used in converting theoretical to actual horsepower, or in converting theoretical to actual circulation of the propeller.

Flow Number

This is probably the most important dimensionless group used to represent the actual flow during mixing in a vessel. Flow Number, N_Q (or pumping number):

$$N_Q = Q'/(N_m\, D^3) \qquad (5\text{-}2)$$

where N_m = impeller speed of rotation, rev per min
Q' = flow rate or pumping capacity, cu ft/min
D = impeller diameter, ft

N_Q is strongly dependent on the flow regime, Reynolds Number, N_{Re}, and installation geometry of the impeller. The flow from an impeller is *only* that produced by the impeller and does not include the entrained flow, which can be a major part of the total "motion" flow from the impeller. The entrained flow refers to fluid set in motion by the turbulence of the impeller output stream [27]. To compare different impellers, it is important to define the type of flows being considered.

It is important to recognize that in the system:

"Process Result" p Flow

Figure 5-11 [28] presents an analysis of pumping number versus Reynolds Number for various vessel dimensional relationships, for *turbine* mixers.

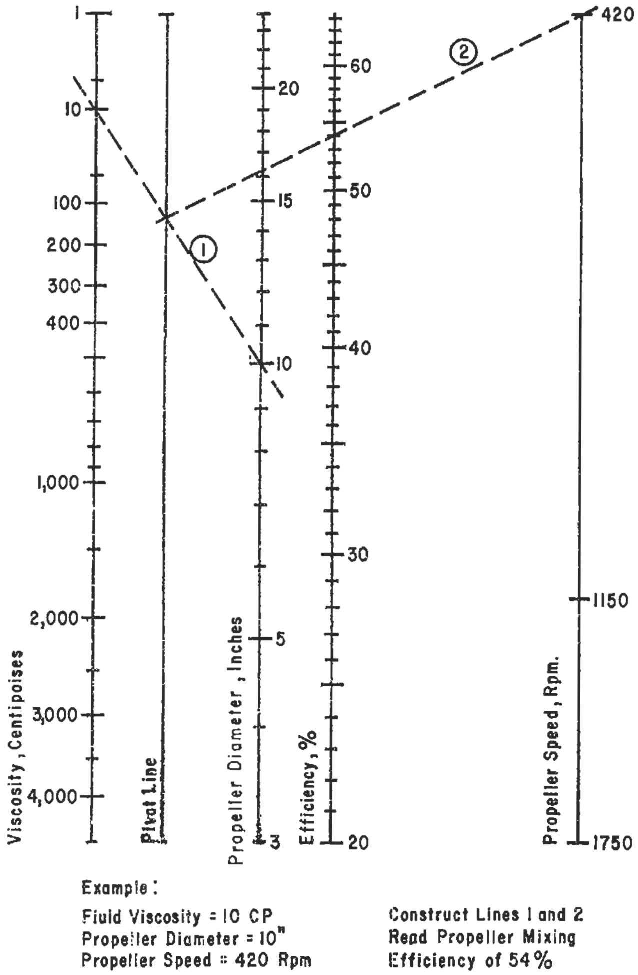


Figure 5-10. Propeller circulation efficiency. This is used with theoretical propeller capacity to determine actual capacity. By permission, *Fluid Agitation Handbook,* Chemineer, Inc.

Power, P; Power Number, P_o; and Reynolds Number, N_{Re}

Power

Power is the external measure of the mixer performance. The power put into the system must be absorbed through friction in viscous and turbulent shear stresses and dissipated as heat. The power requirement of a system is a function of the impeller shape, size, speed of rotation, fluid density and viscosity, vessel dimensions and internal attachments, and position of the impeller in this enclosed system.

The power requirements cannot always be calculated for any system with a great degree of reliability. However, for those systems and/or configurations with known data, good correlation is the result. The relations are [21]:

$$P = Q\rho H \tag{5-3}$$

Viscous flow, N_{Re} less than 10 to 300 is expressed:

$$P = \frac{K_2}{g}\mu(N_s)^2(D)^3 \tag{5-4}$$

K_2 = from Table 5-1
P = power, *not* power number, P_o

Fully developed turbulent flow, N_{Re} over 10,000, in a tank containing four equally spaced baffles having a width of 10% of the tank diameter:

$$P = \frac{K_3}{g}\rho(N_s)^3(D)^5 \tag{5-5}$$

K_3 = from Table 5-1

where g = conversion factor, 32.2 lb mass-ft/lb force/(sec) (sec)
H = total potential head during flow, ft
P = power, ft-lb/sec
W = impeller blade width, ft
μ = viscosity, lb/ft-sec
ρ = density, pounds/cu ft
N_s = revolutions/sec

$$\text{Horsepower, (HP)} = P/550 \tag{5-6}$$

$$\text{Imp. HP} = \frac{N_p N_m^3 D_i^5 \, Sg}{1.523\ 3\ 10^{13}} \text{ Turbine} \tag{5-6A}$$

(Symbols below)
[29]

Table 5-1 shows that in a cylinder tank, four baffles, each 1/12 tank diameter above flat bottom, liquid depth is equal to tank diameter, impeller shaft is vertical and at centerline of tank.

The Reynolds number N_{Re} for mixing is: (dimensionless)

$$N_{Re} = \frac{D^2 N_m \rho}{\mu} \tag{5-7}$$

or, [29],

$$N_{Re} = (10.754\ N_m D_i^2\ Sg)/(\mu') \tag{5-8}$$

where for these units, above equation

D = impeller diameter, ft
N_m = impeller speed, rpm

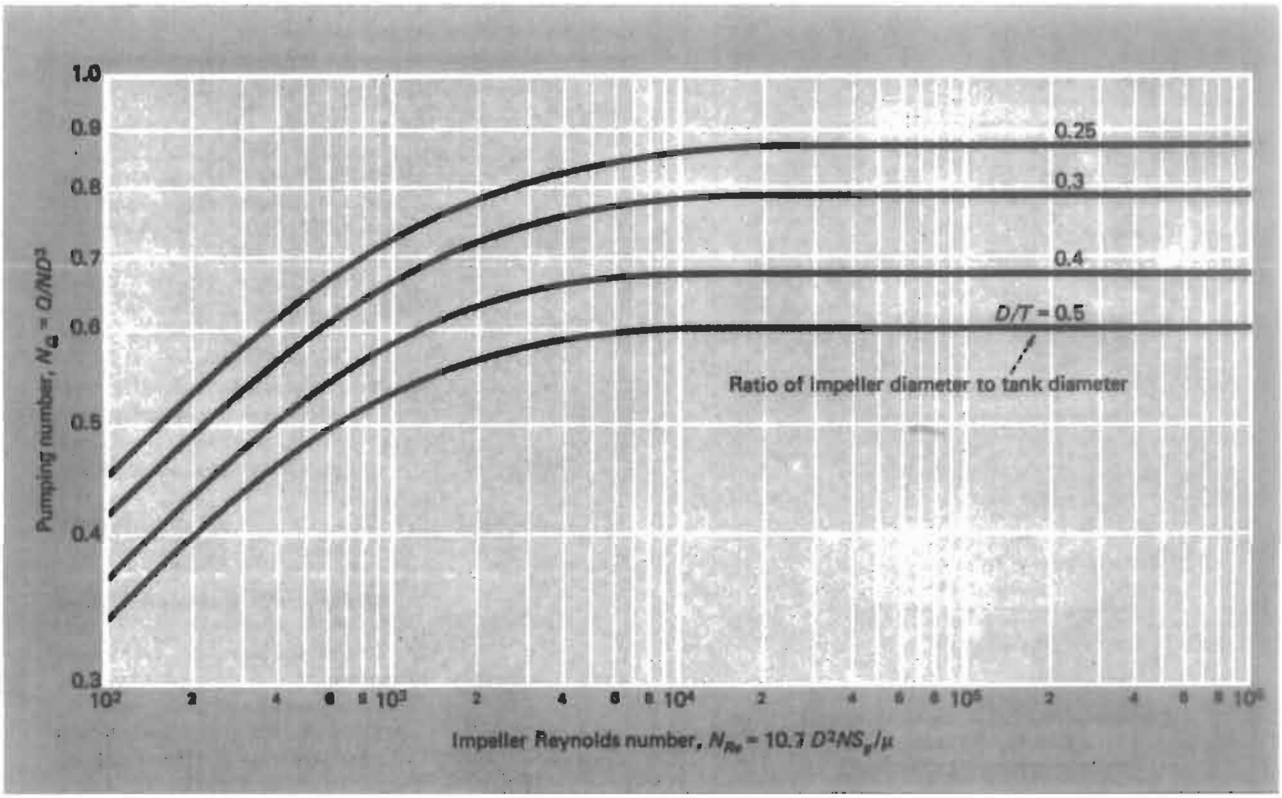


Figure 5-11. Pumping number is the basis for design procedures involving blending and motion. By permission, Hicks, R. W., *et. al., Chem. Eng.*, Apr. 26, 1976, p. 104 [28].

D_i = impeller diameter, in.
μ' = fluid viscosity, centipoise
μ = fluid viscosity, lb/sec ft
ρ = fluid density, lb/cu ft
Sg = fluid specific gravity, (not density)
N_p = power no.

The Froude number is [33]:

$$N_{Fr} = \frac{DN^2}{g} \quad (5-9)$$

g = gravitational constant, 32-ft/sec-sec

Estimated turbine impeller diameter [28]:

$$D_T = 394\ [(H_p)/(n\ S_g\ N_m{}^3]^{1/5} \quad (5-10)$$

Calculate Reynolds number, N_{Re}, from Equation 5-8, then correct for viscosity effects.

$$D_{cor}{}^{\bullet} = D_T\ C_F \quad (5-11)$$

Representative C_F values from [28] are:

N_{Re}	C_F, in.
700	1.0
400	0.98
200	0.95
100	0.91
70	0.89
60	0.88
50	0.87

In general, below a Reynolds number of 50, all impellers give viscous flow mixing; between 50 and 1,000 the pattern is in the transition range; and from N_{Re} above 1000 the action is turbulent.

For $N_{Rc} \leqq 10$, the liquid motion moves with the impeller, and off from the impeller, the fluid is stagnant [34]. The Froude number accounts for the force of gravity when it has a part in determining the motion of the fluid. The Froude numbers must be equal in scale-up situations for the new design to have similar flow when gravity controls the motion [16].

Table 5-1
Baffled Cylindrical Tanks

	K_2 Viscous	K_3 Turbulent
Propeller, 3-blade, pitch = diameter....	41.0	0.32
Propeller, 3-blade, pitch = 2 diameters..	43.5	1.00
Turbine, flat blade, 4 blades..........	70.0	4.50
Turbine, flat blade, 6 blades..........	71.0	6.30
Turbine, flat blade, 8 blades..........	72.0	7.80
Fan turbine, blades at 45°, 6 blades....	70.0	1.65
Shrouded turbine, stator ring..........	172.5	1.12
Flat paddles, 2 blades (single paddle), D/W = 4	43.0	2.25
Flat paddles, 2 blades, D/W = 6......	36.5	1.60
Flat paddles, 2 blades, D/W = 8......	33.0	1.15
Flat paddles, 4 blades, D/W = 6......	49.0	2.75
Flat paddles, 6 blades, D/W = 6......	71.0	3.82

*By permission, R. H. Rushton and J. Y. Oldshue, Chem. Eng. Prog. 49, 161 (1953)

Oldshue [29] points out that to identify the turbulent range as beginning at a specific N_{Re} may not be exactly correct, as it actually varies with different impeller designs. This range may vary from $N_{Re} \cong 10^3$ to $N_{Re} \cong 10^5$, so for common use $N_{Re} = 10^5$ is taken as the turbulent range for all impellers.

Power Relationship

For same family design/styles of impellers [29], see Figure 5-12:

$$P \propto N^3 \tag{5-12}$$

$$P \propto N^3 D^5 \tag{5-13}$$

$$P \propto \rho \tag{5-14}$$

$$P \propto N^3 D^5 \rho \tag{5-15}$$

$$P \propto D^5 \tag{5-16}$$

$$P \propto QH\rho \tag{5-17}$$

Note: (Horsepower) (33,000) = ft lb/min
(Horsepower) (550) = ft lb/sec

The power number, P_o (dimensionless)

$$P_o = f(N_{Re}) \tag{5-18}$$

$$P_o = P\, g_c/(\rho N^3 D^5) \tag{5-19}$$

$$P_o = 1.523\, P\, (10^{13})/(N^3 D^5 \rho) \tag{5-20}$$

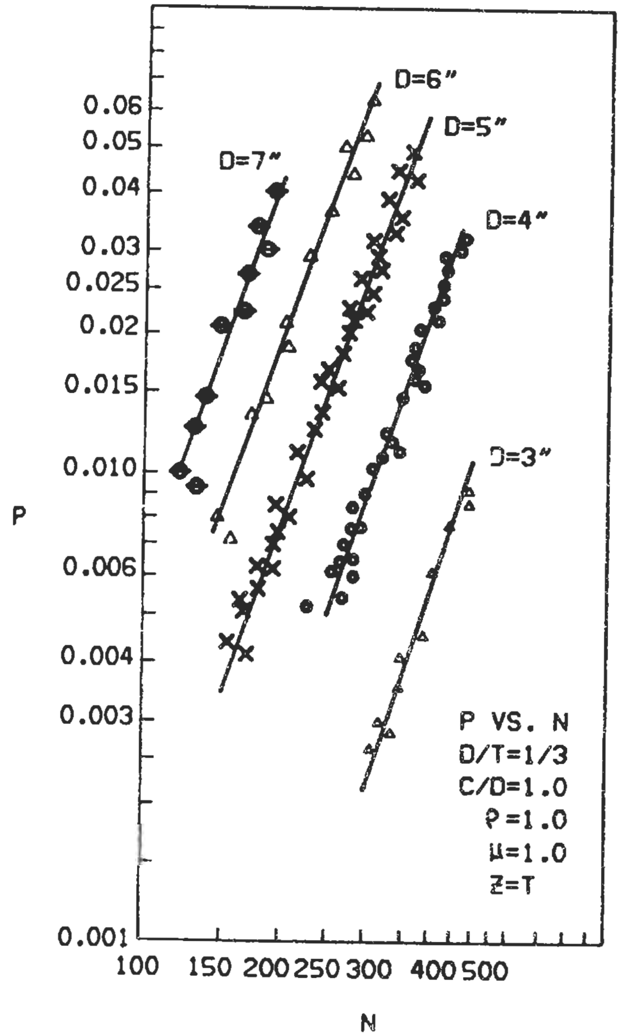


Figure 5-12. Power vs. RPM with impeller diameter parameters. Illustration of impeller input power versus speed for a family of impeller designs, but only of various diameters, showing uniformity of performance. By permission, Oldshue, J. Y., *Fluid Mixing Technology,* 1983, Chemical Engineering, McGraw-Hill Publications Co. [29].

is used in most correlations to represent the relationship to system performance for turbulent flow in a baffled tank. For tanks containing no baffles, the fluid motion remains swirling and a vortex develops. These conditions are characterized by the lower curves in Figures 5-13, 5-14, and 5-15, which include the Froude effect. This effect is not prominent in baffled tanks.

For unbaffled tanks:

$$P = \frac{\Phi(\rho N^3 D^5)}{g}\left(\frac{N^2 D}{g}\right)^{\frac{a - \log N_{Re}}{b}} \tag{5-21}$$

$$\Phi = P_o = N_p \tag{5-22}$$

where Φ is read from the charts and the constants a and b are given in Figure 5-16.

Figure 5-17 is useful for determination of horsepower during turbulent flow for various types of impellers, and Figure 5-18 is useful for laminar flow. Also see Figure 5-19.

Flow and power numbers each decrease as the Reynolds number increases. In unbaffled tanks, a vortex forms that takes over the flow regime and does not allow the usual relationship to describe the performance of the mixing operation. It is proper and good practice to provide baffles in all vessels (see later description for the physical configurations).

At high N_{Re}, the power number, P_o, stays reasonably constant, thus, viscosity has little effect on the power requirements. When moving to lower N_{Re} through the laminar region into the viscous region, the viscosity effect increases. In the laminar range [29]

$$P_o \text{ p } 1/N_{Re};\ N_{Re} < 50 \tag{5-23}$$

$$\text{or, } P_o \text{ p } \mu \tag{5-24}$$

for all other parameters constant.

For $50 < N_{Re} < 1000$ [29] is the transition range. In the immediate impeller area, the flow is fully turbulent; however, in the extremities of a vessel, the motion would be laminar. In this case, as in all others, the tank baffling is a major factor for performance of the system and the power and flow results.

For $N_{Re} > 1000$, the properly baffled tank is turbulent throughout. N_Q and P_o are independent of N_{Re}. If the tank is not baffled, a "forced vortex" dominates the flow in the vessel.

For $N_{Re} > 1000$, in fully baffled tank is turbulent.

$$N_p = P/(N^3\,D_i^5)(\rho) \tag{5-25}$$

Pumping effectiveness or pumping per power is important for flow controlled processes [29].

The shape, size, and baffling of a specific mixing vessel significantly influences the Reynolds number, flow, and power numbers.

$$D_i = 394\ (HP/n\ S_g\ N_m^3)^{1/5} \tag{5-25A}$$

Other relationships [29] for one type of impeller (not different types)

$$\frac{Q}{P} = \frac{N_Q}{P_o}\left(\frac{1}{\rho\, N^2\, D^2}\right), \text{ ratio of flow to power} \tag{5-26}$$

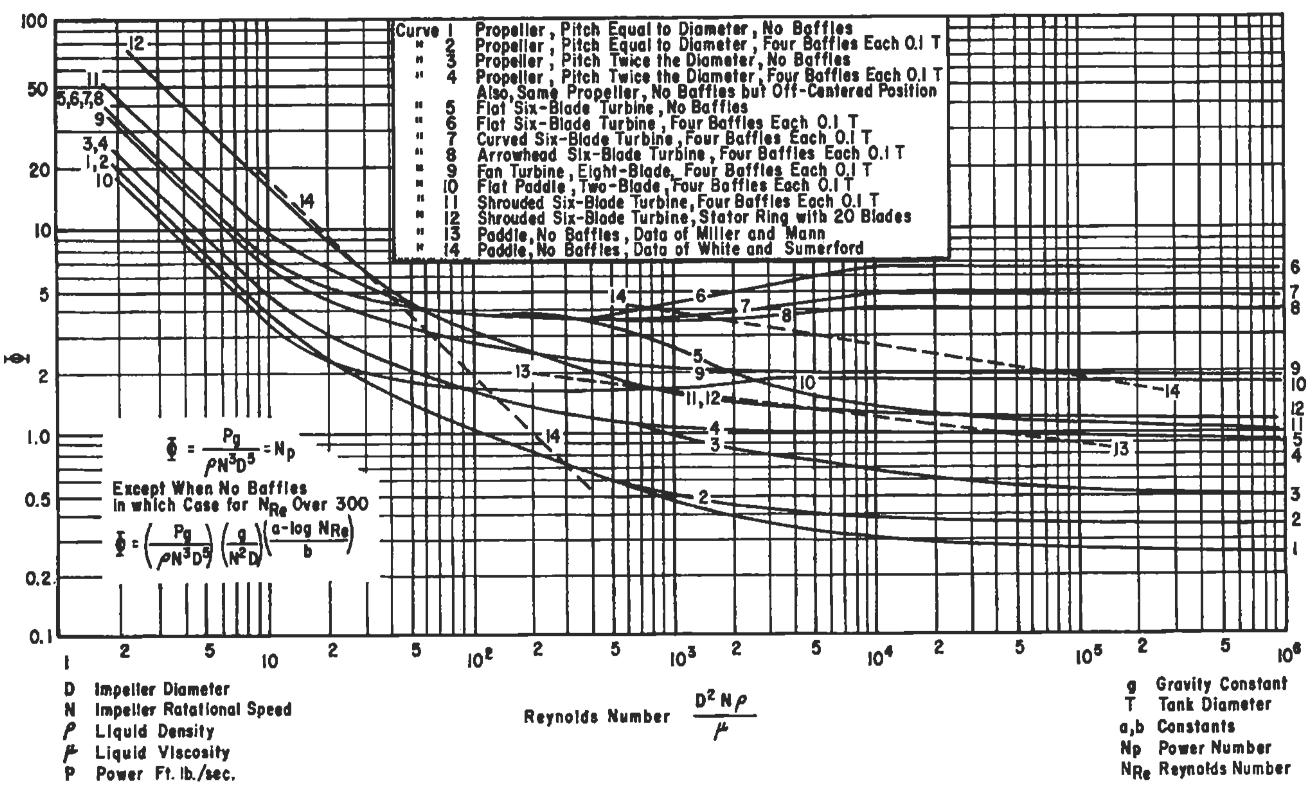


Figure 5-13. Power consumption of impellers. By permission, Rushton, J. H., Costich, E. W. and Everett, H. L., *Chem. Engr. Prog.*, V. 46, No. 8 and No. 9, 1950 [18].

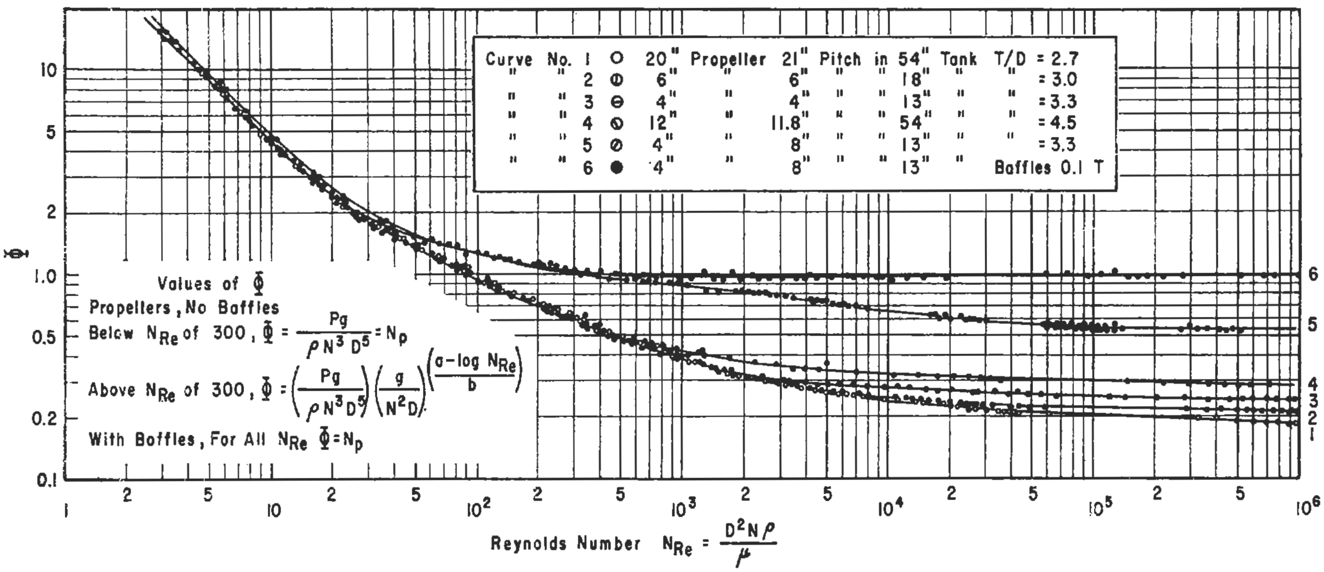


Figure 5-14. Reynolds number correlation for propellers. By permission, Rushton, J. H. *et al.* [18].

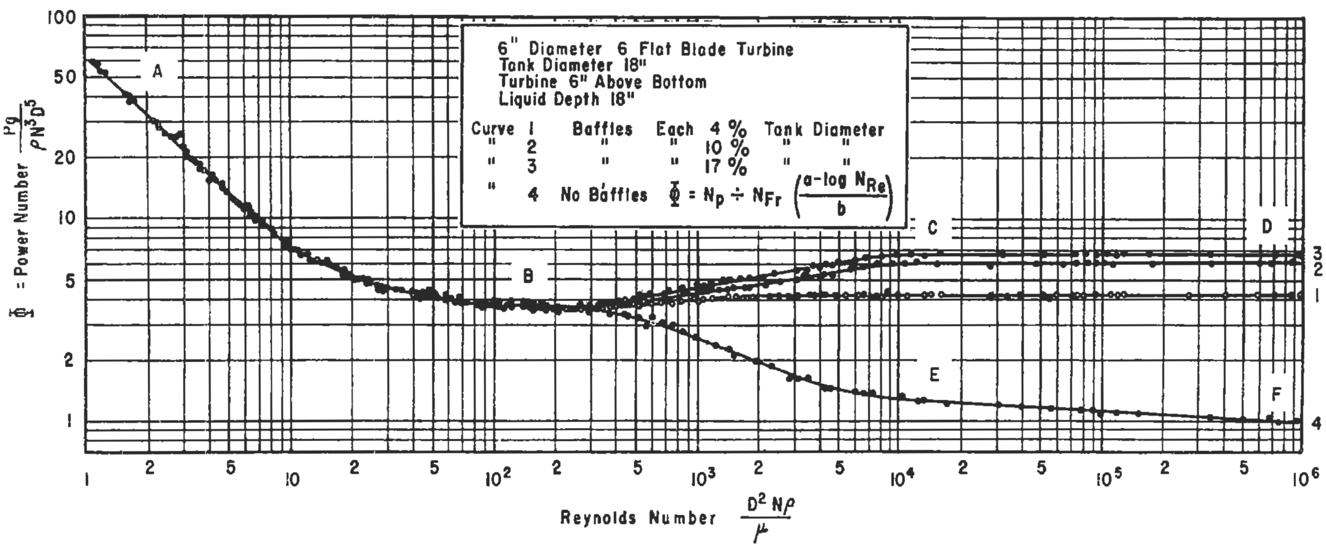


Figure 5-15. Reynolds number correlation for a flat blade turbine. By permission, Rushton, J. H. *et al.* [18].

This is dependent on the impeller type, speed, diameter, and the geometry of the installation.

Torque; τ, $\tau = N_p\,(\rho N^2 D^5)/2\pi$ (5-27)

Lateral fluid forces on mixer; $F = N_F \rho\, N^2 D^4$ (5-28)

For two different impellers, comparing performance of power, flow and flow per power at constant flow and speed [29] in terms of flow, Q, at speed, N:

Fluid forces, $F = [N_F\,(\rho\, N^{2/3} Q^{4/3})]/N_Q^{4/3}$ (5-29)

Torque, $\tau = [N_p \rho\, N^{1/3}\, Q^{5/3}]/(N_Q^{5/3}\,(2\pi))$ (5-30)

Flow per power input:

$$Q/P = \left[\frac{N_Q^{\;5/3}}{N_p}\right] / \,(N^{4/3} Q^{2/3} \rho) \quad (5\text{-}31)$$

Power, input, net to impeller shaft:

$P = (N_p/N_Q^{5/3})\;(\rho\, N^{4/3} Q^{5/3})$ (5-32)

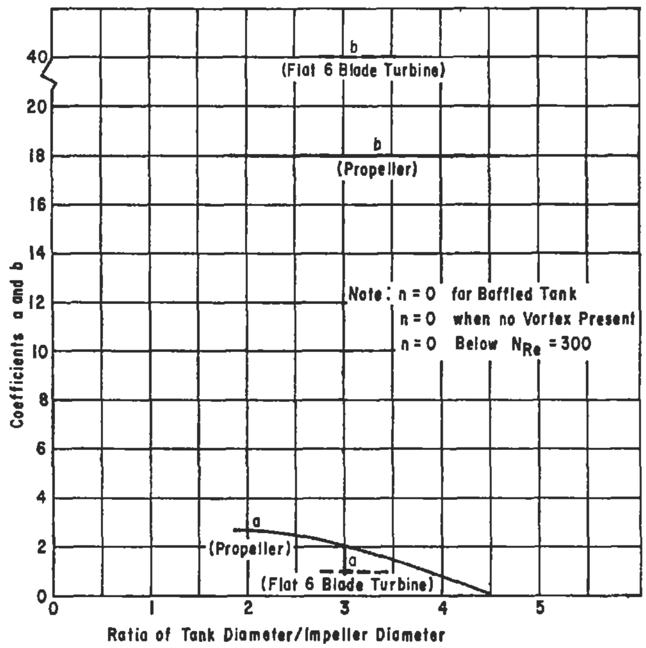


Figure 5-16. Factors in Froude number exponent, n. By permission, Rushton, J. H., *et al.* [18].

$$\text{Diameter, } D = \frac{1}{N_Q^{1/3}}\left(\frac{Q^{1/3}}{N^{1/3}}\right) \tag{5-33}$$

Where using consistent units:

P = impeller power draw, F L/t or ML^2/t^3
t = time
L = length
F = fluid force on turbine, perpendicular to shaft, ML/t^2
D = impeller diameter, L
Q = volumetric flow, L^3/t
T = tank diameter, L
ρ = fluid density, M/L^3
μ = fluid viscosity, M/(Lt)
τ = torque, FL, or ML^2/t^2
Z = liquid depth
N_p = power number, dimensionless,

$$P_o \cong N_p \cong P/(N^3D^5\rho) \tag{5-25}$$

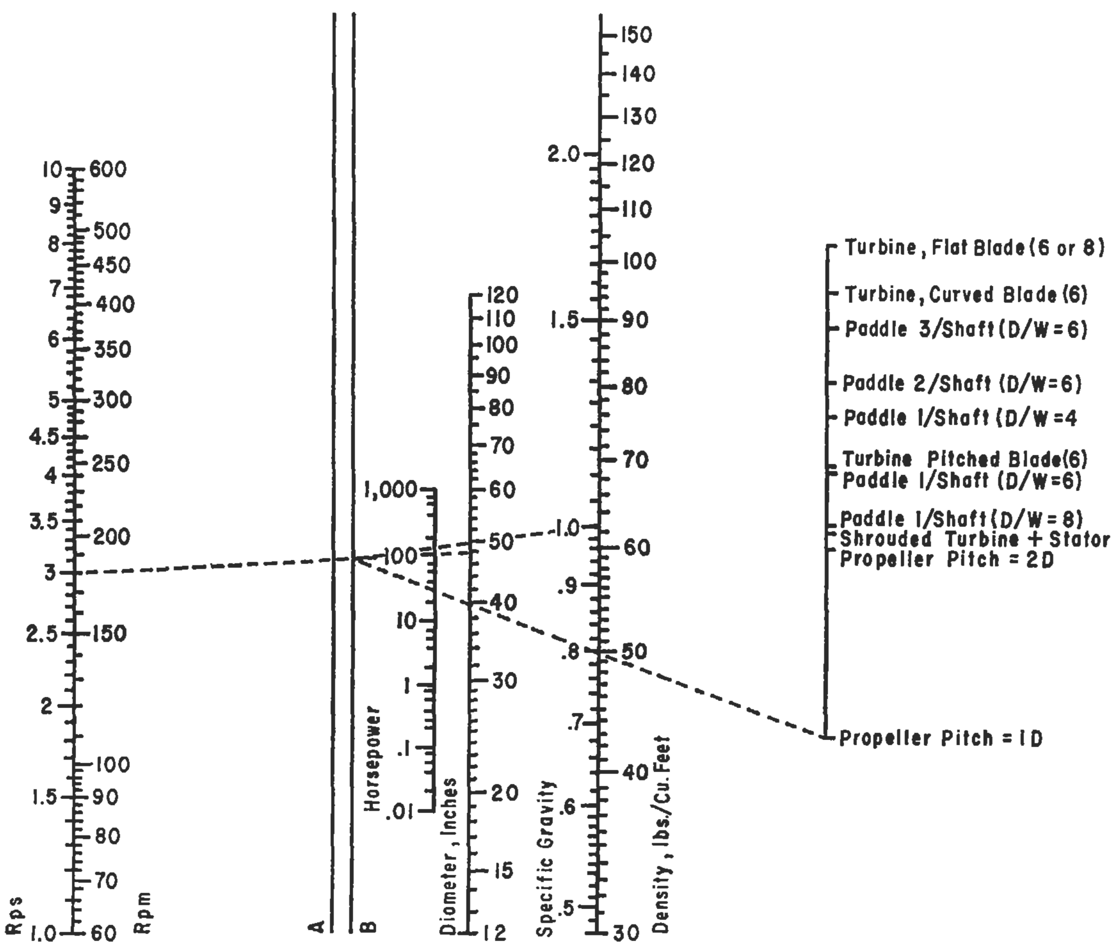


Figure 5-17. Power consumption by impeller type/dimensions for turbulent flow conditions. Knowing impeller type, diameter, speed and batch density; connect RPM with diameter. The intersection with "A," connected to the density scale, makes an intersection on "B." A line from this point to the impeller scale intersects the horsepower scale at the correct value. By permission, Quillen, C. S., *Chem. Engr.*, June 1954, p. 177 [15].

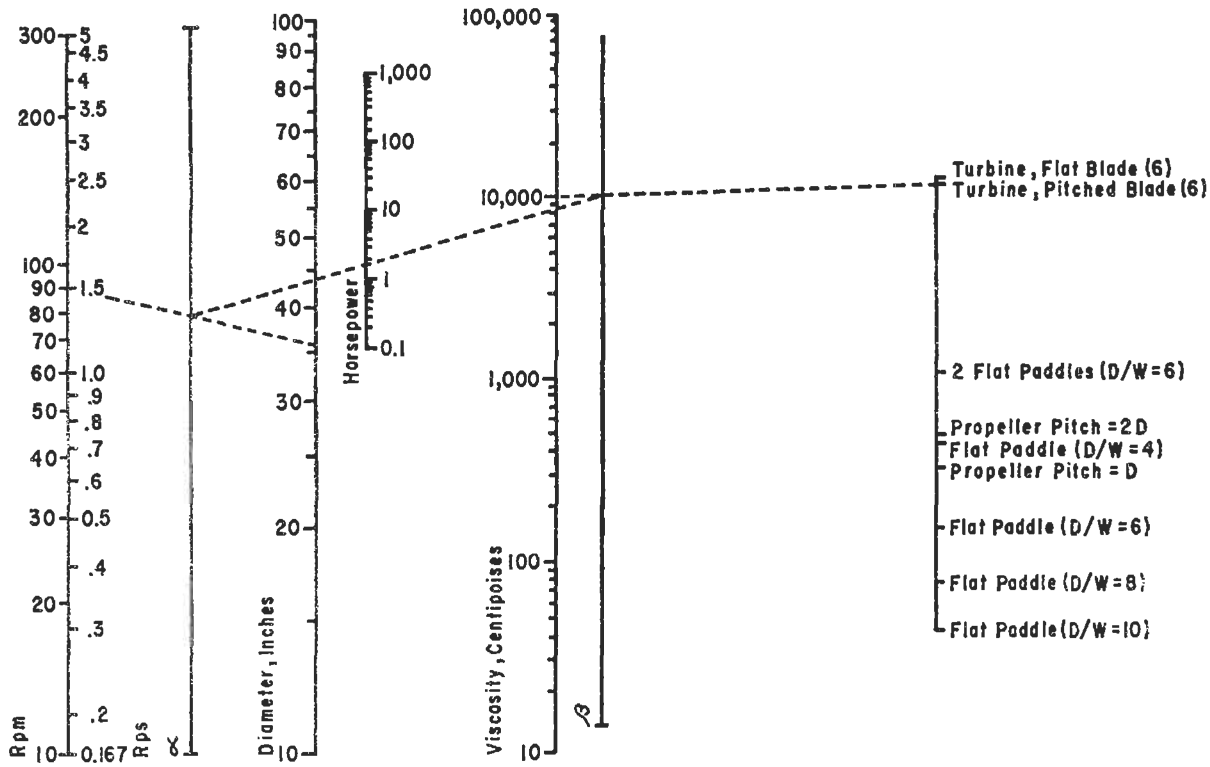


Figure 5-18. Laminar flow mixing. For known impeller type, diameter, speed, and viscosity, this nomograph will give power consumption. Connect RPM and diameter, also viscosity and impeller scale. The intersection of these two separate lines with alpha and beta respectively is then connected to give horsepower on the HP scale. By permission, Quillen, C. S., *Chem. Engr.,* June 1954, p. 177 [15].

N_Q = flow number, dimensionless,

$$N_Q \cong Q'/(N_m D^3) \quad (5\text{-}2)$$

N_{Re} = Reynolds number, dimensionless

$$N_{Re} = \rho\, N D^2/\mu \quad (5\text{-}7)$$

$$N_F = \text{force number}, N_F \cong F/(PN^2D^4) \quad (5\text{-}34)$$

when M = mass
L = length
T = temperature
t = time
F = force, ML/t^2

Oldshue [29] expresses pumping effectiveness as pumping per power and recognizes it as a key function for processes that are flow controlled or need more flow than head or shear.

$$Q = N_Q\, ND^3 \quad (5\text{-}35)$$

$$P = N_p\, \rho\, N^3\, D^5 \quad (5\text{-}36)$$

$$\text{Then } Q/P = (N_Q/N_p)\,(1/\rho\, N^2D^2) \quad (5\text{-}37)$$

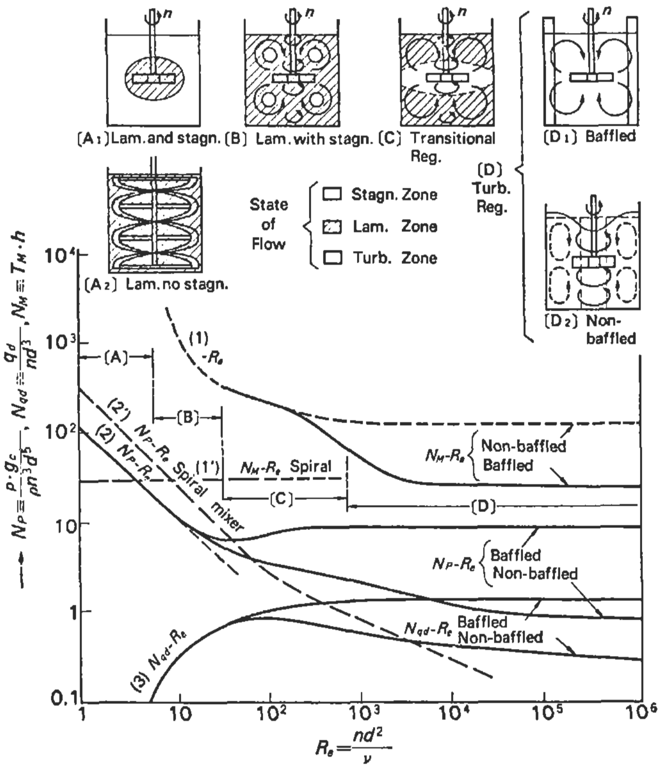
When comparing flow (or pumping) per power, we determine that it is dependent on the impeller type, speed, diameter, and geometry of the installation. The mixer is not fully specified until torque, τ, and lateral loads (fluid force, F) are included in the analysis [29].

$$\tau = N_p\, \rho\, N^2D^5/(2\pi) \quad (5\text{-}38)$$

$$F = N_F\, \rho\, N^2D^4 \quad (5\text{-}39)$$

Table 5-2 presents the effects of expected performance on various parameters or relationships for mixing. To actually calculate a numerical result of comparing impeller performances, the dimensionless numbers for flow power and force are needed. Note that in Table 5-2 the constant basis is across the horizontal top of the chart and the function to be examined or compared is along the vertical left side. The functions in the body of the table are used as ratios for condition (1) and condition (2), holding the basis constant.

For example, referring to Table 5-2, if power input (P) and impeller diameter, D, are kept constant, then speed,



Np = power number
D = tank diameter
d = impeller diameter
Z = height of liquid in tank
n = revolutions per minute
T_M = time

R_e = Reynolds number = No = $d^2 n\rho/\mu$
N_{qd} = discharge flow number = q_d/nd^3
p = pressure
q_d = discharge flow rate
θ_m = mixing time

Figure 5-19. Characteristic curves: flow, power, and mixing time. (H/D = 1, 8-blade paddle H/D = ½, d/D = 1/10, H_D/D = ½) By permission, Nagata, S., *Mixing Principles and Applications,* Halsted Press, John Wiley & Sons, Kodasnsha Scientific. p. 125 [34].

N, is proportional to $1/N_p^{1/3}$; or holding flow (Q) and speed, N, constant then Q/P is proportional to

$$N_Q^{5/3}/N_p,$$

$$\text{thus } \frac{(Q/P)_1}{(Q/P)_2} = \frac{(N_Q^{5/3}/N_p^{3/4})_1}{(N_Q^{5/3}/N_p^{3/4})_2} \tag{5-40}$$

This is a valuable relationship as expressed in Table 5-2, because it expresses the working relationship between all the important variables. Note that as one variable changes all others are changed. One variable cannot be changed alone without affecting the others.

Shaft

The proper size of shaft is very important to avoid whip and vibration, destruction of bearings, gears, and damage to the vessel. The manufacturers usually take a conservative view of this problem, nevertheless it is well to understand the expected operating conditions for any installation. Normally an impeller-shaft system should operate at about 40% of the critical speed. However, the turbine with bottom side stabilizer can go as high as 80% of critical. The manufacturer should provide this information for the specific system.

Table 5-2
Performance Relationships for Mixing Variables: More Than One Variable Changing or Held Constant

	Basis					
Function	N, D	*Q, N	P, N, t	P,D	Q,P	Q,D
N~	1	1	1	$\frac{1}{N_P^{1/3}}$	$\frac{N_Q^{5/4}}{N_P^{3/4}}$	$\frac{1}{N_Q}$
D~	1	$\frac{1}{N_Q^{1/3}}$	$\frac{1}{N_P^{1/5}}$	1	$\frac{N_P^{14}}{N_Q^{3/4}}$	1
Q~	N_Q	1	$\frac{N_Q}{N_P^{3/5}}$	$\frac{N_Q}{N_P^{1/3}}$	1	1
P~	N_P	$\frac{N_P}{N_q^{5/3}}$	1	1	1	$\frac{N_P}{N_Q^3}$
t~	N_P	$\frac{N_P}{N_Q^{5/3}}$	1	$N_P^{1/3}$	$\frac{N_P^{3/4}}{N_Q^{5/4}}$	$\frac{N_P}{N_Q^2}$
$\frac{*Q}{P}$~	$\frac{N_Q}{N_P}$	$\frac{*N_Q^{5/3}}{N_P}$	$\frac{N_Q}{N_P^{3/5}}$	$\frac{N_Q}{N_P^{1/3}}$	1	$\frac{N_Q^3}{N_P}$
F~	N_F	$\frac{N_F}{N_Q^{4/3}}$	$\frac{N_F}{N_Q^{4/5}}$	$\frac{N_F}{N_P^{2/3}}$	$\frac{N_F}{N_Q^{1/2}N_P^{1/2}}$	$\frac{N_F}{N_Q^2}$

*Example: Q/P is proportional to $N_Q^{5/3}/N_P$ on a comparison basis of keeping flow Q and speed N constant.

By permission, Oldshue, "Fluid Mixing Technology," *Chemical Engineering,* McGraw-Hill Publications Co. Inc., 1983 [29].

Drive and Gears

Most mixers are driven by electric motors, or in some cases by mechanical turbines, with gears ratioed to give the proper performance speed of the impeller. A variable or 2-speed driver or gear system often proves worth the extra cost, since it is difficult to predict the exact speed requirements for some new installations. This is particularly true in continuous chemical processes where the general nature of the fluid remains constant but the viscosity, density or solid particle content may change as the plant progress from 'just erected' to steady production and even on to new and different products. This transition may take from a few months to several years and should be economically evaluated. The gear mechanism is not a place to reduce costs for this equipment, since improper application can create costly maintenance.

Usual practice, particularly for good estimating is to assume that the gear drive requires 5% of the impeller horsepower and that system "surging or variations" require a minimum of 10% of this impeller horsepower. Thus

$$\text{Actual motor hp} = \text{impeller required hp}/0.85 \quad \text{(minimum)} \tag{5-41}$$

When the actual *maximum* gear box horsepower is known from the manufacturer, it should be used as long as it is equal to or greater than the 5% allowance noted above. The impeller/fluid horsepower allowable variation should still be 10% or greater. For example, if the calculated required motor drive (or turbine drive) hp = 23, (i.e., 19.55/0.85), the next *standard* motor is 25 hp, so use this, never less than the 23 indicated above because 23 hp is non-standard, and no such motor hp exists.

Figure 5-20 illustrates a vertical propeller mixer assembly, with vertical mounting with gear box and motor. Figure 5-21 is a typical right angle, vertical impeller shaft with horizontal gear and motor drive.

The mixer manufacturer should always be consulted for proper mechanical features design and strength characteristics, such as horsepower, gear rating AGA, shaft diameter, shaft deflection, critical speeds, bottom steady bearing, and side shaft bearings.

Steady Bearings

The installation of mixers on long shafts in tall tanks may become a problem if "whip" of the shaft develops. To reduce this possibility, a bearing support in the bottom of the tank will hold the shaft steady. Lubrication is by the tank fluid. Therefore this has limited application if abrasive particles are present. Normally the manufacturers' designs avoid this extra bearing. Sometimes a guide bearing is installed about midway in the tanks to steady the shaft at this point. Again it is preferable to avoid this, if possible, and the manufacturer should make recommendations for the installation.

Figure 5-20. Portable Vektor(TM) vertical propeller mixer assembly. By permission, Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

Materials of Construction

In general, just about any material that can be worked into the impeller design is available, including steel, stainless alloys, copper alloys, nickel and alloys, hard rubber, and lead, rubber and plastic coatings on impellers and shafts.

Design

Normally the proper impeller selections and horsepower requirements are handled in a cooperative manner with the manufacturer of this equipment in order to obtain the best analysis of a given application. There is no substitute for performing the *proper* test runs to evaluate

1. Motor bracket
2. Standard foot-mounted motor
3. Fabricated housing
4. Lifting eyes
5. High-speed shaft
6. Change pinion
7. Change gear
8. Change gear cover
9. Drain plug
10. Spiral bevel pinion cartridge
11. Timken tapered roller bearings
12. Removable low-speed coupling half
13. Dry well oil seal
14. Spiral bevel gear
15. Low-speed shaft

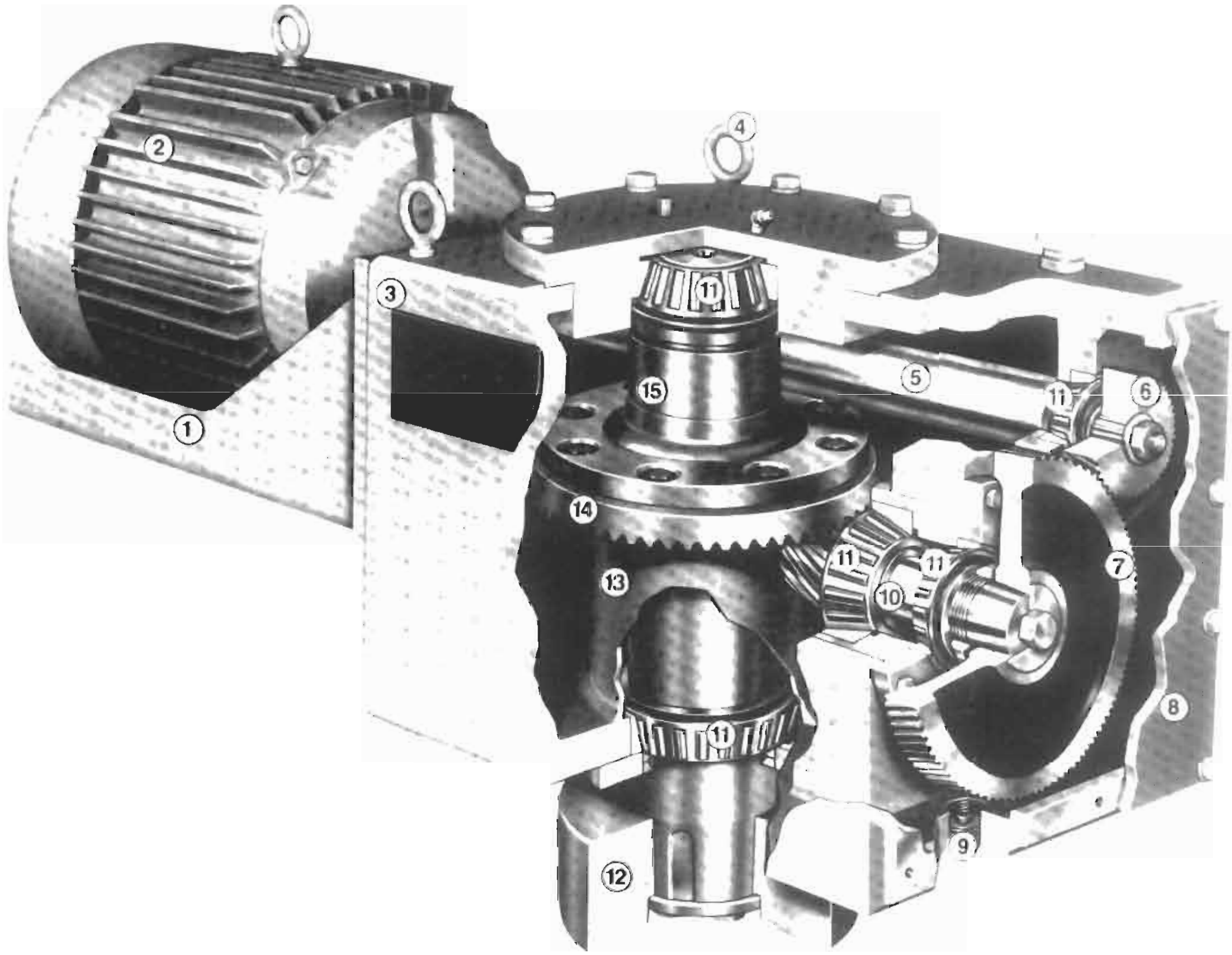


Figure 5-21. Right angle drive for vertical impeller shaft. By permission, Chemineer, Inc. Bulletin 711.

one or more types of impellers in a particular application. Even is this is carried out on a small scale, properly evaluated it can lead to the correct selection of impeller and horsepower. The horsepower seems to be the factor that is missed in some evaluations. Any foreseeable process changes, or ranges of operation, must be considered in order to have sufficient power for start-up as well as normal running.

Specifications

The suggested specification sheet of Figure 5-22 is helpful as a general checklist for the mixing inquiry and can be used in setting forth the known as the desired information with a manufacturer. In general, the specification sheet should not be expected to establish the whole story or information on a mixing problem, unless the problem is known to be fairly straight forward or data is known which can be given to the manufacturer (for example, blending, dispersing or dissolving crystals, etc.). For the unique problems, one-of-a-kind, laboratory data should be taken under the guidance of technical advice from the manufacturer, or other qualified authority in order that adequate scale-up data will be taken and evaluated. It is important to either describe and give dimensions for the vessel to be used, or request the mixer manufacturer to recommend the type best suited to the service.

Flow Patterns

The pattern of the fluid motion is a function of the fluid system, impeller, vessel configuration, and location of the impeller in the fluid system relative to the vessel walls and/or bottom. The patterns illustrated in Figures 5-23A–5-23K indicate that almost any pattern can be established provided the particular impeller type is located in the proper position. This is easier to accomplish in some systems than others.

The use of vertical side wall baffles usually destroys the rotary and swirling motion in vertical tanks. This also can be accomplished to a degree by setting the mixer off center. These baffles should be $\frac{1}{10}$ to $\frac{1}{12}$ of the width or diameter of the tank. Six baffles generally give slightly better performance than four; although four is the usual number, with three not being as good for most situations.

Draft Tubes

The application of draft tubes as related to various mixing operations is shown in Figures 5-23I and 5-24A–5-24I. The draft tubes are basically a tube or shell around the shaft of the mixer including the usual axial impeller, which allows a special or top-to-bottom fixed flow pattern to be set up in the fluid system. The size and location of the tube are related to both the mechanical and mixing performance characteristics as well as peculiar problems of the system. Usually they are used to ensure a mixing flow pattern that cannot or will not develop in the system. Weber gives the following points for draft tubes [23]:

With a draft tube inserted in a tank, no sidewall baffles are required, and, the flow into the axial impeller mounted inside the tube is flooded to give a uniform and high flow pattern into the inlet to the impeller. The upflow in the annulus around the tube has sufficient velocity to keep particles in suspension, if necessary.

A. Increase mixing efficiency
   1. Prevent short circuiting of fluid, define a specific path.
   2. Improve heat transfer coefficient by forcing flow past coil surfaces.
   3. Provide more complete reaction in a gas-liquid system by recirculation of unreacted gases.
   4. Minimize areas of inadequate turbulence in vessel.
   5. Accentuate the direct mechanical shearing action of the mixing impeller upon the fluid.
   6. Amplify mixing action by effectively increasing the ratio of mixer to container diameter.

B. Decrease Design Problems
   1. Reduce required shaft diameter and length, while maintaining complete mixing effectiveness.
   2. Limit or eliminate the need for submerged or internal guide bearings.

Entrainment

Entrainment is an important element in the mixing operation and involves incorporation of low velocity fluid into the mass of the fluid stream or jet issuing from a source such as a mixing impeller. The axial flow from a propeller under proper physical conditions serves as a circular cross-section jet to produce mixing by turbulence and entrainment. The flat-blade turbine issues a jet for entrainment at the top and bottom areas of the ring [2]. It is significant to estimate the relative amount of liquid involved due to entrainment, as this helps to describe the effectiveness of the operation.

From a propeller, the entrainment by circular jet is [9]:

$$Q_e = Q\left[0.23\left(\frac{X}{D_o}\right) - 1\right] \tag{5-42}$$

where Q_e = volume entrained, cu ft/sec
X = distance from impeller source, not to exceed 100 jet diameters, ft
D_o = diameter of jet at origin, ft

This relation is sufficiently accurate for large scale design.

The maximum $Q/P^{1/3}$ for a circular jet is at $X = 17.1\ D_o$ (Refs. [9, 21]) or in other words the optimum jet origin diameter is 1/17.1 of the distance desired for effective entrainment. Since the entrainment efficiency does not fall off too rapidly, it is not necessary to use only the ratio given, but rather to stay in close proximity, say ± 25–35 percent. Large diameter jet streams are more effective for the same power than small streams [17]. Data on flat blade turbines has not been fully evaluated.

Batch or Continuous Mixing

Often pilot plant or research data for developing a process are obtained on a batch operation. Later, a continuous process will usually prove that smaller equipment can be used and that the operation will be more economical. Normally batch mixing requires 10%–25% more power than continuous [29] for stable conditions; however, the *reaction time* for continuous flow is always longer than the reaction time for batch flow, but the practical result may show batch time cycle is increased by filling,

(text continued on page 312)

SPEC. DWG. NO.

A-

Page of Pages	
Unit Price	
No .Units	
Item No.	

Job No. ____________

B/M No. ____________

MIXING EQUIPMENT SPECIFICATIONS

PERFORMANCE

Component	Wt. %	Sp. Gr.	Viscosity, Cp	Temp. °F

Sp. Gr. of Mixture ____________ Viscosity of Mixture ____________

Solids: ☐ Soluble ☐ Insoluble ☐ Abrasive ☐ Sticky ☐ Crystalline ☐ Fluffy

Lbs/Gal. Mixture ____________ Sp. Gr. ____________ Particle Size ____________

Suspension Sp. Gr. ____________ Settling Velocity ____________ FPM

Class: ☐ Blending ☐ Dissolving ☐ Suspending Solids ☐ Cooking ☐ Emulsifying ☐ Heat Transfer ☐ Gas Dispersion

☐ ____________ Time Req'd ____________

Mixing Type: ☐ Violent ☐ Medium ☐ Mild Foam: ☐ Slight ☐ Average ☐ Bad

Cycle: ☐ Batch: Smallest ____________ Gal. Normal ____________ Gal. Largest ____________ Gal.

☐ Continuous: Rate ____________ GPM

Mixer ☐ Will ☐ Will Not be operated during filling. Sequence of Addition ____________

Vessel Specs: Dwg. No. ____________

MATERIALS OF CONSTRUCTION

Vessel ____________ Shaft ____________ Impeller ____________

Mtg. Flg. ____________ Stuffing Box ____________ Steady Bearings ____________

Packing ____________ Other Wetted Parts ____________

SELECTION

Manufacturer ____________ Model ____________

Req'd Vessel Opening Size ____________ Pressure Class ____________ Facing ____________

Mixer Location on Vessel ____________

Mixer Angle ____________ ° with ____________

DESIGN

Impeller: Diameter ____________ Type ____________ No. ____________ Speed ____________ RPM

Normal BHP (Excluding Gear) ____________

Type Bearings ____________ Steady Bearing Req'd? ☐ Yes ☐ No. Guide Bearings Req'd? ☐ Yes ☐ No

Shaft Seal: ☐ Packing ☐ Mechanical. Make ____________ Type ____________

Seal Coolant ____________ Stuffing Box Lubrication ____________

Shaft Coupling: Type ____________ Make ____________

Gear: Manufacturer ____________ Type ____________

Size ____________ Red. Ratio ____________ Rated H.P. ____________ Max. BHP ____________

Mech. Eff. ____________ %. No Reductions ____________ Output ____________ RPM Spec'd Changable ☐ Yes ☐ No

Driver: Manufacturer ____________ Type ____________ Speed ____________ RPM

Elect. Power: ____________ Volts ____________ Phase ____________ Cycle. BHP ____________

Service Factor ____________ Frame ____________

REMARKS

By	Chk'd	App.	Rev.	Rev.	Rev.
Date					

P.O. To: ____________

Figure 5-22. Mixing equipment specifications.

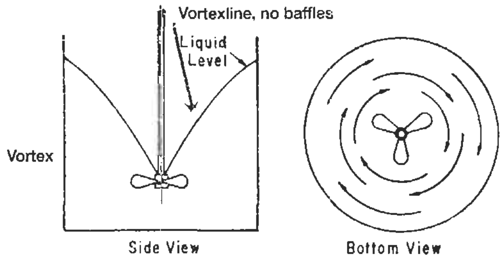


Figure 5-23A. Fluid flow pattern for propeller mounted center with *no* baffles. Note vortex formation. By permission, Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

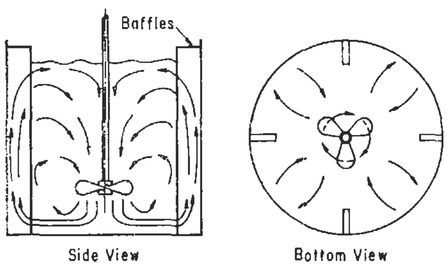


Figure 5-23B. Fluid flow pattern for propeller mounted center with baffles, axial flow pattern. By permission, Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

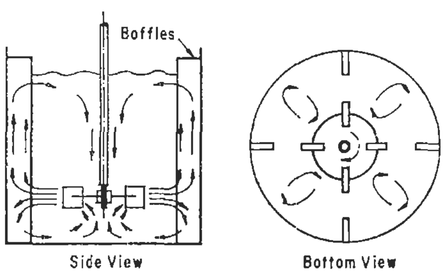


Figure 5-23C. Fluid flow pattern for turbine mounted on-center with baffles, radial flow. By permission, Lightnin (formerly, Mixing Equipment Co.), a unit of General Signal.

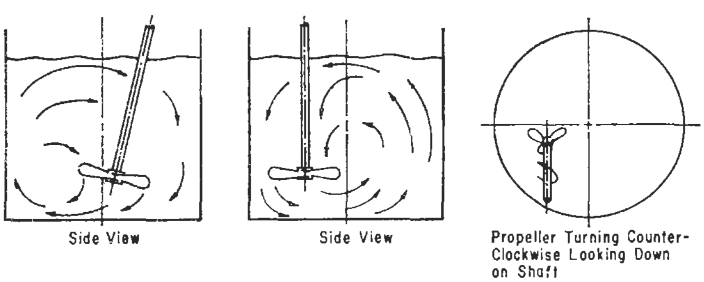


Figure 5-23D. Fluid flow pattern for propeller mounted in angular-off-center position. By permission, Lightnin (formerly, Mixing Equipment Co.), a unit of General Signal.

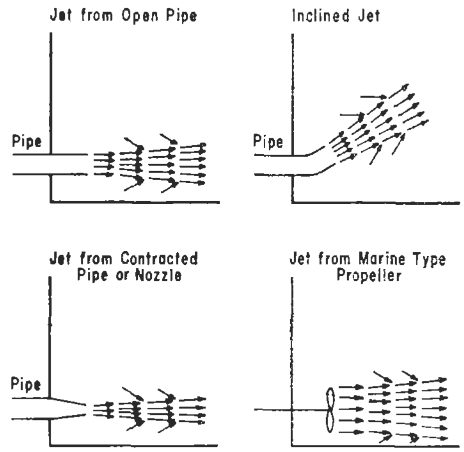


Figure 5-23E. Entraining mixing jets. By permission, Rushton, J. H., *Petroleum Refiner,* V. 33, 1954, p. 101 [17].

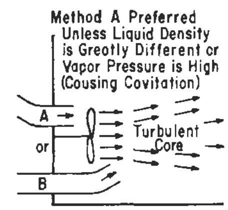


Figure 5-23F. Introducing liquid during mixer operation. By permission, Rushton, J. H., *Petroleum Refiner,* V. 33, 1954, p. 101 [17].

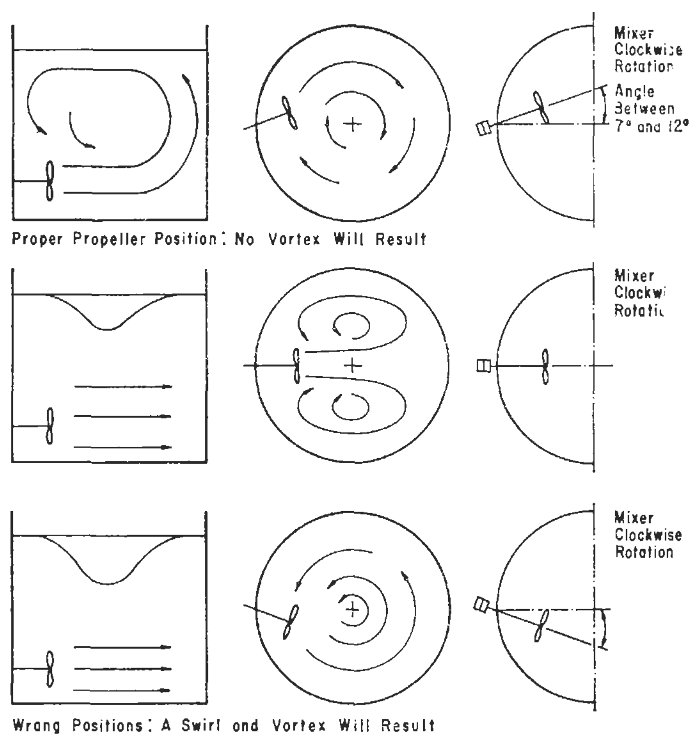


Figure 5-23G. Side-entering propeller mixer position, large tanks. By permission, Rushton, J. H., *Petroleum Refiner,* V. 33, 1954, p. 101 [17].

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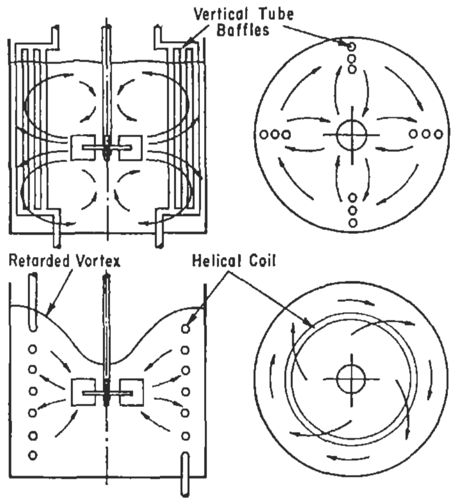


Figure 5-23H. Liquid motion patterns. A.) Vertical-tube baffles; B.) Helical coil, no other baffles. By permission, Dunlap, J. R., Jr. and Rushton, J. H., *A.I.Ch.E. Symposium Series, No. 5,* V. 49, 1953, p. 137. American Institute of Chemical Engineers [6].

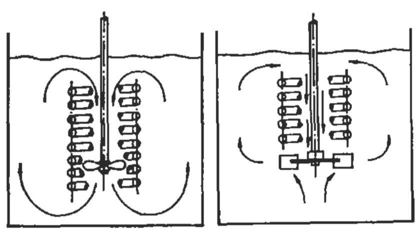


Figure 5-23I. Coil used as draft tube. By permission, Dunlap, J. R., Jr. and Rushton, J. H., *A.I.Ch.E. Symposium Series, No. 5,* V. 49, 1953, p. 137. American Institute of Chemical Engineers [6].

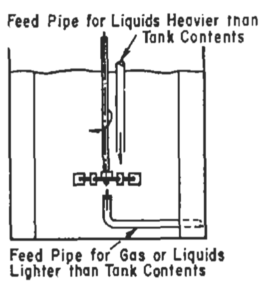


Figure 5-23J. Feed of liquids and gases to turbine. By permission, Dunlap, J. R., Jr. and Rushton, J. H., *A.I.Ch.E. Symposium Series, No. 5,* V. 49, 1953, p. 137. American Institute of Chemical Engineers [6].

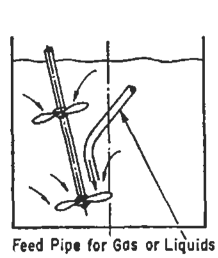


Figure 5-23K. Feed of liquids and gases to dual propellers. By permission, Dunlap, J. R., Jr. and Rushton, J. H., *A.I.Ch.E. Symposium Series, No. 5,* V. 49, 1953, p. 137. American Institute of Chemical Engineers [6].

(text continued from page 309)

cleaning, and emptying the reactor (see Figures 5-25A and B).

In batch operations, mixing takes place until a desired composition or concentration of chemical products or solids/crystals is achieved. For continuous operation, the feed, intermediate, and exit streams will not necessarily be of the same composition, but the objective is for the end/exit stream to be of constant composition as a result of the blending, mixing, chemical reaction, solids suspension, gas dispension, or other operations of the process. "Perfect" mixing is rarely totally achieved, but represents *the* instantaneous conversion of the feed to the final bulk and exit composition (see Figure 5-26).

When conducting pilot plant testing to develop a process involving mixing, which later may be used in the design of a large scale plant, it is wise to discuss the testing with a mixing specialist and outline the needed pilot data required to later scale-up the process, generally from batch pilot plant to continuous commercial process.

Scale-Up and Interpretation

Scale-up techniques for using the results of pilot plant or bench scale test work to establish the equivalent process results for a commercial or large scale plant mixing system design require careful specialized considerations and usually are best handled by the mixer manufacturer's specialist. The methods to accomplish scale-up will vary considerably, depending on whether the actual operation is one of blending, chemical reaction with product concentrations, gas dispersions, heat transfer, solids suspensions, or others.

These scale-up methods will necessarily at times include fundamental concepts, dimensional analysis, empirical correlations, test data, and experience [32].

Similarity concepts use physical and mathematical relations between variables to compare the expected performance of mixing/agitation in different sized systems [33]. This is usually only a part answer to the scale-up problem.

Geometric similarity is often considered the most important feature to establishing similarity in mixing, basing the scaled-up larger unit on the smaller initial model or test unit.

The scale-up of mixing data has been treated with a variety of approaches, some to rather disastrous results. The principles are now well established, and it is a matter of truly understanding the particular systems that poses

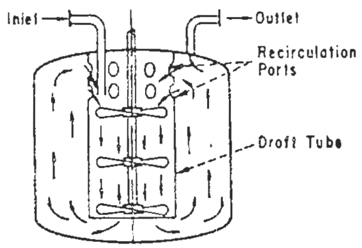


Figure 5-24A. Draft tubes prevent short-circuiting of liquid from inlet to outlet in a continuous mixing vessel. By permission, Weber, A. P., *Chem. Engr.*, Oct. 1953, p. 183 [23].

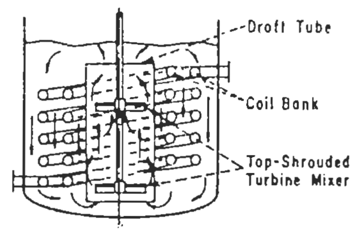


Figure 5-24B. Forced convection past heat transfer surfaces improves the overall coefficient of heat transfer. By permission, Weber, A. P., *Chem. Engr.*, Oct. 1953, p. 183 [23].

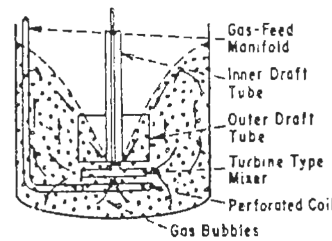


Figure 5-24C. Gas-liquid mixing is more complete when concentric draft tubes are used to recirculate gases. By permission, Weber, A. P., *Chem. Engr.*, Oct. 1953, p. 183 [23].

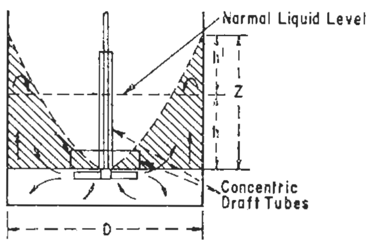


Figure 5-24D. Capacity of a draft tube assembly to suck in gases is a function of the liquid height above the rotor hub. By permission, Weber, A. P., *Chem. Engr.*, Oct. 1953, p. 183 [23].

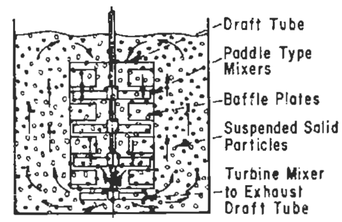


Figure 5-24E. Baffles positioned in the draft tube accentuate the direct mechanical action of low speed mixing elements. By permission, Weber, A. P., *Chem. Engr.*, Oct. 1953, p. 183 [23].

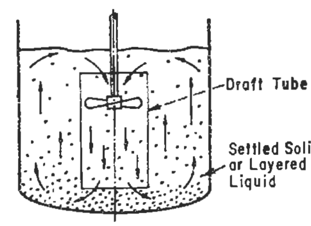


Figure 5-24F. Settled solids or layered liquids are quickly dispersed by the directionalized flow from the draft tube. By permission. Weber, A. P., *Chem. Engr.*, Oct. 1953, p. 183 [23].

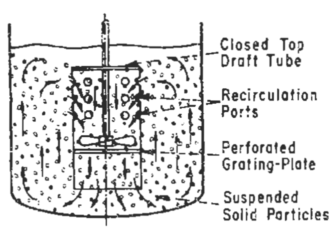


Figure 5-24G. Direct mechanical action can be increased by the addition of a grating plate to the draft tube. By permission, Weber, A. P., *Chem. Engr.*, Oct. 1953, p. 183 [23].

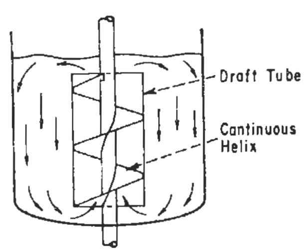


Figure 5-24H. Helix-in-draft-tube assemblies are effective for crutching pastry or fibrous materials. By permission, Weber, A. P., *Chem. Engr.*, Oct. 1953, p. 183 [23].

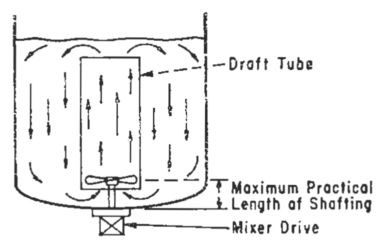


Figure 5-24I. Mechanical design problems may be solved by using a draft tube to amplify the action of the mixer. By permission, Weber, A. P., *Chem. Engr.*, Oct. 1953, p. 183 [23].

the real problem. The important similitude concept involves the following:

1. Geometric similarity requires all corresponding dimensions of a new system to have the same ratio with a test model which has proven acceptable. These dimensions should include vessel diameter and liquid level, baffle width and number in vessel, impeller diameter, number of blades and width ratio. For example, a tank four times the diameter of the original model also requires a turbine ten times the diameter of the original turbine.

2. Kinematic similarity requires geometric similarity *and* requires corresponding points in the system to have the same velocity ratios and move in the same direction between the new system and the model.

3. Dynamic similarity requires geometric *and* kinematic similarity in addition to force ratios at corresponding points being equal, involving properties of gravitation, surface tension, viscosity and inertia [8, 21]. With proper and careful application of this principle scale-up from test model to large scale systems is often feasible and quite successful. Tables 5-

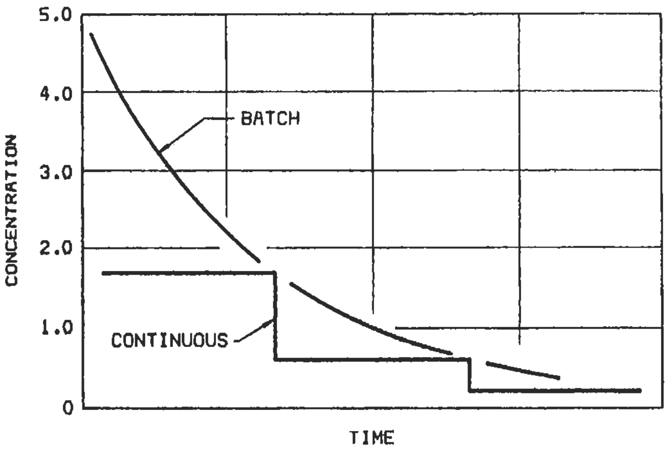


Figure 5-25A. Illustration of chemical reaction in a batch system in which concentration decreases with time, with a three-stage continuous mixing system superimposed. By permission, Oldshue, J. Y., *Fluid Mixing Technology,* 1983, Chemical Engineering McGraw-Hill Publications Co., Inc., p. 340, 347, 348 [29].

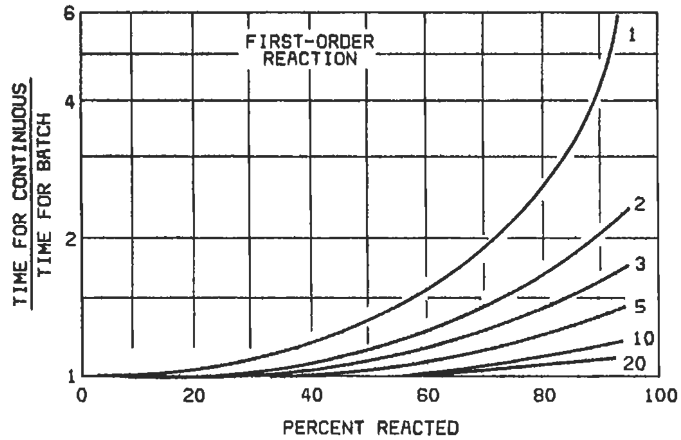


Figure 5-25B. For a first order reaction, the ratio of time in a continuous tank to the time in a batch tank for various percentages of reaction completion. By permission, Oldshue, J. Y. [29].

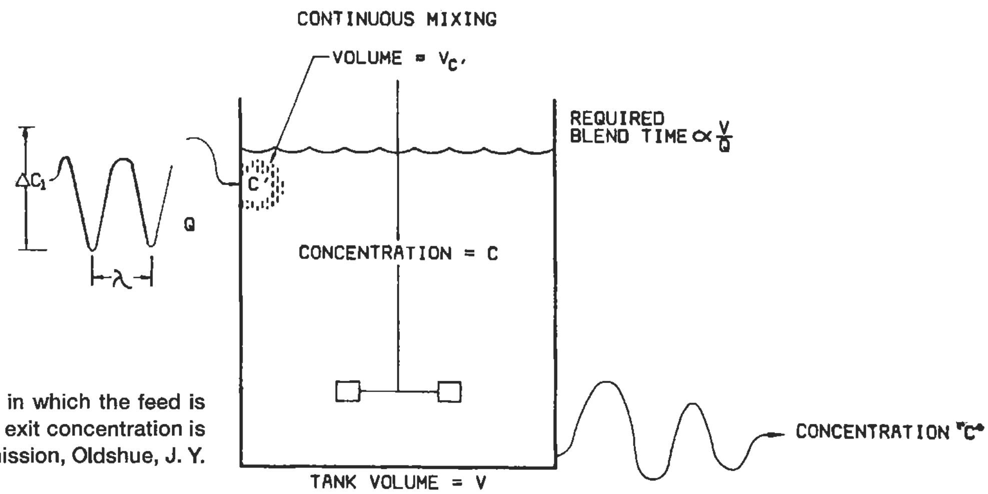


Figure 5-26. Concept of perfect mixing, in which the feed is dispersed instantly into the tank and the exit concentration is equal to the tank concentration. By permission, Oldshue, J. Y. [29].

Table 5-3
Impeller and Flow Characteristics For Turbulent, Baffled Systems Simple Ratio Relationships

At Constant ↓	P_r	D_r	N_r	Q_r	H_r	$(Q/H)_r$
Impeller Diameter, D	N_r^3	.....	$P_r^{1/3}$	N_r	N_r^2	N_r^{-1}
Speed, N	D_r^5	$P_r^{1/5}$	.....	D_r^3	D_r^2	D_r
Power, P	...	$N_r^{-3/5}$	$D_r^{-5/3}$	$N_r^{-4/5}$ or $D_r^{4/3}$	$N_r^{4/5}$ or $D_r^{-4/3}$	$N_r^{-8/5}$ or $D_r^{8/3}$

By permission: Chemineer, Inc., Dayton, Ohio[8].

Table 5-4
Impeller and Flow Characteristics—Viscous, Baffled or Unbaffled Systems Simple Ratio Relationships

At Constant	P_r	D_r	N_r	Q_r	H_r
Impeller Diameter, D	N_r^2	....	$P_r^{1/2}$	N_r	N_r
Speed, N	D_r^3	$P_r^{1/3}$	....	D_r^3	....
Power, P	....	$N_r^{-2/3}$	$D_r^{-3/2}$	N_r^{-1} or $D_r^{3/2}$	$D^{-3/2}$ or N_r

By permission: Chemineer, Inc., Dayton, Ohio[8].

3 and 5-4 present the relationships of the major variables for the two most important cases of mixing.

Often, exact or true kinematic and dynamic similarity cannot be achieved in a system requiring small scale testing to determine the effect of the design, or flexibility in design to allow for final design "trimming." Consideration should definitely be given to such flexibility as (a) mixing impeller designs that can be modified without excessive cost, or the need to build a completely new/larger/smaller unit, (b) multiple gear ratios for the gear drive, with spare ratio gears to adjust speeds, and (c) either variable speed driver or oversized driver to allow for horsepower adjustments.

The dynamic response used to describe fluid motion in the system is bulk velocity. Kinematic similarity exists with geometric similarity in turbulent agitation [32]. To duplicate a velocity in the kinematically similar system, the known velocity must be held constant, for example, the velocity of the tip speed of the impeller must be constant. Ultimately, the process result should be duplicated in the scaled-up design. Therefore, the geometric similarity goes a long way in achieving this for some processes, and the achievement of dynamic and/or kinematic similarity is sometimes not that essential.

For scale-up the "shear-rate" of the fluid, which is a velocity gradient that can be calculated from velocity profiles at any point in the mixing tank [29], is an important concept. The shear rate is the slope of the velocity versus distance curve. Using the time average velocity yields shear rate values between the adjacent layers of fluid that operate on large particles of about 200 micron or greater. In Figure 5-27, usually a maximum shear rate will exist at the impeller jet boundary. The average shear rate is primarily a function of the time average velocity and impeller speed, and is not a function of any geometric type of impeller or the impeller diameter [29]. The maximum shear rate exists at the jet boundary and is a *direct function of impeller diameter and speed,* which is related to the peripheral speed of the impeller. Thus, on scale-up, the maximum impeller zone shear rate tends to increase while the average impeller zone shear rate tends to decrease [29].

The fluid shear stress actually brings about the mixing process, and is the multiplication of fluid shear rate and viscosity of the fluid [29].

The pumping capacity of the impeller is important in establishing the shear rate due to the flow of the fluid from the impeller.

There is no constant scale-up factor for each specific mixing system/process [29]. The two independent impeller variables come from speed, diameter, or power, because once the impeller type/style has been selected, the two variables can be established. The third variable is tied through the power curves (plot of power number v. N_{Re}, see Figures 5-13, 5-14, and 5-15). Figure 5-28 shows that geometric and dynamic similarity can develop useful relationships for some situations [29], but not all, and it is not truly possible to prepare one dimensionless group expressing a process relationship. This suggests that care must be used in resorting only to a dimensionless number for process correlations. Also see Figures 5-29 and 5-30.

Because the most common impeller type is the turbine, most scale-up published studies have been devoted to that unit. Almost all scale-up situations require duplication of *process results* from the initial scale to the second scaled unit. Therefore, this is the objective of the outline to follow, from Reference [32]. The dynamic response is used as a reference for agitation/mixer behavior for a defined set of process results. For turbulent mixing, kinematic similarity occurs with geometric similarity, meaning fixed ratios exist between corresponding velocities.

For scale-up procedure, refer to Figure 5-31, which outlines the steps involved in selecting commercial or industrial mechanical agitation equipment when based on test data.

- Test data should be planned by knowledgeable specialists in order to obtain the range, accuracy, and scope of needed data to achieve a pre-established mixing process result.
- While obtaining test data, scale-up calculations should be made regularly to determine if the end results will be practical, particularly from the available mixing hardware, motor power, etc.

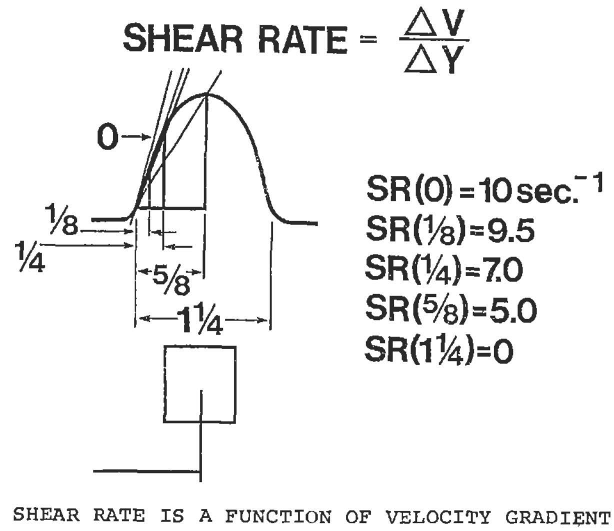


Figure 5-27. Shear rate is a function of velocity gradient. By permission, *Lightnin Technology;* Lightnin Technology Seminar, 3rd ed., 1982, Lightnin (formerly Mixing Equipment Co.), a unit of General Signal, p. 1, Section 2A [27].

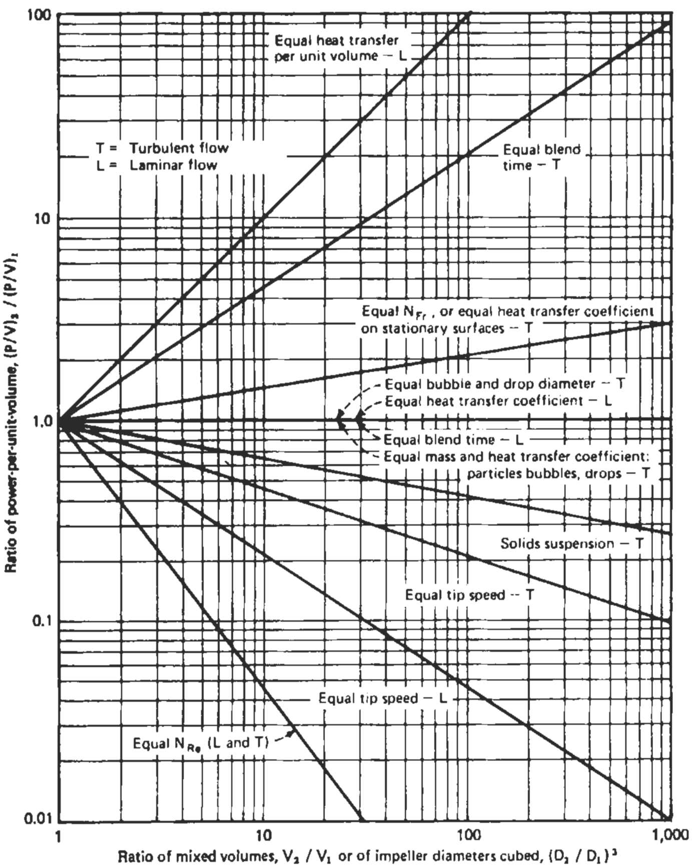


Figure 5-28. Mixing scale-up factors referenced to experienced ratios of power per unit volume. By permission, Penny, W. R., *Chem. Engr.*, Mar. 22, 1971, McGraw-Hill, Inc., p. 88 [31].

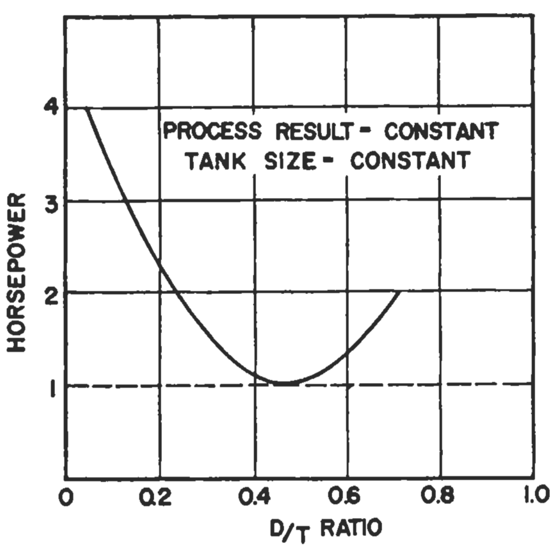


Figure 5-29. Effect of D/T on power requirement for a given process result. By permission, *Fluid Mixing*, Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

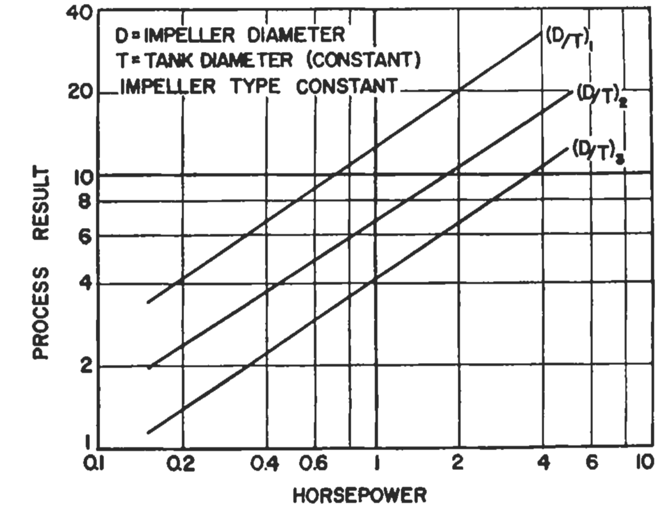
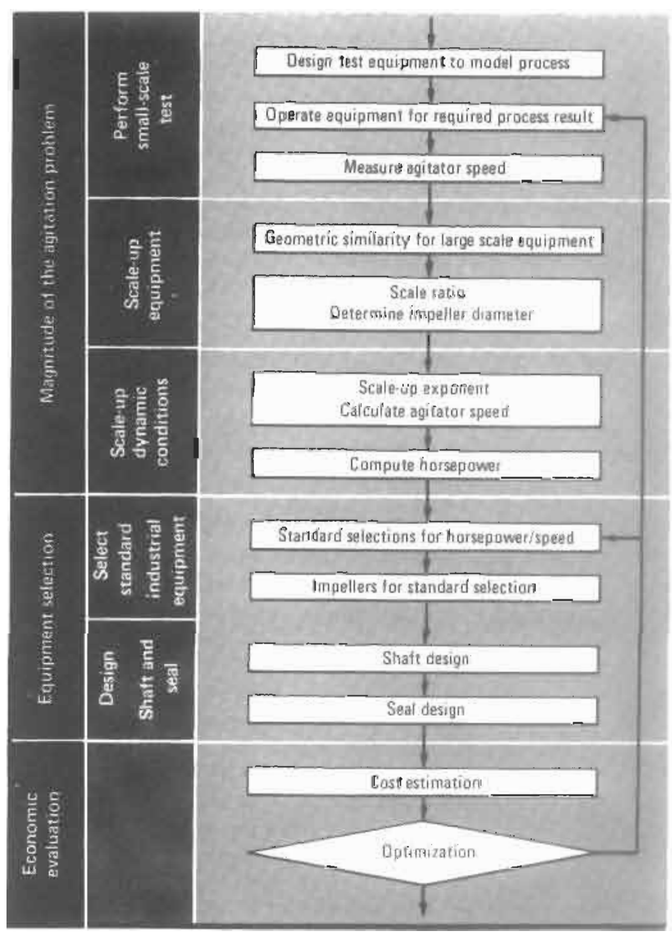


Figure 5-30. Effect of power on process result with constant D/T ratio. By permission, *Fluid Mixing*, Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.



Design procedure for agitator scale-up **Fig. 1**

Figure 5-31. Design procedure for agitator scale-up. By permission, Rautzen, R. R., *et al.*, *Chem. Engr.*, Oct. 25, 1976, p. 119 [32].

- Determine geometric similarity to develop a single scale ratio R, for the relative magnitudes for all linear dimensions [32].

$$D_1/T_1 = D_2/T_2 \tag{5-43}$$

$$W_1/D_1 = W_2/D_2 \tag{5-44}$$

$$Z_1/T_1 = Z_2/T_2 \tag{5-45}$$

when (1) = small size data unit
(2) = proposed larger scaled-up unit

$$\text{then } R = \text{scale ratio} = D_2/D_1 = T_2/T_1 = W_2/W_1 = Z_2/Z_1 \tag{5-46}$$

To select a turbine, there must also be geometric similarities for the type of turbine, blade width, number of blades, impeller diameter, etc. From the geometric similarity determination of the turbine diameter, the mixer speed can be established to duplicate. The "Scale Ratio R,"

$$N_2 = N_1\ (1/R)^n = N_1\ (D_1/D_2)^n \tag{5-47}$$

where n is based on theometrical and empirial considerations and varies with the specific type of mixing problem [32]. See Figure 5-32.

Often the scaled-up design provides equipment or speeds that are non-standard, which then require adjustments by back-calculating from the nearest standard mixer diameter or gear speed to be able to use the industry or manufacturer's standard.

The scale-up exponent, n, is given for typical mixing conditions in Figure 5-32.

"Rules of thumb" regarding scale-up and good design practice are not suitable for determining cost of performance design or physical capital cost.

For geometric similarity of liquid motion (n = 1.0) the linear scale ratio of volume is

$$R = (V_2/V_1)^{1/3} \tag{5-48}$$

The volume ratio can be related to a speed ratio for a given scale-up exponent, see Figure 5-32. The usual range for the scale-up exponent is between 0.67 and 1.0. To select a scale-up exponent, the following provides a guide:

A. n = 1; equal liquid motion

Liquid blending, equivalent *liquid* motion, corresponding velocities are about equal. Similar results obtained with equal tip speed or torque per unit volume.

B. n = 0.75; equal solids suspension

Equal suspension of particles referenced to visual appearances and physical sample testing. Empirical correlations generalized to apply to most problems.

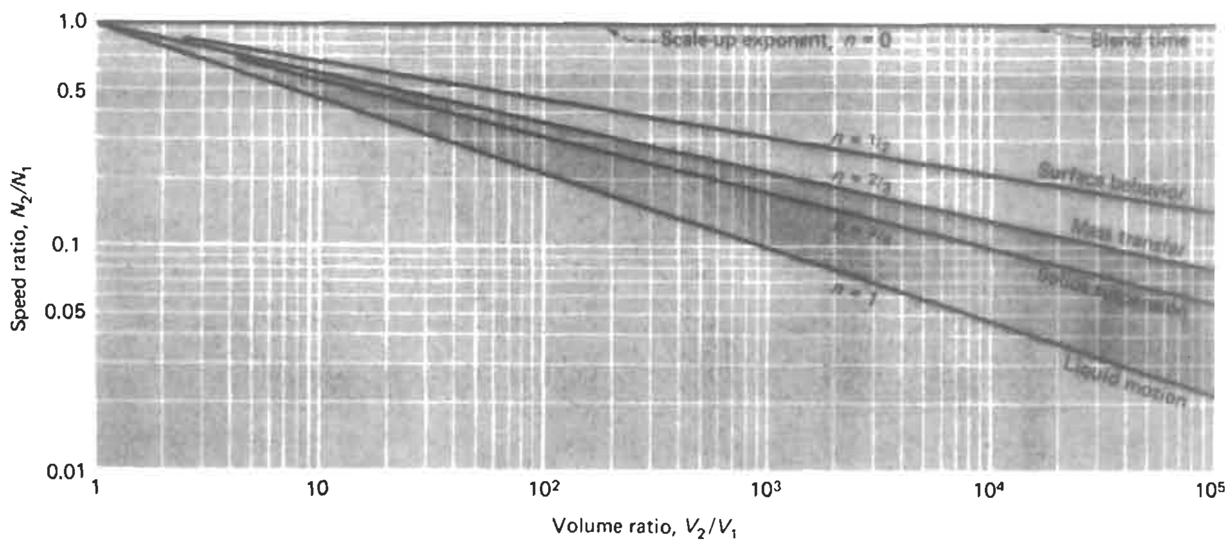


Figure 5-32. Scale-up exponent characterizes the desired type of agitation in order to determine speed-volume ratios. By permission, Rautzen, R. R., *et al.*, *Chem. Engr.*, Oct. 25, 1976, p. 119 [32].

C. n = 0.667; equal mass transfer rate
More directly related to turbulence and motion at the interface. Includes scale-up for rate of dissolving of solids or mass transfer between liquid phases. Using geometric similarity and equal power per volume results in same n value.

D. n = 0.5; equal surface motion
Related to vortexes formation, depth of vortex related to geometric similarity and equal Froude number.

E. N = 0; equal blend time
Rarely used due to large equipment required to hold speed constant for larger units.

For turbulent power, (constant power number, P_o):

$$P\ p\ N^3 D^5 \tag{5-49}$$

$$V\ p\ D^3 \tag{5-50}$$

$$P/V\ p\ N^3 D^2 \text{ (power per volume)} \tag{5-51}$$

For P/V constant in two different systems, then

$$(N_1)^3(D_1)^2 = (N_2)^3(D_2)^2 \tag{5-52}$$

$$\text{or, } N_2 = N_1(D_1/D_2)^{2/3} = N_1(1/R)^{2/3} \tag{5-53}$$

Figure 5-33 provides a selection grid for establishing the industrial standard for the driving equipment, that is, input horsepower of motor and speed of agitator. Select the calculated point in the grid, then move to the nearest speed and *input* horsepower (not motor hp). Usually adjustments from actual design will be required, but often the incremental adjustments will not be great. To aid in these adjustments, equal torque will give equal liquid motion or solids suspension over a narrow range. To use equal torque, set up diagonals from lower left to upper right for the available equipment. Adjust or back-calculate (if necessary) the expected performance and dimensions based on the adjustment. Higher speeds require higher horsepower. For equivalent mass transfer, equal power changes are used.

To recheck final design, the diameter of a single, pitched blade turbine, for turbulent conditions:

$$D_T = 394\,(P_{hp}/S_g\,N^3)^{1/5}\text{, in.} \tag{5-54}$$

D_T = impeller diameter, turbulent operation, in.
P_{hp} = horsepower used by system (not motor horsepower of driver)
S_g = specific gravity of fluid
N = agitator impeller speed, rpm

Of course economics enters into the solution, so some alternate designs may be helpful.

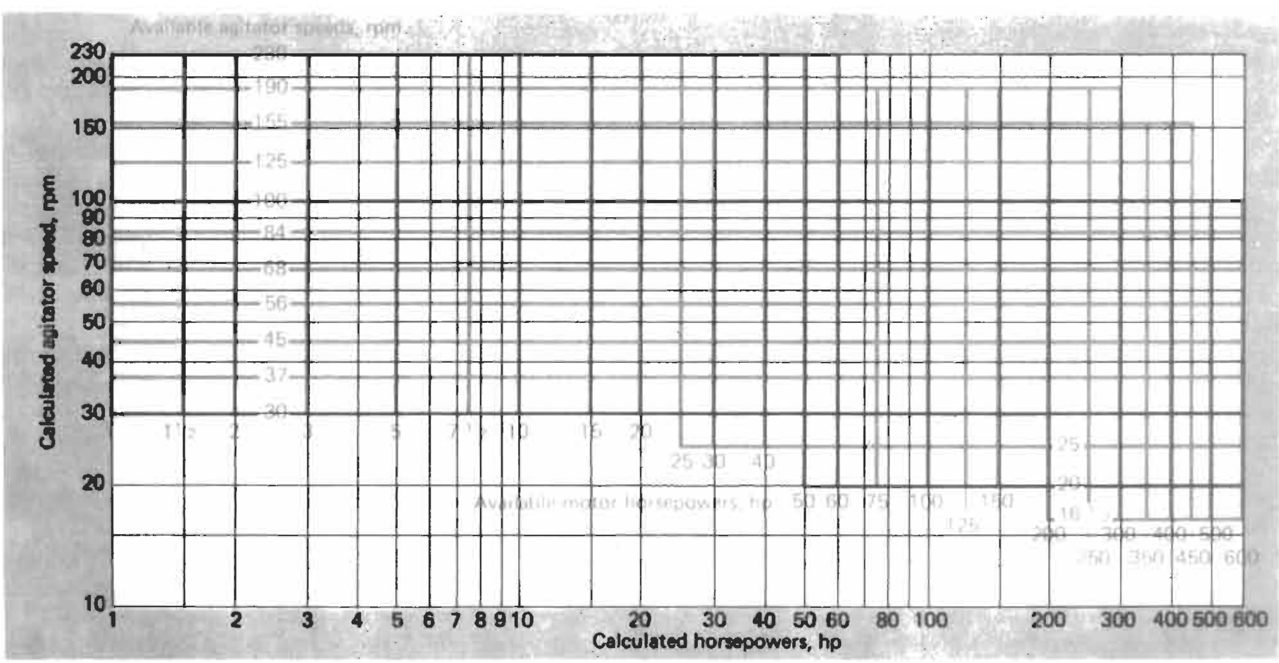


Figure 5-33. Scaled-up horsepower/speed requirement for an agitator system is readily related to industrial equipment. By permission, Rautzen, R. R., *et al., Chem. Engr.,* Oct. 25, 1976, p. 119 [32].

Five forces that can usually be used for scale-up are:

- Input force from mixer, function of [25]
  impeller speed
  impeller diameter
- Opposing forces, functions of
  viscosity
  density of fluid
  surface tension
- Dynamic similarity requires that the ratio of input force, viscosity, density, and surface tension be equal. For the same fluid, only two of these four forces need be equal, because the density and viscosity will be the same [34, 29].

  The geometric and dynamic similarity can use dimensionless groupings.
- Geometric [29]:

$$X_m/X_p = X_R, \text{ common ratio} \tag{5-55}$$

X_m, X_p = dimension of model, and scale-up unit, respectively
X_R = ratio of dimensions

- Dynamic [29]:

$$\frac{(F_I)_m}{(F_I)_p} = \frac{(F_v)_m}{(F_v)_p} = \frac{(F_G)_m}{(F_G)_p} = \frac{(F_\sigma)_m}{(F_\sigma)_p} = F_R \tag{5-56}$$

F = force
Subscripts: I = inertia force
v = viscosity force
G = gravity
σ = interfacial tension
R = ratio
m = model
p = prototype

Force ratios:

$$\frac{F_I}{F_v} = N_{Re} = ND^2\rho/\mu \tag{5-57}$$

$$\frac{F_I}{F_G} = N_{Fr} = N^2D/g \tag{5-58}$$

N_{Fr} = Froude number

$$\frac{F_I}{F_\sigma} = N_{We} = N^2D^3\rho/\sigma \tag{5-59}$$

N_{we} = Weber number

Heat Transfer, Hydraulic Similarity [29]:

$$h = f(N, D, \rho, \mu, cp, k, d) \tag{5-60}$$

$$\left(\frac{\text{result}}{\text{system conductivity}}\right) = f\left(\frac{\text{applied force}}{\text{resisting force}}\right) \tag{5-61}$$

$$\left(\frac{hD}{k}\right) = \left(\frac{ND^2\rho}{\mu}\right)^x \left(\frac{c_p\mu}{k}\right)^y \left(\frac{D}{d}\right)^z \tag{5-62}$$

x, y, z are empirical exponents
k = thermal conductivity
d = heat transfer tube diameter
ρ = density of fluid or specific gravity
μ = viscosity of fluid

Blending, Hydraulic Similarity:

$$\theta = f(N, D, \rho, \mu, T) \tag{5-63}$$

where θ = time
T = tank diameter

(Result/system conductivity) = f (applied force/resistancy force)

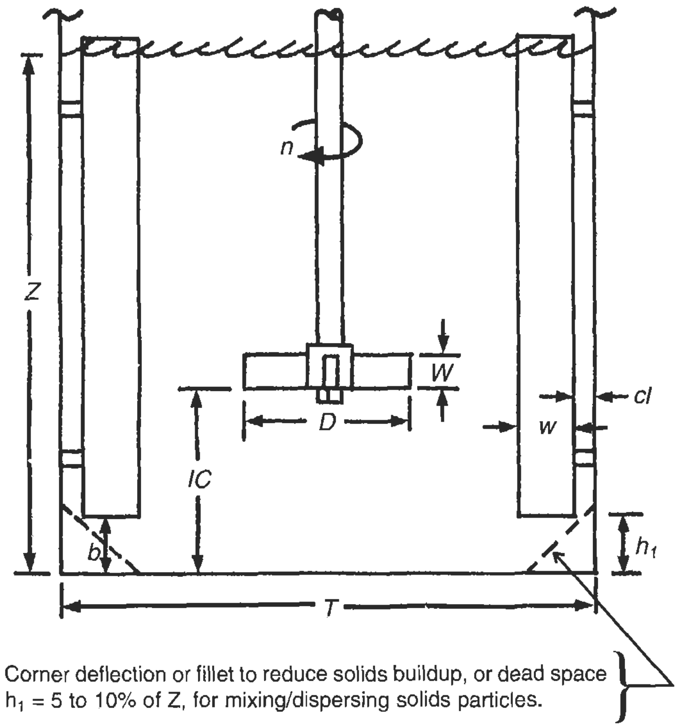
$$\theta N\rho \ ([ND^2\rho/\mu])^x (D/T)^z \tag{5-64}$$

x and z are empirical coefficients.

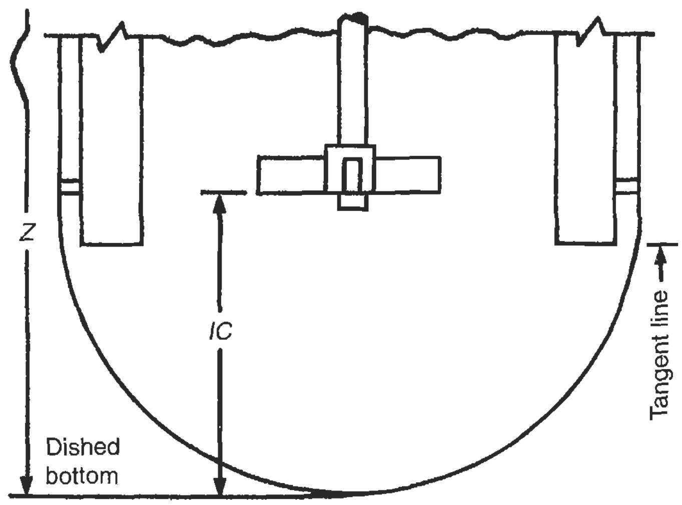
Example 5-1: Scale-up from Small Test Unit [32], See Figure 5-34

Follow the example of Reference [32], using scale-up rules. A pilot plant test run has been conducted using a laboratory equipped test vessel. Design equivalent process results for a 10,000 gallon tank are:

Data from Test Unit	*Proposal Vessel*
Vessel dia, T_1 = 12 in.	T_2 = 144 in.
Vessel liquid level, Z_1 = 12 in.	Z_2 = 144 in.
Batch volume, V_1 = 6 gal.	V_2 = 10,000 gal.
Impeller dia, D_1 = 4 in.	D_2 = 48 in.
Impeller shaft speed, N_1 = 450 rpm	N_2 = 90 (calc.)
Impeller width, W_1 = 1 in.	W_2 = 12 in.
Baffle width, (4) B_1 = 1 in.	B_2 = 12 in.
Distance to impeller, C_1 = 4 in.	C_2 = 48 in.
HP input to shaft, HP_1 = 0.0098 (calc.)	HP input HP_2 = 19.5 (calc.)



a. Flat-bottom tanks.

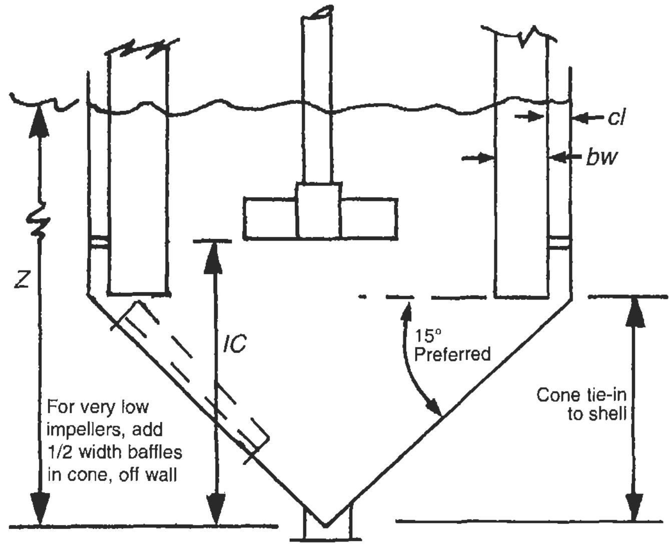


b. Dished-bottom tanks. Most efficient tank design.

Figure 5-34. Typical vessel baffles to improve mixing performance. Adapted/modified by permission, Casto, L. V., *Chem. Engr.*, Jan. 1, 1972, p. 97 [30].

From scale-up ratio:

$R = (V_2/V_1)^{1/3} = (10{,}000/6)^{1/3} = 11.85$, round to 12



c. Cone-bottom tank, four baffles, 90° apart. Least efficient tank design.

Then, using relationships of volume ratios to geometric similarity,

$T_1/T_2 = 12/R$
$T_2 = (12 \text{ in.})\ (12) = 144$ in.
$Z_2 = (12 \text{ in.})\ (12) = 144$ in.
$D_1 = (4 \text{ in.})\ (12) = 48$ in.
$W_2 = (1 \text{ in.})\ (12) = 12$ in.
$B_2 = (1 \text{ in.})\ (12) = 12$ in.
$C_2 = (4 \text{ in.})\ (12) = 48$ in.
$N_2 = ?$

For example blend time as the criterion, the scale-up exponent = 0, and no change in speed is required for the larger scale equipment, requiring that the longer 48-inch diameter turbine operate at 450 rpm,

$HP = (D_T/394)^5(S_g)(N)^3$
$= (48/394)^5(1)(450)^3$
$= 2445.4$ hp input to shaft of impeller, *impossible to use.*

Alternating, scaling up for equal mass transfer, with n = 0.667, reading Figure 5-32 speed ratio, $N_1/N_2 = 0.2$, at $V_2/V_1 = 10{,}000/6 = 1{,}666$

Then, $N_2 = 0.2(450) = 90$ rpm shaft speed

and, $HP = (D_2/394)^5\ (S_g)(N)^3$
$HP = (48/394)^5(1)(90)^3 = 19.5$ hp input to shaft

Actual motor hp: $19.5/0.85 = 22.9$

Using the grid in Figure 5-33, calculated shaft hp = 19.5, and agitator speed = 90 rpm.

Read closest motor hp = 20. However, the 0.5 hp difference between the 19.5 and 20 may not be sufficient to handle the power loss in the gear box. Most industrial practice is to take the closest standard *motor* hp to the 22.9 hp determined above, which is 25 hp. The gear box must have an output speed of 90 rpm and will use only the hp determined by the impeller shaft even if the motor is larger, that is, 25 hp. It will only put out the net hp required, that is, the sum of impeller shaft and losses through the gear box.

Referring to Table 5-3 for turbulent, baffled systems, if power is held constant and the system has too large a shear characteristic and apparently too small a volume or flow, the impeller can be increased 20% and the new speed at constant Power, P, will be:

$$N_r = D_r^{-5/3} \quad (5\text{-}65)$$

$$\frac{N_1}{N_2} = \left(\frac{D_1}{D_2}\right)^{-5/3} \quad (5\text{-}66)$$

$$N_2 = N_1\left(\frac{D_2}{D_1}\right)^{-5/3} = N_1\left(\frac{1.2}{1.0}\right)^{-5/3} = 0.738\ (N_1) \quad (5\text{-}67)$$

(Use Table 5-4 for viscous systems.)
or the new speed will be 73.8% of the original, using the 20% larger diameter impeller. This is true for geometrically similar systems. However, there is little power change over a wide ratio of impeller-to-tank diameter.

For a constant amount of power available to a system, the flow and turbulence effects and ratios can be changed by replacing one impeller by another dimensionally similar. Figures 5-29 and 5-30 illustrate the type of studies which should be made in evaluating a system. If the density or viscosity of a fluid changes during scale-up, then in the turbulent range the horsepower is directly proportional to density, and thus viscosity has very little effect. In viscous flow the density has no effect while the horsepower is proportional to viscosity [8]. The effect is small in the range from 1 to 1000 centipoise, but amounts to a factor of 1.4 when changing from 1000 to 10,000 centipoise. Above this point the change is quite large and should not be handled by proportion.

Figure 5-28 summarizes the scale-up relationships for many of the important and controlling functions, depending upon the nature of the process equipment. The figure identifies which curves apply to turbulent (T) and laminar (L) flow patterns in the fluids being subjected to the mixing operation. Note that the Froude number, N_{Fr} is a function. This scale-up chart applies to systems of similar geometry. When the geometry is different, special and specific analysis of the system must be made, as the chart will not apply.

Baffles

Vertical side-wall baffles (Figures 5-23B and C and 5-34) projecting about 1/10 to 1/12 of the tank diameter into the vessel perform a helpful purpose in controlling vortex action. The baffles are set off from the tank wall a few inches to prevent build-up of particles.

The important dimensional features and/or ratios for a center-mounted mixer unit (vertical) are [29, 30]:

1. Flat bottom tank
   a. Number of vertical baffles in a vertical vessel: four (more than four provides little, if any, benefit)
   b. Width of baffles: 1/10 to 1/12 tank diameter, w
   c. Distance baffles off wall, cl: 3 in. to 6 in.
   d. Baffle spacing: on 90° around tank
   e. Distance of baffles off flat vessel bottom: 4 in. to 6 in., b
   f. Height of liquid in tank: Z
   g. Height of impeller off flat bottom: IC., equal to impeller (turbine) diameter, or IC = D; sometimes IC = 2/3 D is suggested
   h. Liquid depth over top of impeller or turbine: Z-C should be 2D
   i. Baffles extend above liquid level
2. Dished bottom tank, center mounted mixer, 4 baffles
   a. Essentially same criteria as for flat bottom tank
   b. Impeller distance off bottom: essentially same *referenced to vessel tangent line* as for flat bottom vessel
3. Cone bottom tank. Handle as for dished bottom above.
4. For fluids with viscosity up to 5000 cp, and even up to 30,000 cp for some situations, use standard baffles described above. The baffle widths can be reduced as the viscosity increases from 5,000 cp to 12,000 cp. [30], and may be eliminated completely for viscosities over 12,000 cp. There are exceptions, such as mixing wide range of fluids of low and high viscosities.
5. Special baffles. For certain mixing problems, various baffling arrangements have been found to be advantageous (see Reference [30]).
6. Baffles can be omitted when propeller mixers are top mounted at an angular off-center position (see Figure 5-23D) and vortex swirling is prevented. This is not recommended for large power systems on large tanks, due to shaft fatigue.

Baffles that extend from the liquid level down, but not to, the tank bottom allow heavy swirling action in the bottom of the tank, but no vortex at the top. When baffles extend from the bottom up, but not to the liquid level, some vortex and swirling action will take place at the top.

In this case foam can be re-entered into the mixture by this action, and solids or liquids added will enter the impeller rather rapidly. The deeper the liquid above the baffle the greater the rotating action. Bottom swirling action allows the segregation of heavy solid particles [21].

In general, some sidewall baffles are desirable in most mixing operations. Baffles allow the system to absorb relatively large amounts of power which is needed for development of mixing turbulence, and still avoid vortex and swirling action, that is, the tank fluid stays under control. This is indicated typically by Figure 5-15 for the flat blade impeller in area C D (or the similar region on other figures). Use 4 vertical baffles, Figures 5-23B, C and 5-34.

A large number of mixing problems operate in this region and can be easily interpreted. Here large amounts of power can be added to the system for greater volume and/or shear forces by simply increasing the speed. This is one reason for variable speed drives. However, in the portion of Figure 5-15 lettered EF as well as AB, BC, and BE, the power changes exponentially with the speed. The advantage of using baffles is that the flow pattern is fixed to follow the portion CD of the curves.

When fluid swirling is prominent, it is difficult or impossible to reproduce the same mass transfer effect in any other size vessel, and hence cannot be reproduced by geometric similarity [21].

Mixing operations are not limited to a flat bottom tank, but in general, for each system there is a tank configuration which is optimum. The difference between them is small in many instances.

A tank bottom of dished or spherical shapes is usually better than a flat bottom as it requires less horsepower for the system.

Impeller Location and Spacing: Top Center Entering

For base tank dimensions, Z/T, (height/tank diameter) equal to 1.0 [29]:

A. For blending and solid suspension, use Z/T for minimum power at about 0.6 to 0.7, but recognize that this may not be the most economical.
B. One or two impellers can operate on one shaft at liquid coverages (Z-IC) over the impeller of at least 3D, see Figure 5-34.
C. Draw-off of the mixed liquids influences the location of the impeller. For blending, the preferred impeller location in a Z/T = 1.0 Vessel is at the midpoint of liquid depth for a continuous flow process, but may not be for a batch system.
D. It is important to avoid zone mixing, when each impeller on a shaft (of more than one impeller) mixes its own zone, and then a stagnant or less-mixed region develops between the impellers. Multiple axial-flow impellers have less tendency in this regard than do multiple radial flow impellers. As viscosity of one fluid increases, the flow pattern becomes more radial, thus the tendency for zone mixing increases. When axial flow impellers are too close together on one shaft, they tend to behave as a single larger impeller, with decrease in power draw but with a decrease in pumping capacity also.
E. For radial flow turbine, locate 1.5D to 2.0D apart, with liquid coverage over the top impeller of minimum ½ to 3.0D, depending on surface motion desired.
F. Polyethylene polymer autoclave type reactors usually contain 8 to 120 impellers of the same or different circulation designs on a single shaft to ensure rapid total homogeneous mixing in the reactor, which contains a gas at about 30,000 psi and, hence, the fluid is neither a gas or a liquid because the densities are about the same.

Reference [30] suggests the depth distances in a tank over which the turbines are effective mixers:

Liquid Viscosity, Cp	*Turbine Diameters (Vertical Spacing)*
<5,000	3.0–4.0
5,000	2.5
15,000	2.0
25,000	1.7
50,000	1.4

The turbine should be positioned to give 35% of the liquid below and 65% above the turbine. For depths exceeding these values, multiple impellers should be used. For total liquid depths less than the effective height of a turbine impeller, the turbine should be located 0.35 Z from the tank bottom, unless there is an overriding restraint.

The top entering mixer units are better for producing flow at constant power than the side entering units [29]. The radial flow impellers require more horsepower when compared to the marine impellers.

If a propeller is located quite close to the bottom of a tank, the flow becomes radial like that of the flat blade turbine. In a properly baffled system the propeller flow is axial. When dynamic similarity is accomplished, the systems are similar [21]. For a first approximation, placing the impeller at ⅙ of liquid height off the bottom is good.

When possible it is best to withdraw the fluid from a propeller-mixed system directly below the propeller. This allows removal of all solids and mixed liquids.

For a turbine the preferred location for withdrawal of mixed fluid is at the side opposite the turbine impeller. A study of the flow pattern of the system should be made to be certain that the proper fluid mixture is sampled or removed.

Side entering mixers (usually propellers) as shown in Figure 5-23G are placed 18–24 inches above a flat tank floor with the shaft horizontal, and at a 10° horizontal angle with a vertical plane through the tank centerline. This equipment is used for fluids up to 500 centipoise [21]. For fluids from 500–5000 c.p., the mixer is usually top entering.

Viewing from behind the mixer, for a clockwise rotation of the impeller, the mixer must be angled to the left at the 7°–10° angle to the vertical plane noted above. Side-entering mixers usually run at 280 to 420 rpm, compared to 30 to 100 rpm for top entering units. The side-entering units are primarily used for large blending tanks of gasoline, oils, chemicals (watch the shaft seal through the tank here), paper stock blending, and similar large systems, but they do not necessarily have to be limited to large systems.

The proper placement of the impeller for specific applications is necessary for good mixing performance; therefore, a thorough discussion with a mixing company specialist is useful.

In gas dispersion systems the gas inlet should normally be directly below the impeller inlet, or on a circular pattern at the periphery of the impeller.

In order to achieve uniform solid suspension or pick-up of solid particles off the bottom, the upward velocities of the fluid streams in all portions of the vessel must exceed the terminal settling velocity of the particular particles. This can be determined by small scale tests.

For solids which float, or which are added from the top, a vortex action helps to draw the material down into the impeller. Often a draft tube is used to serve as a suction entrance for the impeller. Uniform suspensions are difficult to maintain when the tank liquid height is much greater than the tank diameter. The impeller is normally placed ⅙ of liquid depth off the bottom [21].

Process Results

The effect on the process of a change in operation of the mixer system (impeller, baffles, etc.) is the final measurement of performance. Thus, operations such as blending, uniform particle suspension, reaction, gas absorption, etc., may be acceptable under one physical system and not so to the same degree under a slightly modified one. The ratio per unit volume on scale-up must be determined experimentally.

Generally as system size increases the impeller flow per fixed power input will increase faster than will the turbulence of the system. Even the same degree of turbulence is no insurance that the rates of mixing, mass transfer, etc., will be the same in the larger system. Once it can be determined whether the process result is controlled, more or less, by flow or turbulence, the scale-up can be more intelligently analyzed with the use of the relations previously presented; noting that head, H, is the turbulence indicator. Generally speaking, for fixed power input the relative proportions of flow and turbulence in a given system vary as indicated in Table 5-5.

Table 5-5
Expected Proportions of Flow and Turbulence in a Mixing System

Relative Impeller Diameter	Percent Flow	Percent Turbulence	Relative Speed
Large	High, > 50	Low, < 50	Low
Medium	About 50	About 50	Medium
Small	Low, < 50	High, > 50	High

Operations such as blending, solids-suspension, dissolving, heat transfer and liquid-liquid extraction are typical of systems requiring high flow relative to turbulence, while gas-liquid reactions and some liquid-liquid contacting require high turbulence relative to flow. The case of (1) 100% of suspension—requires head to keep particles suspended and (2) 100% uniformity of distribution of particles—requires head for suspension plus flow for distribution.

In the case of heat transfer using coils in the tank, it is generally necessary to increase the horsepower per unit volume from small scale to full size equipment.

For some scale-up situations, particularly when the change in size of the system is not great and the fluid properties remain unchanged, the use of horsepower per unit of liquid volume is an acceptable scale-up tool.

The horsepower per unit volume is fairly constant with increasing tank volume, actually falling off slightly at large volumes. Therefore, a usually safe scale-up is to maintain a constant HP/volume.

Bates [8] describes the handling of process results for reaction completion, gas absorption, phase distribution when related to power, as shown in the log-log plot of Figure 5-35.

The slope of the line is significant and serves as a guide to the type of mixing mechanism required. The irregular line to the left of point U indicates non-uniformity of tank contents. At U, homogeneity is accomplished. The System C line indicates no change is required in horsepower in order to achieve better or different results. This would be typical of a blending operation. For System B or any system between C and B where the slope of the line is between 0 and 0.1, the action is only slightly influenced by

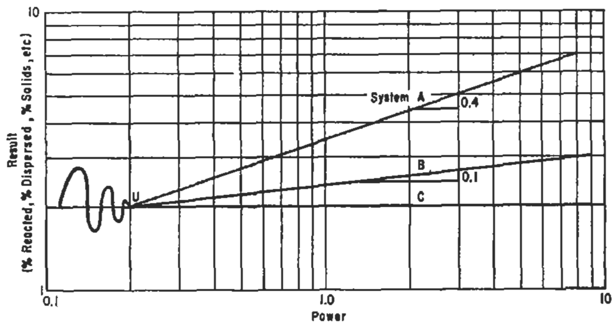


Figure 5-35. Process result as function of power. By permission, *Fluid Agitation Handbook*, Chemineer, Inc.

mixing, and for scale-up the system will be more controlled by the process and fluid properties than by increased mixing action. A change in D/T will only slightly influence the process result.

For system A and generally those in between systems with slope 0.1 to 0.4, the process result is primarily dependent upon impeller flow and will be influenced by a change in D/T. For those systems with slopes greater than 0.4, the process result is significantly influenced by fluid head and shear and by the ratio of impeller flow to shear. The determination of the proper D/T is worth close study [8].

Blending

Blending of two or more fluids into a uniform mixture is quite common in the finishing of many chemical and petroleum products. This includes the addition of additives as well as upgrading off-specifications with above-specification material to yield a salable product. Rushton [17] describes blending in large tanks and Oldshue *et al.*, [14] evaluate factors for effective blending. The 7°–12° angle with a flat bottom tank plan centerline as shown in Figure 5-23G has been shown to be optimum for efficient blending in small and large tanks. The angle should be to the left of the centerline and the propeller should rotate clockwise when viewed from the shaft or driver end. The results for low viscosity fluids (0.3 to 1.0 cp) relate time for a complete blend using a side-entering 3-blade propeller mixer starting with a full stratified tank (no baffles) of two liquids [21].

$$\theta = k'\left(\frac{\rho_h - \rho_l}{\rho_h}\right)^{0.9}\left(\frac{D}{T}\right)^{-2.3}(H.P.)^{-1} \qquad (5\text{-}68)$$

This time is considerably longer than for the arrangement with the second fluid entering the suction of the operating propeller.

For side entering horizontal mixers not limited to blending operations, there are some differences in recommendations concerning the physical location of the impeller:

1. The impeller should be located ½ to 1½ times the impeller diameter away from the tank wall in plan.
2. The impeller centerline should be ¾ to 2D off the tank bottom.
3. The impeller shaft should make a plan angle of 8° to 30° (10° optimum) to the left of a centerline of the tank.

For blending design and selection of mixing impellers, the fluids are divided into those below and above 50,000 cp [29]. Different impellers must be selected for the various ranges of viscosities, even within the 50,000 cp limits.

Blending is usually involved in developing uniform viscosities, densities, and temperatures. For best blending performance, the mixer should be operating while a second fluid is added to an initial tank of original fluid. This aids in preventing the tank contents from stratifying. To aid the manufacturer of the mixing equipment, the owner's engineer should provide viscosity and shear rate data for each of the fluids to be blended.

Jet mixing using liquid (recirculating or direct feed, see Figure 5-36A, B) through single or a multiple jet nozzle arrangement (Figure 5-5W) have been studied for tank blending operations. Reference [29] points out that a mixer can be considerably more efficient than jets producing the same flow of liquids. The jets are reported to be useful for mixing thick slurries, where settling with the agitator, not running, can pose a real problem for start-up [40, 41].

Emulsions

Emulsions require high shear in the mixing operation with high speed and low D/T ratio.

Extraction

The mass transfer in extraction equipment using mixers requires careful study before scale-up.

Gas-Liquid Contacting

This is an important system in chemical processing. The effect of apparent density (liquid plus gas) as the fluid mixture enters the impeller is quite pronounced on the system horsepower. The horsepower falls off with increased gas flow which may lead to the danger of underpowering the unit. The absorption coefficient is a func-

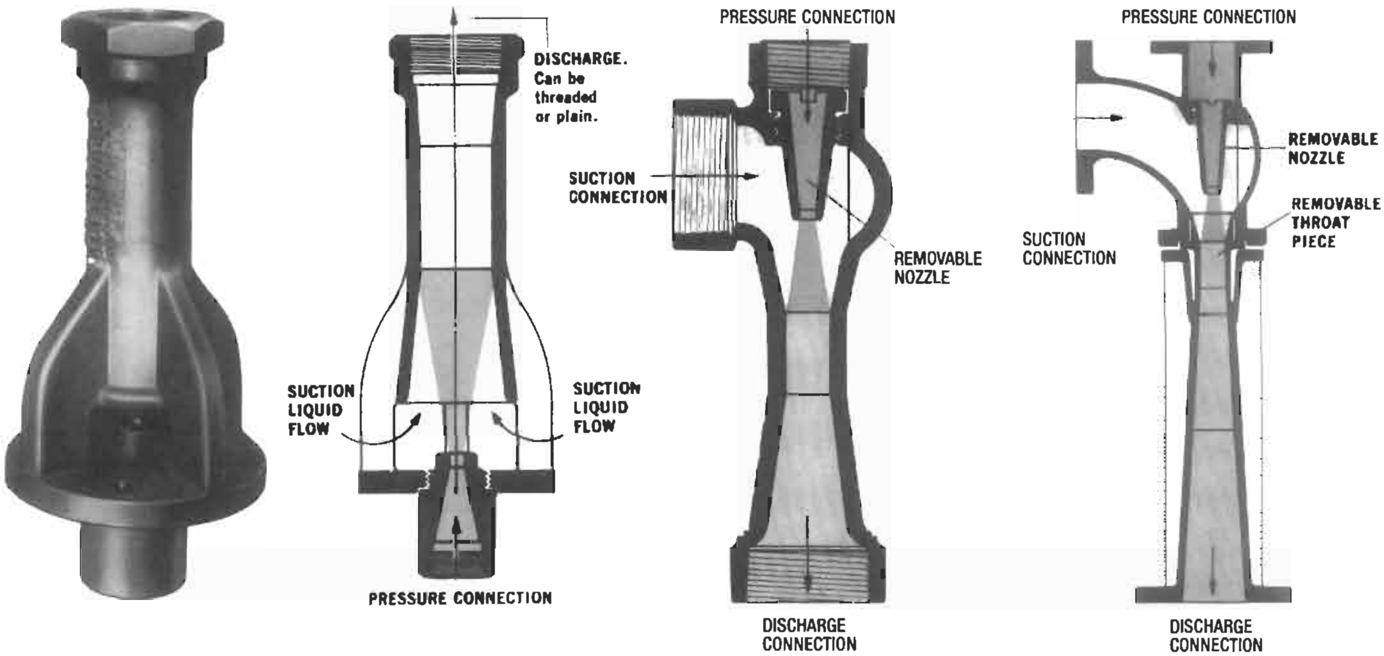


Figure 5-36A. Liquid mixing jets. By permission, Ketema, Schutte and Koerting Div.

tion of the impeller size, its speed and the inlet gas flow rate. For scale-up they should be handled in the form of the Sherwood number k_1 (D/D_V), which can be related to power [21].

Gas-Liquid Mixing or Dispersion

This is another common processing operation, usually for chemical reactions and neutralizations or other mass transfer functions. Pilot plant or research data are needed to accomplish a proper design or scale-up. Therefore, generalizations can only assist in alerting the designer as to what type of mixing system to expect.

This dispersion of the gas passes through several stages depending on the gas feed rate to the underside of the impeller and the horsepower to the impeller, varying from inadequate dispersion at low flow to total gas bubble dispersion throughout the vessel. The open, without disk, radial flow type impeller is the preferred dispersing unit because it requires lower horsepower than the axial flow impeller. The impeller determines the bubble size and interfacial area.

The gas dispersion ring or sparger can be a special design with holes or a single pipe entering the underside of the impeller, and there will be very little differences in mass transfer performance. References [25] and [29] provide valuable detail for considering design for gas dispersion/mass transfer.

Heat Transfer: Coils in Tank, Liquid Agitated

Heat transfer during mixing of fluids in a tank depends to some extent on the degree of mixing, turbulence, etc., affecting the heat transfer coefficient on the process side of the system and flowing against the coils, plates, or other surfaces for transfer. However, sizing an impeller or selecting an impeller to achieve a particular heat transfer coefficient has been proven to be impractical, because the coefficient is relatively independent of impeller speed [29, 35]. The heat transfer in a mixing vessel is by forced connection, and its heat transfer coefficient is usually one of the controlling factors to heat transfer. The other factors are cooling/heating side film coefficient, except when condensing steam, the scaling or fouling factors on the process side, and coolant/heating medium on the opposite side.

Despite the technical study and examination of this subject, it is important to recognize that because of the variety of factors noted earlier, the designer should not expect precise results and should allow considerable flexibility in the physical/mechanical design in order to adjust the system to achieve the required results.

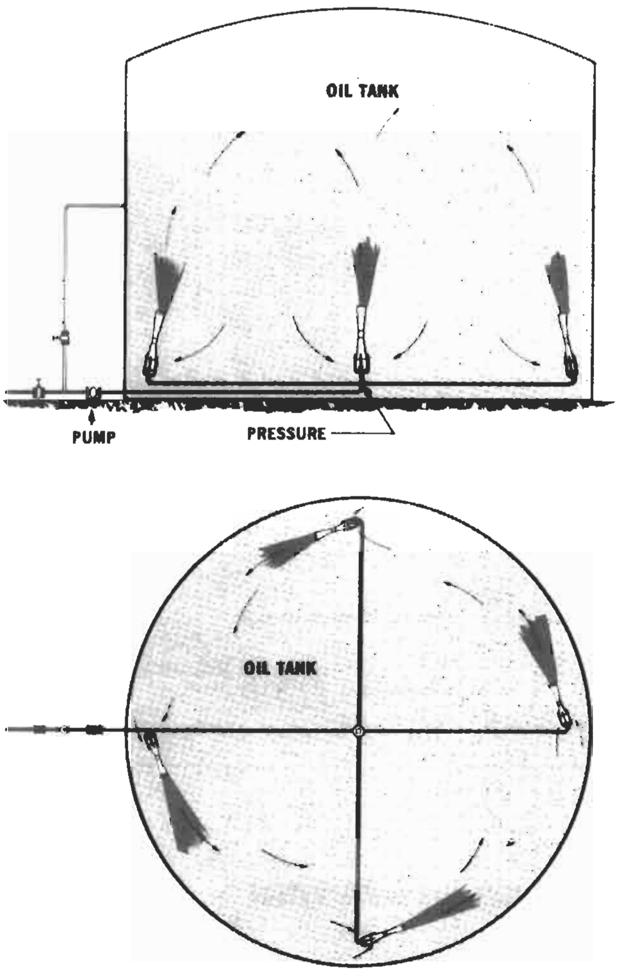
Effects of viscosity on Process Fluid Heat Transfer Film Coefficient

Figure 5-37 presents a typical heating and cooling chart for the changes in process side film coefficients, h_o, as a function of bulk viscosity for organic chemicals.

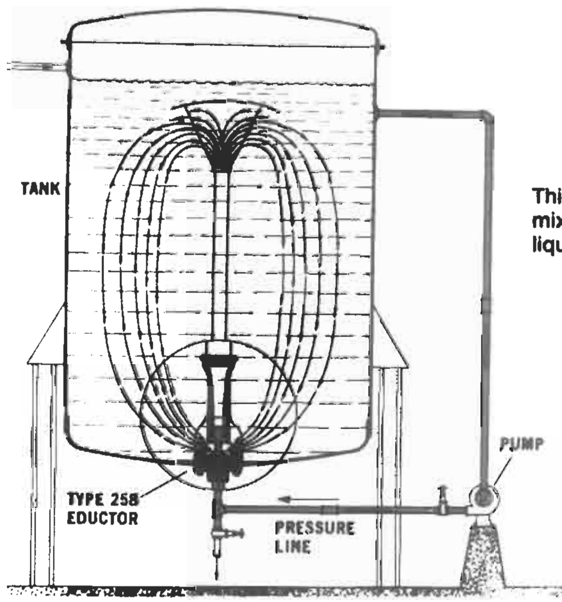
The common arrangements for transferring heat to liquids in a tank are:

1. vertical tube banks
2. helical coils
3. external jackets.

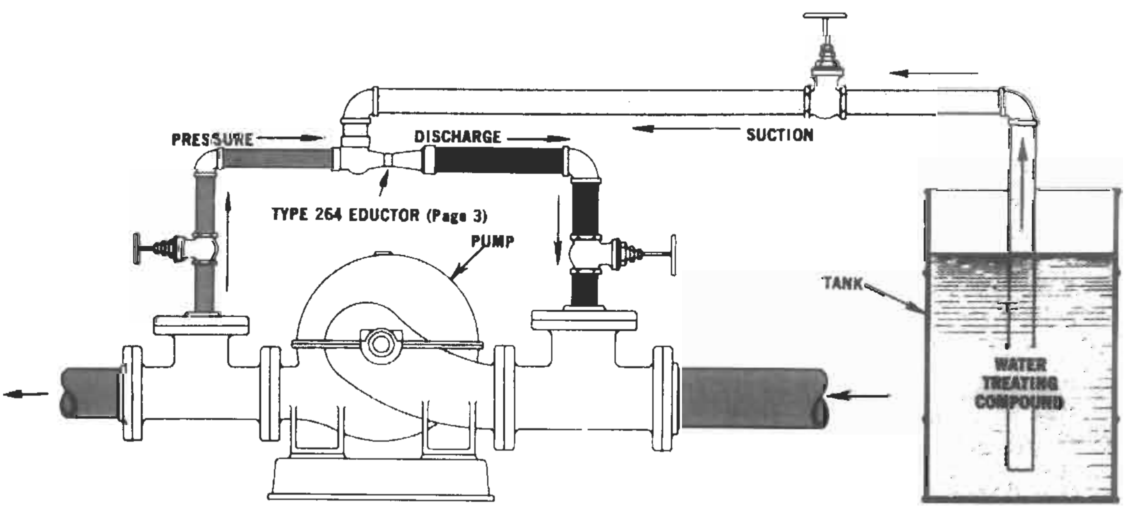
The range of chemical, petrochemical, and refining processes requires some type of heat transfer involving



This arrangement has proved satisfactory for the tank blending of oils.

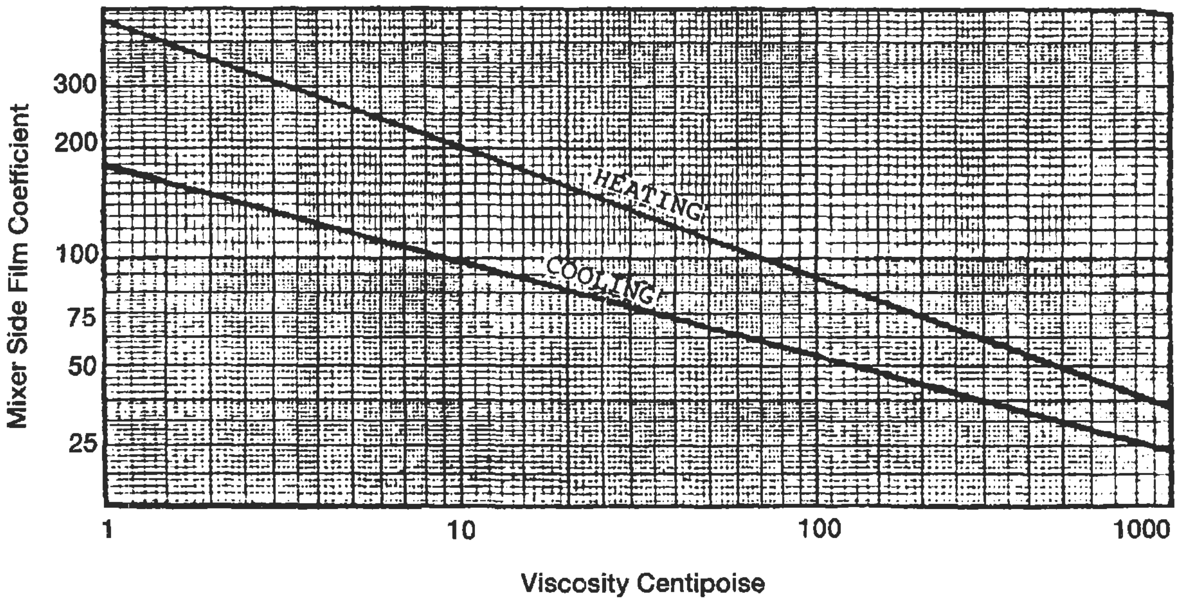


This illustrates the batch mixing of two or more liquids.



An Eductor is used to introduce a water treating compound into boiler feed water.

Figure 5-36B. Illustration of jet mixing for blending of oils by circulation within the tank. Oil from the top is drawn down and entrains the oil in the bottom of the tank through the eductor nozzle (jet). By permission, Ketema, Schutte and Koerting Div.



Thermal Effectiveness of Heat Transfer Surfaces

Tank Jacket	1.0
Vertical Tubes	1.54
Helical Coils (First Bank)	1.54
Helical Coils (Second Bank)	1.31
Helical Coils (Third Bank)	1.08
Plate Coils (or Panel Coils)	1.12

Figure 5-37. Typical mixer-side film coefficients for heating and cooling outside vertical or single helical coils with organic chemicals being mixed and heated/cooled. (Thermal effectiveness = 1.54 compared to tank jacket = 1.0). By permission, *Lightnin Technology,* Lightnin Technology Seminar, 3rd ed., 1982, p. 5, section 2D, Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.

the mixing operation at the same time that some heat transfer is required. Typically, these are:

- reactions in mixing vessels with heat transfer to add or remove heat
- hydrogenation
- fermentation
- polymerizations
- general cooling
- general heating
- wide variety of processes including pharmaceuticals

When heat transfer occurs in the presence of suspended solid particles, the chemical reaction may require mass transfer, etc. However, the mixing part of the operation must be designed to accomplish the process results, which may be those named above (and others) plus the required heat transfer affecting the outside film coefficient, h_o, across/around any coils, flat surfaces, or wall jackets.

Heat transfer in these systems conforms to the general laws of other types of heat transfer, because heat is still being transferred through a barrier wall between two fluids in motion to determine the effective individual film coefficients.

$$q = U_o A \Delta T \tag{5-69}$$

where U_o = overall heat-transfer coefficient referenced to outside heat transfer surface area, Btu/hr/sq ft/°F

h_o = outside, process side in tank, film coefficient, Btu/hr/sq ft/°F

k_w = thermal conductivity of material of heat transfer wall, Btu/hr (sq ft)/(°F)(ft)

A_n = area of outside coil or heat transfer barrier, sq ft/ft

A_i = area of inside of surface for heat transfer, such as coil, flat surface, or other barrier, sq ft/ft

h_i = inside heat transfer fluid side coefficient, in coil, flat plate, or other barrier, Btu/hr/sq ft/°F

r_o = fouling resistance (factor) associated with fluid on outside (tank process side) of heat transfer barrier such as tube or flat plate/wall of vessel, hr (sq ft) (°F) Btu

r_i = fouling resistance (factor) associated with fluid on inside (heat transfer fluid) of heat transfer barrier such as tube or flat plate/wall of vessel, hr (sq ft) (°F) Btu

A_{avg} = average of inside and outside tube surface area, sq ft/ft

$$\frac{1}{U_o} = \frac{1}{h_o} + r_o + \left(\frac{L_w}{k_w}\right)\left(\frac{A_n}{A_{avg}}\right) + r_i + \frac{1}{h_i\left(\frac{A_i}{A_o}\right)} \quad (5\text{-}70)$$

r_w = resistance of tube wall, L_w/k_w; hr (sq ft) (°F)/Btu

L_w = thickness of tube wall, ft

For estimating or even practical purposes, some of the components of the equation can be simplified. The ratio, A_i/A_o, can be used as (D_i/D_o). Note that consistent units must be used for k_w and L_w.

In mixing, the moving liquid in a vessel establishes a heat transfer film coefficient on the surface of the heat transfer barrier such as tube coils, or the internal vessel shell wall with a jacket exterior to this wall for circulating heating and cooling fluid. This film becomes a function of the movement of the fluid against and/or next to the heat transfer barrier surface. Thus, the thinner the film, the better the heat transfer. Therefore, the selection of the type of impeller, its rotational speed, and the fluid properties all influence the actual flow of heat through the film (see Figure 5-38 and 5-39). Oldshue [29] identifies features of the system (vessel, impeller, fluid properties) that influence the resulting heat transfer coefficient. The generalized representation is:

$$N_{NU} = \frac{h_o d}{k} = 0.17(N_{Re})^{0.67}(N_{PR})^{0.37}\left(\frac{D}{T}\right)^{0.1}\left(\frac{d}{T}\right)^{0.5}\left(\frac{\mu}{\mu_w}\right)^m \quad (5\text{-}71)$$

N_{NU} = Nusselt number

N_{Re} = Reynolds number = $\left(\frac{D^2 N\rho}{\mu}\right) \cong \frac{10.754 D_i^2 N_m S_g}{\mu}$

N_{PR} = Prandtl number = $\left(\frac{c_p \mu}{k}\right)$

$$h_o \propto \mu^{-3} D^{1.44} N^{0.67} T^{-0.6} k_w^{0.63} d^{-0.6} \rho^{0.67} cp^{0.37} \quad (5\text{-}72)$$

where D = impeller diameter, ft

N = impeller speed, rpm

ρ = density, lb/cu ft

μ = bulk viscosity of fluid

μ_w = wall viscosity at film process fluid temperature at heat transfer surface

m = experimentally determined exponent, depending on bulk viscosity

n_o = mean mixer side film coefficient of tank temperature

k = thermal conductivity of fluid, Btu/hr/sq ft/°F/ft

d = d_t = tube OD, ft

T = tank diameter, ft

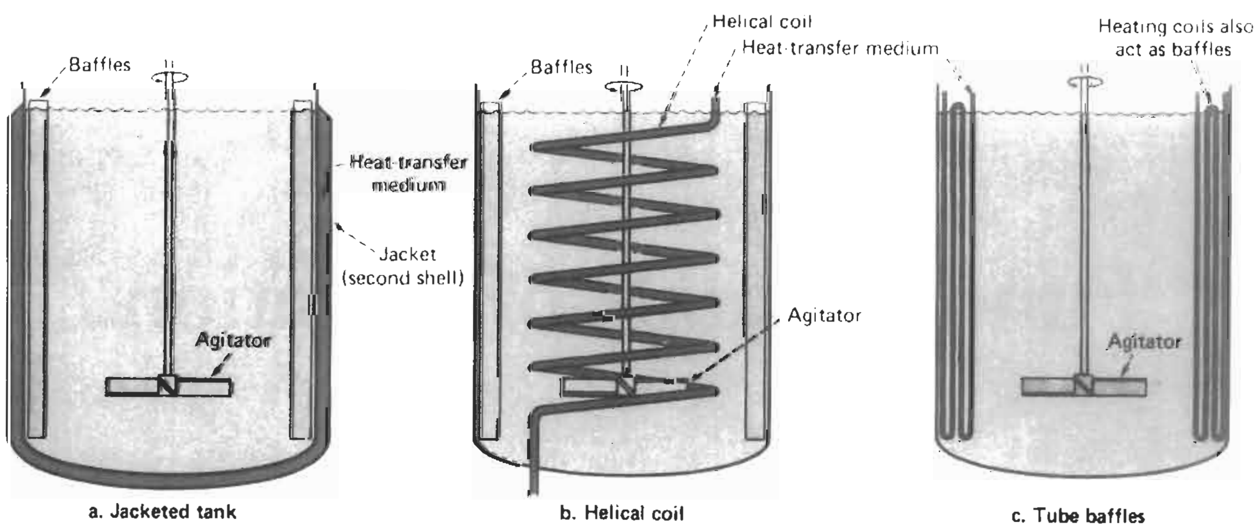


Figure 5-38. Heat transfer surfaces in agitated tanks may be the actual wall of the vessel or immersed tubes. By permission, Dickey, D. S. and Hicks, R. W., *Chem. Engr.*, Feb. 2, 1976, p. 93 [35].

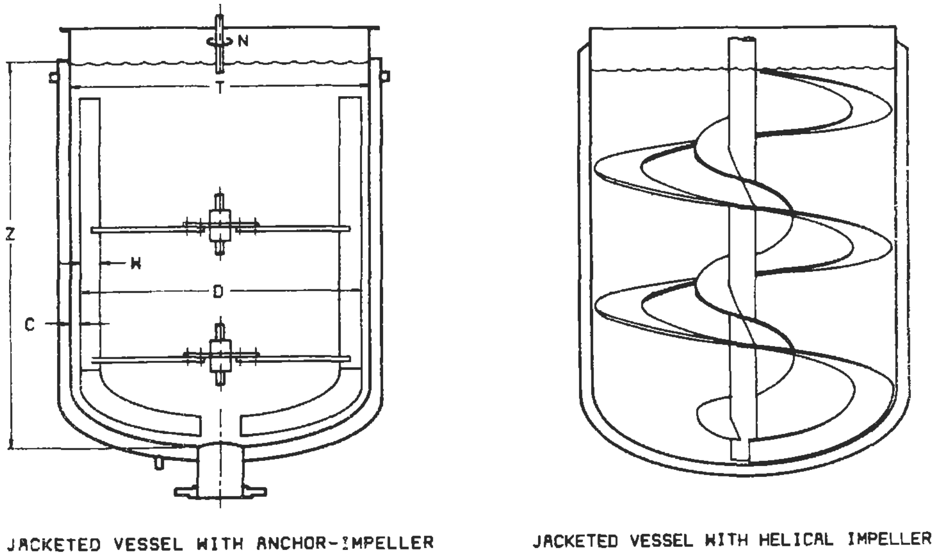


Figure 5-39. Close-clearance anchor and helical impellers. By permission, Oldshue, J. Y., *Fluid Mixing Technology,* 1983, Chemical Engineering McGraw-Hill Publications Co., Inc. [29].

The vertical tubes serve as baffles to a certain extent, but not enough to prevent some vortex formation. The helical coil installations may have sidewall baffles (usually four 1/10 or 1/12 dia.), or baffles assembled with the coil itself. (See Figures 5-23H, 5-23I, 5-38 and 5-39.)

Figure 5-40 gives the heat transfer relations for a flat paddle turbine and for anchors in a jacketed vessel and also in coil-tank arrangements. The data of Cummings and West [5] are based on large equipment and give results 16% higher for the coil and 11% higher for the jacket than the results of Chilton, *et al.* [3].

Heat transfer data appear to be no better than ±20% when trying to compare several investigators and the basic fundamentals of their systems.

The exponents applying to each system are given on the figure.

The work of Uhl [22] gave particular emphasis to viscous materials in jacketed vessels, and the correlating equations are:

1. Paddles, with or without baffles; $R_e = 20$ to 4,000

$$\frac{hT}{k} = 0.415\left(\frac{D^2 N\rho}{\mu}\right)^{0.67}\left(\frac{c\mu}{k}\right)^{0.33}\left(\frac{\mu_w}{\mu}\right)^{-0.24} \qquad (5\text{-}73)$$

2. Turbine, no baffles; $R_e = 20$ to 200

Relation same as for paddle, coefficient changes from 0.415 to 0.535 when the turbine is about two-thirds of the distance down from top to bottom of the vessel. When the turbine is very near the bottom the coefficient is 0.44.

3. Anchor, no baffles
   (a) $R_e = 300$ to 4,000, coefficient is 0.38 and μ_w/μ is raised to -0.18 power.
   (b) $R_e = 30$ to 300, coefficient is 1.00 and $(\mu_w/\mu)^{-0.18}$ and Reynolds number term is raised to 0.50 power.

The correlations of Cummings and West [5, 29] for turbine mixers in vessels with jackets and coils are:

1. Vertical Helical Coils, Multiple Coils [29]

$$h_o\,(\text{coil})\frac{d_t}{k} = 0.17\left(\frac{D^2 N\rho}{\mu}\right)^{0.67}\left(\frac{c_p\mu}{k}\right)^{0.37}\left(\frac{D}{T}\right)^{0.1}\left(\frac{d_t}{T}\right)^{0.5}\left(\frac{\mu}{\mu_w}\right)^{m} \qquad (5\text{-}74)$$

m = can vary from 0.1 to 1.0
$m = 0.1\,(\mu\,8.621 \times 10^{-5})^{-0.21}$

The tube diameter, d_t, and tube spacing influence the film coefficient, h_o. The work of Oldshue [29] covers a practical range about as large and as small as is industrially used. The tube spacing of 2 to 4 tube diameters produced consistent results.

where D = impeller diameter, ft
T = tank diameter, ft
μ = bulk viscosity
μ_w = wall viscosity at bulk process fluid temperature, d_t
d_t = tube diameter, ft, O.D.

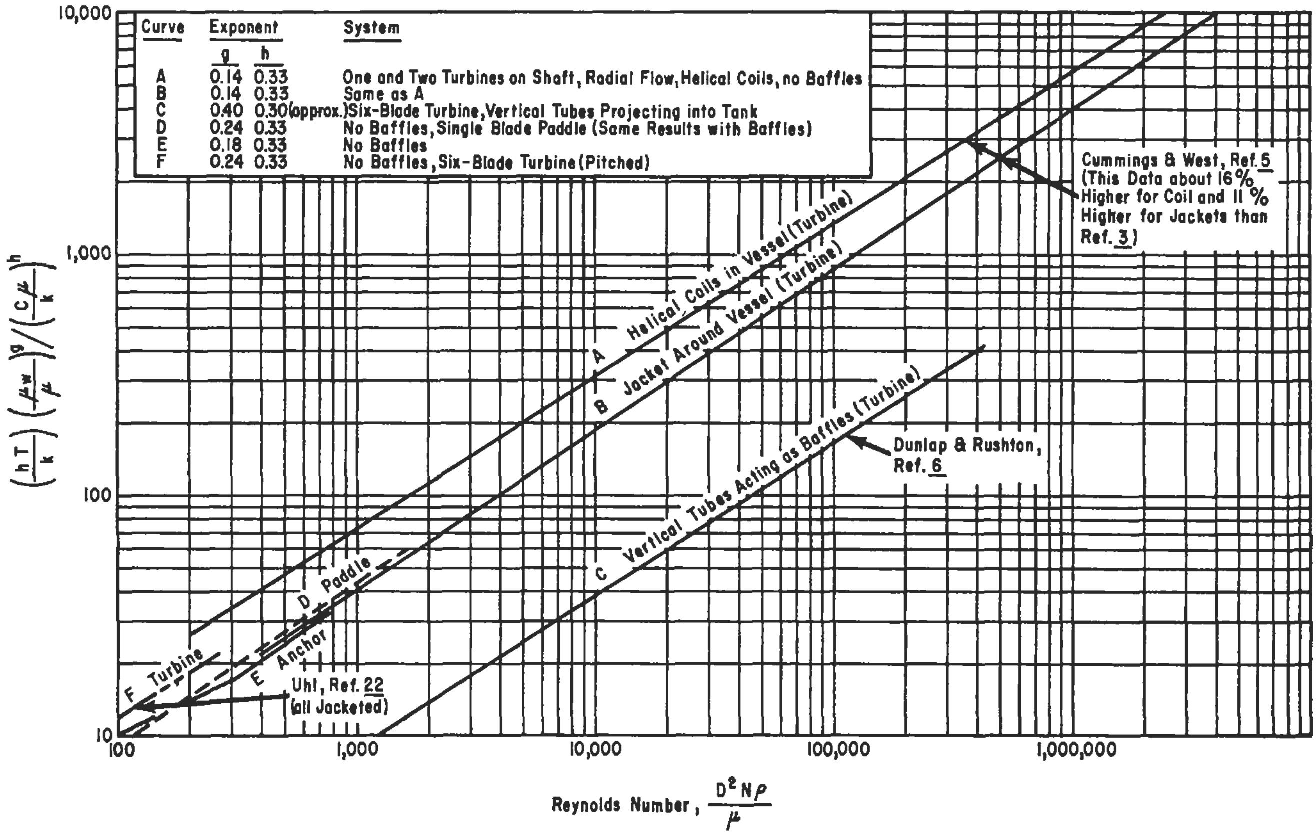


Figure 5-40. Liquid mixing heat transfer. Compiled by permission from References 3, 5, 6, and 22.

The helical coil relationship was developed using four vertical baffles of 1/12 T, located either outside the coil or inside the coil [29].

The addition of a second bank of similar coils for additional heat transfer area, either outside or inside the first bank of coils will not provide twice the transfer of heat; but the effectiveness is estimated to be 70%–90% of the first bank of duplicate coils (assuming the same heat transfer area per bank) [29]. The additional coils will provide additional baffling; therefore, the need for the dimension of the vertical baffles is reduced, or they might be removed entirely.

The value of, m, (an experimental exponent) [29] will run between 0.1 and 1.0 for μ ranging from 600 to 0.3 cp.

The particular work of Oldshue and Gretton [13] on heat transfer outside helical coils is of considerable general application. Refer to Equation 5-74.

This is considered applicable to all sizes of tanks using 6-blade flat blade turbines, even with baffles on wall or inside single helical coil, providing that tube diameters are of $0.018 \leqq d/T \leqq 0.036$, with viscosity range probably to 10,000 centipoise. Tube spacing of coil wraps of 2 to 4 tube diameters indicated no appreciable effect on the coefficient. The wider spacing gives a lower coefficient for materials above 50 cp. For example, the ratio of coefficients at 4d and 2d for water of 0.4 cp and oil of 50 cp were 0.96 and 0.88 respectively. The placement of baffles directly adjacent to the wall gives only a 5 percent better coefficient than if placed 1 inch off the wall or inside the helical coil. However, the power is about 10 percent lower when the baffles are off the wall.

2. Vertical Tubes [29].

Data of Dunlap and Rushton is given in [6]. This tube design can prevent the need for vertical baffles in a tank, and the heat transfer is good.

$$h_o \text{ (tubes) } d_t/k = 0.09\left(\frac{D^2 N\rho}{\mu}\right)^{0.65}\left(\frac{c_p \mu}{k_w}\right)^{0.3}\left(\frac{D}{T}\right)^{0.33}\left(\frac{2}{n_b}\right)^{0.2}\left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (5\text{-}75)$$

Table 5-6
Mixing Correlation Exponents For Various Systems

Tank Configuration	Slope x of Correlation Line	Reference
Jacketed cast iron hemispherical bottom vessel Propeller—no baffles T/D = 2.5 U-Type impeller—no baffles. T/D = 1.05	0.67	1
Helical coil, 9.6 in. diam. ½ in. tubing in 1 ft. diam. tank. Flat paddle, T/D = 1.66 close to bottom	0.62	3
Liquid depth equal to tank diam. No baffles T/d = 24	0.67	3
Helical coil, 18 in. diam. 1 in. tubing in a 24 in. diam. tank.	0.62	5
2 curved blade turbines. No baffles. T/D = 2.5; T/d = 30	0.67	5
4 vertical tube baffles, 1½ in. tubes. One flat blade turbine. T/D = 3 Turbine position one-half liquid depth. T/d = 25.3	0.65	6
4 vertical tube baffles, 1 in. tubes. One flat blade turbine. T/D = 3; T/d = 37.0	0.90	19
Helical coil 34 in. diam. 1¾ in. tube in 4 ft. diam. tank. One flat blade turbine, 4 baffles each 1/12 T. T/D = 3 Turbine position 1/3 liquid depth. T/D = 27.5.	0.67	13
Paddle 14⅛ in. x 2⅜ in. close to bottom. T = 14⅛ in. No baffles and 4 at ⅛ T. Oils from 100 to 46,000 centistokes	0.67	22
Fan turbine 6 blades 12 in., 45° Pitch Tank and liquids like above No baffles	0.67	22
Same as above	0.67	22
Anchor impeller 22½ in. Diam. Tank and liquids like above	0.5 up to N_{Re} = 300 0.67 above N_{Re} = 300	22

Extracted in part from J. H. Rushton and J. Y. Oldshue, Chem. Eng. Prog. 49, 273 (1953) and ibid., presented at Philadelphia meeting A.I.Ch.E. (1958), Ref. 20 and 21 resp., by permission.
Note: Reference numbers refer to published article cited.

where n_b = number of tube baffles (vertical), i.e., four or six banks of vertical tubes with three tubes per bank
k = liquid thermal conductivity

Although the outside coefficient of a vertical coil is some 13% higher than for a helical coil, the inside coefficient is quite often lower due to the physical arrangement and the lower coefficient if gases are evolved and venting is required. The over-all coefficient may end up about the same as the helical coil. The outside film coefficient for a system varies with $(HP)^{0.22}$ in the turbulent region. Thus

$$\frac{h_2}{h_1} = \left[\frac{(HP)_2}{(HP)_1}\right]^{0.22}$$

The power required for vertical tubes in a vessel is 75 percent of that for standard wall baffles [13]. It is sometimes difficult to physically place as much vertical coil surface in a tank as helical coil surface. Dunlap studied vertical coils and the results are correlated for dimensionally similar systems by [6] [29]

This is shown in Figure 5-40 with certain simplifications to facilitate plotting with the other data. The 4-blade turbine mixer was centered in the tank about ⅓ of the fluid depth from the flat bottom. The vertical coils extended out into the tank in groups of three. The liquid depth was equal to the tank diameter.

Table 5-8 gives the order of magnitude for coil-in-tank heat transfer.

3. Vertical Plate Coils

The results of Petree and Small are summarized in [29]. These coils present a solid vertical face, with the "coils" vertical but impressed in the plates for flow of the heating or cooling medium. They take the place of vertical baffles, and are more solid obstructions to "through flow" in the vessel than individual vertical coils. Usually four or six banks are used.

For $N_{Re} < 1.4 \times 10^3$ in fluid bulk in tank:

$$h_o \text{ (plate coil) } (P_{pcw}/k) = 0.1788\left(\frac{ND^2\rho}{\mu}\right)^{0.448}\left(\frac{c_p\mu}{k_w}\right)^{0.33}\left(\frac{\mu}{\mu_f}\right)^{0.50} \quad (5\text{-}76)$$

μ_f = viscosity of fluid film at mean film temperature
μ = viscosity of fluid bulk at bulk temperature
P_{pcw} = plate coil width, one plate, ft

Table 5-7
Example Scale-Up of Two, D_2/D_1, For Varying Mixing Slopes[16]

Mixing Slope, x	N_2/N_1	P_2/P_1	V_2/V_1	$(P/V)_2/(P/V)_1$
0.5	1.0	32.0	8	4.0
0.6	0.8	16.0	8	2.0
0.75	0.63	8.0	8	1.0
0.9	0.54	5.0	8	0.62
1.0	0.50	4.0	8	0.50

By permission; Rushton, J. H., Chem. Eng. Prog. V 47, (1951), pg. 485 [16].

For $N_{Re} > 4 \times 10^3$ in fluid bulk in tank

$$h_o \text{ (plate coil) } (P_{pcw}/k) = 0.0317\left(\frac{ND^2\rho}{\mu}\right)^{0.658}\left(\frac{c_p\,\mu}{k_w}\right)^{0.33}\left(\frac{\mu}{\mu_f}\right)^{0.50} \quad (5\text{-}77)$$

Some scale-up heat transfer relationships [29]:

a) Laminar flow, $R_e < 100$
   $h_o \propto$ HP (horsepower at the impeller shaft)
b) For a constant impeller diameter, turbulent flow:
   $h_o \propto (HP)^{0.22}$
c) For constant speed, same impeller type (family)
   $h_o \propto (HP)^{0.29}$

Table 5-8
Order of Magnitude of Outside Film Coefficients, h Turbulent Range

Fluid	Helical Coils BTU/(Hr.) (sq. ft.) (°F. between tank fluid and coil fluid)	Vertical Coils
Water, 0.38 cp	500–1100	500–1100
Oil, 10 cp	....	20–70
Oil, 52 cp	40–70	10–30

4. Jacket [35] [29]

$$\frac{h_o T}{k} = 0.85\left(\frac{D^2 N\rho}{\mu}\right)^{0.66}\left(\frac{c_p\mu}{k}\right)^{0.33}\left(\frac{\mu}{\mu_w}\right)^{0.14}\left(\frac{Z}{T}\right)^{-0.56}\left(\frac{D}{T}\right)^{0.13} \quad (5\text{-}78)$$

Design Application

A few typical overall heat transfer coefficients, U_o, are presented in Table 5-9 [27] [29].

Figure 5-41 indicates the mixing correlation exponent, x, as related to power per unit volume ratio for heat transfer scale-up. The exponent x is given in Table 5-6 for the systems shown, and is the exponent of the Reynolds number term, or the slope of the

Table 5-9
Approximate Overall Heat Transfer Coefficients for Jacketed Mixing Vessels

U Expressed in BTU/hr/sq ft/°F

FLUID INSIDE JACKET	FLUID IN VESSEL	WALL MATERIAL	AGITATION	U
Steam	Water	Enameled C.I.*	0–400 R.P.M.	96–120 [25]
Steam	Milk	Enameled C.I.	None	200
Steam	Milk	Enameled C.I.	Stirring	300
Steam	Milk boiling	Enameled C.I.	None	500
Steam	Boiling water	Steel	None	187
Hot water	Warm water	Enameled C.I.	None	70
Cold water	Cold water	Enameled C.I.	None	43
Steam	Water	Copper	None	148
Steam	Water	Copper	Simple stirring	244
Steam	Boiling water	Copper	None	250
Steam	Paraffin wax	Copper	None	27.4
Steam	Paraffin wax	Cast iron	Scraper	107
Water	Paraffin wax	Copper	None	24.4
Water	Paraffin wax	Cast iron	Scraper	72.3

*C.I. = Cast Iron
**= or Glass-lined on steel
Adapted by permission: Lightnin/Unit of General Signal [27].

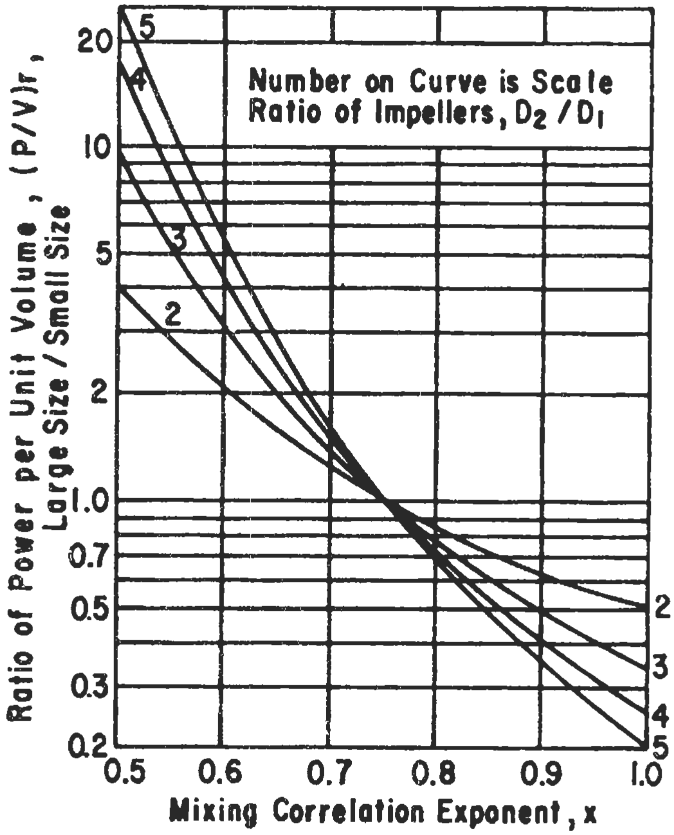


Figure 5-41. Scale-up relation for power, volume, and mixing slope. By permission, Rushton, J. H. and Oldshue, J. Y., *Chem. Engr. Prog.*, V. 49, No. 4, 1953, p. 161 [20].

lines as given in Figure 5-40. It should be determined on any system not considered dimensionally similar or otherwise applicable to cases shown in Table 5-6 or Table 5-7.

Maintaining $(h_o)_1 = (h_o)_2$ on scale-up:
For dimensionally similar systems [16]:

1. Determine from experimental data the Reynolds number exponent for the system, or use information in Figure 5-40 if the systems of Table 5-8 can be considered similar, use proper coefficients and solve for outside film coefficient, h_o.

2. Referring to Figure 5-41, read the new approximate horsepower per unit volume ratio for the exponent x and the impeller ratio, D_2/D_1, where D_2 is the larger impeller diameter. Calculate:

$$\left(\frac{P}{V}\right)_2 \Big/ \left(\frac{P}{V}\right)_1 \quad \text{Ratio from curve, Figure 5-41} \qquad (5\text{-}79)$$

3. Determine impeller speed:

$$N_{2s} = N_{1s}\left(\frac{D_1}{D_2}\right)^{(2x-1)/x} \qquad (5\text{-}80)$$

4. Determine power using x and compare with step 5 below:

$$P_2 = P_1\left(\frac{D_2}{D_1}\right)^{(3-x)/x} \qquad (5\text{-}81)$$

5. Determine tank volume:

$$V_2 = V_1\left(\frac{D_2}{D_1}\right)^3 \qquad (5\text{-}81A)$$

and substitute in step 2, solving for P_2.

For a scale up of impeller diameters of 2, $D_2/D_1 = 2$, Table 5-7 illustrates the changes to be expected. Such scale up has been found to be quite reliable.

In-line, Static or Motionless Mixing

This mixing device contains no moving parts, is relatively simple, and its cost can be quite reasonable when compared to mechanical driven mixers.

Static or motionless mixers are a relatively new development and have proven to be effective for many specific and valuable process applications. Although useful for a wide range of viscosity fluids, some of the units have performed exceptionally well in the mixing of molten polymers. This type of unit is particularly useful in liquid-liquid mixing, although some units can be designed for solid-liquid dispersion and for gas-gas and gas-liquid mixing/dispersion. Some units are also reported suitable for solid-solid blending. The three commonly applied units are from Chemineer (Kenics U.S. Patent 3,286,992), Charles Ross and Son (Interfacial Surface Generator, [Dow Chemical, U.S. Patent 3,168,390]) and Koch Engineering Co. (Sulzer Bros. of Winterthur, Switzerland, U.S.

Patent pending). Some designs work well on mixing powders. There are over 30 different models of static mixers worldwide [38].

The concept of the motionless mixer is to achieve a uniform composition and temperature distribution in fluids flowing through the device. Originally, this objective dealt with molten polymer mixing, as an alternative to the dynamic mixing of an extruder screw [38]. The units are now used widely for just about all fluid mixing, blending, dispersion, etc., by multiple splitting of the flowing streams. The concept is built around a stationary, rigid element placed in a pipe or cylinder that uses the energy of the fluid flowing past to produce the mixing. The unit becomes more efficient by adding additional static elements (see Figures 5-42, 5-43, 5-44A & B, 5-45, 5-46 and 5-47) causing the flowing fluid elements to split, rearrange, recombine and split again and repeat the process "x" times until a uniform or homogeneous flowing stream is produced at the discharge. The pressure drop varies with the mixer elements design, number per unit length, fluid flowing (density and viscosity). The shear on the fluid carries forward the mixing process.

The better mixing occurs under turbulent flow conditions.

Residence time in the units can be varied or adjusted, which makes them suitable to serve in certain types of reactions.

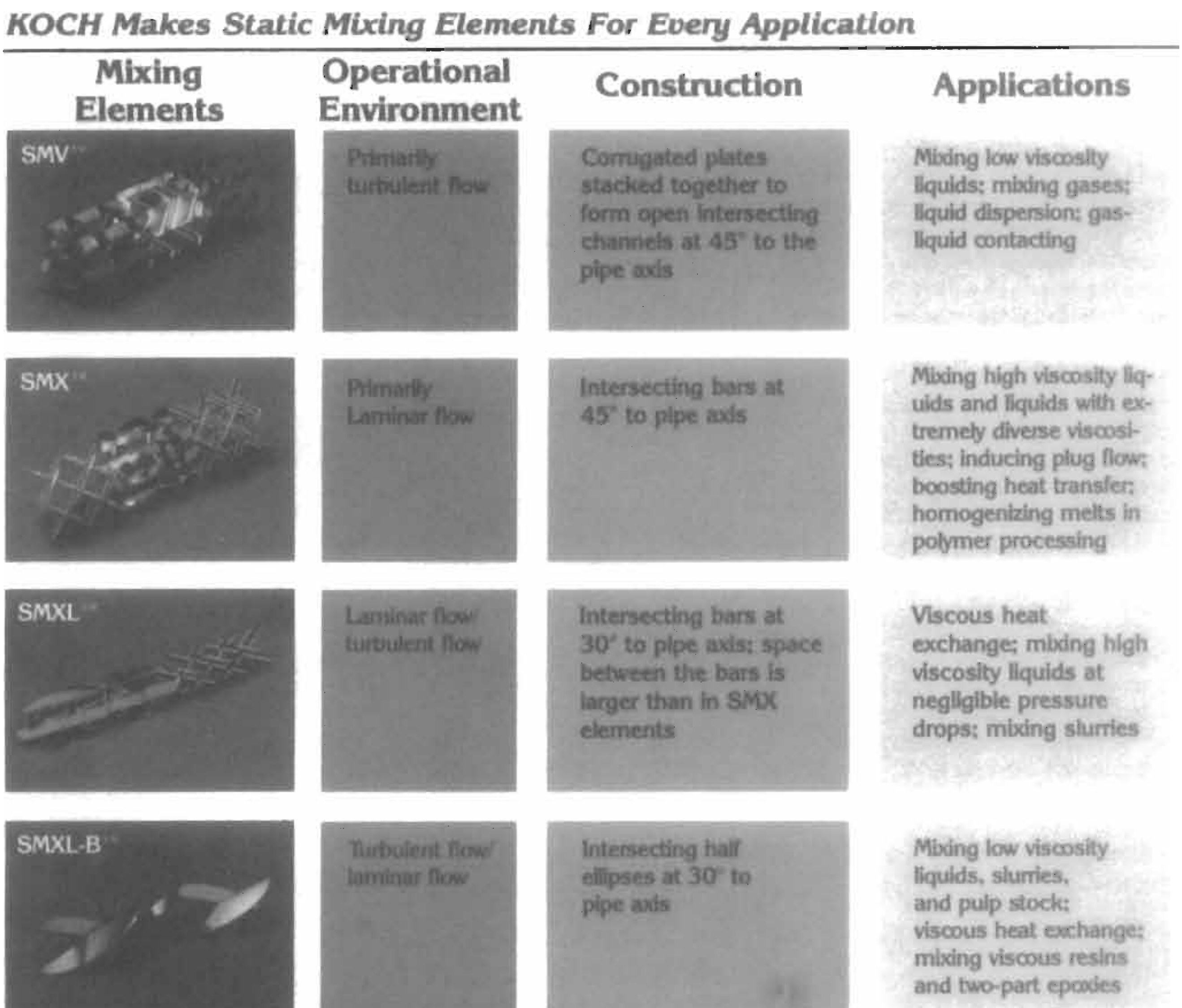


Figure 5-42. Several types of static mixing elements usually adapted into pipe or fabricated casings. By permission, Koch Engineering Co.

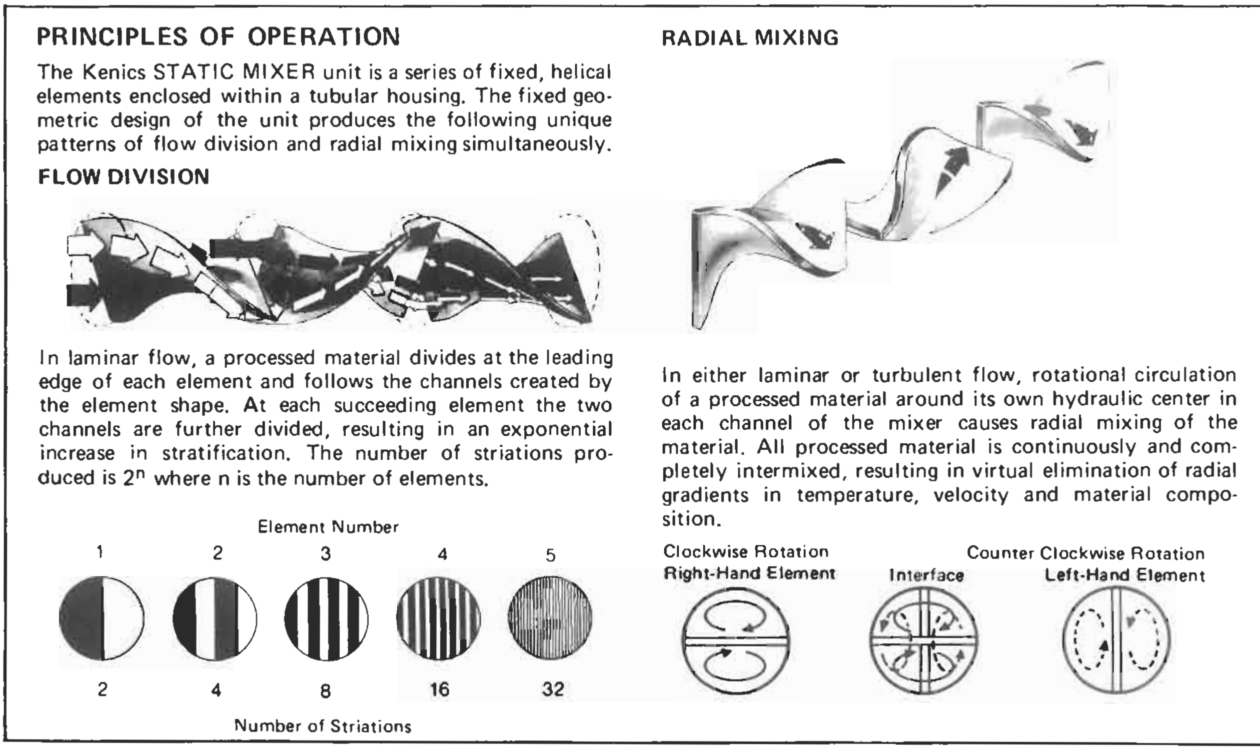


Figure 5-43. Principles of operation of static mixer modules. By permission, Kenics Corp., Div. Chemineer, Inc.

Figure 5-44A. Two details of the Dow motionless mixer. By permission, Charles Ross and Son Co.

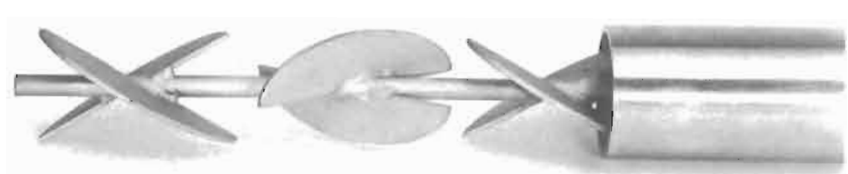


Figure 5-44B. The Dow interfacial surface generator is mathematically predictable. By permission, Charles Ross and Son Co.

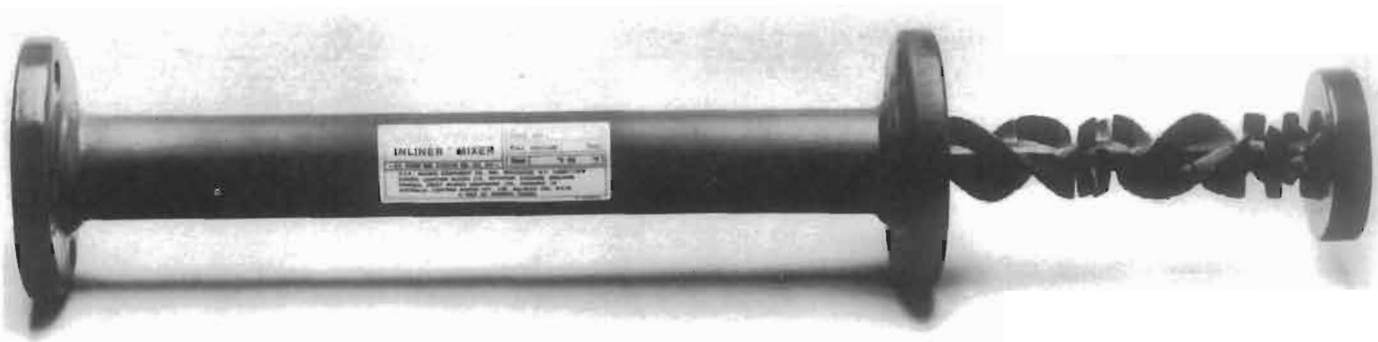
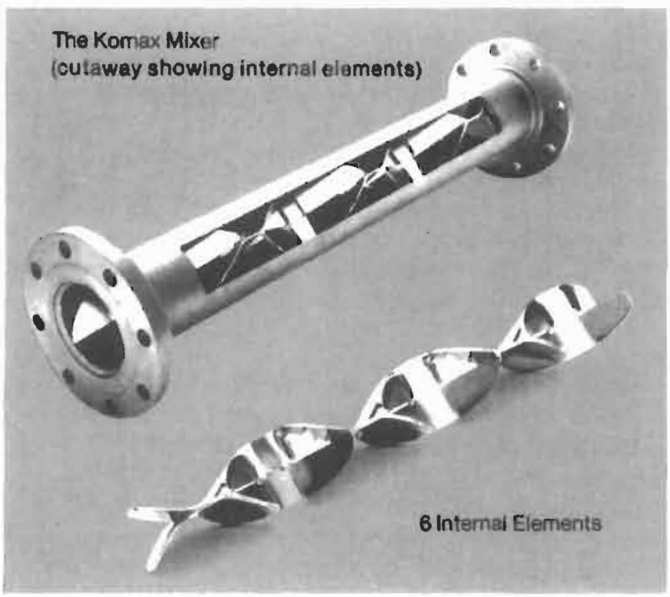


Figure 5-45. Lightnin® Inliner(TM) motionless mixer. By permission, Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.



Quite often the mixing units or elements are installed in a circular pipe; however, they can be adapted to rectangular or other arrangements. Pressure drop through the units varies depending on design and whether flow is laminar or turbulent. Because of the special data required, pressure drops should be determined with the assistance of the manufacturer. Of course, pressure drops must be expected to be several multiples of conventional pipe pressure drop.

Applications

In addition to the general classification of applications previously mentioned, Tables 5-10A and 5-10B give typical applications. Although the number of modules or elements referred to is somewhat specific to the manufacturer, the tables give a general description of similar systems from other manufacturers.

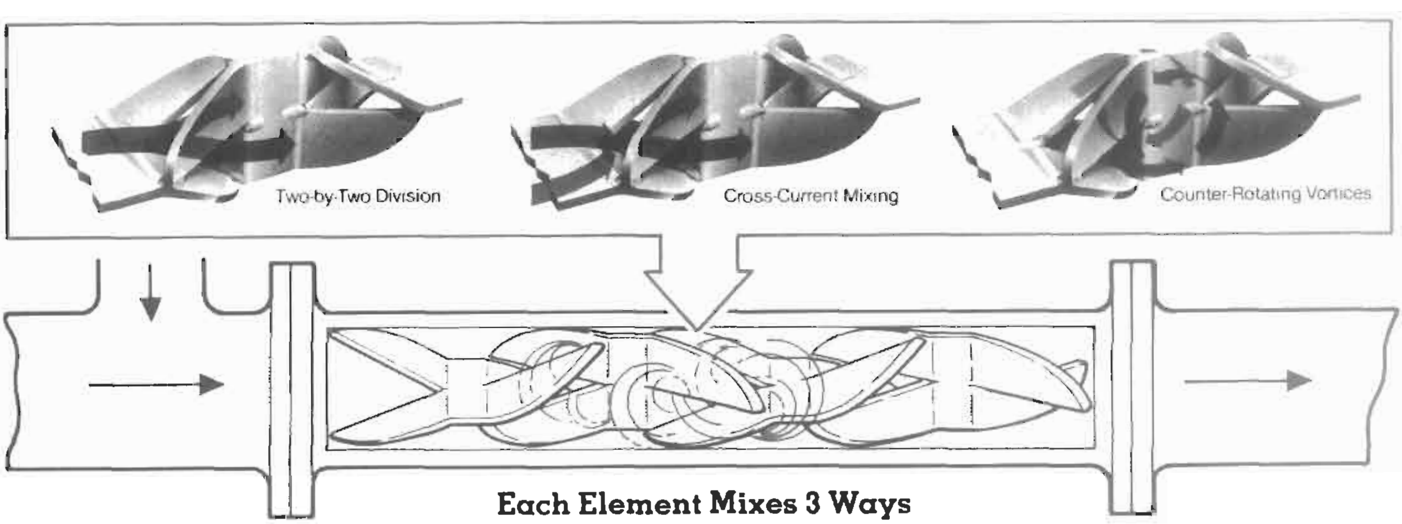


Figure 5-46. Komax(TM) motionless mixer. By permission, Komax Systems, Inc.

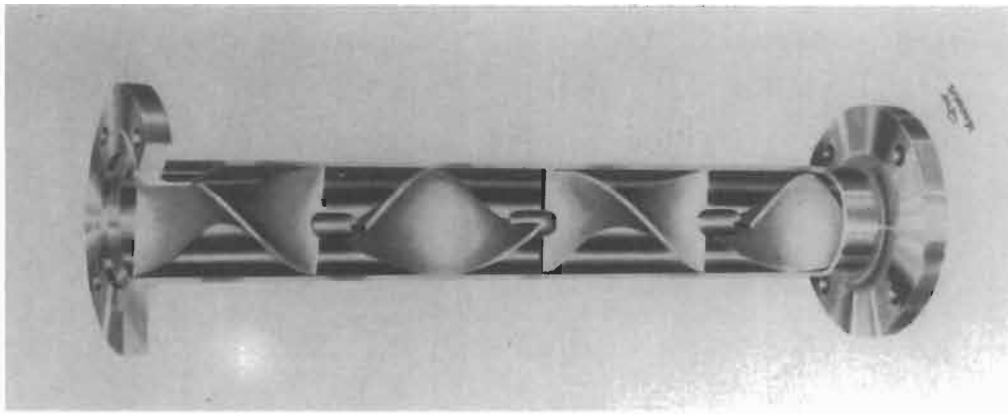


Figure 5-47. Luwa Blendrex(TM) motionless mixer. By permission, Luwa Corporation.

Materials of Construction

These elements that insert in a pipe or specially fabricated cylindrical holder can usually be fabricated from any workable and weldable metal or alloy. In addition, most plastic that can be fabricated by molding, cutting, heat welding, or even bolting can be used. This wide array of fabrication materials allows the units to fit an extremely wide range of corrosive applications.

Mixer Design and Solution

For some fluid systems, the motionless mixer may not practically achieve total homogeneity. In some situations of widely diverse fluid densities, the centrifugal motion created may "throw" some of the fluid to the outside of the flow path when it emerges from the unit. These are concepts to examine with the manufacturer, as only the manufacturer's data can properly predict performance, and the design engineer should not attempt to actually physically design a unit.

Mixing fluids with viscosity ratios of over 1,000:1 is one of the most difficult applications [39]. Mixing highly viscous fluids is also quite difficult.

The statistical measure of homogeneity is expressed as a function of the element geometry and the length of the unit (i.e., the number of elements in the mixer assembly).

$$\sigma = \sqrt{\frac{\sum_{i=1}^{n}\left(x_i - \bar{x}\right)^2}{n-1}} \text{ and, } \bar{x} = \frac{\sum_{i=1}^{n} x_i}{n} \qquad (5\text{-}82)$$

x_i = temperature, concentration or some other measurable variable

$\frac{\sigma}{\bar{x}}$ = the lower this value, the more homogeneous the mixture. Generally does not need to be lower than 0.05, i.e., 5% standard deviation from the arithmetic mean.

σ = standard deviation
$\bar{x}$ = arithmetic mean
n = number of samples

The results of these calculations depends entirely on the manufacturer's design.

Pressure Drop

The energy to drive the fluid through a static mixer comes from the fluid pressure itself, creating a loss in pressure (usually small) as the fluid flows through the unit.

For laminar flow

$$\Delta p_1 = 8.9 \times 10^{-8} \left(N_e \, Re_D\right) \frac{\mu M * (L/D')}{\rho (D')^3} \qquad (5\text{-}83)$$

N_e = Newton number, depends on unit design. Can range from 0.8 to 1.9, for example.
M = mass flow rate, lb/hr
μ = absolute viscosity, Cp
ρ = density, lb/cu ft
L = mixer length, in.
D′ = inside pipe diameter, in.
Re_D = Reynolds number related to inside diameter of pipe
Q_g = flow rate, gal/min

For Turbulent Flow

$$\Delta p_i = (3.6 \times 10^{-5})(N_e)(\rho Q_g^2/(D')^4(L/D') \qquad (5\text{-}84)$$

Table 10A
Typical Applications for the Motionless Mixers

Application	Type of Flow	Number of Modules
1. Blend one grade of oil (or gasoline) into another oil.	Turbulent	1 six-element module
2. Generate liquid-liquid dispersions (droplets)	Highly Turbulent $N_{Re} > 100{,}000$ Low Turbulence $N_{Re} < 100{,}00$	2 six-element modules 3 six-element modules
3. Blend out thermal gradient in a viscous stream	Laminar	1 six-element module
4. Blend two resins to form a homogeneous mixture	Laminar	4 six-element modules
5. Dilution of molasses stream with water	Low Turbulence	1 or 2 six-element modules Number of modules depends on the flow rate and viscosity ratio.
6. pH control. Neutralization and treatment of waste water streams	Turbulent	1 four- or six-element module depending on reaction conditions.
7. BOD treatment of water	Turbulent	1 six-element module
8. Gas-liquid dispersions	Turbulent	1 or 2 six-element modules
9. Solvent dilution	Turbulent	1 six-element module
10. Pipeline reactor for gaseous or liquid phase reactions	Turbulent	1 or 2 six-element modules
11. Gas dispersion in a liquid	Turbulent	Varies
12. Gas-gas mixing and/or dispersion	Turbulent	Varies

Table 10B
Typical Applications for the Motionless Mixers

Application	Type of Flow	Number of Elements
1. Blending catalyst, dye or additive into a viscous fluid	Laminar	Usually 10-14. The exact number of elements will depend on the viscosity ratio.
2. Delustering of polymer dope	Laminar	10
3. Blend out thermal gradients of polymer melt stream from extruder or heat exchanger	Laminar	4 - 6
4. Disperse solid particles in a viscous fluid	Laminar	10
5. Liquid-Liquid blending to a homogenous product	Turbulent Flow	4 - 6
6. Solid-Solid Blending of food products, explosives	up to 2500 lbs/hr	6 elements placed in a vertical position.
7. Waste water neutralization	Turbulent	4 - 6
8. Pipeline reactor to provide selectivity of product	Laminar	10
9. Multi-component epoxy dispensing systems	Laminar	10
10. Concrete or clay mixing	Laminar	10
11. Manufacture of powder coating	Laminar	4 - 6. Mixer located downstream of extruder
12. Thermal and color dispersion in blow molding machinery	Laminar	4 - 6. Mixer located downstream of extruder
13. Manufacture of liquid or solid foods requiring sanitary construction	Laminar or Turbulent	4 - 10

Horsepower Requirement

The power requirements for the mixer element or mixing unit is expressed:

$$\text{Theoretical HP} = (5.83 \times 10^{-4})(Q_g)(\Delta P) \quad (5\text{-}85)$$

ΔP = pressure drop, psi
Q_g = flow rate, gpm

Reynolds number [25]

Compute the open pipe N_{Re}:

$$N_{Re} = 3157\, Q_g\, Sg/\mu' D' \quad (5\text{-}86)$$

where Sg = SpGr, dimensionless
μ' = viscosity, cp
D' = inside pipe diameter, in.
ΔP = pressure drop, psi
k = viscosity correction factor for turbulent flow

For laminar flow: $N_{Re} < 2000$
ΔP_o for equivalent length of open pipe

$$\Delta P_o = 2.73 \times 10^{-4}\, \mu' L\, Q_g/(D')^4, \text{ for one module} \quad (5\text{-}87)$$

L = length, ft
Sg = SpGr

Q_g = gpm
μ' = viscosity, cp

For a laminar static mixer:

$$\Delta P = (7.4 + 0.07\ N_{Re})\ (\Delta P_o) = \Delta P_L \qquad (5\text{-}88)$$

For ΔP (total), multiply (ΔP_L) (no. modules)

For turbulent flow: $N_{Re} > 2000$

ΔP_o for open pipe, and f = Re vs f chart (Fluid Flow, Chapter 2) for pipe

$$\Delta P_o = 0.0135\ [(f)(L)(Sg)(Q_g)^2/D'^5] \qquad (5\text{-}89)$$

$$\Delta P_t = 66.5\ (N_{Re})^{0.086}\ \mu'^{0.064}\ (\Delta P_o) \qquad (5\text{-}90)$$

ΔP = open pipe pressure drop
ΔP_t = static mixer pressure drop in turbulent flow, psi

For fluid velocities > 2.5 fps, and when lower volume fraction to be mixed is greater than 25% of total flow, use a dual turbulent module, i.e., ΔP_t would be 2x for one module.

The static mixer is also useful for direct contact heat transfer between fluids, two phase contacting, and other useful applications such as mass transfer.

For final design details and selection of mixer elements, refer directly to the manufacturers, as each design is different and may not perform like a competitor's.

Nomenclature

A = Heat transfer area, referenced to inside, i, or outside, o, sq ft
A_n = Area of outside coil or heat transfer barrier, sq ft/ft
A_i = Area of inside surface for heat transfer, such as coils, flat surfaces, or other barrier, sq ft/ft
A_{avg} = Average of inside and outside tube surface area, sq ft/ft
a = Constant in Froude number exponent equation
a = Constant in power number equation, Figure 5-13
B = Number of vertical wall baffles
b = Constant in power number equation, Figure 5-13
b = Constant in Froude number exponent equation
b = Distance baffle off flat vessel bottom, Figure 5-34
ba = Baffle clearance off tank bottom, Figure 5-34
bw = Baffle width, in., see Figure 5-34.
c, c_p = Specific heat Btu/(lb)(°F)
cp = Viscosity, centipoise
cl = Distance baffles off wall, Figure 5-34
C_F = Correction factor for Reynolds number for viscosity effects
D_i = Impeller diameter, in.
D = Impeller diameter, ft or L, Figure 5-34
D' = Inside diameter of pipe, in.
D_o = Diameter of jet from propeller mixer at origin, ft
D_T = Diameter of single pitch blade turbine, under turbulent conditions, in.
D_{cor} = Impeller diameter for turbulent regime, corrected for viscosity effect, in.
D_v = Diffusivity, sq ft/sec
d = Tube outside diameter, ft
d' = Pipe inside diameter, ft
d_t = Tube diameter, O.D., ft
F = Force (fluid) on turbine, perpendicular to shaft, ML/t^2
Fr = Froude number
f = function of ______.
g_c = Acceleration of gravity, 32.2 ft/(sec) (sec)
g = Gravitational conversional factor, 32.2 lb mass-ft/(lb force) (sec) (sec)
H = Total potential head during flow, ft of liquid
H_p = Motor, horsepower
HP = Impeller horsepower used by the system, ft lbs/sec, or HP
h = Film coefficient of heat transfer, kettle liquid to jacket wall or to coil, Btu/(hr) (sq ft) (°F)
IC = Clearance of impeller off tank bottom, in., Figure 5-34, equal to impeller (turbine) diameter, or IC = D; sometimes IC = ⅔ D is suggested
K = Absorption coefficient
K_1 = Proportionality constant, a function of the impeller shape, = 0.4 for three blade propeller in water, Equation 5-1
K_2 = Correlating factor for viscous flow power, Table 5-1
K_3 = Mixing factors, turbulent flow power, Table 5-1
k = Viscosity correction factor for turbulent flow (static mixer)
or, k = Thermal conductivity of heat transfer fluid (liquid), BTU/(hr)(ft^2)(°F/ft)
k' = Proportionality constant depending upon system, for blending
k_w = Thermal conductivity heat transfer wall, Btu/(hr) (sq ft) (°F/ft)
k_1 = Liquid film mass transfer coefficient (lb moles) (cu ft)/(sec) (sq ft) (lb mole)
L = Static mixer length, in., or length of pipe in ft
L_w = Thickness of heat transfer wall, ft
M* = Mass flow rate, lb/hr
m = Function of fluid properties, such as μ, k and cp
N_s = Shaft speed of rotation, revolutions per second
N_e = Newton number, depends on design
N_F = Force number, consistent units, dimensionless = $F/(PN^2D^4)$
N = N_m = Impeller speed of rotation, rpm
P_o = N_p = Power number, dimensionless
N_{Pr} = Prandl number (heat transfer)
N_Q = Flow number
N_{Re} = Reynolds number, dimensionless
N_U = Nusselt number, (heat transfer)
N_3 = Correlating factor for turbulent flow power
N_{we} = Weber number
n = Exponent in scale-up equation, describing type/degree of mixing required, Figure 5-32, or number of samples in statistics
or, n = Number of impellers
n_b = Number of tube baffles (vertical)
P = Power input to impeller, ft-lb/sec (see Equation 5-19) or if in a ratio can be as horsepower
P_{hp} = P' = Power, horsepower used by impeller mixing system

$P_o = N_p$ = Power number, dimensionless, Equation 5-19
P_{pcw} = Plate coil width, one plate, ft
Δp = Pressure drop, psi
ΔP_o = Pressure drop for open pipe, psi
ΔP_t = Static mixer pressure drop in turbulent flow, psi
Q = Flow rate or pumping capacity from impeller, cu ft/sec, or L^3/t
Q' = Flow rate or pumping capacity from impeller, cu ft/min
Q_e = Volume entrained into circular jet from propeller mixer, cu ft/sec
Q_g = Flow, gal/min
R = Scale-up ratio
r_w = Resistance of heat transfer barrier wall hr/(sq ft) (°F)/BTU, $= L_w'/k_w$
S_g = Fluid specific gravity (not density), referenced to water = 1.0
s = Exponent of Schmidt group
T = Tank diameter, ft, or L (consistent units), Figure 5-34
t = Residence or holding time, sec, or time of mixing
U_o = Overall heat transfer coefficient, bulk mixing liquid to transfer fluid on opposite side of heat transfer wall (coil, plate, jacket), Btu/hr/sq ft/°F
u = Velocity of mixed fluids through mixer, ft/sec
V = Volume, consistent units
W = Physical depth or height of turbine mixer, ft or in., consistent with other dimensions, Figure 5-34
or, W = Impeller blade width, ft
w = width of baffles in vertical tank, Figure 5-34.
X = Distance from impeller source, not to exceed 100 jet diameters, ft
x = Mixing correlation exponent, or empirical constant
$\overline{x}$ = Arithmetic mean (statistics)
x_i = Concentration of measurable variable
x_m = Dimension of model
x_p = Dimension of scale-up unit
x_R = Ratio of dimensions on scale-up
Z = Overall liquid vertical height of mixing vessel, from top liquid level to *bottom* (flat or dished or elliptical), ft or in., consistent with other components of equations, see Figure 5-34
z = Empirical constant

Subscripts

h = Heavy fluid
l = Light fluid
r = Ratio of values of two conditions
′ = prime, to designate a different use of similar symbol
1 = Initial condition
2 = Second condition
f = Film
i = Inside surface (heat transfer)
o = Outside surface (heat transfer)
G = Gravity
I = Inertia
R = Ratio
v = Viscosity
t = tube

Greek Symbols

θ = Blending time, min
μ_w = Viscosity of liquids at wall surface, lb/(sec) (ft)
μ = Viscosity in body liquid, lb/(sec) (ft)
μ' = Fluid viscosity, centipoise, cp
ρ = Density, lb/cu ft
σ = Standard deviation (statistics), or interfacial tension
τ = Torque on shaft, consistent units, FL or ML^2/t^2
$\phi = N_P = P_o$ = Power number, dimensionless
Φ = Power number, P_o, or ratio of power number to Froude number, N_{Fr}, to exponential power, n

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Chapter

6

Ejectors and Mechanical Vacuum Systems

Ejectors

Industry has grouped pressures below atmospheric into [24]:

	Range of Pressure
Rough vacuum	760 torr to 1 torr
Medium vacuum	1 torr to 0.001 torr
High vacuum	0.001 torr to 10^{-7} torr
Ultra-high vacuum	10^{-7} torr and below

Rough vacuum is used in about 90% of the chemical and petrochemical and other processing industries. This range generally includes vacuum distillation, filtration, crystallization, drying, reaction, and others. Medium vacuum is most applicable to molten metals degassing, molecules distillation, freeze drying, and others. High and ultra-high ranges are most useful for thin films, research, and space simulation.

For reference, note that unit of vacuum measurement is the torr.

1.00 torr = 1 mm mercury (mm Hg), abs
25.400 torr = 1 in. mercury (in. Hg)
750.1 torr = 1 bar
1.868 torr = 1 in. water at 4°C (in H_2O)
760.0 torr = 1 standard atmosphere (atm) (at sea level)
Absolute pressure = barometric pressure at location—vacuum

Statements about *vacuum* can be misleading when a clarification is not included. Vacuum refers to the "degree of emptiness" of a process system. A perfect vacuum represents an absolute zero of pressure, which is technically unmaintainable. A vacuum system indicates a system that can be a matter of the degree to which the system approaches absolute zero pressure. To create a vacuum in a fixed system, it is necessary to draw out or pump out the air in the volume. When part of the air is removed, the system has a partial vacuum. For example, when 15-inch Hg vacuum is referenced to a 29-inch Hg barometer, then the absolute pressure is 29 − 15 = 14-inch Hg absolute. The 15-inch Hg vacuum can be considered a negative gauge reading. See Chapter 2 for a diagrammatic relationship of pressures.

For quick reference, the listing below presents the most commonly used types of vacuum pumping equipment:

- liquid piston/ring
- centrifugal
- axial
- two-impeller straight lobe
- helical lobe
- reciprocating
- sliding-vane rotary
- ejector
- rotary oil-sealed
  - rotary piston type
  - vane type
- diffusion (not used for industrial/commercial application)

Vacuum Safety

Safety around mechanical vacuum pumps is possibly no different than that for other process mechanical rotating machinery. However, there is a *decided danger* of an implosion (collapse) of a tank, reactor, other process equipment operating below atmospheric pressure if:

1. It is not designed to satisfy the ASME codes for total or "full" vacuum, *regardless* of the expected actual operating vacuum on the equipment, vessel, etc.
2. There are none or inadequate vacuum relief devices on the equipment or system being evacuated.
3. Block valves are installed to allow the blocking off of equipment (vessels, tanks, etc.) thereby pulling a higher vacuum than design, if not for "full" vacuum.

   The implosion or collapse danger is real even for a tank, for example, that is not designed for vacuum (such as an API large storage tank), and liquid is pumped out of the tanks thereby creating a negative pressure, or vacuum, which collapses the roof and/or sidewalls, because no or inadequate vacuum relief was installed to allow in-flow of air as the liquid is removed (see Chapter 7).

4. Air in leakage, depending on the quantity, can create an explosive mixture in some process reaction systems; therefore, the system should be tested for air leaks and kept as tight as practical.
5. Also see Wintner [32].

Typical Range Performance of Vacuum Producers

A useful summary of the typical equipment used for developing and maintaining process system vacuum is presented in Table 6-1. Also see Birgenheier [33]. The positive displacement type vacuum pumps can handle an overload in capacity and still maintain essentially the same pressure (vacuum), while the ejectors are much more limited in this performance and cannot maintain the vacuum. The liquid ring unit is more like the positive displacement pump, but it does develop increased suction pressure (higher vacuum) when the inlet load is increased at the lower end of the pressure performance curve. The shapes of these performance curves is important in evaluating the system flexibility. See later discussion.

A simplified alternate to the previously cited procedures is suggested by Gomez [29] for calculating air in-leakage, but it is not presented in detail here.

The two most common ejectors are operated by water (or process liquid) or steam. The liquid ejectors are used for creating a modest vacuum or for mixing liquids. The steam ejector is important in creating and holding a vacuum in a system. Ejectors have no moving parts and operate by the action of one high pressure stream entraining air and other vapors (or liquids) at a lower pressure into the moving stream and thereby removing them from the process system at an intermediate pressure. Figure 6-1 illustrates the major components and the principle of operation. Since the steam jet ejector is the unit most commonly used for many process applications, it will be discussed in the greatest detail.

Referring to Figure 6-1, the high pressure steam enters the steam chest and expands in passing through the steam nozzle, leaving the nozzle at high velocity. Air, gas or vapor, or liquid mixture enters the ejector through the

Table 6-1
Typical Capacities and Operating Ranges for Vacuum Equipment

Type	Lowest recommended suction pressure	Capacity range, ft^3/min
Steam ejectors		
One-stage	75 torr	10–1,000,000
Two-stage	12 torr	
Three-stage	1 torr	
Four-stage	200 micron*	
Five-stage	20 micron	
Six-stage	3 micron	
Liquid-ring pumps		
60°F water-sealed		
One-stage	75 torr	3–10,000
Two-stage	40 torr	
Oil-sealed	10 torr	
Air-ejector first stage	10 torr	
Rotary-piston pumps		
One-stage	20 micron	3–800
Two-stage	1 micron	
Rotary-vane pumps		
Operated as a dry compressor	50 torr	20–6,000
Oil-sealed	1 torr	50–800
Oil-sealed, spring-loaded vanes		
One-stage	20 micron	3–50
Two-stage	1 micron	
Rotary blowers		
One-stage	300 torr	30–30,000
Two stage	60 torr	
Integrated pumping systems		
Ejector—liquid-ring pump	150 micron	100–100,000
Rotary-blower—liquid-ring pump	1 torr	100–10,000
Rotary-blower—rotary-piston pump	0.001 micron	100–30,000
Rotary-blower—rotary-vane pump	100 micron†	100–30,000

*1 micron = 0.001 torr.
†Based on two-stage, oil-sealed rotary-vane design that relies on centrifugal force to throw the vanes against the casing wall.
By permission, Ryans and Croll [22].

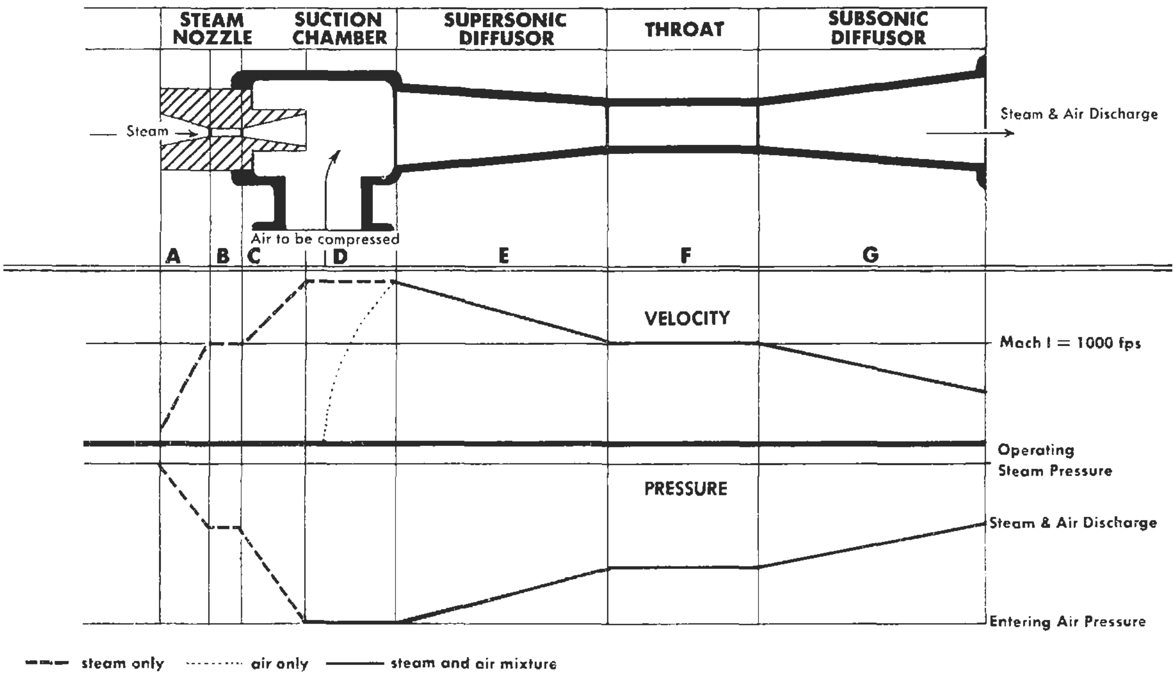


Figure 6-1. Basic ejector components and diagram of energy conversion in nozzle and diffuser. By permission, Ingersoll-Rand Co.

suction nozzle or vapor inlet, passing into the suction chamber. Here the air or other mixture is entrained by and into the high velocity steam. This new mixture enters the upper (or inlet) portion of the diffuser, passes through the diffuser throat (center narrow portion) and exits through the outlet end of the diffuser. In the diffuser the velocity head of the mixture is converted back to a pressure which is higher than the air-mixture suction, but considerably less than the inlet steam pressure.

Features

Ejectors have the following features that make them good choices for continuously producing economical vacuum conditions:

1. They handle wet, dry, or corrosive vapor mixtures.
2. They develop any reasonable vacuum needed for industrial operations.
3. All sizes are available to match any small or large capacity requirements.
4. Their efficiencies are reasonable to good.
5. They have no moving parts, hence, maintenance is low and operation is fairly constant when corrosion is not a factor.
6. Quiet operation.
7. Stable operation within design range.
8. Installation costs are relatively low when compared to mechanical vacuum pumps. Space requirements are small.
9. Simple operation.

Ejectors are rather versatile when applied to a wide variety of processing movement, compression, mixing, etc., operations. A brief listing of some useful functions are (By permission [30]):

- **Pumping and Lifting Liquids**
  - **Using steam as the motive fluid**
    - Steam jet syphons
    - Steam jet exhausters
    - Single-stage vacuum pumps
  - **Using air as the motive fluid**
    - Air jet syphons
    - Air jet exhausters
  - **Using liquids as the motive fluid**
    - Water jet eductors
    - Water jet exhausters
- **Heating Liquids (by Direct Contact)**
  - **Tank type heaters**
    - Steam jet heaters
  - **Pipeline type heaters**
    - Steam jet heaters
    - Steam jet heaters (large capacity)
    - Steam jet syphons

Open type heaters
Steam jet heaters (large capacity)

- **Moving Air and Gases (and Pump Priming)**

Using steam as the motive fluid
Steam jet blowers
Steam jet exhausters
Steam jet thermo-compressors
Single-stage vacuum pumps
Multi-stage vacuum pumps

Using air as the motive fluid
Air jet blowers
Air jet exhausters
Single-stage vacuum pumps
Air jet compressors

Using gas as the motive fluid
Gas jet compressors

Using liquid as the motive fluid
Water jet exhausters
Barometric condensers
Low level condensers
Water jet eductors (small capacities)

- **Handling Slurries and Granular Solids**

Using steam as the motive fluid
Steam jet syphons
Steam jet slurry heater
Single-stage vacuum pumps

Using air as the motive fluid
Air jet exhausters

Using liquid as the motive fluid
Water jet eductors

Types

Ejectors may be single or multi-stage and also multi-jet inside a single housing or stage. The extra stages, with or without interstage condensing of steam, allow the system to operate at lower absolute pressures than a single stage unit. Various combinations of series of jets with no inter-condensing can be connected to jets with intercondensers or aftercondensers to obtain various types of operation and steam economy. The condensers may be barometric or surface type.

Figure 6-2 suggests a few of the many uses to which ejector type units are used in industry.

Figure 6-3 illustrates a single-stage non-condensing ejector. In this type of installation the steam outlet from the ejector is either exhausted to atmosphere or on top of water in a sump.

Figure 6-4 shows two individual single-stage ejectors discharging into a common surface after-condenser. The steam condensate can be re-used from this installation.

Figures 6-5 and 6-6 illustrate two-stage ejector installations with barometric and surface type inter-after condensers respectively. The discharge of the steam non-condensables from the second stage jet of Figure 6-5 is exhausted to the atmosphere, while in Figure 6-6 the steam is condensed in the aftercondenser and, essentially, only non-condensables leave the vent of the aftercondenser. Figure 6-7A indicates a diagram of a three-stage barometric type installation.

Figure 6-7B illustrate a barometric refrigeration unit, generating chilled water in the range of 34°F to 55°F for process cooling. It uses steam ejectors to lower the chill tank's vapor pressure to establish boiling/evaporation of the water in the tanks and condenses the vapors released by plant cooling water in the barometric condensing unit, which is sealed through the vacuum leg into a "hot" well.

Figure 6-8 illustrates various arrangements of ejectors with inter and aftercondensers. The condensers can be barometric or nonbarometric types.

Note that in Figure 6-8 and Table 6-2 the letter designations of the stages conform to the latest *Standards of the Heat Exchange Institute for Steam Jet Vacuum Systems* [11]. The letter designates the jet's stage position in the system.

Precondensers are recommended for any ejector system when the pressure conditions and coolant temperature will allow condensation of vapors, thus reducing the required design and operating load on the ejectors. This is usually the situation when operating a distillation column under vacuum. The overhead vapors are condensed in a unit designed to operate at top column pressure, with only the non-condensables and vapors remaining after condensation passing to the ejector system.

Intercondensers are used to condense the steam from a preceding ejector stage, thus reducing the inlet quantity of vapor mixture to the following stage. This is a means of increasing steam economy.

Aftercondensers operate at atmospheric pressure. They do not affect the steam economy or ejector performance, but they do avoid the nuisance of exhausting steam to the atmosphere, thus, they allow steam to be recovered. They also serve as silencers on the ejectors, and with barometric types they can absorb odors and corrosive vapors.

Condenser tail pipes, used with any condenser, are sealed with a 34-foot leg into a sump, or with a condensate pump operating under vacuum on suction. With surface-type condensers, the level may be sealed in a receiver with a float or other type of level control.

Thermocompressors are steam jet ejectors used to boost low pressure or waste steam to a higher intermediate pres-

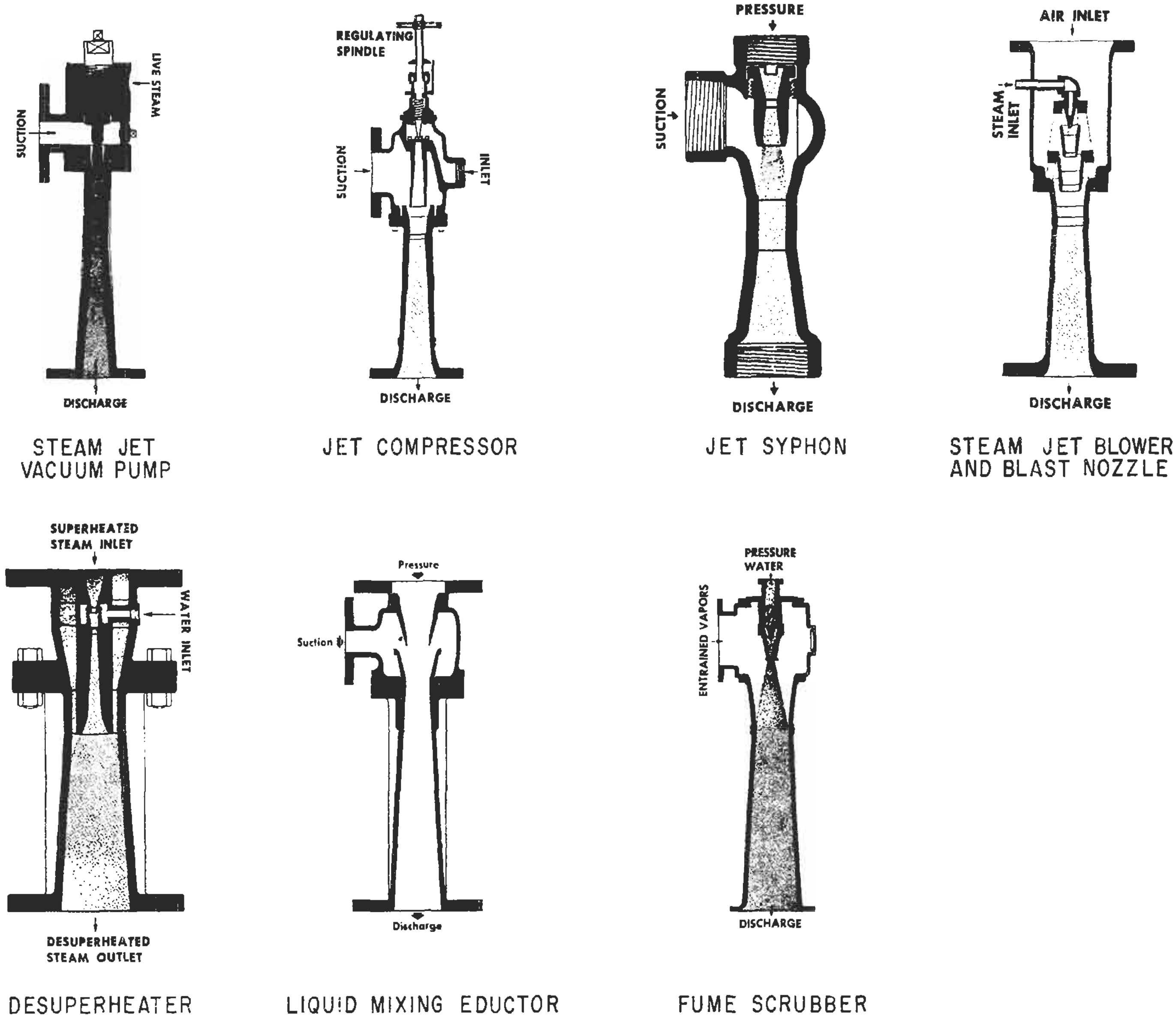


Figure 6-2. Steam, air, gas, and liquid ejectors. By permission, Ketema, Schutte & Koerting Division.

sure. Single-stage units are usually not used for compression ratios (ratio of absolute discharge to suction pressures) greater than three [16]. This type of pressure increase for low pressure steam is usually uneconomical when the final discharge pressure exceeds one-third of the high pressure motive steam [16]. These units are usually limited to single-stage installations based on steam economy.

Materials of Construction

Because the ejector is basically simple in construction, it is available in many materials suitable for handling corrosive vapors. Standard materials include cast iron, Meehanite, cast steel, stainless steel, Monel®, Hastelloy®, titanium, Teflon®, Haveg®, rubber-lined steel, graphite-lined, polyvinyl chloride (PVC), fiberglass reinforced plastic (FRP), and bronze for the body and diffuser depending on the pressure and temperature rating. The nozzle is usually stainless steel or monel. Other materials of construction include: porcelain, carbon, graphite, impregnated graphite, synthetic resins, glass, and special metals of all types. Intercondensers and aftercondensers are sometimes made of these same materials and may include random-packed surface-type, graphite or glass tubes, etc.

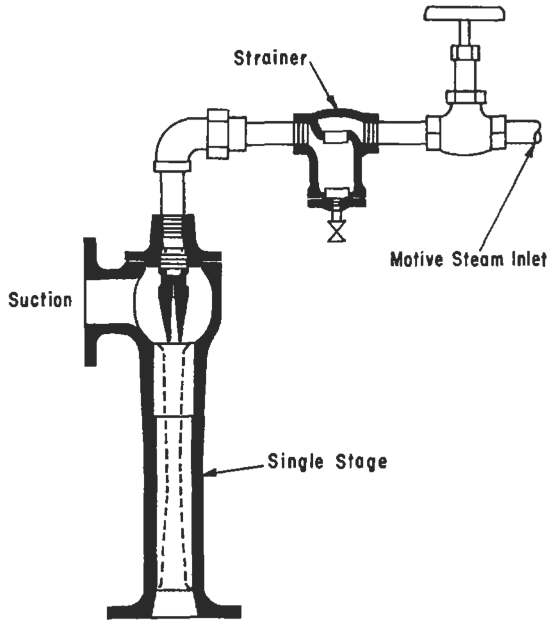


Figure 6-3. Single-stage non-condensing ejector. By permission, C. H. Wheeler Mfg. Co.

Vacuum Range Guide

It is necessary to consult manufacturers for final and specific selections. However, the following guide data is reliable and should serve to check recommendations or to specify a system. It is advisable to try to accomplish the specific operation with as few ejectors as possible, because this leads to the most economical operation and lowest first cost in the majority of cases. Figures 6-9A, B, and C are a basic comparison guide for vacuum systems.

The ranges shown for various numbers of ejector stages provide a reasonable operational guide with understandable variations between various manufacturers, even in combinations of each manufacturer's specific ejectors used to attain lower or upper range of the chart. For example, at zero load or shutoff pressure, Reference [18] indicates the approximate values for evacuation pressures (lowest):

Stage No. in System	*Lowest Absolute Pressure, mm Hg abs*
Single	50
Two	4 to 10
Three	0.8 to 1.5
Four	0.1 to 0.2
Five	0.01 to 0.02
Six	0.001 to 0.003

Figure 6-10 is a summary of operating pressure ranges for a variety of processes and vapor mixtures.

Table 6-1 and Table 6-3 gives the usual industrial application ranges for ejector stages.

Figures 6-11A, B, and C indicate the capacity of various ejector-condenser combinations for variable suction pressures when using the same quantity of 100 psig motive steam. Each point on these curves represents a point of maximum efficiency, and thus any one curve may represent the performance of many different size ejectors each operating at maximum efficiency [1]. Good efficiency may be expected from 50%–115% of a design capacity. Note that the performance range for the same type of ejector may vary widely depending upon design conditions.

Pressure Terminology

For design purposes it is necessary to use absolute pressures. In plant operation pressures are often used as "vacuum." It is important to eliminate confusion before making a proper performance analysis. See Tables 6-4 and 6-5.

If pressure is expressed as inches of mercury vacuum, the reading of the local barometer (or a reference barometer) is necessary to establish the absolute suction pressure, or pressure in the vacuum system.

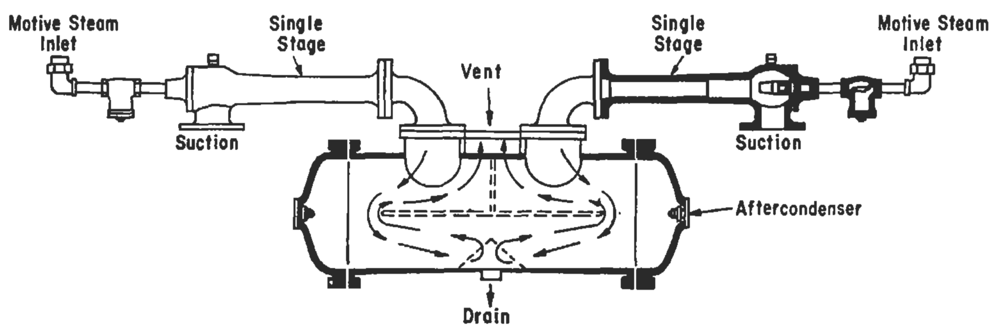


Figure 6-4. Twin single-stage ejectors with surface after-condenser. By permission, C. H. Wheeler Mfg. Co.

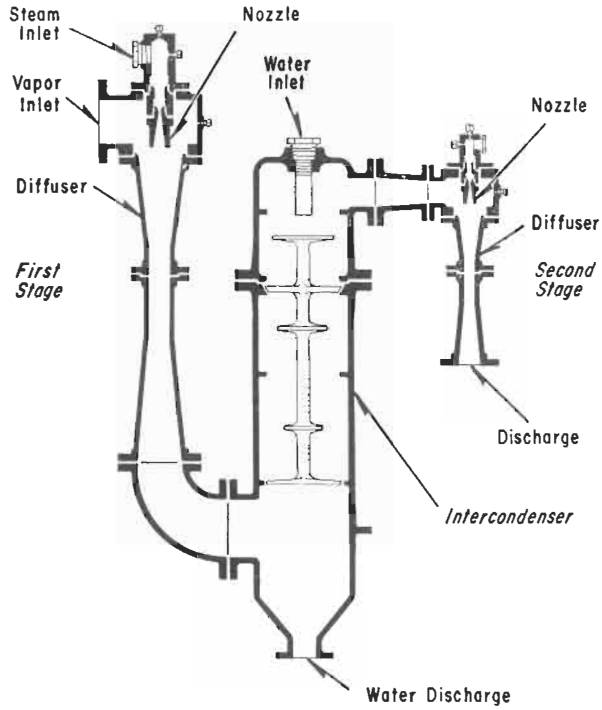


Figure 6-5. Two-stage ejector using barometric type intercondenser. By permission, Elliott Co.

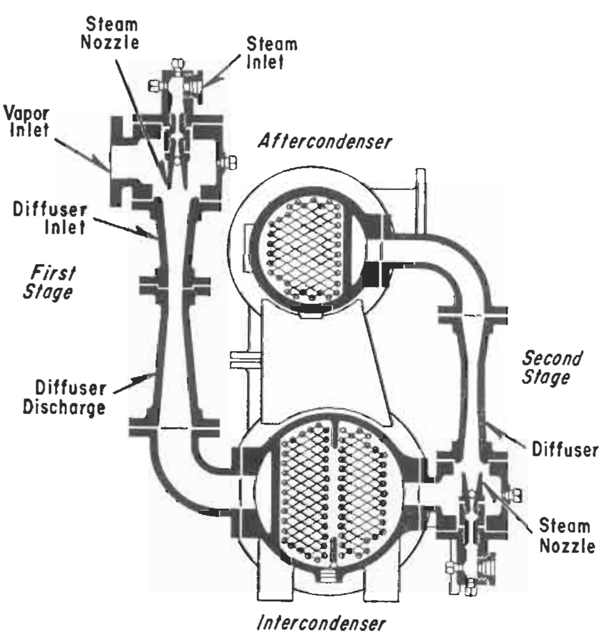


Figure 6-6. Two-stage ejector employing surface type inter-and-after-condenser. By permission, Elliott Co.

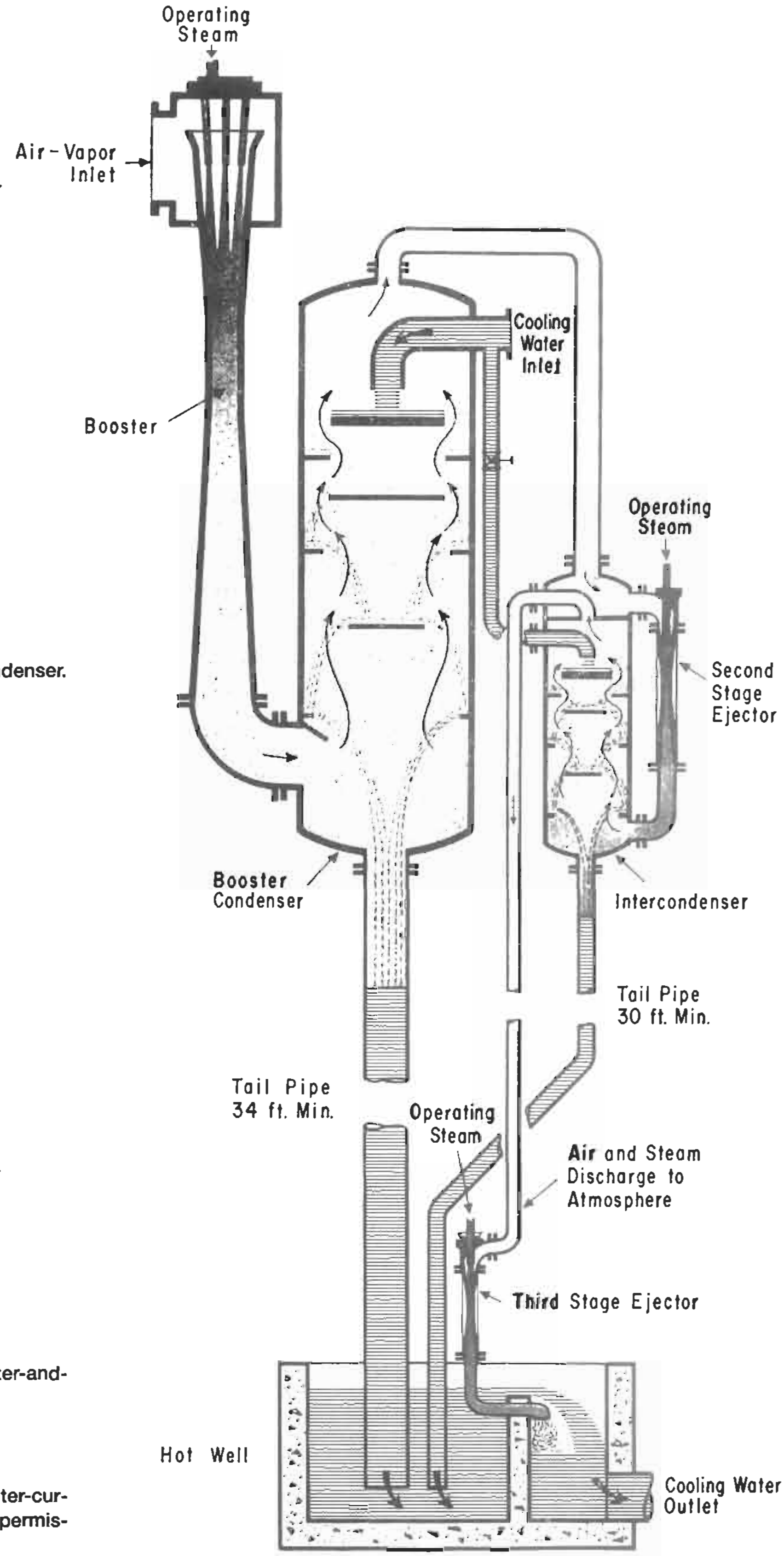


Figure 6-7A. Flow diagram of three-stage ejector with counter-current barometric booster condenser and intercondenser. By permission, Ingersoll-Rand Co.

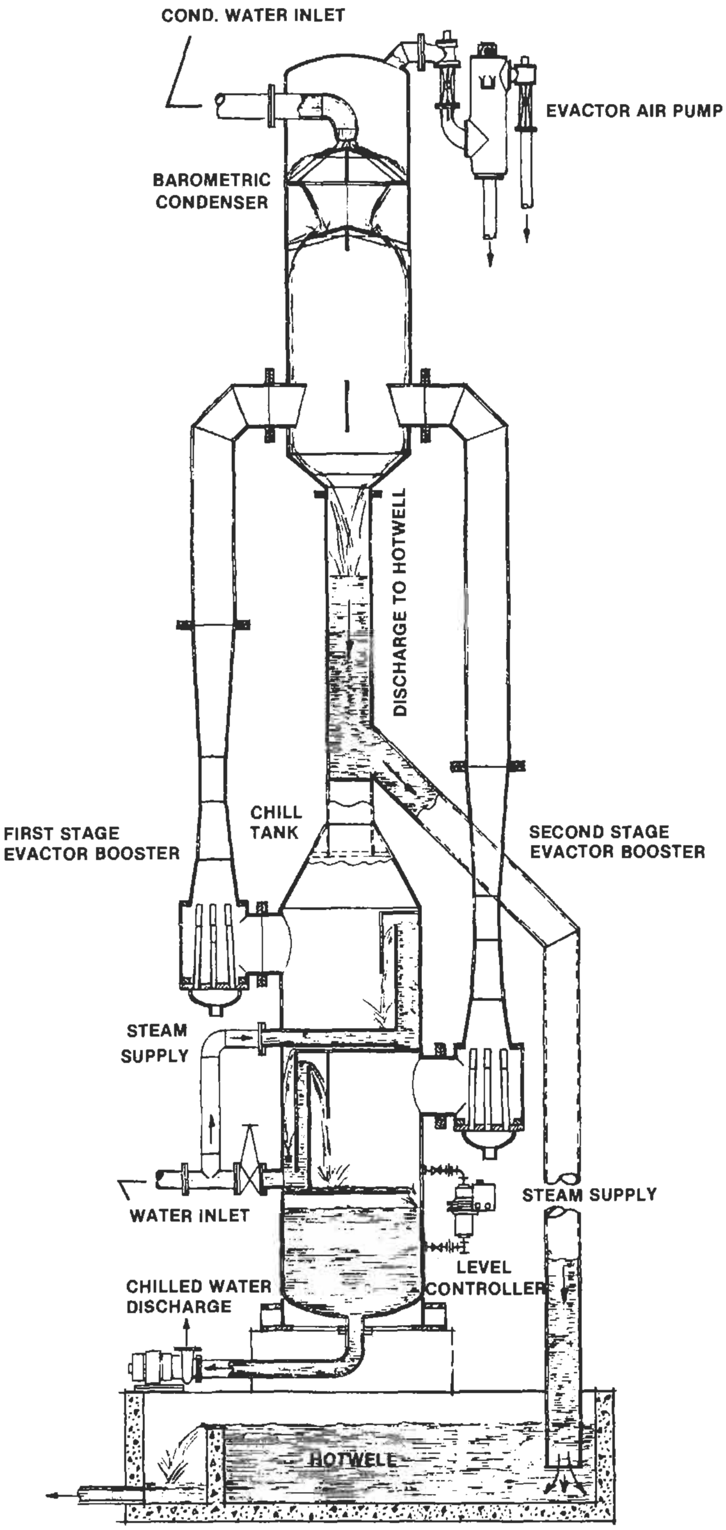


Figure 6-7B. Chilled water refrigeration unit using steam jet ejectors. By permission, Croll-Reynolds Co., Inc.

Absolute Measurement Relations
(Reference to Mercury)

One inch mercury = 25.4 millimeters mercury = 0.491 psi abs.

One inch	= 25,400 microns (micrometers)
One millimeter	= 0.03937 inch
One millimeter	= 1,000 microns
One micron	= 0.001 millimeter
One micron	= 0.00003937 inch
One kilogram/sq centimeter	= 28.96 inches
One kilogram/sq centimeter	= 14.22 psi abs
One kilogram/sq centimeter	= 735.6 millimeter

Table 6-2
Standard Ejector Units Designations Conforming to Heat Exchange Institute

Letter No.	Position in Series	Normal Range of Suction Pressures (Hg. Abs.)	Normal Range of Disch. Pressures (Hg. Abs.)
Z	Atmospheric stages	3″–12″	30″–32″
Y	1st of two stages	.5″–4″	4″–10″
X	1st of three stages	.1″–1″	1″–3″
W	1st of four stages	.2 mm–4 mm	2 mm–20 mm
V	1st of five stages	.02 mm–.4 mm	.4 mm–3 mm
U	1st of six stages	.01 mm–.08 mm	.08 mm–.4 mm

The different types of condensing equipment used with the various series are identified by the following letters:

B—Barometric Counter-Flow Condenser, Intercondenser and Aftercondenser

S—Surface Type Condenser, Intercondenser and Aftercondenser

J—Atmospheric Jet Condenser, Intercondenser and Aftercondenser.

C—Surface Coil Type Condenser, Intercondenser and Aftercondenser

N—Signifies no condenser in the series.

The operating range of the condensing equipment determines the nomenclature. Here are the basic divisions.

Condenser	1.5″ Hg–4″ Hg abs
Intercondenser	4″ Hg–10″ Hg abs
Aftercondenser	30″ Hg–32″ Hg abs

By permission, Croll-Reynolds Co., Inc.

Example 6-1: Conversion of Inches Vacuum to Absolute

A distillation column is operating at 27.5 inches mercury vacuum, referenced to a 30-inch barometer. This is the pressure at the inlet to the ejector. Due to pressure drop through a vapor condenser and trays of a distillation column, the column bottoms pressure is 23 inches vacu-

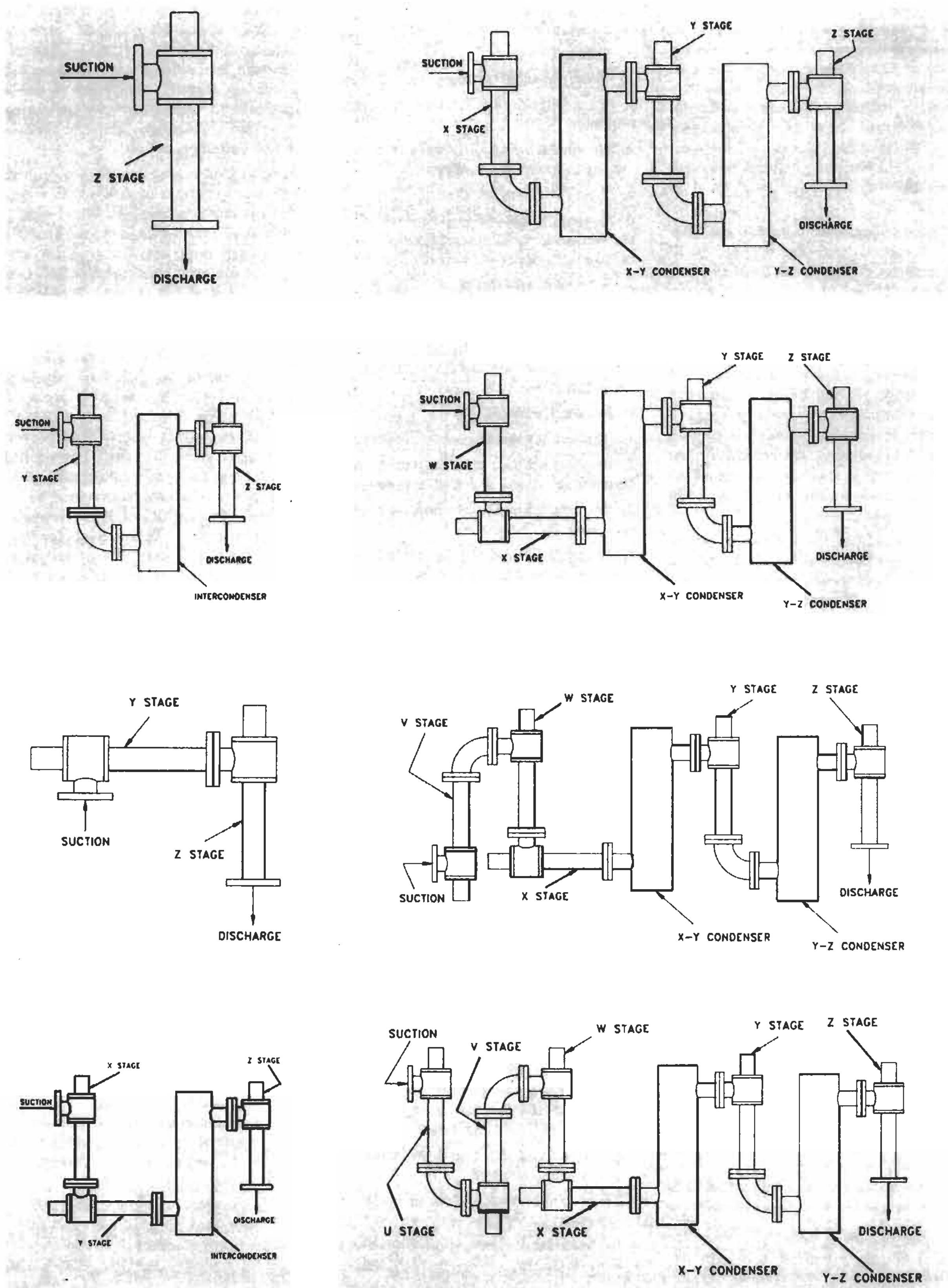
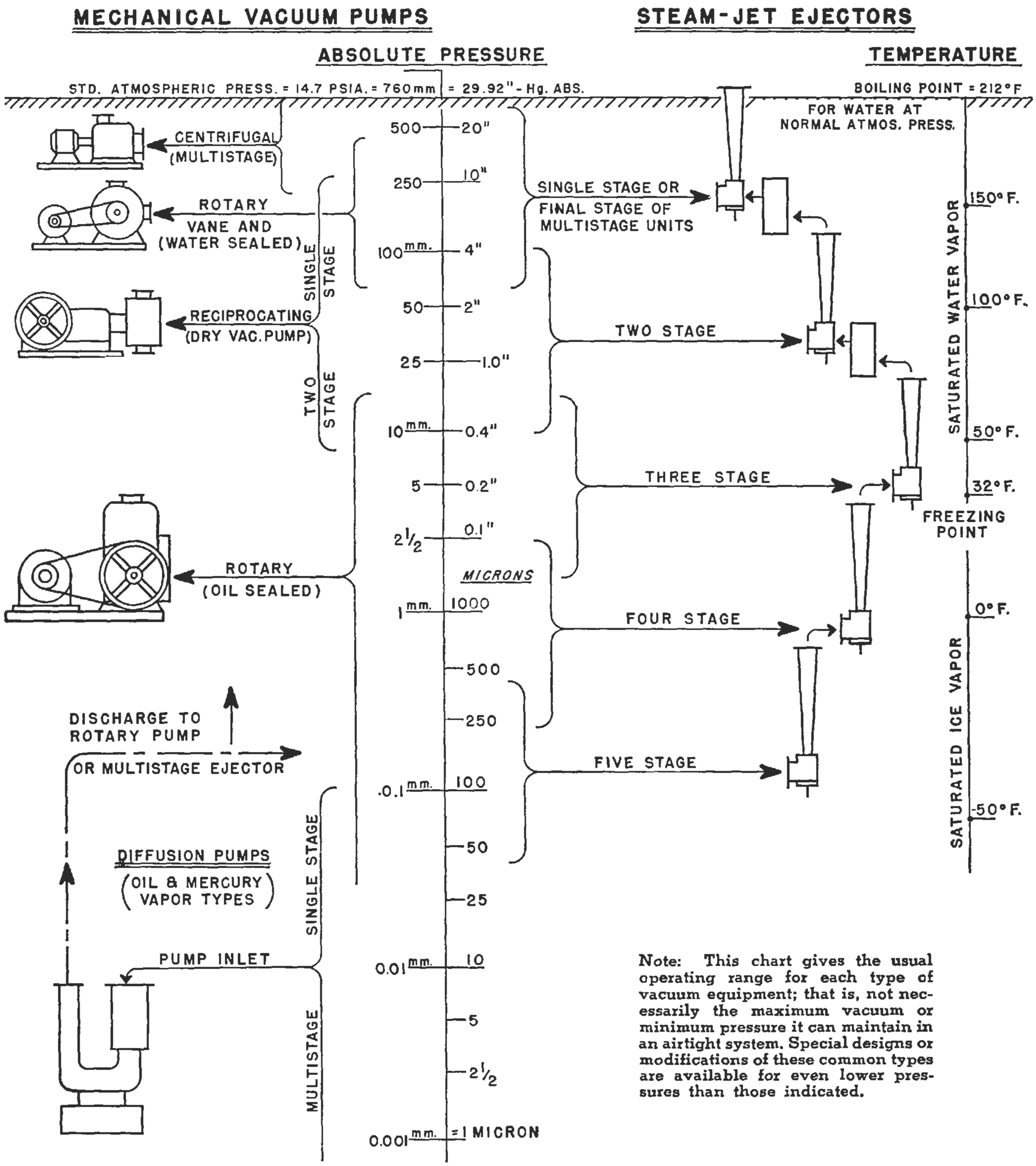


Figure 6-8. Steam jet arrangements with inter-and-after condensers. By permission, Croll-Reynolds Co., Inc.



WHERE VACUUM EQUIPMENT APPLIES

The common types of vacuum-producing equipment used in commercial processes are indicated on this chart, together with the approximate operating range of each one. The central logarithmic scale shows absolute pressures in terms of both millimeters and inches of mercury. The right-hand scale gives the temperatures at which water or ice vaporizes at the corresponding pressures. Combinations of equipment are necessary to obtain extremely low pressures.

Figure 6-9A. Where vacuum equipment applies. By permission, Ingersoll-Rand Co.

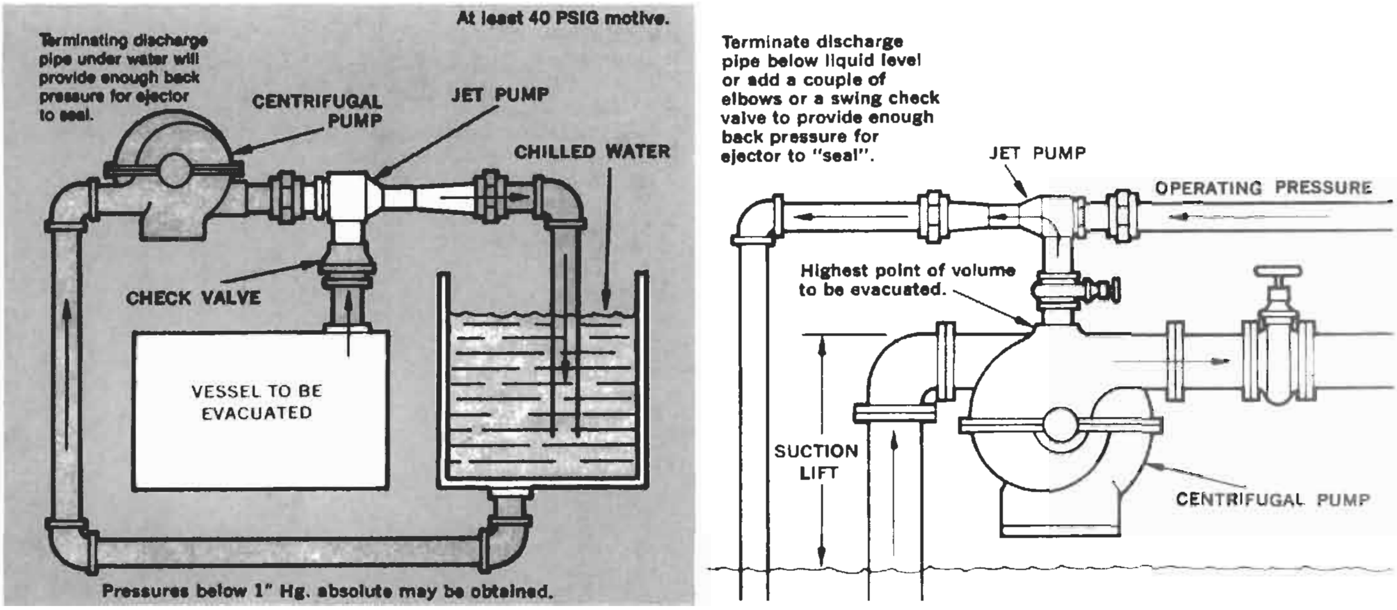


Figure 6-9B. Typical jet (gas) system application. By permission, Penberthy, Inc.

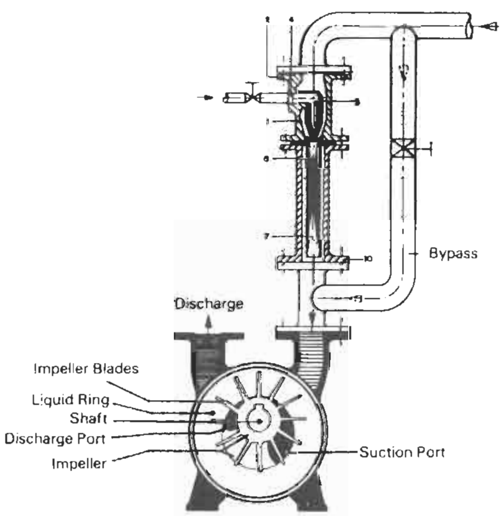


Figure 6-9C. Improving the maximum vacuum obtainable by staging an ejector ahead of the suction of a liquid ring pump or any other device that can handle water into the unit, unless dry air is used. By permission, Graham Manufacturing Co., Inc.

um. Determine the absolute pressures at the entrance to the ejector and at the bottom of the column.

Ejector inlet, absolute pressure = 30 − 27.5 = 2.5 in. mercury

Columns bottom, absolute pressure = 30 − 23 = 7 in. mercury

Pressure Drop at Low Absolute Pressures

Refer to Chapter 2, Flow of Fluids

Performance Factors

Steam Pressure

The motive steam design pressure must be selected as the lowest expected pressure at the ejector steam nozzle. The unit will not operate stably on steam pressures below the design pressure [16].

Recommended steam design pressure:

= minimum expected line pressure at ejector nozzle minus 10 psi

This design basis allows for stable operation under minor pressure fluctuations.

An increase in steam pressure over design will not increase vapor handling capacity for the usual "fixed capacity" ejector. The increased pressure usually decreases capacity due to the extra steam in the diffuser. The best ejector steam economy is attained when the steam nozzle and diffuser are proportioned for a specified performance [8]. This is the reason it is difficult to keep so-called standard ejectors in stock and expect to have the equivalent of a custom designed unit. The "throttling type" ejector has a family of performance curves depending upon the motive steam pressure. This type has a lower compression ratio across the ejector than the fixed-type. The fixed-type unit is of the most concern in this presentation.

For a given ejector, an increase in steam pressure over the design value will increase the steam flow through the nozzle in direct proportion to the increase in absolute

Table 6-3

General Pressure Ranges for Ejectors

No. Stages	Minimum Practical Absolute Pressures, mm. Hg.*	Range Operating Suction Pressure, mm. Hg. (1) (2)	Closed Test Pressure mm. Hg. (2)
1.........	50	75, (3″) and up	37–50
2.........	5	10–100	5
3.........	2	1–25	1
4.........	0.2	0.25–3	0.05–0.1
5.........	0.03	0.03–0.3	0.005–0.01
6.........	0.003		
7.........	0.001 to 0.0005**		

* Linck, C. G., Selecting Ejectors for High Vacuum, Chem. Eng. Jan. 13, pg. 145 (1958) Ref. (9)

** Berkeley, F. D., Ejectors Have a Wide Range of Uses, Pet. Ref. 37, No. 12, pg. 95 (1958), Ref. (1)

(1) Worthington Corp. Bul. W-205-E21 (1955), Ref. (14)

(2) The Jet-Vac Corp., Bulletin, Ref. (15).

Table 6-4

Low Absolute Pressure Equivalents (References to Mercury)

Microns	Millimeters	Inches	Inches Vac. Referred to 30″ Barometer
10....	0.01	.000394	29.999606
100....	0.10	.003937	29.996063
200....	0.20	.007874	29.992126
300....	0.30	.011811	29.988189
400....	0.40	.015748	29.984252
500....	0.50	.019685	29.980315
600....	0.60	.023622	29.976378
700....	0.70	.027559	29.972441
800....	0.80	.031496	29.968504
900....	0.90	.035433	29.964567
1000....	1.00	.039370	29.960630

Table 6-5

Absolute Pressure Conversion Table Millimeters to Inches Mercury

Milli-meters	Inches	Milli-meters	Inches	Milli-meters	Inches
1	0.0394	26	1.0236	170	6.6929
2	0.0787	27	1.0630	180	7.0866
3	0.1181	28	1.1024	190	7.4803
4	0.1575	29	1.1417	200	7.8740
5	0.1969	30	1.1811	210	8.2677
6	0.2362	35	1.3780	220	8.6614
7	0.2756	40	1.5748	230	9.0551
8	0.3150	45	1.7717	240	9.4488
9	0.3543	50	1.9685	250	9.8425
10	0.3937	55	2.1653	260	10.236
11	0.4331	60	2.3622	270	10.630
12	0.4724	65	2.5590	280	11.024
13	0.5118	70	2.7559	290	11.417
14	0.5512	75	2.9528	300	11.811
15	0.5906	80	3.1496	325	12.795
16	0.6299	85	3.3465	350	13.780
17	0.6693	90	3.5433	375	14.764
18	0.7087	95	3.7402	400	15.748
19	0.7480	100	3.9370	450	17.717
20	0.7874	110	4.3307	500	19.685
21	0.8268	120	4.7244	550	21.653
22	0.8661	130	5.1181	600	23.622
23	0.9055	140	5.5118	650	25.590
24	0.9449	150	5.9055	700	27.559
25	0.9843	160	6.2992	750	29.528

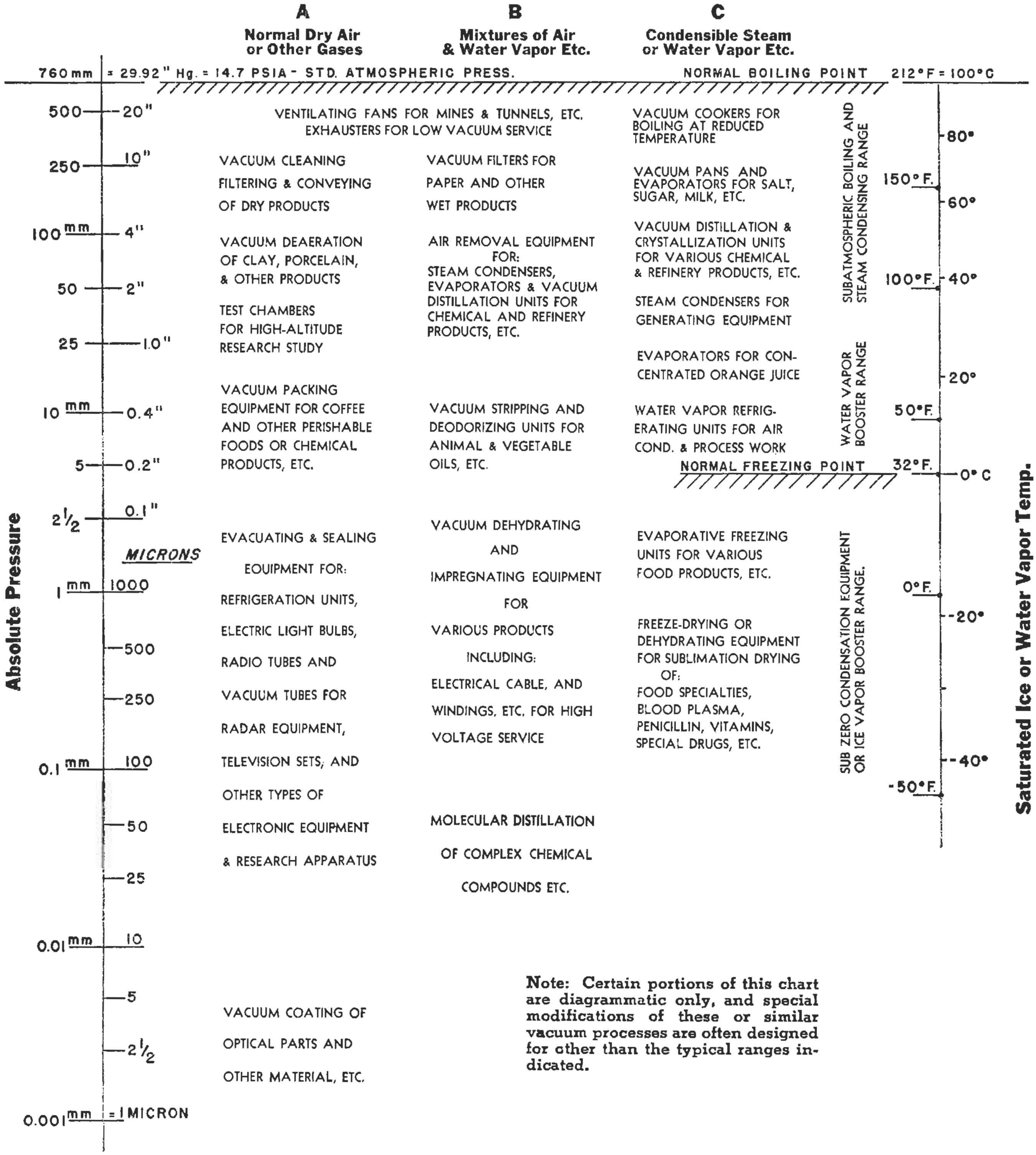
Note: To change above values to pressure in pounds per square inch absolute multiply by the following factors:
Multiply millimeters of mercury by 0.01934
Multiply inches of mercury by 0.4912

Courtesy C. H. Wheeler Mfg. Co., Philadelphia, Pa.

steam pressure [16]. The higher the actual design pressure of an ejector the lower the steam consumption. This is more pronounced on one- and two-stage ejectors. When this pressure is above about 350 psig, the decrease in steam requirements will be negligible. As the absolute suction pressure decreases, the advantages of high pressure steam becomes less. In very small units the physical size of the steam nozzle may place a lower ceiling on steam pressures. Figure 6-12 illustrates the effect of excess steam pressures on ejector capacity for single- and two-stage units.

For ejectors discharging to the atmosphere, steam pressures below 60 psig at the ejector are generally uneconomical [16]. If the discharge pressure is lower as in multistage units, the steam pressure at the inlet can be lower. Single-stage ejectors designed for pressures below 200 mm Hg. abs., cannot operate efficiently on steam pressures below 25 psig [1]. The first stage or two of a multistage system can be designed (although perhaps not economically) to use steam pressures below one psig.

To ensure stable operations the steam pressure must be above a minimum value. This minimum is called the motive steam pickup pressure [1] when the pressure is being increased from the unstable region. Figure 6-13 indicates both this point and the second lower break pressure which is reached as the pressure is lowered from a stable region. As the pressure is reduced along line 5-3-1, the operation is stable until point 1 is reached. At this point the ejector capacity falls off rapidly alone line 1-2. As the steam pressure is increased, stable operation is not resumed until point 4 is reached and the capacity rises along line 4-3. With further increases it rises along 3-5. This is the stable region. Operation in the region 3-1 is unstable and the least drop in pressure can cause the sys-



OPERATING RANGES OF VACUUM PROCESSES

Most of the principal industrial processes that necessitate the use of subatmospheric pressures are listed here, together with the typical pressure range for each one. The processes involve the evacuation or removal of various combinations of air, other gases, and water vapor, as shown under the three headings A, B, and C. The charts on this page and the preceding one were compiled by J. F. Plummer, Jr., of Ingersoll-Rand Company, Phillipsburg, N. J.

Figure 6-10. Operating ranges of vacuum processes. By permission, Ingersoll-Rand Co.

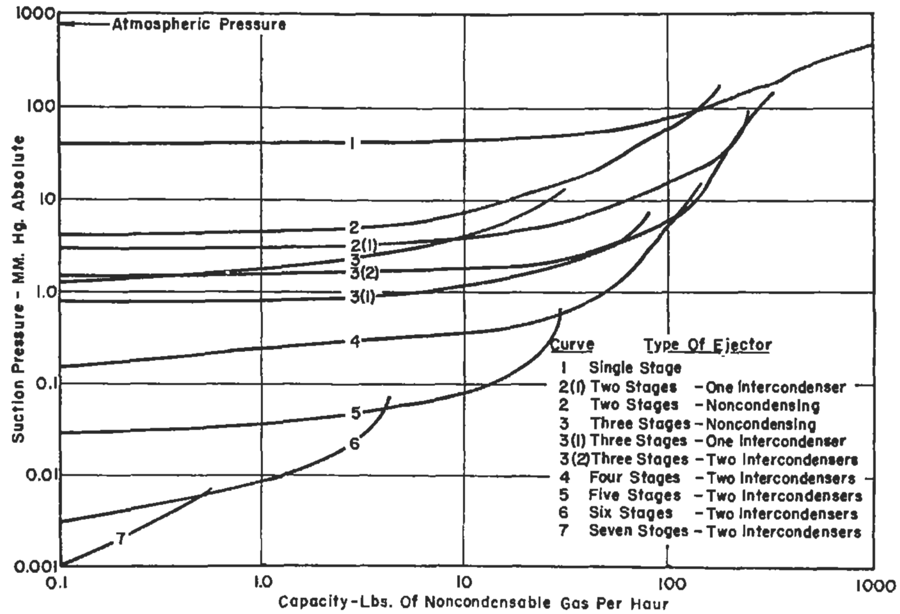


Figure 6-11A. Comparison guide for steam ejector performance. As absolute pressure is reduced, the number of stages increases for a given capacity. The same steam consumption is used for each design. By permission, Berkley, F. D. [1].

tem to lose vacuum. The relative location of points 3 and 1 can be controlled to some extent by ejector design; and the points may not even exist for ejectors with low ratios of compression.

Figure 6-14 indicates the change in region of stable performance as reflected in changes in the backpressure on the ejector and the variation in steam pressure. This system backpressure might represent a variation in barometric pressure for a unit discharging to the atmosphere, or the variation in a feedwater (or other) heater operating pressure if the ejector discharges into a closed system or condenser. Figure 6-14 numerically represents the latter situation, although the principle is the same.

The three motive steam pressure curves, 100%-90%-80%, are obtained from the ejector manufacturer as is the performance curve of suction pressure versus percent of ejector design capacity. This latter curve for an actual installation would show actual absolute suction pressures versus pounds per hour or cubic feet per minute of air or percent design capacity.

The backpressure is represented by the straight lines labeled minimum, normal and maximum. Only one capacity curve is shown since the increase in capacity resulting from the lower steam pressure is negligible [4].

Curves 1, 2 and 3 represent the maximum safe discharge pressure, as the system will operate along the capacity curve as long as the system discharge pressure from the ejector is less than the maximum value of the curve, all for a given suction pressure [4]. The slopes of the curves are a function of the type of ejector, its physical design and relative pressure conditions. Whenever the discharge backpressure exceeds the maximum safe discharge pressure as represented by one of the curves, the ejector operates in the "break" unstable region.

In Figure 6-14 the 100% pressure curve does not cross any of the system backpressure lines (minimum, normal or maximum) and the ejector would be expected to operate stably over its entire range, down to shut-off. Following the 90% steam pressure curve, the ejector is stable at 100% design suction pressure and 100% design capacity at the maximum back pressure. It is unstable below design load unless the heater pressure is reduced. Note that its break occurs at 20 psia and 100% design suction pressure. If the discharge pressure is reduced to 19 psia, the unit will be stable to shut-off (zero capacity). The 80% steam pressure will allow stable operation from shut-off up through the full capacity range as long as the backpressure does not exceed 18 psia. This type of analysis is necessary to properly evaluate ejector performance with varying system conditions.

A unit is said to have 50% overload capacity when it blanks off (zero load) at a stable absolute pressure and has an operating curve which stably handles 1.5 times the design conditions of flow.

Effect of Wet Steam

Wet steam erodes the ejector nozzle and interferes with performance by clogging the nozzle with water droplets [16]. The effect on performance is significant and is usually reflected in fluctuating vacuum.

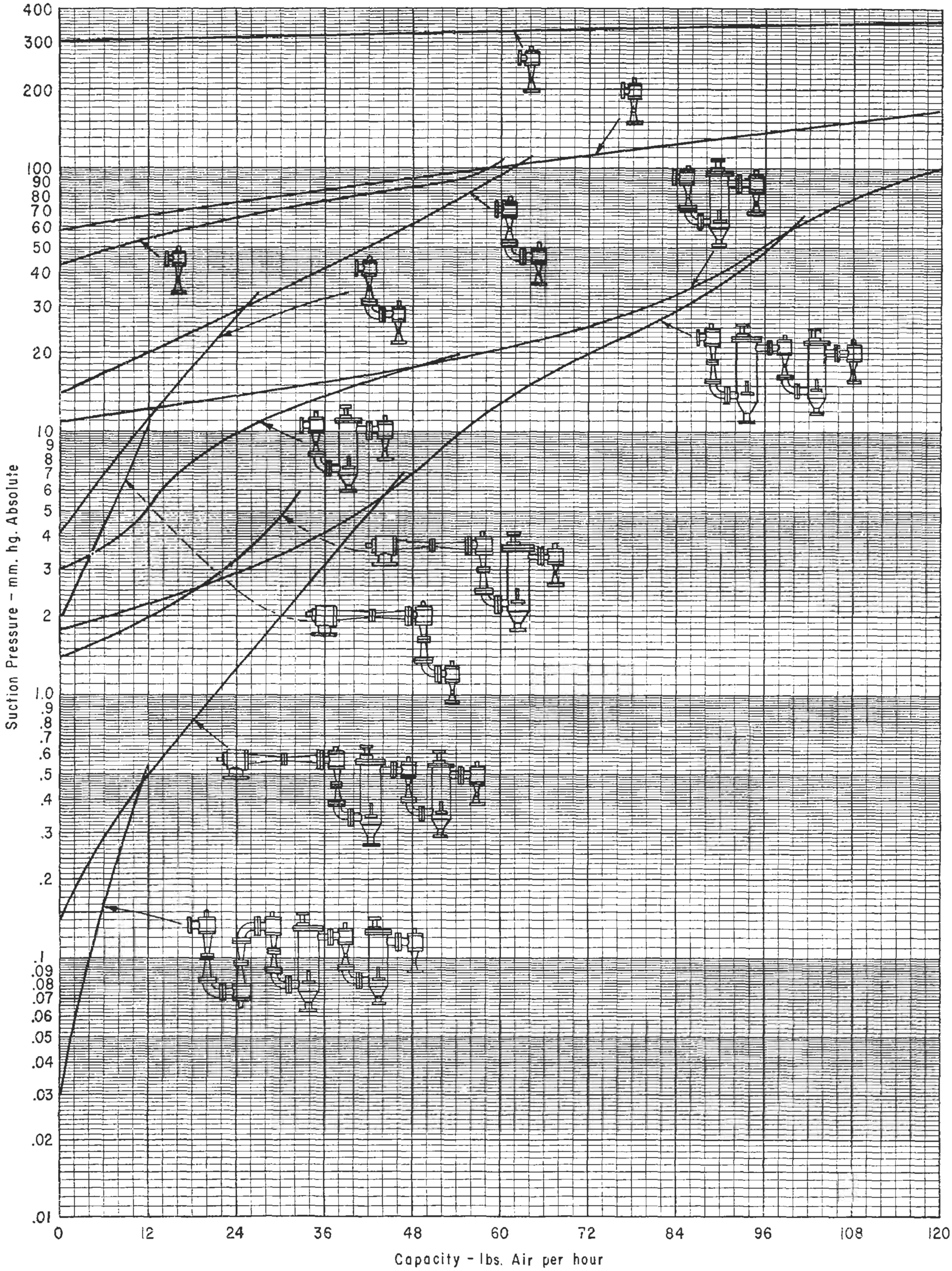


Figure 6-11B. A typical relative comparison of various designs of steam jet ejectors. Based on same steam consumption, 100 psig steam pressure and 85°F water. Curves represent the capacity of ejectors designed for maximum air handling capacity at any one particular suction pressure. By permission, Graham Manufacturing Co.

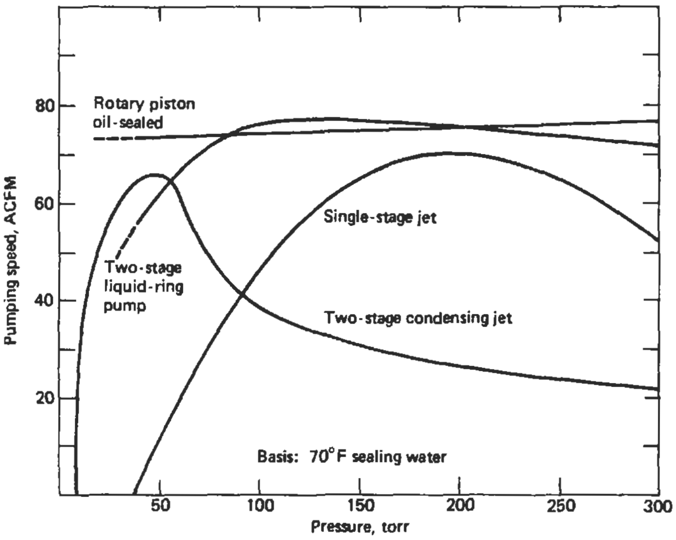


Figure 6-11C. Typical performance curves for steam jet ejectors, liquid ring pumps, and rotary piston oil-sealed pumps. By permission, Ryans, J. L. and Roper, D. L. [24].

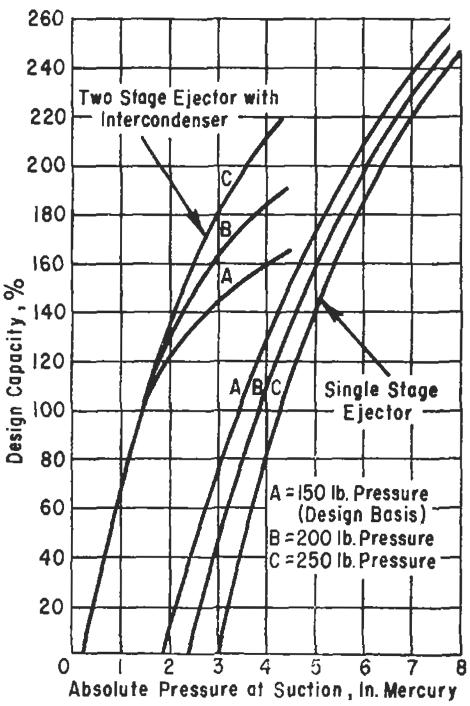


Figure 6-12. Effects of excess steam pressure on ejector capacity. By permission, C. H. Wheeler Mfg. Co.

Effect of Superheated Steam

A few degrees of superheat are recommended (5–15°F), but if superheated steam is to be used, its effect must be considered in the ejector design. A high degree of superheat is of no advantage because the increase in available energy is offset by the decrease in steam density [16].

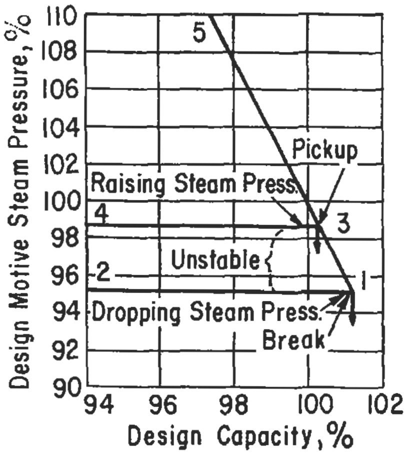


Figure 6-13. Effect of steam pressure on capacity for constant system suction and back-pressure. By permission, P. Freneau, [4].

Suction Pressure

The suction pressure of an ejector is expressed in absolute units. If it is given as inches of vacuum it must be converted to absolute units by using the local or reference barometer. The suction pressure follows the ejector capacity curve, varying with the non-condensable and vapor load to the unit.

Discharge Pressure

As indicated, performance of an ejector is a function of backpressure. Most manufacturers design atmospheric discharge ejectors for a pressure of 0.5 to 1.0 psig in order to insure proper performance. The pressure drop through any discharge piping and aftercooler must be taken into consideration. Discharge piping should not have pockets for condensation collection.

Figure 6-15 indicates the effect of increasing the single-stage ejector backpressure for various suction pressures. Figure 6-16 illustrates the effect of increasing the motive steam pressure to overcome backpressure effects. When this pressure cannot be increased, the nozzle may be redesigned to operate at the higher backpressure.

Capacity

The capacity of an ejector is expressed as pounds per hour total of non-condensable plus condensables to the inlet flange of the unit. For multistage units, the total capacity must be separated into pounds per hour of condensables and non-condensables. The final stages are only required to handle the non-condensable portion of the load plus the saturation moisture leaving the intercondensers.

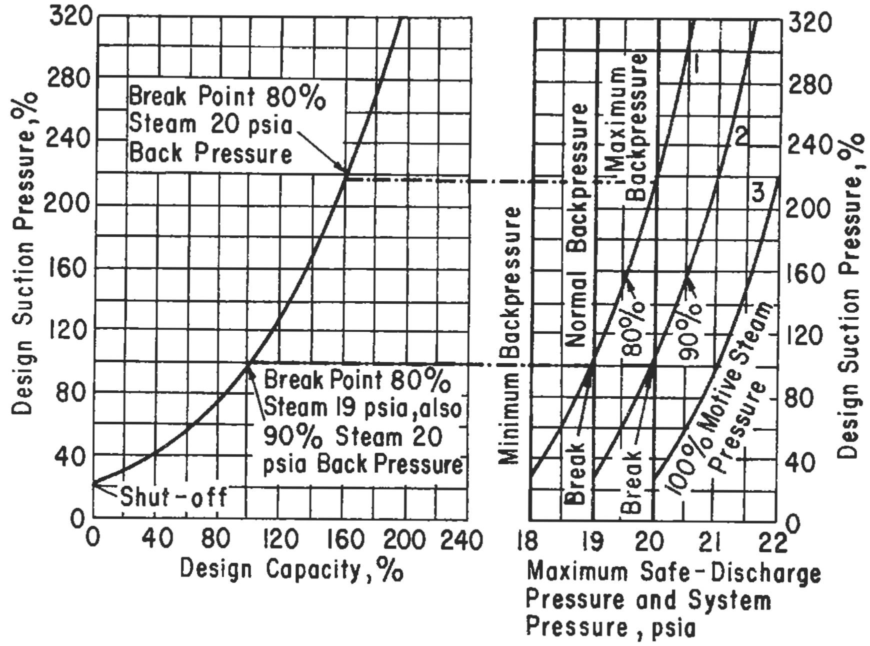


Figure 6-14. Effect of steam back-pressure variations with steam pressure. By permission, Freneau, P. [4].

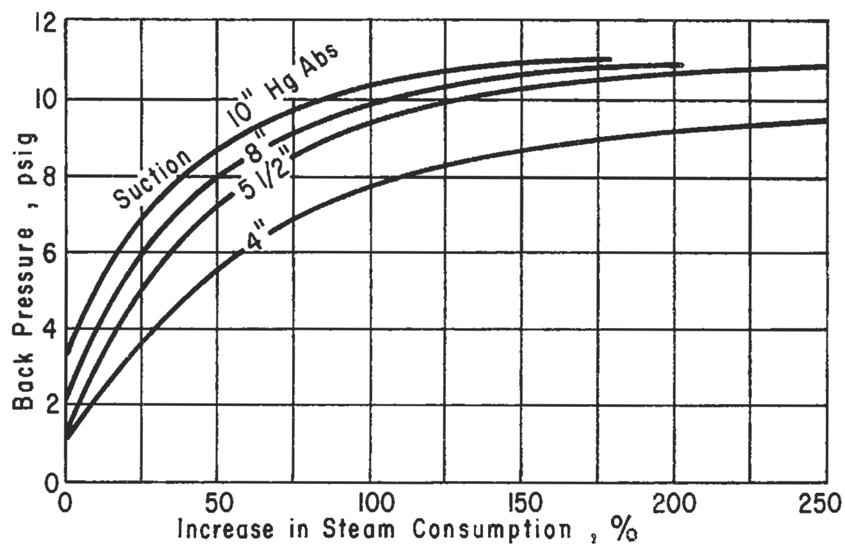


Figure 6-15. Effect of back-pressure on single-stage exhausters. By permission, Ketema, Schutte & Koerting Div. and Ketterer, S. G. and Blatchley, C. G. [8].

The non-condensables leaving a surface condenser are saturated with water vapor at the temperature corresponding to the pressure. For a process condenser the vapor corresponds to the process fluid.

1. Surface Condenser

To provide for sufficient total capacity, the temperature at the air outlet of a well designed surface condenser is generally assumed to be about 7.5°F below the temperature of saturated steam at the absolute pressure in the condenser [12].

2. Barometric or Low Level Jet Condenser

In this case the temperature at the air outlet of this type condenser is generally assumed to be 5°F above the inlet temperature of the cooling water. In addition to the normally expected air leakage, an allowance must be made for air liberated from the injection water [10].

Types of Loads

Air Plus Water Vapor Mixtures

The Heat Exchanger Institute [11] references all tests and calculations for jet performance to 70°F air equiva-

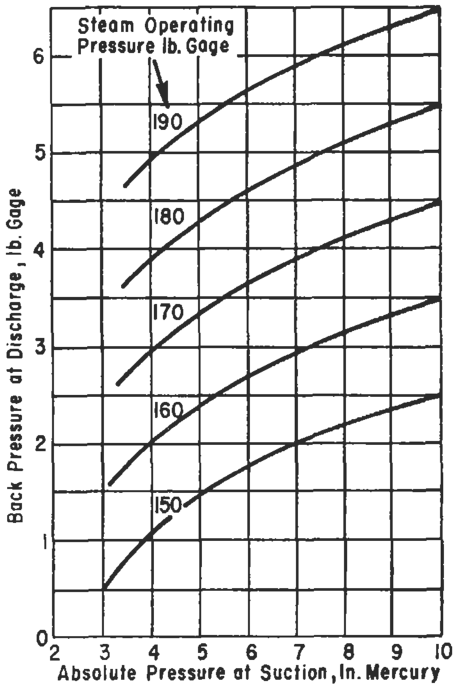


Figure 6-16. Effect of high back-pressure on ejector operation. By permission, C. H. Wheeler Mfg. Co.

lent. Figures 6-17 and 6-18 are used to handle the evaluation on an equivalent air basis. If actual performance curves are available for the temperature and vapor mixture in question, conversion to an equivalent basis is not necessary except for test purposes.

Example 6-2: 70°F Air Equivalent for Air-Water Vapor Mixture

What is the 70°F air equivalent for 500 pounds per hour of a mixture containing 150 pounds per hour of air and 350 pounds per hour of water vapor if it is at 350°F.

From Figure 6-17, the entrainment ratio for air is 0.93 lbs. air at 70°F/lbs air at 350°F.

For air: 70°F air equivalent = 150/0.93 = 161.3 lbs/hr

For steam: From Figure 6-17, entrainment ratio = 0.908 lbs steam at 70°F/lbs steam at 350°F

70°F steam equivalent = 350/0.908 = 385 lbs/hr

From Figure 6-18 at molecular weight = 18, ratio = 0.81

The 70°F air equivalent of the steam equivalent = 385/0.81 = 476 lbs/hr

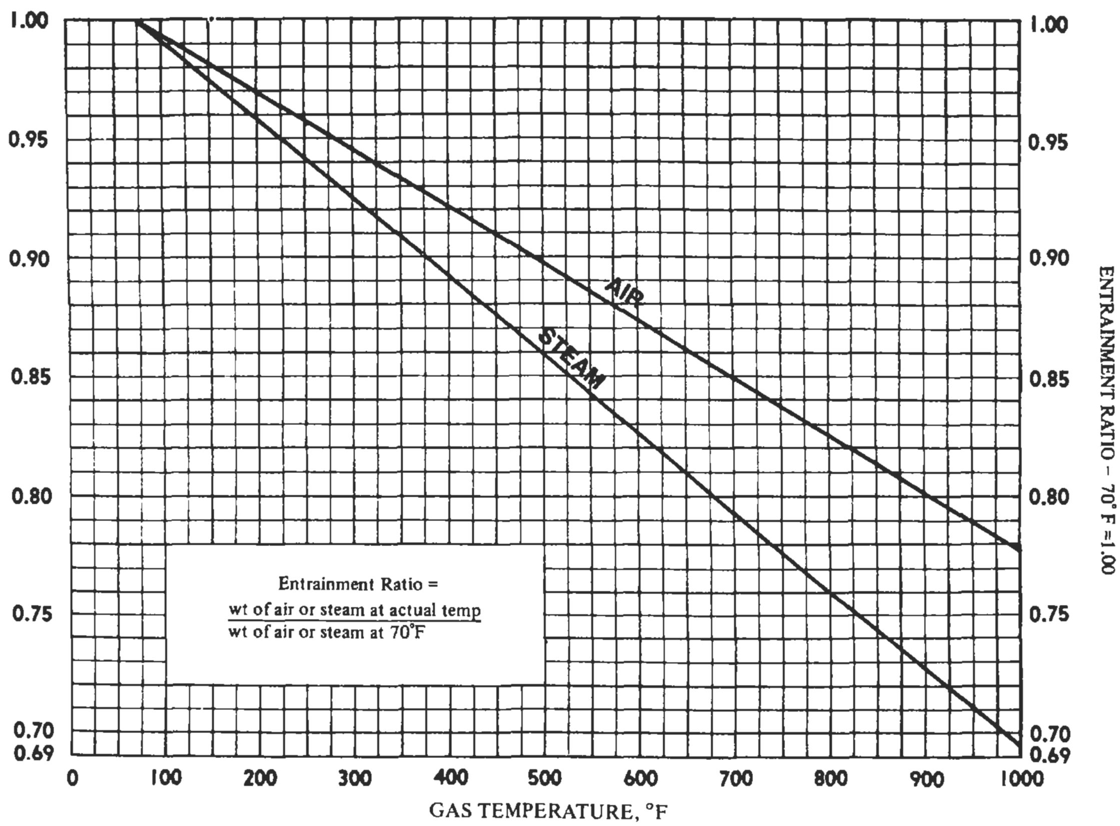


Figure 6-17. Temperature entrainment ratio curve. Reprinted by permission, *Standards for Steam Jet Vacuum Systems* 4th Ed., Heat Exchange Institute, Inc., 1988.

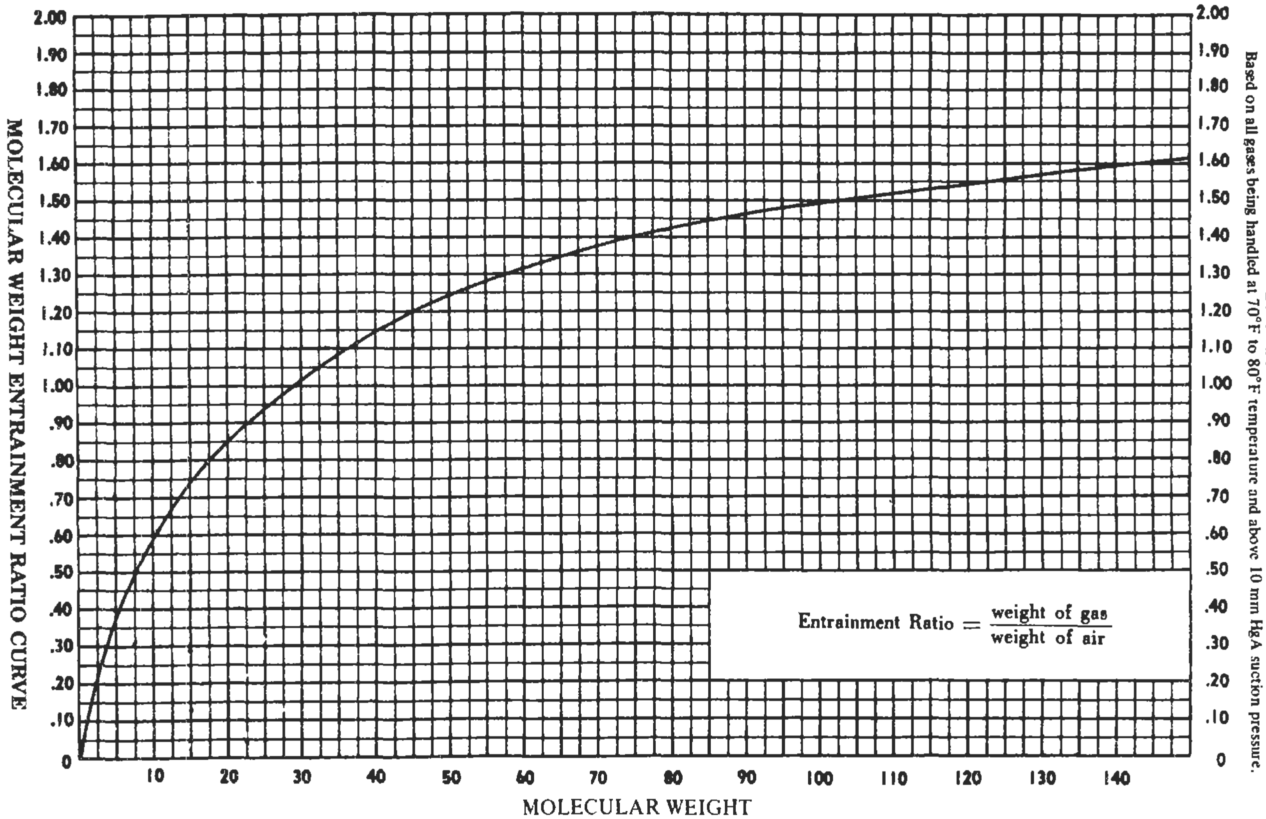


Figure 6-18. Molecular weight entrainment ratio curve. Reprinted by permission, *Standards for Steam Jet Ejectors,* 3rd Ed., Heat Exchange Institute, 1956.

Mixture: The mixture 70°F air equivalent = 161.3 + 476 = 637.3 lbs/hr

Example No. 6-3: Actual Air Capacity For Air-Water Vapor Mixture

Some manufacturers furnish 70°F air equivalent curves to allow the purchaser to convert performance to actual plant conditions. It is almost impossible to operate on one design point.

What is the expected actual capacity at 500°F of a mixture rated at 70°F?

Air	= 325 lbs/hr	
Water vapor	= 300 lbs/hr	
Total	525 lbs/hr	

For air: From Figure 6-17, ratio = 0.897 actual air capacity = 225 (0.897) = 202 lbs/hr

For water vapor: From Figure 6-17, ratio = 0.859 actual water vapor capacity = 300 (0.859) = 257 lbs/hr

Total mixture actual capacity = 202 + 257 = 459 lbs/hr

Note: if the data for this example had been given as *air equivalent,* then the water vapor portion would have been corrected for molecular weight, using Figure 6-18.

If there is only one of the components alone in the system its corrections are made as illustrated in the preceding examples.

Steam and Air Mixture Temperature

For a mixture of steam and air handled by an ejector, the temperature of the mixture in the ejector mixing chamber is calculated by [11]:

$$t_m = \frac{W_s C_{ps} t_s + W_a C_{pa} t_a}{W_s C_{ps} + W_a C_{pa}} \qquad (6\text{-}1)$$

t_m = temp of mixture at ejector suction, °F
W_s = steam flow rate, lbs/hr
C_{ps} = specific heat of steam at constant pressure corresponding to downstream absolute pressure (0.45 approx.)
t_s = temp of steam on downstream side of nozzle, °F
W_a = Air flowrate, lbs/hr
C_{ps} = specific heat of air at constant pressure (0.24 approx.)
t_a = ambient air temp °F

Example 6-4: Steam Air Mixture Temperature in Ejector

Steam used to draw air out of a vessel is:
240 psig (255 psia) @ 440°F total temp
This is: 440 − 402 sat temp = 38 superheat
Temp of air from vessel: 75°F
Steam flow: 475 lbs/hr
Air flow: 175 lbs/hr
Ejector suction pressure: 1.5 in Hg abs
Then: Enthalpy of steam = 1226.5 Btu/lb (from superheat vapor tables @ 255 psia)

Corresponding steam temperature at 1.5 in. Hg abs and enthalpy of 1226.5 = 366°F (interpolation on superheated steam tables at 1.5 in. Hg abs)

$$t_m = \frac{(475)\ (0.45)\ (366) + (175)\ (0.24)\ (.75)}{(475)\ (0.45) + (175)\ (0.24)}$$
$$= 318° F$$

Reference [11] provides a complete procedure for testing ejector units in vacuum service, and the charts and calculation procedures for the tests.

Another design approach for calculating saturated gas loads for vacuum systems is given in Reference [28].

Total Weight of a Saturated Mixture of Two Vapors: One Being Condensable

Often when the non-condensable quantity is known or estimated, it is important to state whether these gases are in the presence of water or other process liquid. In this case, the amount of condensable vapor above the liquid must be considered as it also will enter the ejector suction.

$$W_v = \frac{W_n M_v P_v}{M_n P_n} \qquad (6\text{-}2)$$

where
n refers to the non-condensable component
v refers to the condensable vapor.

Non-Condensables Plus Process Vapor Mixture

Many process systems fall in this group. They are handled in a similar manner to the other systems, correcting for temperature and molecular weight.

1. Calculate the average molecular of the mixture.
2. The air equivalent is determined from Figure 6-18 using the average molecular weight.

$$\text{Air Equivalent} = \frac{\text{lbs/hr of mixture}}{\text{Ratio, from Fig. 6-18 (uncorrected for temperature)}}$$

3. The 70°F air equivalent correcting for temperature is found as previously described, using the air curve of Figure 6-17.

Example 6-5: Actual Capacity For Process Vapor Plus Non-Condensable

A distillation system is to operate with a horizontal overhead condenser, Figure 6-19, and pressures are as marked. The estimated air leakage into the system is 7 lbs/hr. The molecular weight of the product vapor going out the condenser into the ejector (at 80°F) is 53. The vapor pressure of the condensing vapors is 3 mm Hg abs at 80°F.

Partial pressure air = 5 − 3 = 2 mm Hg (See Figure 6-19)

Vapor required to saturate at 80°F and 5 mm abs total pressure.

$$W_v = \frac{W_n M_v P_v}{M_n P_n} = \frac{7\ (53)\ (3)}{(29)\ (2)} = 19.2 \text{ lbs / hr} \qquad (6\text{-}2)$$

Average molecular weight of mixture:

Air	7 lbs/hr	= 0.241 mols/hr
Process vapor	19.2	= 0.362 mols/hr
Total vapor	26.2 lbs/hr	= 0.603 mols/hr

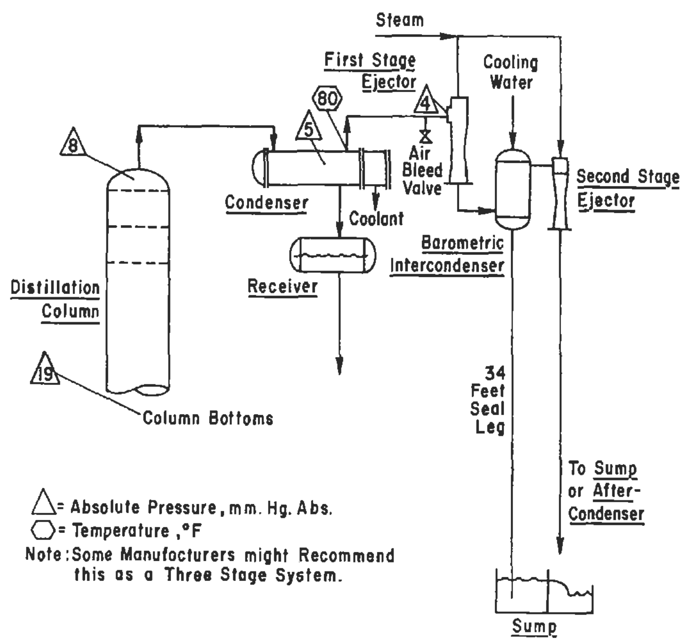


Figure 6-19. Vacuum system for distillation.

Mixture avg mol wt = 26.2/0.603 =43.4
Molecular weight correction (From Figure 6-18) = 1.18

$$\text{Air equivalent (at } 80°\text{F)} = \frac{26.2}{1.18} = 22.2 \text{ lbs / hr}$$

Temperature correction (Figure 6-17, using air curve) = 0.999
70°F air equivalent for mixture = 22.2/0.999 = 22.2 lbs/hr

This is the value to compare with a standard manufacturer's test or performance curve at 70°F.

The air bleed is used to maintain a constant condition. However, a control valve may be used instead. Control or hand valves in the lower pressure vapor lines to an ejector are not recommended, as they must be paid for in system pressure drop and ejector utility requirements.

Non-Condensables Plus Water Vapor Mixture

This is also a frequent process situation. To determine the 70°F air equivalent, the non-condensable are determined as in Example 6-5 and the water vapor as in Example 6-2. The total for the mixture is the sum of these two values.

Total Pressure of System at Suction to Ejector

$$P = P_n + P'_{v1} + P'_{v2} + \ldots \qquad (6\text{-}3)$$

Air-Water Vapor Mixture Percent Curves

For saturated air-water vapor systems, Figures 6-20A, B, C, and D are useful in solving for the pounds of water vapor per pound of air (Dalton's Law, Equation 6-2).

Example No. 6-6: Use of Water Vapor-Air Mixture Curves

A system handles 50 lbs/hr of air that is saturated with water vapor at 3 inch Hg abs and 95°F. Find the total amount of water vapor.

For 3 inch Hg abs and 95°F saturation, the fraction of water vapor from Figure 6-20 is 0.77 lbs water vapor/lb air.

Total water vapor = (50) (0.77) = 38.5 lbs/hr
Total mixture = 50 + 38.5 = 88.5 lbs/hr

Weight of Air and Water Vapor Mixture

$$W_m = W_a + \frac{0.62\,(W_a)(P_v)}{P_a} \qquad (6\text{-}4)$$

Example 6-7: Total Weight of Mixture

Calculate the total weight of mixture to be handled when evacuating 25 pounds per hour of air from 28.5 inches vacuum with a mixture temperature at 80°F. Barometer = 30 inches Hg.

Absolute pressure = 30.0 − 28.5 = 1.5 in. Hg abs
P_v = 1.034 in. Hg abs (water vapor at 80°F)
P_a = 1.50 − 1.034 = 0.466 in. Hg abs (air)
W_m = 25 + 0.62(25) (1.034)/0.466 = 59.4 lbs mixture/hr

Total Volume of a Mixture

The total fixed volume of a mixture of gases and vapors at a given condition is the same as the volume of any one component (gas laws), and its pressure is composed of the sum of the individual partial pressures of each component.

Example 6-8: Saturated Water Vapor-Air Mixture

An air-water vapor mixture is saturated with water vapor at 80°F and 2 in. Hg abs total pressure. The air in the mixture is 60 lbs/hr. Determine the volume at these conditions.

Vapor pressure water at 80°F = 1.034 in. Hg abs
Partial pressure air = 2.0 − 1.034 = 0.966 in. Hg abs
Weight of mixture:
W_m = 60 + 0.62(60) (1.034)/0.966 = 99.8 lbs/hr
Weight of water vapor = 39.8 lbs/hr (from above)
Volume of air under its condition in the mixture:

$$\text{Vol of air} = \text{vol of mixture} = V = \frac{WRT}{70.73\,P_a}, \text{ use in. Hg} \qquad (6\text{-}5)$$

$$\text{Gas constant, } R = 1544/MW = 1544/29 = 53.3 \qquad (6\text{-}6)$$

$$V = \frac{60(53.3)\,(460 + 80)}{70.73\,(0.966)} = 25{,}300 \text{ cu ft/hr}$$

= 422 cu ft/min at 80°F

As a check:
From steam tables, specific volume of water vapor at 80°F and 1.034 in. Hg abs = 633.8 cu ft/lb

Volume = (39.8) (633.8) = 25,200 cu ft/hr
= 421 cu ft/min

As an alternate method, total mols could be calculated and converted to volume at 80°F and 2 mm Hg abs.

(text continued on page 366)

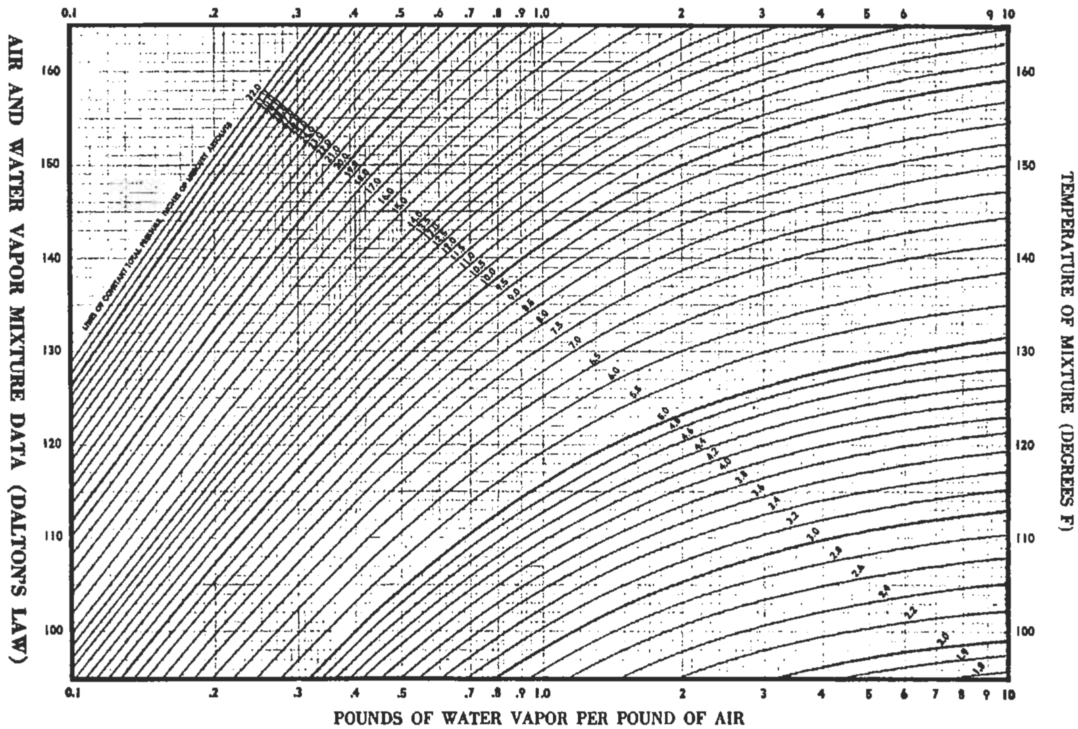


Figure 6-20A. Air and water vapor mixture data (Dalton's Law)—saturated. Reprinted by permission, *Standards for Steam Jet Ejectors,* 3rd. Ed., Heat Exchange Institute, 1956 [11].

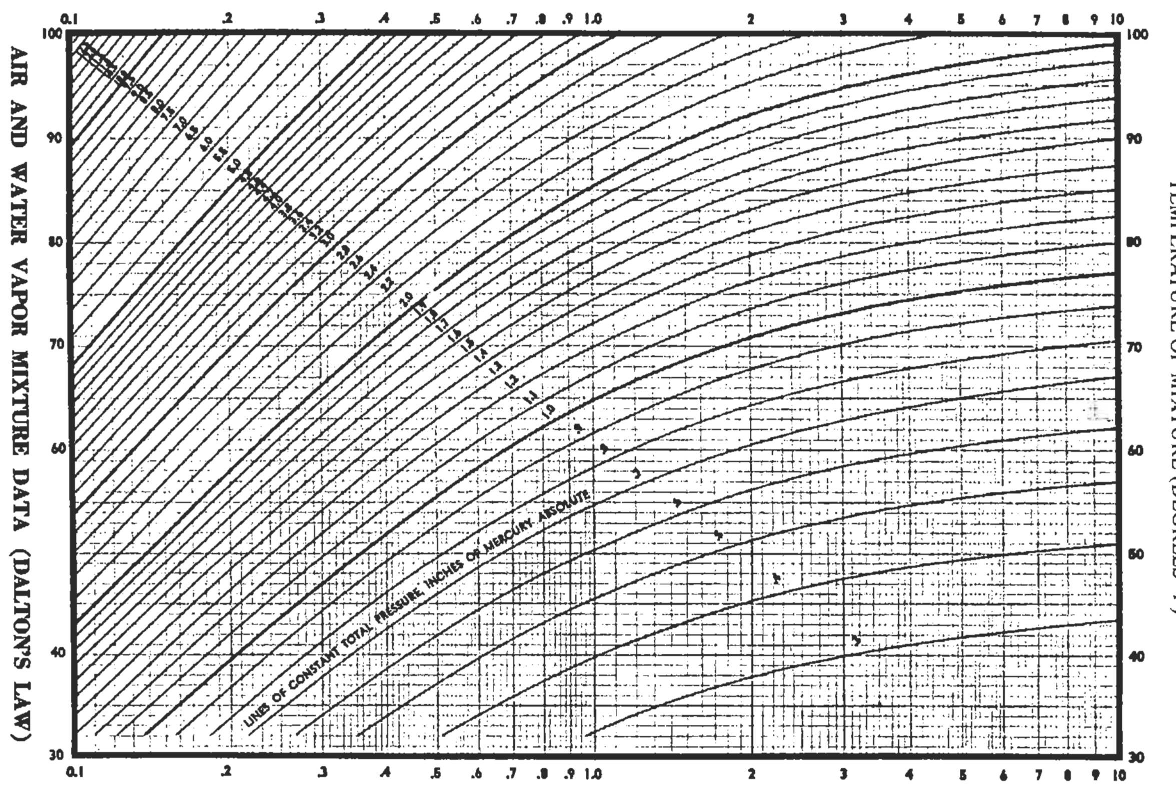


Figure 6-20B. Air and water vapor mixture data (Dalton's Law)—saturated, (continued). Reprinted by permission, *Standards for Steam Jet Ejectors,* 3rd. Ed., Heat Exchange Institute, 1956 [11].

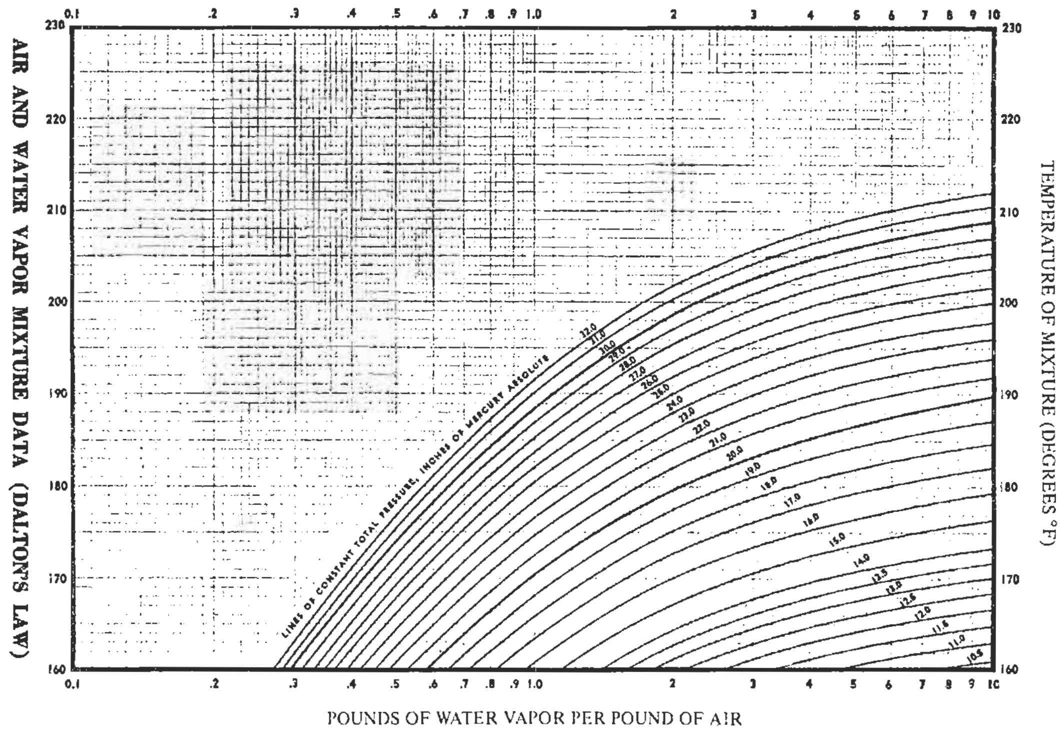


Figure 6-20C. Air and water vapor mixture data (Dalton's Law)—saturated, (continued). Reprinted by permission, *Standards for Ejectors,* 3rd. Ed., Heat Exchange Institute, 1956 [11].

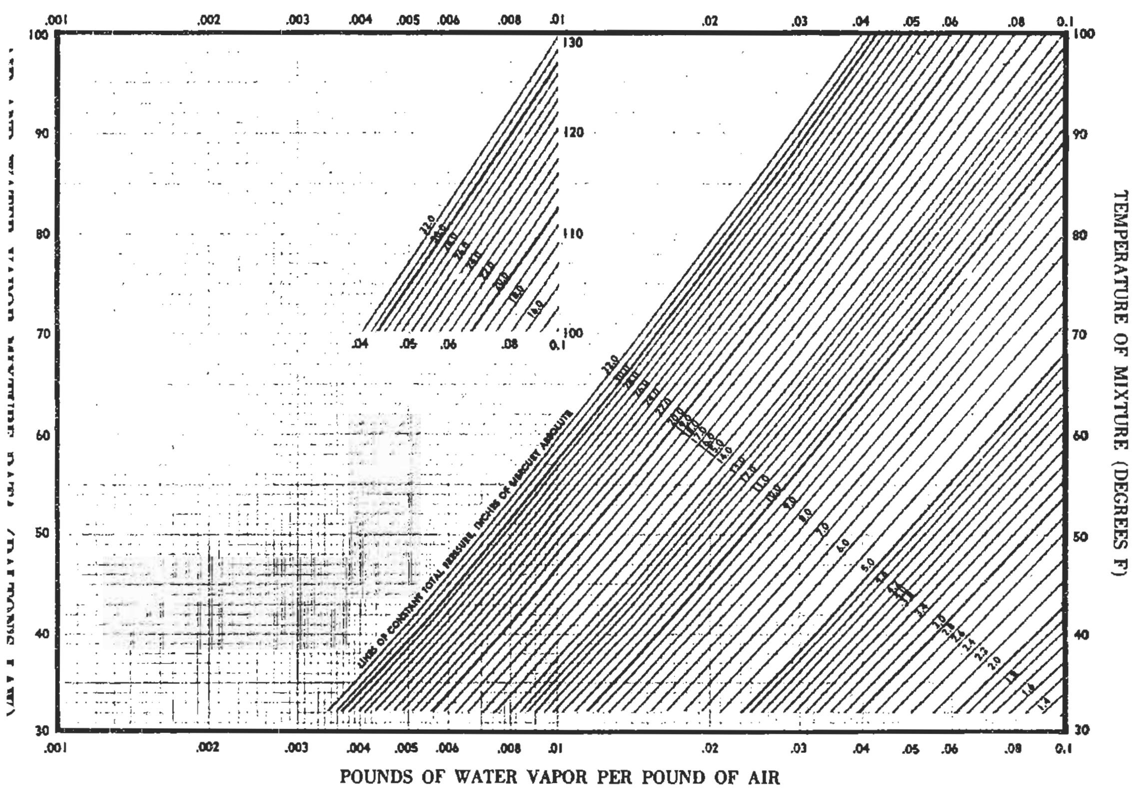


Figure 6-20D. Air and water vapor mixture data (Dalton's Law)—saturated, (continued). Reprinted by permission, *Standards for Steam Jet Ejectors,* 3rd. Ed., Heat Exchange Institute, 1956 [11].

(text continued from page 363)

Air Inleakage into System

Few vacuum systems are completely airtight, although some may have extremely low leakage rates. For the ideal system the only load for the ejector is the non-condensables of the process (absorbed gases, air, etc.) plus the saturated vapor pressure equivalent of the process fluid. Practice has proven that allowance must be made for air leakage. Considering the air and non-condensables. For "base" ejector capacity determine inert gases only by:

(Pounds/hr air + non-condensables + process released air + process released non-condensables)/Hr ≅ Air in leakage, lb/hr (6-7)

For design of a new system, it is recommended that the results of the summation above be multiplied by 2 or 3 to establish the jet system inert (noncondensables) capacity, and add to this the non-condensed process vapors that are released into the jet suction system.

Air leakage occurs at piping connections (flanges, screwed fittings, valves), stuffing boxes, mechanical equipment seals, etc. Whenever possible a system should be tested to determine the air leakage [12, 14], but for new designs and those situations where tests cannot be made, the recommended values of the Heat Exchange Institute are given in Table 6-6 for ejectors serving surface condensers, and are minimum safe values. A very tight system will show better performance.

Figure 6-21 gives maximum air leakage values for commercially tight process systems which do not include any agitator equipment. For design purposes the ejector is usually purchased to operate on a load at about twice these values. For systems with agitators and ordinary shaft seals, the system leakage should be increased by 5 pounds of air per hour [12] per agitator. If special seals are used this value may be reduced to 1 or 2. The more rotating shafts which must be sealed to the outside atmosphere, the more likely will be the possibilities of increased leakage.

An alternate design for air inleakage used by some manufacturers and process engineers is Figure 6-21 plus the summation obtained by examining the process system using the factors of Table 6-7. This method is considered to be conservative, however, as in general the incremental cost may be very small between a unit barely large enough and one which has ample capacity to take surges in air leakage.

Since the determination of air inleakage involves considerable knowledge of vacuum systems and judgment, no empirical method can be expected to yield exact and correct values. Most manufacturers use one of the methods presented here, together with a factor to account for the basic type of plant, maintenance practices, operational techniques of the production personnel, and other related items. Thus, for a tight and efficient plant, the leakage values of Figure 6-21 may sometimes be reduced to 0.75 of the values read, while for a sloppy, loose-run plant the values might be multiplied by 2 or 3,

Table 6-6
Vacuum Pump Capacities From Steam Surface Condensers

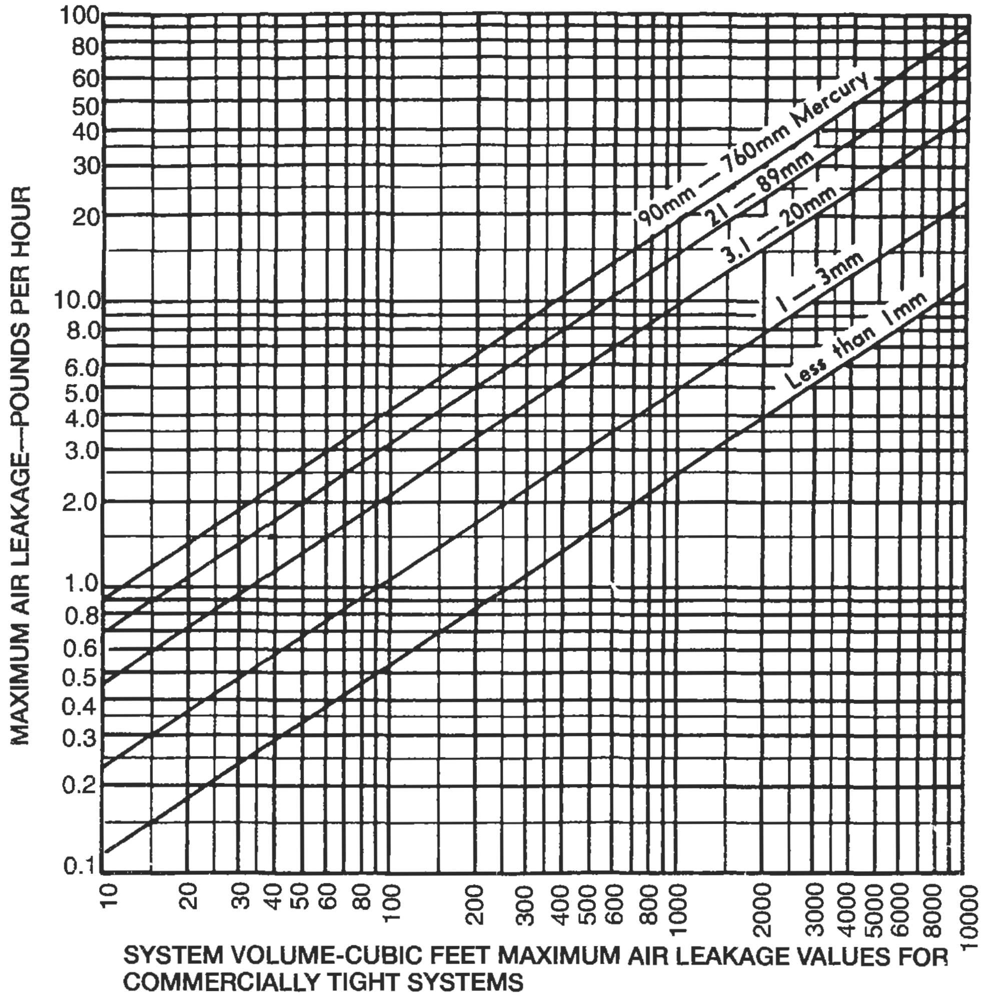
Maximum Steam Condensed, Lb. Per Hr.	DRY AIR AT 70° F.			
	Serving Turbines		Serving Engines	
	SCFM	Lbs./Hr.	SCFM	Lbs./Hr.
Up to 25,000.....	3.0	13.5	6.0	27.0
25,001 to 50,000.....	4.0	18.0	8.0	36.0
50,001 to 100,000.....	5.0	22.5	10.0	45.0
100,001 to 250,000.....	7.5	33.7	15.0	67.4
250,001 to 500,000.....	10.0	45.0	....	....
500,001 and up........	12.5	56.2	....	....
	Rapid Evacuator Capacities, Dry Air, cfm at 70° F., 15 in. Hg. abs.			
Up to 75,000.....	150			
75,000 to 250,000.....	300			
250,001 to 600,000.....	600			
600,001 and up........	900			

*Standards of Heat Exchange Institute, Steam Surface Condensers, Third Edition, Ref. (12) by permission.

Table 6-7
Estimated Air Leakage Into Equipment Vacuum System

Type Fitting	Estimated Average Air Leakage, Lbs./Hr.
Screwed connections in sizes up to 2 inches........	0.1
Screwed connections in sizes above 2 inches.......	0.2
Flanged connections in sizes up to 6 inches........	0.5
Flanged connections in sizes 6 inches to 24 inches including manholes..................	0.8
Flanged connections in sizes 24 inches to 6 feet....	1.1
Flanged connections in sizes above 6 feet..........	2.0
Packed valves up to ½" stem diameter..........	0.5
Packed valves above ½" stem diameter...........	1.0
Lubricated plug valves.........................	0.1
Petcocks	0.2
Sight glasses	1.0
Gage glasses including gage cocks................	2.0
Liquid sealed stuffing box for shaft of agitators, pumps, etc., per inch shaft diameter...........	0.3
Ordinary stuffing box, per inch of diameter........	1.5
Safety valves and vacuum breakers, per inch of nominal size.............................	1.0

* From D. H. Jackson, Selection and Use of Ejectors, *Chem. Eng. Prog.* **44**, 347 (1948)



Notes:

1.) The air leakage rates indicated are guidelines only. In actual field practice, the air leakage may be significantly larger or smaller depending upon the condition of sealing, maintenance factors, and application.

2.) For comparison purposes, alternate methods of air leakage estimation may be used.

Figure 6-21. Maximum air leakage values for commercially tight systems. Reprinted by permission [11].

or the alternate method using Table 6-7 may be checked, or even multiplied by 2 or 3.

Example 6-9: Ejector Load For Steam Surface Condenser

A surface condenser condensing the steam from a process turbine drive operates at 1.0 in. Hg abs. The condensing load is 85,000 lbs/hr steam. What is the capacity of the ejector?

The temperature of the condensing steam at 1.0 in. Hg is 79°F (from steam-tables).

The saturation pressure corresponds to a temperature of 79° − 7.5°F = 71.5°F, based on condenser-ejector design practice. The pressure from steam tables = 0.78 in. Hg abs.

The water vapor to saturate the air going to the ejector is:

$$W_v' = 0.62 \frac{P_v}{P_a} = (0.62)\ (0.78)/(1 - 0.78) \qquad (6\text{-}8)$$

$$= 2.19 \text{ lbs water vapor/lb air}$$

From Table 6-6 the recommended dry air SCFM = 5.0 The equivalent

lbs/hr = (5.0) (60) (0.075 lb/cu ft) = 22.5

Total water vapor = (2.19) (22.5) = 49.4 lbs/hr

Total vapor mixture to ejector = 49.4 + 22.5 = 71.9 lbs/hr

For ejector design a value of 1.25 times this value is recommended.

Another recommendation based upon years of experience is reasonably conservative [14]:

Suction Pressure In. Hg Abs	Allowance for Air Leakage, Lbs/Hr
8 to 15	30 to 40
5 to 8	25 to 30
3 to 5	20 to 25
1 to 3	10 to 20

For systems with moving sealed parts, make extra allowance and consult the seal manufacturer.

Air leakage into systems operating at or below 0.53 atmosphere, or 15 in. Hg abs, is constant and approximately independent of the process itself. From atmospheric down to 15 in. Hg, the air leakage increases as the pressure decreases.

Dissolved Gases Released From Water

When ejectors pull non-condensables and other vapors from a direct contact water condenser (barometric, low level jet, deaerator) there is also a release of dissolved gases, usually air, from water. This air must be added to the other known load of the ejector. Figure 6-22 presents the data of the Heat Exchange Institute [10] for the amount of air that can be expected to be released when cooling water is sprayed or otherwise injected into open type barometric or similar equipment.

Alternate, for additional air inleakage calculations are presented by Reference [26] based on industrial studies

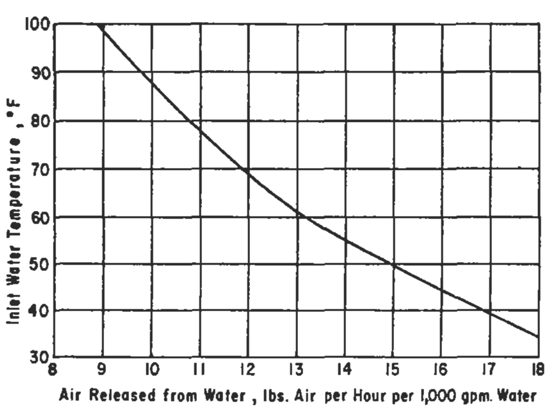


Figure 6-22. Dissolved air released from water on direct contact in vacuum systems. Reprinted by permission, *Standards for Direct Contact Barometric and Low Level Condensers*, 4th Ed., Heat Exchange Institute, 1957.

and experience. The objective is to be able to develop reliable specifications for "rough vacuum" equipment. The procedure is [22]:

1. Estimate air leakage into the system based on possible weld cracks, metal weld, or metal porosity:

$$1 \leq P < 10 \text{ torr}; W'_a = 0.026\, P^{0.34}\, V^{0.60} \quad (6\text{-}9)$$

$$10 \leq P < 100 \text{ torr}; W'_a = 0.032\, P^{0.26}\, V^{0.60} \quad (6\text{-}10)$$

$$100 \leq P < 760 \text{ torr}; W'_a = 0.106\, V^{0.60} \quad (6\text{-}11)$$

where P = system operating pressure, torr
V = system volume, cu ft
W'_a = air inleakage resulting from metal porosity and cracks along weld lines, lb/hr

2. Estimate acceptable air leakages resulting from leakage around static and rotary seals, valves, access ports, and other items of mechanical nature required for process operation from the following equations and the specific leak rates θ indicated in Table 6-8, which is based on $w_a \leq 10$ lb/hr.

Table 6-8
Specific Air Inleakage Rates for Rough Vacuum, for Use with Equations

Component	θ = specific leak rate*, lb/h/in.
Static seals	
O-ring construction	0.002
Conventional gasket seals	0.005
Thermally cycled static seals	
t X 200°F	0.005
200 X t < 400°F	0.018
t x 400°F	0.032
Motion (rotary) seals	
O-ring construction	0.10
Mechanical seals	0.10
Conventional packing	0.25
Threaded connections	0.015
Access ports	0.020
Viewing windows	0.015
Valves used to isolate system	
Ball	0.02
Gate	0.04
Globe	0.02
Plug-cock	0.01
Valves used to throttle control gas into vacuum system	0.25

*Assumes sonic (or critical) flow across the component.
By permission: Ryans and Croll [22].

$$1 \le P < 10 \text{ torr}; w_a = \pi D\theta P^{0.34} \quad (6\text{-}12)$$

$$10 \le P < 100 \text{ torr}; w_a = 1.2\, \pi D\theta P^{0.26} \quad (6\text{-}13)$$

$$100 \le P < 760 \text{ torr}; w_a = 3.98\, \pi D\theta \quad (6\text{-}14)$$

where D = sealed diameter, in. (estimates of nominal diameter, acceptable)
w_a = acceptable air-leakage rate assigned to a system component, lb/hr
θ = specific air leakage rate, lb/hr/in.
P = system operating pressure, torr
$W_{Ta} = W_T$ = total calculated air inleakage, lb/hr

3. Calculate the total acceptable air leakage rate, W_T, lb/hr by adding $\Sigma W'_a$ to the sum of the leak rates assigned to the individual system components, w_a

$$W_T = \Sigma W'_a + \Sigma w_a, \text{ lb/hr} \quad (6\text{-}15)$$

To determine the capacity of the vacuum pump, the values of W_T above should not be used. It is necessary to apply over-design or safety factors to ensure reliability [26] because pump capacity decreases with time and wear and air inleakage surges can occur due to a wide variety of leak developing situations that result in more air or surges of air inleakage. The over-design factor should be applied *only* to the pump inlet throughput specification. The recommended [22] over-design factor should be 1.5 to 2.0 times the air inleakage rate [22], and should also be applied to saturated vapors entering the vacuum equipment at the suction conditions required. Do not put a safety factor on the suction condition of temperature or pressure, provided the worst expected conditions of operation are specified. This requires a close examination of the process flow sheet range of operation. A safety factor of 2.0 is recommended for multistage steam jets with compression ratio above 6:1; while a 1.5 factor is adequate for most mechanical pumps and single-stage jets with a compression ratio of under 4:1 [22]. The above procedure can be simplified for preliminary inleakage calculations [22]:

Multiply W'a × 2

For equipment with rotary seals allow additional 5 lb/hr for each conventional seal (packing type) and 2 lb/hr for each mechanical seal and O-ring.

To account for air inleak a for vessels containing a liquid level (portion of vessel submerged), the following applies.

If there is a large pinhole leak a few inches below the liquid surface, it will behave like a leak above the liquid. However, a small pinhole leak in the same location may have zero inleakage due to capillary effects. The problem becomes complicated as the depth becomes large, and flow transition occurs from critical (sonic) to subsonic flow through any particular leak. These factors can be calculated by conventional fluid flow methods if the size of the leak is estimated, which then is a real problem, or by using the data of Table 6-8 and applying fluid flow head losses.

The hydraulic effects for the submerged portions of the vessel or system can be ignored when the total static head on the submerged portion is not greater than 0.53 times atmospheric pressure plus the hydraulic head [22], i.e., P_s:

$$P_s = (p/760) + (h_L\rho_L/34) \quad (6\text{-}16)$$

where P_s = static pressure, atm
p = atmospheric pressure, mm Hg
h_L = liquid height, ft, below liquid surface
ρ_L = specific gravity of liquid, relative to water = 1.0

Acceptable Air Inleakage Rates [24]

In order to estimate an acceptable air inleakage rate for sizing a vacuum pump for use in the medium to high vacuum system, consider:

100 Microns (0.10 Torr) to 1.0 Torr Range

Estimate W'_a:

$$0.1 \le P < 1 \text{ torr}; W'_a = 0.026P^{0.64}\, V^{0.60} \quad (6\text{-}17)$$

Estimate air inleakage for individual system specific leak rates, θ, from Table 6-8 [22] and from ($w \le 5$ lb/hr)

$$0.1 \le P < 1 \text{ torr}: w_a = \pi\, D\theta P^{0.64} \quad (6\text{-}18)$$

Note that estimating maximum acceptable differs from the design equations for w_a and W'_a.

Calculate total acceptable air inleakage rate, W_T.

$$W_T = \Sigma W'_a + \Sigma w_a \quad (6\text{-}19)$$

A simplified alternate to the previously cited procedures is suggested by Gomez [29] for calculating air inleakage, but it is not presented in detail here.

Total Capacity at Ejector Suction

The total capacity is the sum of all the expected condensables and non-condensable flow quantities (in pounds per hour) which will enter the suction inlet of the ejector. It consists of the following

1. Air leakage from surrounding atmosphere.
2. Non-condensable gases released from gases originally injected into the process for purge, products of reaction, etc.
3. Non-condensable gases, usually air, released from direct contact water injection.
4. Condensable vapors saturating the non-condensables.

Reasonable factors of safety should be applied to the various loads in order to insure adequate capacity. Excess ejector capacity can be handled by pressure control and some adjustment in steam flow and pressure, but insufficient capacity may require ejector replacement. Factors of 2.0 to 3.0 are not uncommon, depending upon the particular type of system and knowledge of similar system operations.

Capacities of Ejector in Multistage System

When the ejector system consists of one or more ejectors and intercondensers in series, the volume as pounds per hour of mixture to each succeeding stage must be evaluated at conditions existing at its suction. Thus, the second stage unit after a first stage barometric intercondenser, handles all of the non-condensables of the system plus the released air from the water injected into the intercondenser, plus any condensable vapors not condensed in the condenser at its temperature and pressure. Normally the condensable material will be removed at this point. If the intercondenser is a surface unit, there will not be any air released to the system from the cooling water.

Booster Ejector

Booster ejectors are designed to handle large volumes of condensable vapors at vacuums higher than that obtainable with standard condensers using cooling water at the maximum available temperature. They are usually used with a barometric (or surface) condenser and standard two-stage ejectors. The booster picks up vapors from the process system at high vacuum (low absolute pressure, around 0.5 in. Hg abs) and discharges them together with its own motivating steam to a lower vacuum condition (compresses the mixture) where the condensable vapors can be removed at the temperature of the condenser water. The non-condensable vapors leave the condenser, passing to the two-stage ejector system. This overall system allows a constant vacuum to be maintained in the process, unaffected by the temperature of the cooling water. Booster ejectors are used with barometric and surface steam jet refrigeration systems, degassing of liquids, high vacuum distillation, evaporation, vacuum cooling and vacuum drying, or other systems where large volumes of condensable materials are to be removed at high vacuum. Figure 6-23 illustrates one application.

Evacuation Ejector

An evacuation booster or "hogging" ejector is sometimes used to remove air from a system on start-ups. Its capacity is set to bring the system pressure down to near operating conditions before the continuous operating ejector system takes over. Figure 6-23 illustrates the installation of such a unit.

When an extra jet for this purpose is not desirable, the secondary jet of a multiple system is often sized to have sufficient air removal capacity to pump down the system in a reasonable time.

Load Variation

Figure 6-24 illustrates three different multistage ejector designs, A, B, and C, which indicate that design A is quite sensitive to changes in load above the design point. Designs B or C are less sensitive. The curve extended toward point D indicates the capacity of the primary or first stage when all the vapor is condensed in the intercondenser; or if handling air or an air-vapor mixture, the performance when the secondary jets have sufficient capacity to take all the non-condensables.

The curve labeled A indicates performance at overload when the air-handling capacity of the secondary stage is limited. This condition arises as a result of design for steam economy. If the capacity of the secondary jets is larger, the performance along curve B or C can be expected. When the secondary jet capacity is limited as curves A, B, or C indicate, a capacity increase brings a rise in suction pressure when the load increase is mainly air or non-condensables. The increase in pressure is less when the load increase is due to condensables. This emphasizes the importance in sizing the secondary jets for ample non-condensable capacity, and the importance of specifying the range and variety of expected conditions which may confront the system.

Once a system has been evacuated to normal operating conditions, it is possible for capacity to fall to almost zero when the only requirement is air inleakage or small quantities of dissolved gases. Under these conditions, it is important to specify an ejector system capable of stable operation down to zero load or "shut-off" capacity. The curve of Figure 6-24 represents such a system.

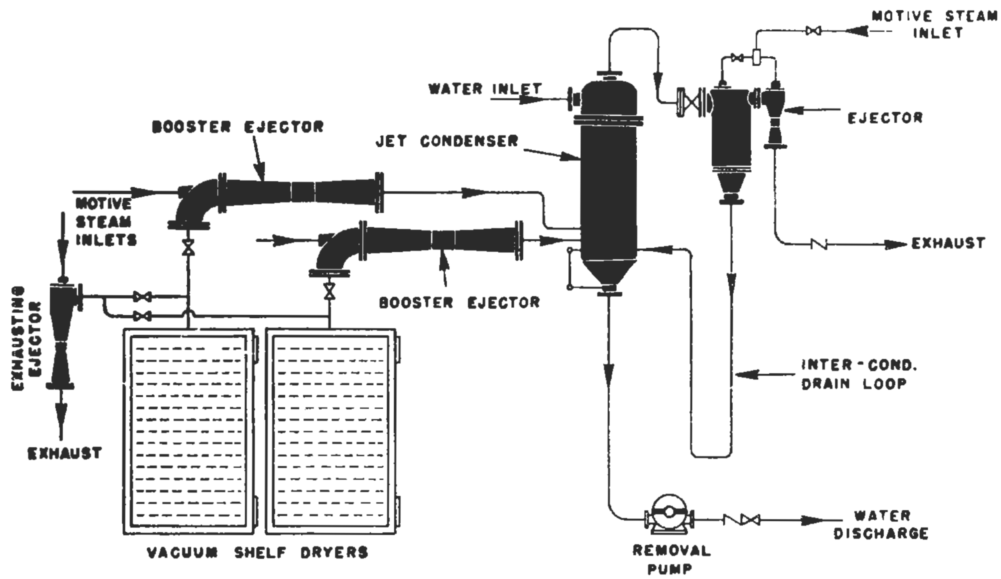


Figure 6-23. Drying-special three-stage assembly serving two high vacuum dryers. By permission, C. H. Wheeler Mfg. Co.

Steam and Water Requirements

Figure 6-25 presents estimated steam requirements for several ejector systems. Exact requirements can be obtained only from the manufacturers, and these will be based on a specific performance.

Figures 6-26A and B give typical good estimating selection curves for single-stage ejectors. Table 6-9 gives evacuation factors.

Size selection: locate size at intersection of ejector suction pressure and capacity on Figure 6-26A.

Steam consumption: read values on curves or interpolate.

$$W_s = W_{s90}F \tag{6-20}$$

Evacuation:

$$W'_m = E\,V/t \tag{6-21}$$

Example 6-10: Size Selection. Utilities and Evacuation Time for Single Stage Ejector

Total mixture to be handled = 60 lbs/hr
Suction pressure: 4 in. Hg abs
Steam pressure: 125 psig
Size selection: Figure 6-26A: 2 inch L
Steam consumption: 440 lbs/hr at 90 psig

at 125 psig, F = 0.88 (Figure 6-26B)
W_s = 440 (0.88) = 397 lbs/hr

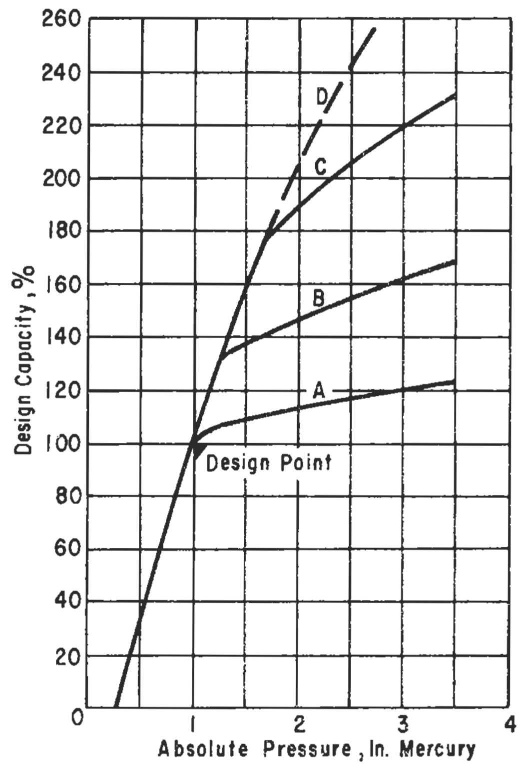


Figure 6-24. Effects of overloads on ejector operation. By permission, C. H. Wheeler Mfg. Co.

Evacuation: system volume = 300 cu ft
E = 1.3 (Table 6-9)
V = 300
W'_m = 60
60 = 1.3 (300)/t
t = 6.5 minutes to evacuate the volume with the 2-inch L ejector

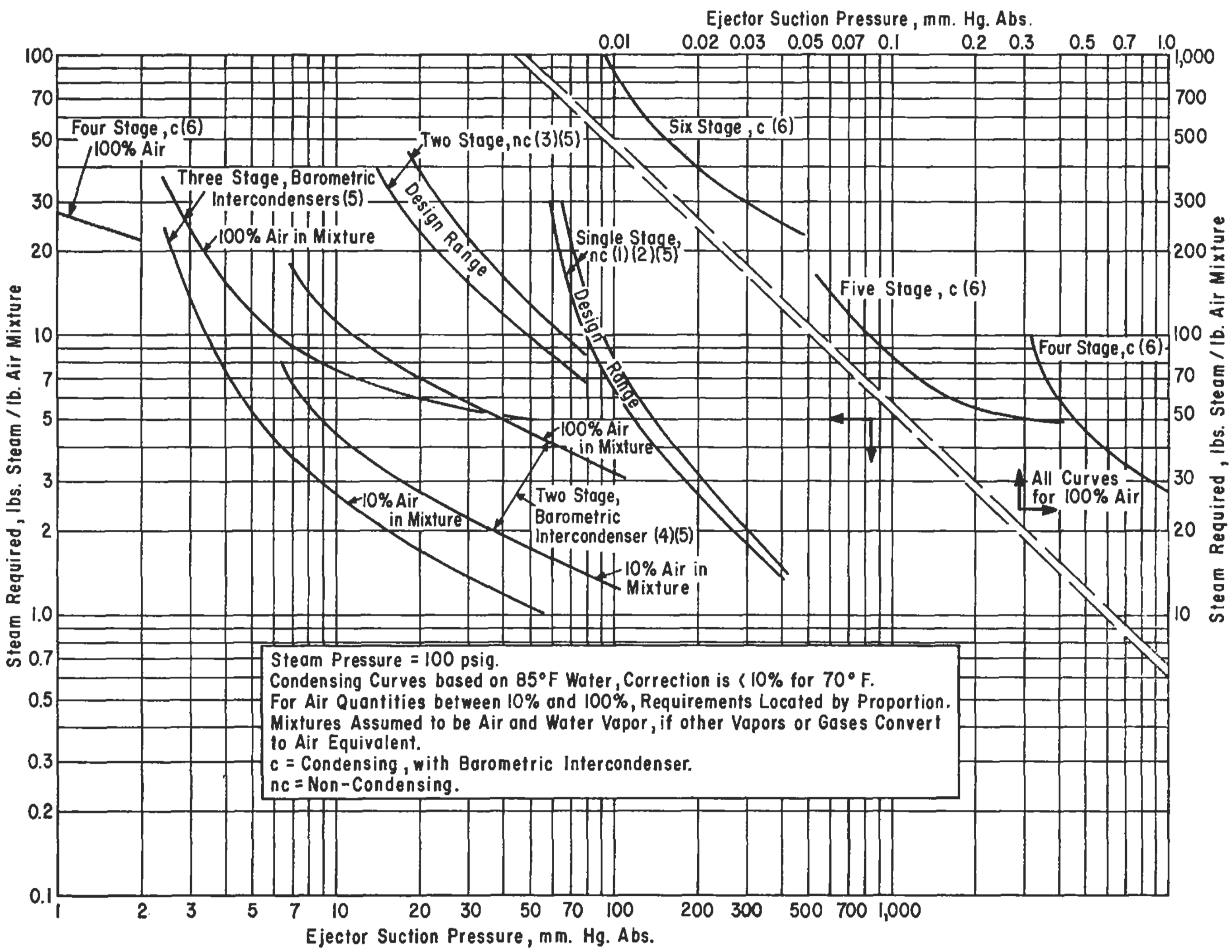


Figure 6-25. Estimating steam requirements for ejectors.

Rating of the two stage non-condensing ejectors is handled in the same manner as for a single stage ejector, using Figures 6-27 A and B, and Table 6-10.

Figures 6-28 A, B, C, D and E give representative estimating data for two stage ejectors with barometric intercondenser.

Size selection: locate size at intersection of ejector suction pressure and capacity on Figure 6-28A.

Steam consumption: Use Figure 6-28B; for K use Figure 6-28C

$$W_s = W'_s W_m K F \qquad (6\text{-}22)$$

Water consumption:

GPM (approximate) = $0.06\ W_s$
Minimum GPM = 10

Example 6-11: Size Selection and Utilities For Two-Stage Ejector With Barometric Intercondenser

Total mixture to be handled = 40 lbs/hr
Pounds of air in mixture = 14 lbs/hr
Suction pressure = 1.5 in. Hg abs
Steam pressure at ejector nozzle = 150 psig

Size selection, using Figure 6-28A: 2 inch (inlet and outlet connections)

Steam consumption:

$W'_s = 5.55$ (Figure 6-28B)
$W_m = 40$
$K = 0.67$, (Figure 6-28C) at $W_a/W_m = 15/40 = 0.375$
$F = 0.88$ (Figure 6-28D)
$W_s = 5.55\ (40)\ (0.67)\ (0.88)$
$= 131$ lbs steam/hr

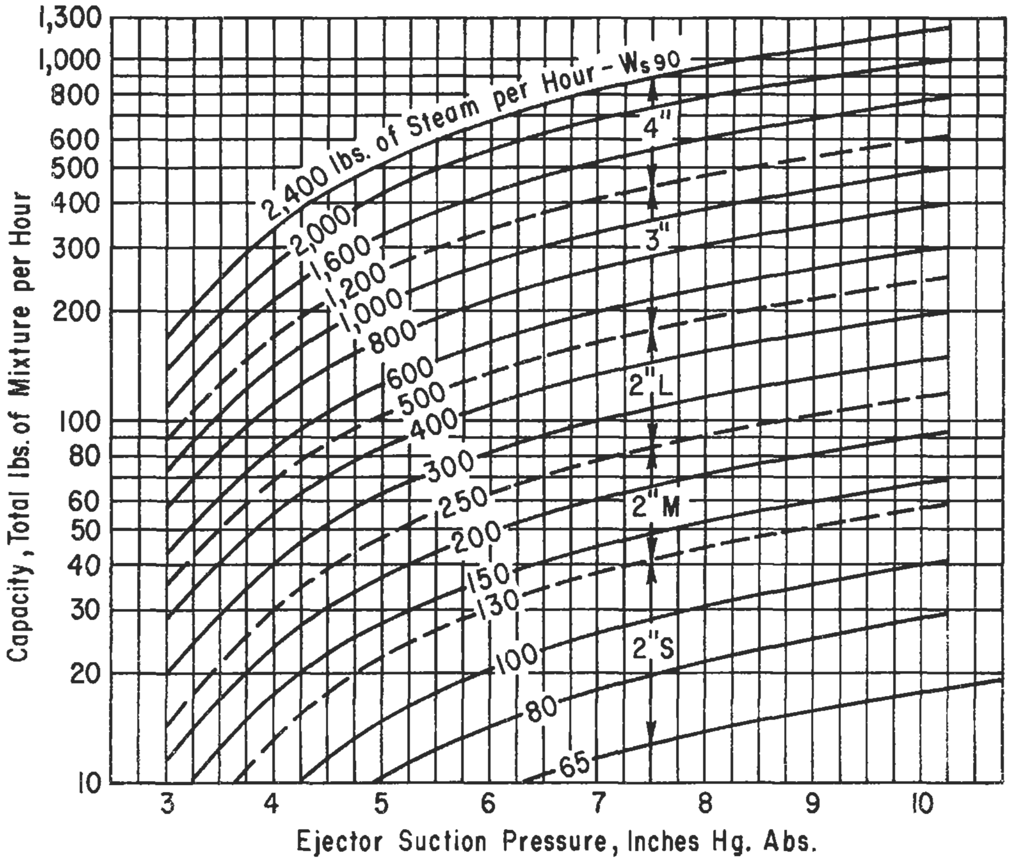


Figure 6-26A. Size ejector for 20 psig steam consumption, single stages, 3″–10″ Hg abs. By permission, Worthington Corp.

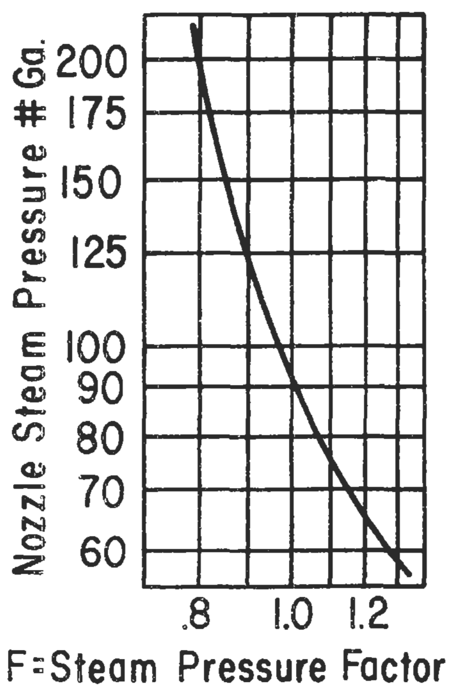


Figure 6-26B. Steam pressure factor for Figure 6-26A. By permission, Worthington Corp.

Water consumption:

GPM = 0.06 (131) = 7.85

Use 10 GPM minimum

Ejector System Specifications

Figure 6-29 is helpful in summarizing specifications to the ejector manufacturer for rating of ejectors and intercondensers to serve a specific process application. Most conditions require a detailed evaluation with the manufacturer's test curves, as very few complicated systems requiring high vacuum can be picked from stock items. The stock items often fit single-stage vacuum requirements for process and such standardized situations as pump priming.

Figure 6-29 is also adaptable to air and water ejector applications.

In all cases it is important to describe the system, its requirements, control and method of operation in the specifications. The manufacturer needs complete data concerning the motive steam (air or water) and the condensable and non-condensable vapors.

Table 6-9
Evacuation Factors for Single Stage Jet

Final Suction Pressure In. Hg. Abs.	Evacuation Factor, E
10	1.9
8	1.0
6	1.5
5	1.0
4	1.3

*By permission, Worthington Corp. Bulletin W-205-S1A.

Utility unit costs as well as any preference for maximum operating economy or minimum first cost should be stated if the manufacturer is to make a selection to best fit the plant system and economics.

Tables 6-11, 6-12, 6-13, and 6-14 provide useful reference data.

Table 6-10
Evacuation Factors For Two Stage Non-Condensing Ejectors

Final Suction Pressure In. Hg. Abs.	Evacuation Factor, E
0.5	0.48
1.0	0.67
1.5	0.81
2.0	0.92
2.5	1.00
3.0	1.10

*By permission, Worthington Corp. Bul. W-205-S10A.

Ejector Selection Procedure

As a guide, the following is a suggested procedure for rating and selecting an ejector system for vacuum operation.

1. Determine vacuum required at the critical process point in system.
2. Calculate pressure drop from this point to the process location of the suction flange of the first stage ejector.

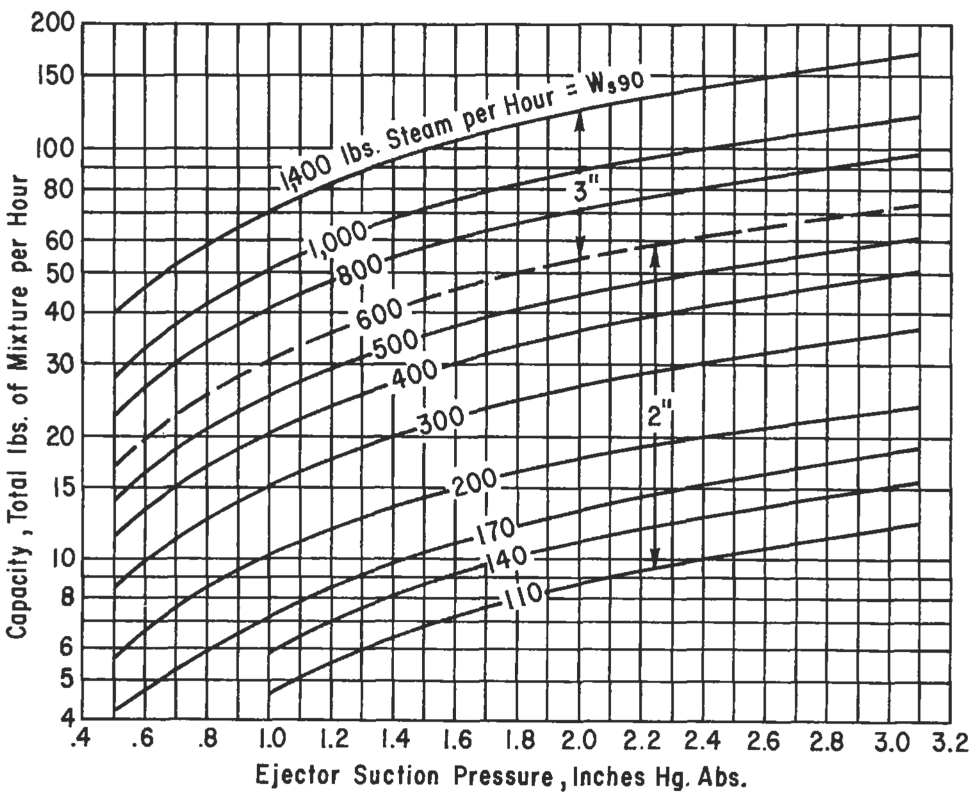


Figure 6-27A. Size ejector for 90 psig steam consumption, two-stage, non-condensing 0.6″–3.0″ Hg abs. By permission, Worthington Corp.

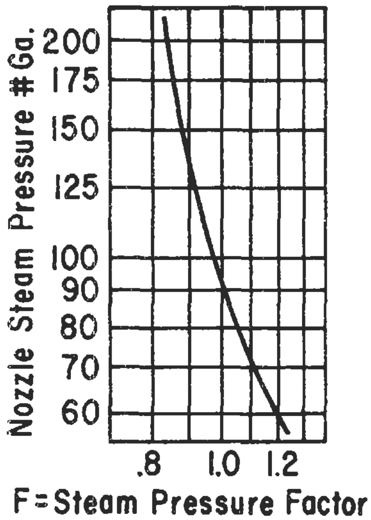


Figure 6-27B. Steam pressure factor for Figure 6-27A. By permission, Worthington Corp.

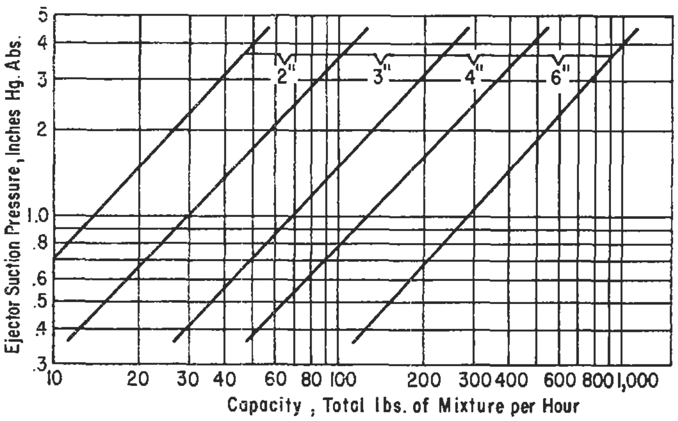


Figure 6-28A. Ejector size, single stage—typical. By permission, Worthington Corp.

3. At ejector suction conditions, determine:
   a. Pounds/hour of condensable vapor
   b. Pounds/hour of non-condensable gases
      (1) dissolved
      (2) injected or carried in process
      (3) formed by reaction
      (4) air inleakage
4. Prepare specification sheet, Figure 6-29 and forward to manufacturers for recommendation.
5. For checking and estimating purposes, follow the guides as to the number of stages, utility requirements, etc. presented herein.

Barometric Condensers

Barometric condensers are direct contact coolers and condensers. They may be counter flow or parallel flow. Good contact direct cooling is an efficient inexpensive design, being considerably cheaper and more efficient than indirect surface or tubular coolers.

Temperature Approach

When serving vacuum equipment, the temperatures are usually set as follows when the non-condensables do not exceed one percent of the total water vapor being condensed. See Figures 6-20 A, B, C, and D.

Terminal Difference, steam temperature corresponding to vacuum less outlet water temperature = 5°F.

Exit Air or Non-Condensables, temperature to be 5°F higher than inlet water temperature to barometric.

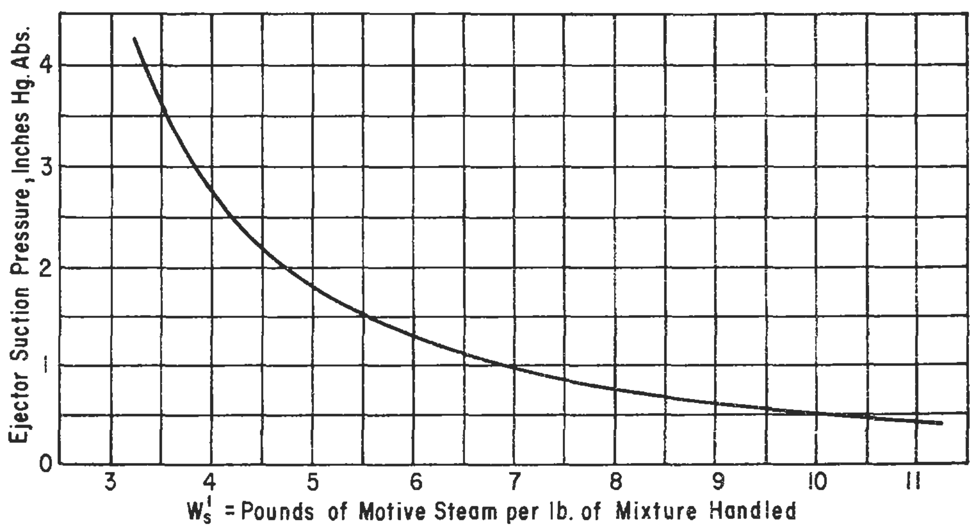


Figure 6-28B. Steam consumption factor. By permission, Worthington Corp.

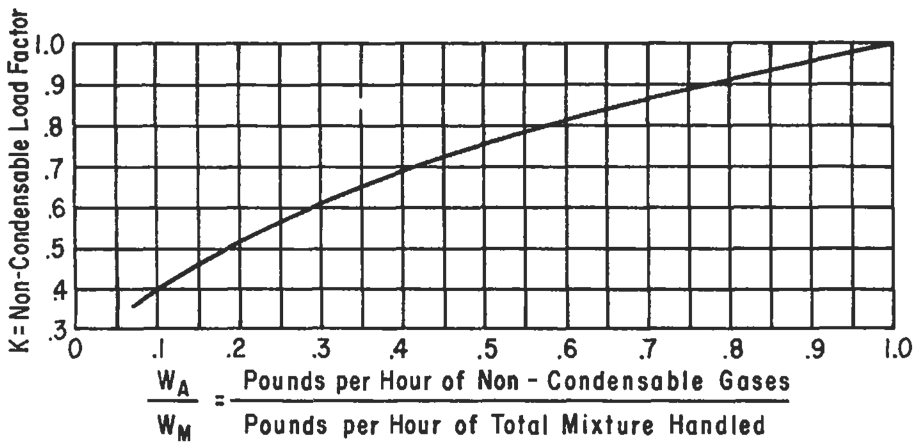


Figure 6-28C. Non-condensable load factor. By permission, Worthington Corp.

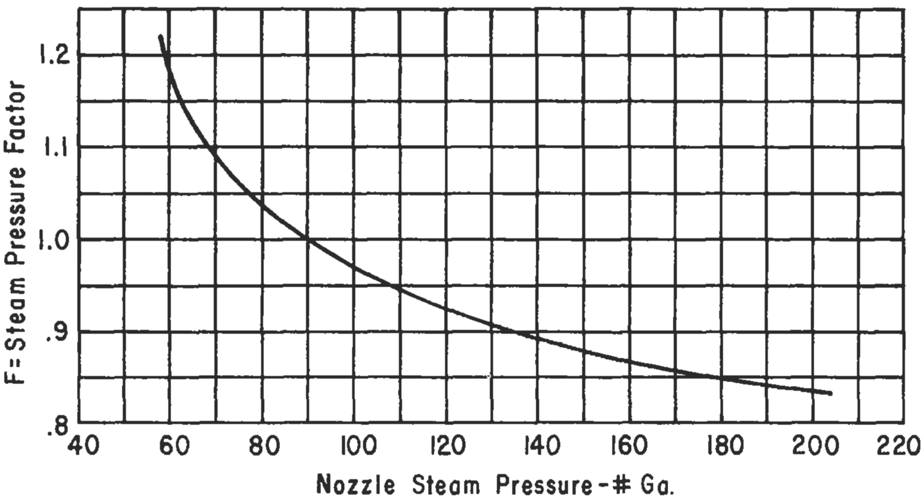


Figure 6-28D. Steam pressure factor. By permission, Worthington Corp.

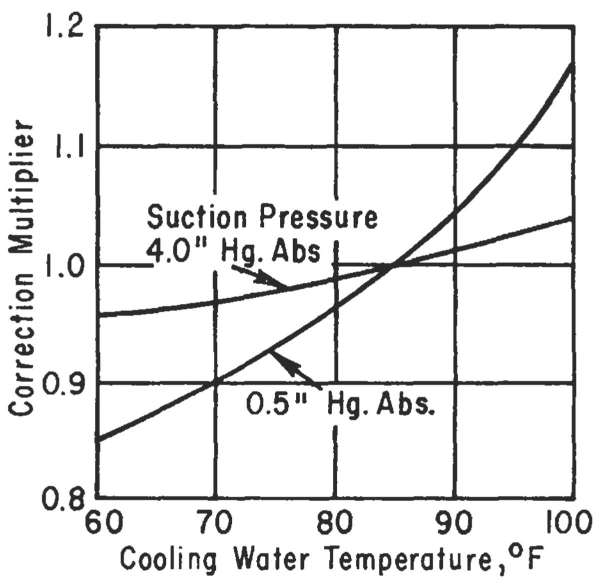


Figure 6-28E. Steam requirement correction for two-stage unit with barometric intercooling. By permission, Fondrk, V. V., *Petroleum Refiner*, V. 37, No. 12, 1958 [3].

$$\text{GPM cooling water required} = W_s L / (\Delta\, t_w\, 500) \quad (6\text{-}23)$$

where W_s is the pounds of steam to condense and L is the latent heat of vaporization, usually taken as 1000 BTU/lb for process applications and 950 BTU/lb for turbine exhaust steam [6, 12].

Example 6-12: Temperatures at Barometric Condenser on Ejector System

A barometric condenser is to condense 8,500 pounds per hour of steam at 3.5 in. Hg abs using 87°F water. The non-condensables are 43 pounds/hr. Note that the non-condensables are less than one percent of the steam.

Steam temperature (steam tables) at 3.5 in. Hg abs	120.6°F
Terminal difference	5
Outlet water temperature from barometric	115.6°F
Inlet water temperature	85
Water temperature rise	30.6°F

SPEC. DWG. NO.
A-

Page of Pages	
Unit Price	
No. Units	
Item No.	

Job No. ____________________

B/M No. ____________________

STEAM EJECTOR SPECIFICATIONS

PERFORMANCE

Make ____________________ Type ____________________

Service ____________________ Condenser: ☐ Barometric ☐ Surface

No. of Stages ____________________ No. of Ejectors per Stage ____________________

Suct. Press ________ MM HgAbs. Suct. Temp ________ °F Max. Disch. Press ________ MM HgAbs.

Steam: Min. Press ________ PSIA. Temp. ________ °F Quality ________ %

Water: Source ____________________ Max. Press ________ PSIA. Max. Temp. ________ °F

Vol. of Evacuated System ____________________ Cu. Ft.

Expected Air Leakage ____________________ Lbs/Hr.

Max. Evacuating Time ____________________ Min.

Ejector Load	Lbs/Hr.	Mol. Wt.	Cp, BTU/Lb-°F	Latent Ht., BTU/Lb.
Condensables				
Non-Condensables				

DESIGN

	1st. Stage	2nd. Stage	3rd. Stage	4th. Stage	5th Stage
Propelling Steam, Lbs/Hr.					
Steam: Inlet Size					
Press. Class & Facing					
Water, GPM					
Water ΔT, °F					
Water: Inlet Size					
Press. Class & Facing					
Water: Exit Size					
Press. Class & Facing					
Suct. Chamber Press. MM HgABS.					
Suct. Chamber Temp. °F					
Condensers: Pre- Inter- After					
Barometric: No. Contact Stages					
Surface: Outside Tube Area Sq. Ft.					

MATERIALS OF CONSTRUCTION

Ejector: Steam Chest ____________________ Steam Nozzles ____________________

Diffuser: Inlet ____________________ Discharge ____________________ Suct. Chamber ____________________

1st. Stage Suct. Chamber Inlet (Size x Pr. Cl. x Facing) ________ X ________ X ________

Barometric Condenser: Shell ____________________ Baffles ____________________ Nozzles ____________________

Surface Condenser: Shell ____________________ Head ____________________

Tubes (O.D. x BWG x L) ________ X ________ X ________ Material ____________________

Tube Sheet ____________________ Baffles ____________________

Steam Strainer ____________________ Shut Off Valves ____________________

REMARKS

Tail Pipes Furnished by ____________________

Interconnecting Piping by ____________________

By	Chk'd.	App.	Rev.	Rev.	Rev.
Date					

P.O. To: ____________________

PURCHASE ORDER NUMBER

Figure 6-29. Steam ejector specifications.

Table 6-11
Air Density Table

Temp. °F.	Density. Lbs./cu. ft.	Lbs./Hr./CFM at 30″ Hg. Abs.
30	0.08105	4.86
40	0.07943	4.76
50	0.07785	4.66
60	0.07635	4.58
70	0.07493	4.50
80	0.07355	4.42
90	0.07225	4.34
100	0.07095	4.25
110	0.06966	4.18
120	0.06845	4.10
130	0.06730	4.04
140	0.06617	3.97
150	0.06510	3.91

Table 6-12
Gas Constants

PV = WRT R = 1544/Molecular Weight

For Use With Units Of:
Cubic Feet, Lbs./Sq. Ft. Abs., °R., Pounds

Gas or Vapor	Formula	Mol. Weight	Gas Constant, R
Hydrogen	H_2	2	772
Carbon Monoxide	CO	28	55.1
Oxygen	O_2	32	48.3
Methane	CH_4	16	96.5
Ethylene	C_2H_4	28	55.1
Nitrogen	N_2	28	55.1
Ammonia	NH_3	17	90.8
Carbon Dioxide	CO_2	44	35.1
Steam (Water Vapor)	H_2O	18	85.8
Sulfur Dioxide	SO_2	64	24.1
Air	....	29	53.3

Air temperature leaving barometric = 85 + 5 = 90°F

$$\text{GPM cooling water required} = \frac{8500\ (1000)}{(30.6)\ (500)} = 556$$

Water Jet Ejectors

Ejectors using water as the motive fluid are designed for reasonable non-condensable loads together with large condensable flows. Water pressures as low as 10–20 psig are usable, while pressures of 40 psig and higher will maintain a vacuum of 1–4 inches of Hg absolute in a single stage unit [1]. Combinations of water and steam ejectors are used to efficiently handle a wide variety of vacuum situations. The water ejector serves to condense the steam from the steam ejector.

Water ejectors and water jet eductors are also used for mixing liquids, lifting liquids, and pumping and mixing suspended solids and slurries. Sizes range from ½ inch to 24 inches. The ejectors are usually used in pumping air or gases while the eductors are used in pumping liquids.

Table 6-13
Temperature—Pressure—Volume of Saturated Water Vapor Over Ice

Temp. °F.	ABSOLUTE PRESSURES Inches Hg	M.M. Hg	Microns	Specific Volume Cu. Ft. Per Lb.
32	0.1803	4.580	4580	3,306
30	0.1645	4.178	4178	3,609
25	0.1303	3.310	3310	4,508
20	0.1028	2.611	2611	5,658
15	0.0806	2.047	2047	7,140
10	0.0629	1.598	1598	9,050
5	0.0489	1.242	1242	11,530
0	0.0377	0.958	958	14,770
— 5	0.0289	0.734	734	19,040
—10	0.0220	0.559	559	24,670
—15	0.0167	0.424	424	32,100
—20	0.0126	0.320	320	42,200
—25	0.0094	0.239	239	55,800
—30	0.0071	0.180	180	74,100
—35	0.0051	0.130	130	99,300
—40	0.0038	0.097	97	133,900

Values obtained from Keenan & Keyes—"Thermodynamic Properties of Steam". John Wiley & Sons, 1936, by permission.

Steam Jet Thermocompressors

Steam jet thermocompressors or steam boosters are used to boost or raise the pressure of low pressure steam to a pressure intermediate between this and the pressure of the motive high pressure steam. These are useful and economical when the steam balance allows the use of the necessary pressure levels. The reuse of exhaust steam from turbines is frequently encountered. The principle of operation is the same as for other ejectors. The position of the nozzle with respect to the diffuser is critical, and care must be used to properly position all gaskets, etc. The thermal efficiency is high as the only heat loss is due to radiation [5].

Ejector Control

Ejectors do not respond to wide fluctuations in operating variables. Therefore, control of these systems must necessarily be through narrow ranges as contrasted to the usual control of most other equipment.

For the single stage ejector, the motive steam flow cannot be decreased below critical flow in the diffuser [2], (Figure 6-30). Units are usually designed for stable opera-

Table 6-14

Pressure—Temperature—Volume of Saturated Steam

Absolute Pressure Ins. Hg	Temperature °F.	Volume cu. ft./lb.	Absolute Pressure Ins. Hg	Temperature °F.	Volume cu. ft./lb.	Absolute Pressure Ins. Hg	Temperature °F.	Volume cu. ft./lb.
0.1803	32	3306	3.0	115.06	231.6	20.0	192.37	39.07
			3.1	116.22	224.5	21.0	194.68	37.32
			3.2	117.35	217.9	22.0	196.90	35.73
			3.3	118.44	211.8	23.0	199.03	34.28
0.20	34.57	2996.0	3.4	119.51	205.9	24.0	201.09	32.94
0.25	40.23	2423.7	3.5	120.56	200.3	25.0	203.08	31.70
0.30	44.96	2039.4	3.6	121.57	195.1	26.0	205.00	30.56
0.35	49.06	1761.0	3.7	122.57	190.1	27.0	206.87	29.50
0.40	52.64	1552.8	3.8	123.53	185.5	28.0	208.67	28.52
0.45	55.89	1387.7	3.9	124.49	181.0	29.0	210.43	27.60
0.50	58.80	1256.4				29.922	212	26.80
0.60	63.96	1057.1				30	212.13	26.74
0.70	68.41	913.8						
0.80	72.32	805.7						
0.90	75.84	720.8				**Lb. Per Sq. In. Abs.**		
			4.0	125.43	176.7			
			4.5	129.78	158.2			
			5.0	133.76	143.25	14.696	212	26.80
			5.5	137.41	131.00	15	213.03	26.29
1.00	79.03	652.3	6.0	140.78	120.72	20	227.96	20.089
1.10	81.96	596.0	6.5	143.92	112.00	30	250.33	13.746
1.20	84.64	549.5	7.0	146.86	104.46	40	267.25	10.498
1.30	87.17	509.1	7.5	149.63	97.92	50	281.01	8.515
1.40	89.51	474.9	8.0	152.24	92.16	60	292.71	7.175
1.50	91.72	444.9	8.5	154.72	87.08	70	302.92	6.206
1.60	93.81	418.5	9.0	157.09	82.52	80	312.03	5.472
1.70	95.78	395.3	9.5	159.48	78.48	90	320.27	4.896
1.80	97.65	374.7				100	327.81	4.432
1.90	99.43	356.2						
						125	344.33	3.587
						150	358.42	3.015
2.00	101.14	339.2	10.0	161.49	74.76	175	370.75	2.602
2.10	102.77	324.0	11.0	165.54	68.38	200	381.79	2.288
2.20	104.33	310.3	12.0	169.28	63.03	225	391.79	2.0422
2.30	105.85	297.4	13.0	172.78	58.47	250	400.95	1.8438
2.40	107.30	285.8	14.0	176.05	54.55	275	409.43	1.6804
2.50	108.71	274.9	15.0	179.14	51.14			
2.60	110.06	265.0	16.0	182.05	48.14			
2.70	111.37	255.7	17.0	184.82	45.48	300	417.33	1.5433
2.80	112.63	247.2	18.0	187.45	43.11	350	431.72	1.3260
2.90	113.86	239.1	19.0	189.96	40.99	400	444.59	1.1613

Values obtained directly or by interpolation from Keenan & Keyes—"Thermodynamic Properties of Steam," John Wiley & Sons, 1936 by permission and Courtesy C. H. Wheeler Co., Philadelphia, Pa.

tion at zero suction flow with the motive fluid maintaining the required volume and energy to produce the necessary diffuser velocity. This is "shut-off" operation. A decrease in motive pressure below the stability point will cause a discontinuity in operation and an increase in suction pressure. If the motive fluid rate increases, the suction pressure will increase or capacity will decrease at a given pressure.

Figure 6-31 illustrates control schemes for the single stage unit which allow greater stability in performance. As the load changes for a fixed suction pressure, the process fluid is replaced by an artificial load (usually air; Figure 6-31, item 1) to maintain constant ejector operation. An artificial pressure drop can be imposed by valve (2), although this is not a preferred scheme. When the addition of air (1) overloads the aftercondenser, the discharge mixture can be recycled to control the pressure [3].

Figure 6-32 illustrates ejector systems with large condensable loads which can be at least partially handled in the precondenser. Controls are used to maintain constant suction pressure at varying loads (air bleed), or to reduce the required cooling water at low process loads or low water temperatures [2]. The cooler water must not be throttled below the minimum (usually 30%–50% of maximum) for proper contact in the condenser. It may be controlled by tailwater temperature, or by the absolute pressure.

The controls for larger systems involve about the same principles unless special performance is under consideration.

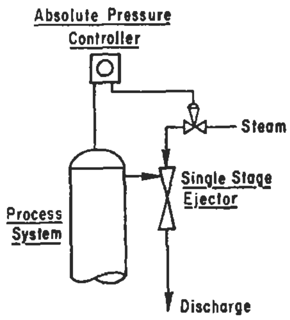


Figure 6-30. Single-stage ejector control.

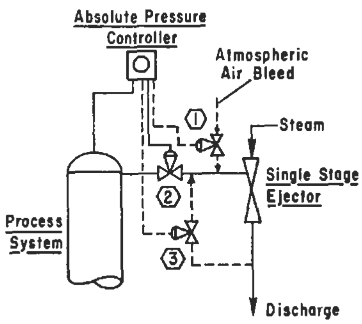


Figure 6-31. Single-stage ejector control with varying load.

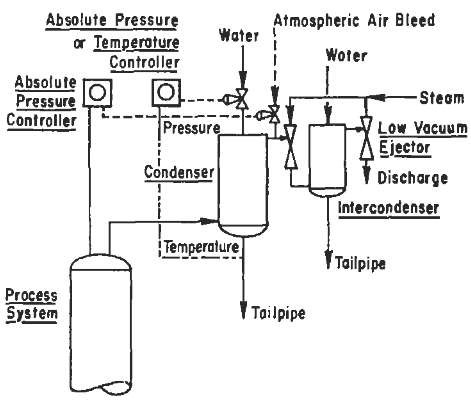


Figure 6-32. System handling large quantities of condensable vapors.

Time Required For System Evacuation

It is difficult to determine the time required to evacuate any particular vessel or process system including piping down to a particular pressure level below atmospheric. When using a constant displacement vacuum pump this is estimated by O'Neil [31]:

$$P_d = \frac{V}{t} \log_e \frac{P_2 - P_c}{P_1 - P_c} \tag{6-24}$$

where P_d = piston displacement cu ft/min
V = system volume, cu ft
t = evacuation time, min
P_2 = absolute discharge pressure of pump, psia
P_c = absolute intake pressure of pump with closed intake
P_1 = absolute intake pressure of pump

The relation above is theoretical, and does not take into account any inleakage while pumping. It is recommended that a liberal multiplier of perhaps 2 or 3 be used to estimate closer to actual time requirements.

An alternate relation for calculating evacuation time is from the Heat Exchange Institute [11]:

$$W_{Ta} = \frac{0.15\,V}{t}\,(P_2 - P_1)_1 \text{ lbs / hr} \tag{6-25}$$

Alternate Pumpdown to a Vacuum Using a Mechanical Pump

For large process systems of vessels, piping, and other equipment, the downtime required to evacuate the system before it is at the pressure (vacuum) level and then to maintain its desired vacuum condition, can become an important consideration during start-up, repair, and restart operations.

Reference [26] suggests an improved calculation:

$$t_s = \frac{\ln\,(P''_n / P''_o)}{R_{ps} \ln\,(V'/[V' + V'_o])} \tag{6-26}$$

where t_s = pumpdown time, sec
p''_n = final pressure in vessel or system, torr
P''_o = starting pressure in vessel or system, torr
R_{ps} = pump speed, rotations (or strokes)/sec
V' = volume of vessel or system, liters
V'_o = volume of pump chamber, liters
S = pump speed, liters/sec
S_o = pump speed, at P''_o, liters/sec
S_n = pump speed at P''_n, liters/sec

Example 6-13: Determine Pump Downtime for a System

Calculate the pump downtime for a system of vessels and piping with a volume of 500 liters. The final pressure is to be 0.01 torr, starting at atmospheric. From the speed-pressure curve of a manufacturer's pump at 0.01 torr, speed is 2.0 liters/sec. At atmospheric pressure, S_o = 2.75 liters/sec with P''_o = 760 torr. From the manufacturer's data, R_{ps} = 15 and V'_o = 0.5 liters.

$$\text{Solving}: t = \frac{\ln\left(\frac{P''_n}{P''_o}\right)}{R_{ps}\ln\left(\frac{V'}{V'+V'_o}\right)} \qquad (6\text{-}26)$$

$$= \frac{\ln\left(\frac{0.01}{760}\right)}{(15)\ln\left(\frac{500}{500+0.5}\right)}$$

$$t = \frac{\ln(0.0000131579)}{(15)\ln(0.999000999)} = \frac{-11.23841862}{15(-0.0009995003)}$$

$$t = 749.6 \text{ sec} = 12.49 \text{ min}$$

Evacuation With Steam Jets

Rough Estimate of System Pumpdown Using Steam Jets [24]

The remarks presented earlier regarding the use of steam jets for pumping down a system apply. The method of power [35] presented by Reference [24] is:

$$t = [2.3 - 0.003\,(P'_s)]\,V/w_j \qquad (6\text{-}27)$$

where t = time required to evacuate a system from atmospheric pressure to the steady state operating pressure, min
P'_s = design suction pressure of ejector, torr
V = free volume of the process system, cu ft
w_j = ejector capacity, 70°F dry air basis, lb/hr

This assumes dry air with no condensables and negligible pressure drop through the system to the ejector. Also, the jet air handling capacity is assumed approximately twice the design capacity, and air inleakage during evacuation is negligible.

When considering time to evacuate a system using a steam jet, first recognize that securing reasonable accuracy is even more difficult than for a positive displacement pump. The efficiency of the ejector varies over its operating range; therefore as the differential pressure across the unit varies, so will the volume handled. Consequently, evacuation time is difficult to establish except in broad ranges. The above relations can be adopted to establish the order of magnitude only.

A recommended evacuation calculation is given in Reference [19]. This is specific to Penberthy equipment, but is considered somewhat typical of other manufacturers.

- Establish suction load of air to be evacuated in cubic feet volume of vessel/system.
- Establish the required time to evacuate, in minutes.
- Determine operating steam pressure, psig.
- Determine/establish required final suction pressure in vessel/system, inches Hg abs.
- Establish discharge pressure required (usually to atmosphere), psig.
- Note that the performance is specific to the ejector used.

Example 6-14: Evacuation of Vessel Using Steam Jet for Pumping Gases

The performance and procedure use the data of Penberthy for this illustration (by permission):

Evacuating—Selection Procedure

Refer to U Evacuation Time chart.

Step 1. Determine evacuation time in minutes per hundred cubic feet.

Step 2. Go to the left-hand column in table, final Suction Pressure (hs). Read across to find evacuation time equal to or less than that determined in Step 1. Read to the top of table and note unit number. See Table 6-15.

Step 3. Read Steam Consumption of unit selected off Capacity Factor Chart. See Table 6-16.

Evacuating—EXAMPLE:

To evacuate 3000 cubic foot vessel full of air at atmospheric pressure:
Operating Steam Pressure, PSIG (hm) 100
Final Suction Pressure, inches Hg abs (hs) 5
Time to evacuate, hrs 2.5
Discharge Pressure (hd) .... atmosphere

Step 1. Determine evacuation time in minutes per hundred cubic feet.

$$\frac{2.5 \text{ hr} \times 60}{30 \text{ (hundred) cu ft}} = 5 \text{ min} / 100 \text{ cu ft}$$

Step 2. Go to the final pressure on left of Evacuation Time chart (5 in. Hg hs). Read across and find evacuation time equal to or less than 5 minutes. See Table 6-15.

The U-2 will evacuate the tank in 5.33 minutes per hundred cubic feet and the U-3 will complete the evacuation in 3.42 minutes per hundred cubic feet.

Step 3. Read steam consumption of selected unit off Capacity Factor Chart. See Table 6-16. The unit to select

Table 6-15
Example Using Penberthy Model U Ejector for Evacuation Time

U MODEL EVACUATION TIME (in minutes per 100 cu ft at 100 PSIG Operating Steam Pressure)																
	Model Number															
Suction Press In. Hg abs. (hs)	U-1H	U-2H	U-3H	U-4H	U-5H	U-6H	U-7H	U-8H	U-9H	U-10H	U-11H	U-12H	U-13H	U-14H	U-15H	U-16H
12″	4.68	3.08	1.98	1.37	1.01	.769	.610	.494	.409	.343	.293	.253	.206	.171	.145	.123
11″	5.06	3.32	2.14	1.48	1.09	.830	.657	.532	.441	.370	.316	.273	.222	.185	.156	.133
10″	5.44	3.57	2.30	1.59	1.17	.894	.707	.572	.474	.398	.340	.293	.239	.198	.168	.143
9″	5.85	3.84	2.46	1.71	1.26	.960	.760	.615	.510	.427	.365	.315	.257	.213	.180	.154
8″	6.29	4.14	2.66	1.84	1.35	1.04	.818	.662	.549	.460	.393	.339	.276	.230	.194	.165
7″	6.76	4.45	2.86	1.98	1.46	1.12	.880	.771	.590	.495	.423	.365	.297	.247	.209	.178
6″	7.35	4.84	3.10	2.15	1.58	1.21	.955	.774	.640	.537	.460	.396	.323	.268	.227	.193
5″	8.10	5.33	3.42	2.37	1.74	1.33	1.06	.853	.706	.592	.507	.437	.356	.295	.250	.213
4″	9.32	6.13	3.94	2.73	2.01	1.54	1.22	.981	.813	.683	.584	.504	.410	.340	.288	.245
3″	11.6	7.60	4.87	3.38	2.48	1.90	1.50	1.22	1.01	.845	.721	.623	.507	.422	.356	.304

By permission, Penberthy Inc.

Table 6-16
Example Ejector Capacity Factor and Steam

U AND L CAPACITY FACTOR AND STEAM CONSUMPTION																
MODEL	L-1H	L-2H	L-3H	L-4H	L-5H	L-6H	L-7H	L-8H	L-9H	L-10H	L-11H	L-12H	L-13H	L-14H	L-15H	L-16H
NUMBER	U-1H	U-2H	U-3H	U-4H	U-5H	U-6H	U-7H	U-8H	U-9H	U-10H	U-11H	U-12H	U-13H	U-14H	U-15H	U-16H
CAPACITY FACTOR OPERATING STEAM	.293	.445	.694	1.00	1.36	1.78	2.25	2.78	3.36	4.0	4.69	5.43	6.66	8.03	9.49	11.12
CONSUMPTION LB. PER HOUR	85	125	195	270	370	480	610	755	910	1090	1280	1480	1820	2190	2580	3030
(Qm)	(Valid at standard nozzle pressure of 80, 100, 120, 140, 160, 180 or 200 PSIG.)															

By permission, Penberthy Inc.

would be the U-3 in this case and its steam consumption is 195 pounds per hour.

There are often useful operations performed by jet equipment, such as pumping air or gases, exhausting systems, heating liquids, mixing of liquids, priming (removal of air) for centrifugal pumps, and many others (See Figures 6-9B and 6-10).

Mechanical Vacuum Pumps

The process designer or mechanical engineer in a process plant is not expected to, nor should he, actually design a mechanical vacuum pump or steam jet, *but* rather he should be knowledgeable enough to establish the process requirements for capacity, pressure drops, etc., and understand the operation and details of equipment available.

Mechanical vacuum pumps are eight to ten times more efficient users of energy than steam jets; although, steam jets are reliable and cost less [23]. See Table 6-17.

These units are mechanical compressors but are designed to operate at low suction pressures absolute. They require special seals to prevent inleakage of air or other vapors that could create suction performance problems. They also require special clearances between the housing and the pressure producing element(s). Figures 6-9A and 6-10 present representative diagrams of operating ranges of vacuum pumps and ejectors.

The chapter on Compression in Volume 3 of this series presents details of several mechanical vacuum units, and this information will not be repeated here. However, more specific vacuum units and system related data is given.

Figure 6-33 diagrams vacuum system arrangements for process systems. It is important to examine the plant economics for each system *plus* the performance reliability for maintaining the desired vacuum for process control.

The most used mechanical vacuum pumps or compressors are reciprocating, liquid-ring, rotary-vane, rotary blower, rotary piston, and diaphragm.

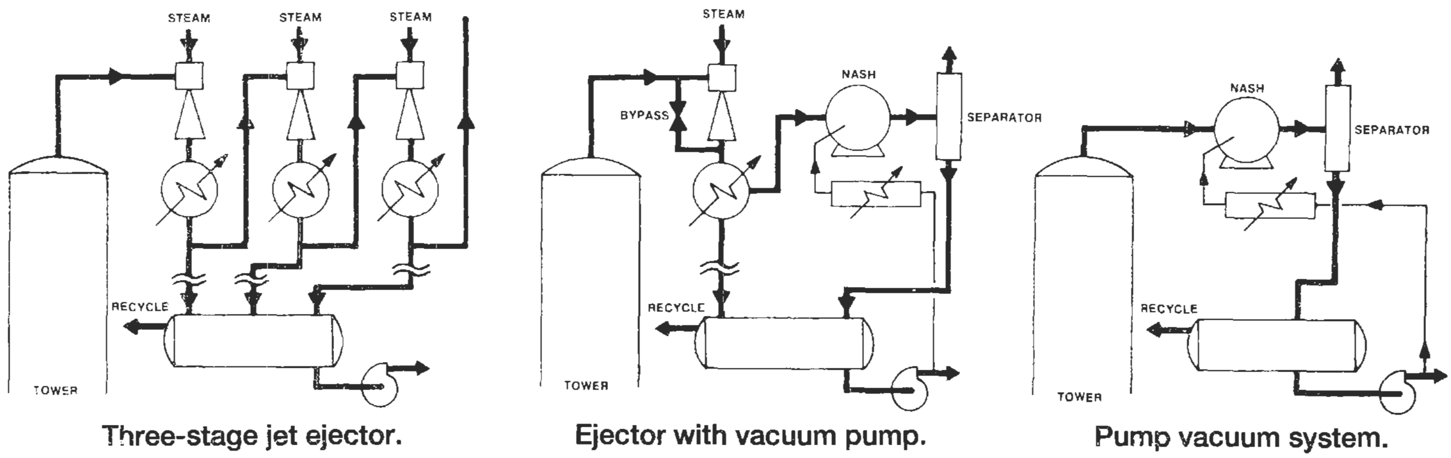


Figure 6-33. Typical vacuum systems holding vacuum on a vessel. By permission, Nash Engineering Co.

Table 6-17
Typical Operating Range of Vacuum Generating Equipment
(For Ejectors, See Figure 6-9A and Figure 6-25)

Type Equipment	Absolute Pressure Range (Discharge, then Lower or Suction) mm Hg Abs	mm Hg Abs
Rotary piston (Liquid)		
Single-stage	760	250
Two-stage	250	150
Rotary Lobe, two impellers		
Single-stage	760	250
Two-stage	250	75
Helical rotary lobe		
Single-stage	760	200
Two-stage	200	75
Centrifugal	760	130
Rotary sliding vane		
Single-stage	760	75
Two-stage	75	10
Reciprocating		
Single-stage	760	30
Two-stage	30	10
Rotary piston, oil sealed		Microns, Abs
Single-stage	760	60
Two-stage	60	1
Rotary vane, oil sealed		
Single-stage	760	7
Two-stage	7	0.08

*Varies with manufacturer's equipment.
Compiled from various published references and manufacturer's literature.

Although the thermal efficiencies of various mechanical vacuum pumps and even steam jet ejectors vary with each manufacturer's design and even size, the curves of Figure 6-34 present a reasonable relative relationship between the types of equipment. Steam jets shown are used for surface intercondensers with 70°F cooling water. For non-condensing ejectors, the efficiency would be lower.

Combinations of steam jet ejectors operating in conjunction with mechanical pumps can significantly improve the overall system efficiency, especially in the lower suction pressure torr range of 1 torr to 100 torr. They can exist beyond the range cited, but tend to fall off above 200 torr. Each system should be examined individually to determine the net result, because the specific manufacturer and the equipment size enter into the overall assessment. Some effective combinations are:

- Steam jet—liquid ring vacuum pump
- Rotary blower—liquid ring vacuum pump
- Rotary blower—rotary vane compressor
- Rotary blower—rotary piston pump

Liquid Ring Vacuum Pumps/Compressor

Figure 6-35 provides a pictorial cross section of this type unit along with an actual photograph of disassembled major components.

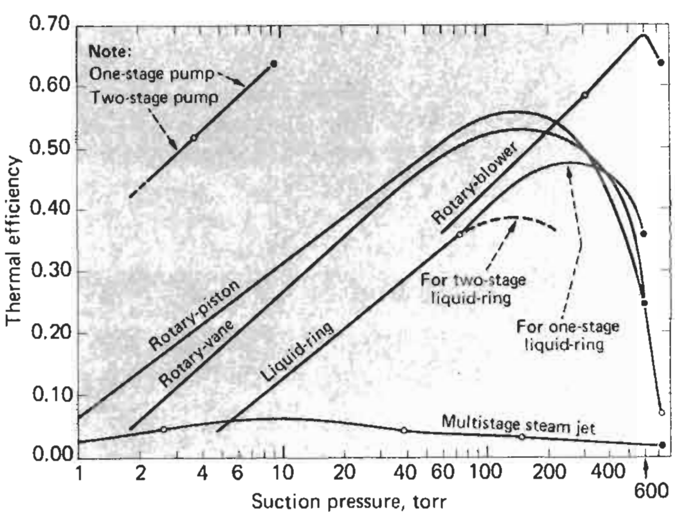


Figure 6-34. Rough estimates of thermal efficiency of various vacuum producing systems. By permission, Ryans, J. L. and Croll, S., *Chem. Eng.*, V. 88, No. 25, 1981, p. 72 [22].

How it Works: Typical of This Class of Pump (By permission of [27])

"The Nash vacuum pump or compressor has only one moving part—a balanced rotor that runs without any internal lubrication. Such simplicity is possible because all functions of mechanical pistons or vanes are performed by a rotating band of liquid compressant.

While power to keep it rotating is transmitted by the rotor, this ring of liquid tends to center itself in the cylindrical body. Rotor axis is offset from body axis. As the schematic diagram in Figure 6-35 shows, liquid compressant almost fills, then partly empties each rotor chamber during a single revolution. That sets up the piston action. Stationary cones inside the rotor have closed sections between ported openings that separate gas inlet and discharge flows.

A portion of the liquid compressant passes out with discharged air or gas. It is usually taken out of the stream by

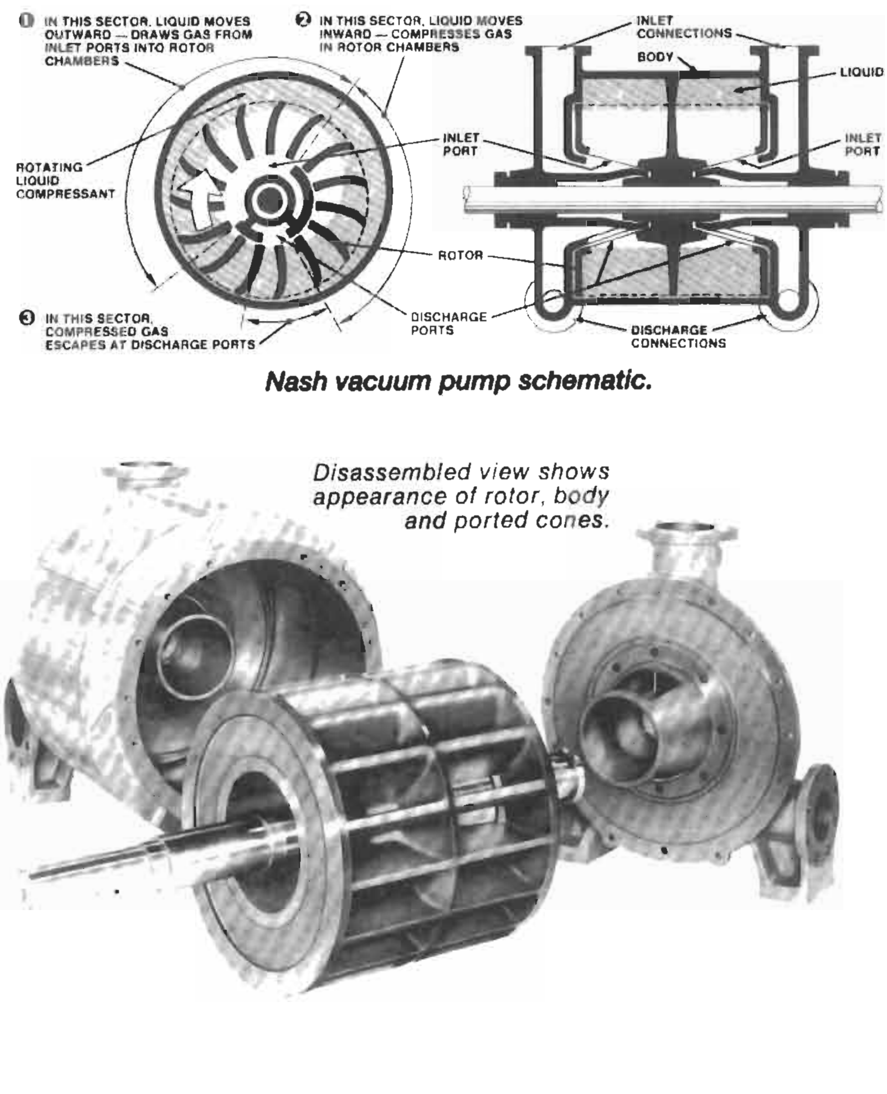


Figure 6-35. Diagram of liquid ring vacuum pump features. By permission, Nash Engineering Co.

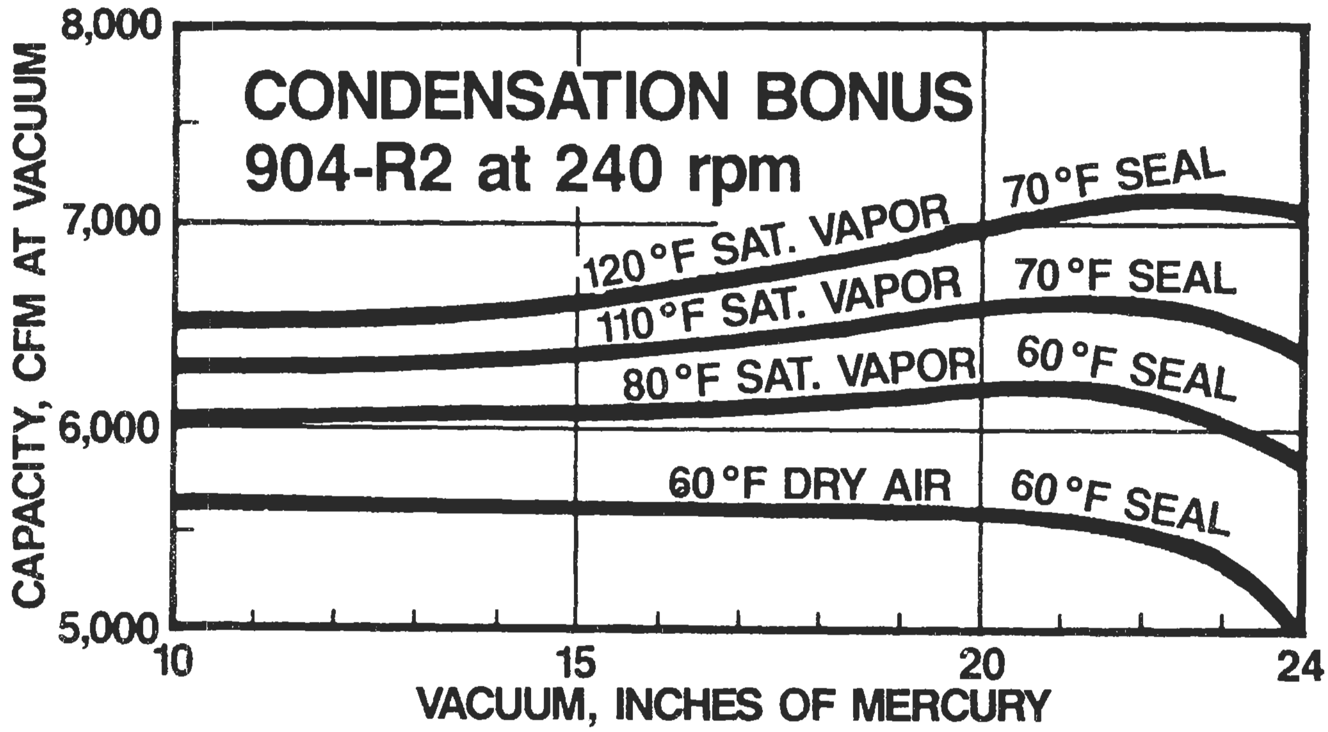


Figure 6-36. Typical capacity increases gained when a vacuum pump sealed with relatively cool water handles air saturated with water vapor. By permission, Nash Engineering Co.

a discharge separator furnished by the manufacturer. Makeup is regulated by orifices and manual valve adjustments. There is an optimum flow rate, but performance is not seriously affected by variations.

Moisture or even slugs of liquid entering the inlet of a liquid-ring vacuum pump will not harm it. Such liquid becomes an addition to the liquid compressant. Vapor is often condensed in a vacuum pump. The condensate is also added to the liquid compressant.

In a typical closed-loop system, liquid from the separator is cooled in a heat exchanger then recirculated. Any excess liquid added by mist and vapor flows out through a level-control valve."

The liquid ring pump/compressor is available from several manufacturers, with about the same operating principle, but differing in mechanical assembly and sealing details as well as ranges of operation. This type unit has only one moving part, i.e., a balanced rotor (See Figure 6-35) that does not require internal lubrication, because the circulating sealing liquid inside the unit provides cooling and lubrication (even when the liquid is water). There is a ring of rotating liquid in the case that is circulated by the rotor. To accomplish the pumping action, the rotor is *offset* from the axis of the case or body. The rotating liquid fills and empties the chamber/case of the rotor during each revolution. There are no inlet or outlet valves. The inlet and outlet ports or openings are ported so the rotary rotor with its stationary cones are closed between the ported openings that separate the gas inlet and discharge gas-liquid mixture to a mechanical separator.

The vapor or gas becomes separated and flows out either to the atmosphere (if air or environmentally acceptable) or a condensable vapor can be condensed inside the pump by using recirculated chilled coolant directly as the circulating liquid. The excess liquid from the condensation can be drawn off the separator.

These types of units can accept wet vapors or gases coming into the suction, as well as corrosive vapors when the proper circulating liquid compressant is selected. Some types of entrained solids can be pumped through the unit, while abrasive solids will naturally do some erosive damage. An important significance of this type unit is that the liquid used to compress the incoming vapors can be selected to be compatible with the vapors, and does not have to be water. For dry air applications, these units normally operate with 60°F seal compressing water. This keeps the water loss low. For saturated air or other vapors, the chilled recirculating liquid reduces the volume that the pump must handle, and thereby, increases pump capacity. Figure 6-36 illustrates for a particular pump the relative increase

(text continued on page 393)

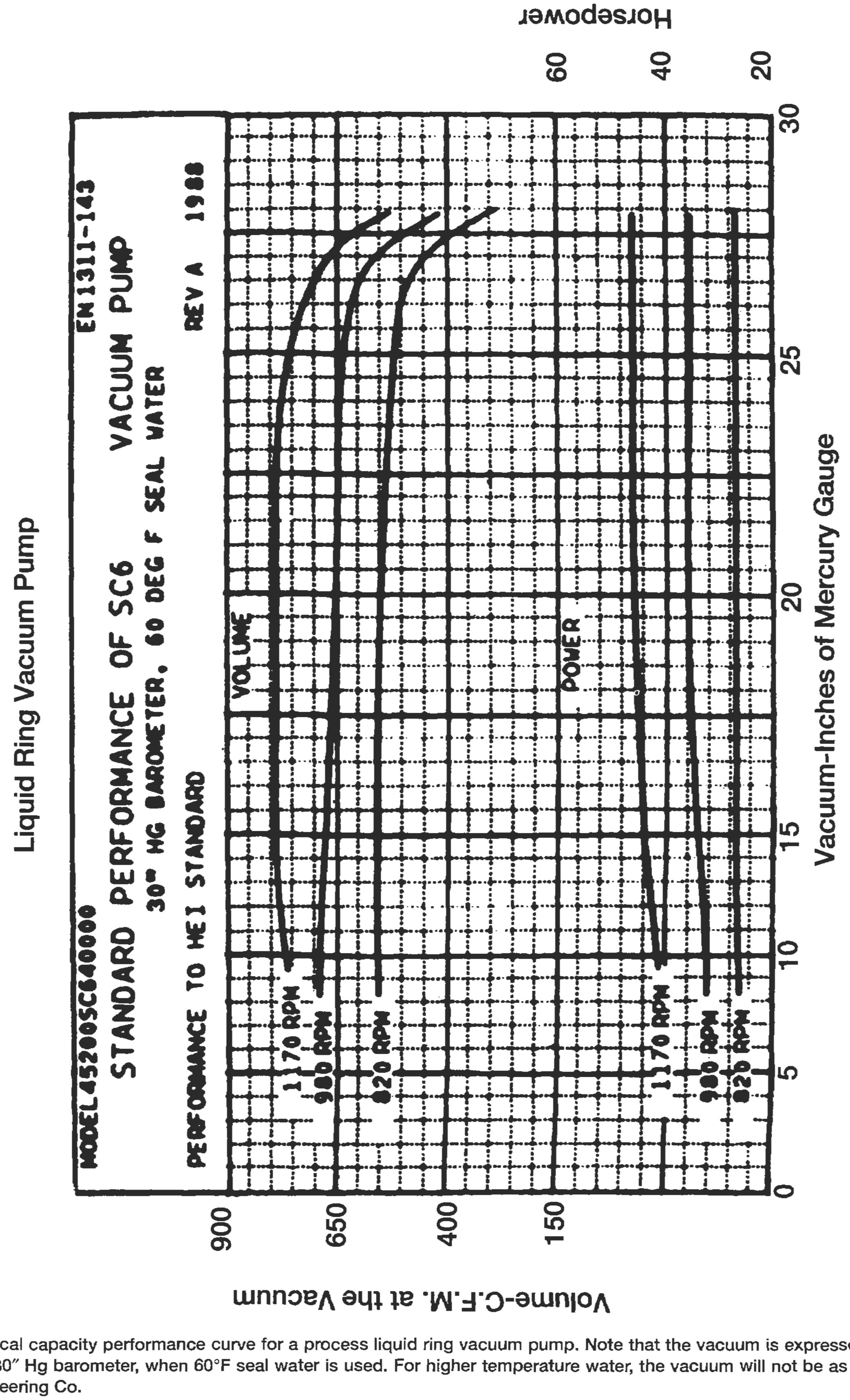
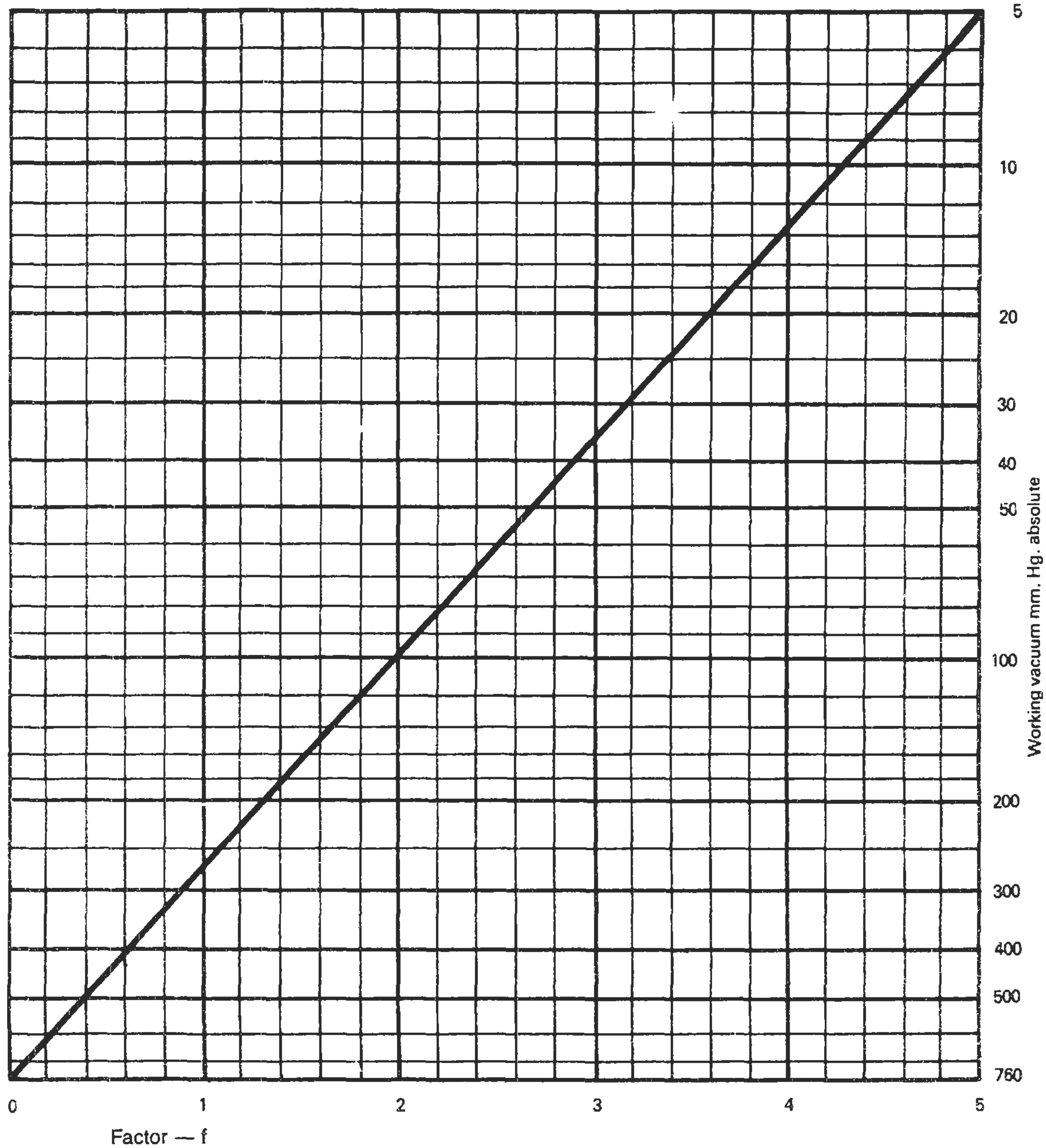


Figure 6-37. Typical capacity performance curve for a process liquid ring vacuum pump. Note that the vacuum is expressed here as gauge, referenced to a 30″ Hg barometer, when 60°F seal water is used. For higher temperature water, the vacuum will not be as great. By permission, Nash Engineering Co.



Total volume to be extracted = Volume of vessel x f
(f at the working vacuum from the above chart)

$$\text{Evacuating time} = \frac{\text{Total volume to be extracted}}{\text{Capacity of the vacuum pump at the working vacuum}}$$

Example: Vessel size = 200 ft^3

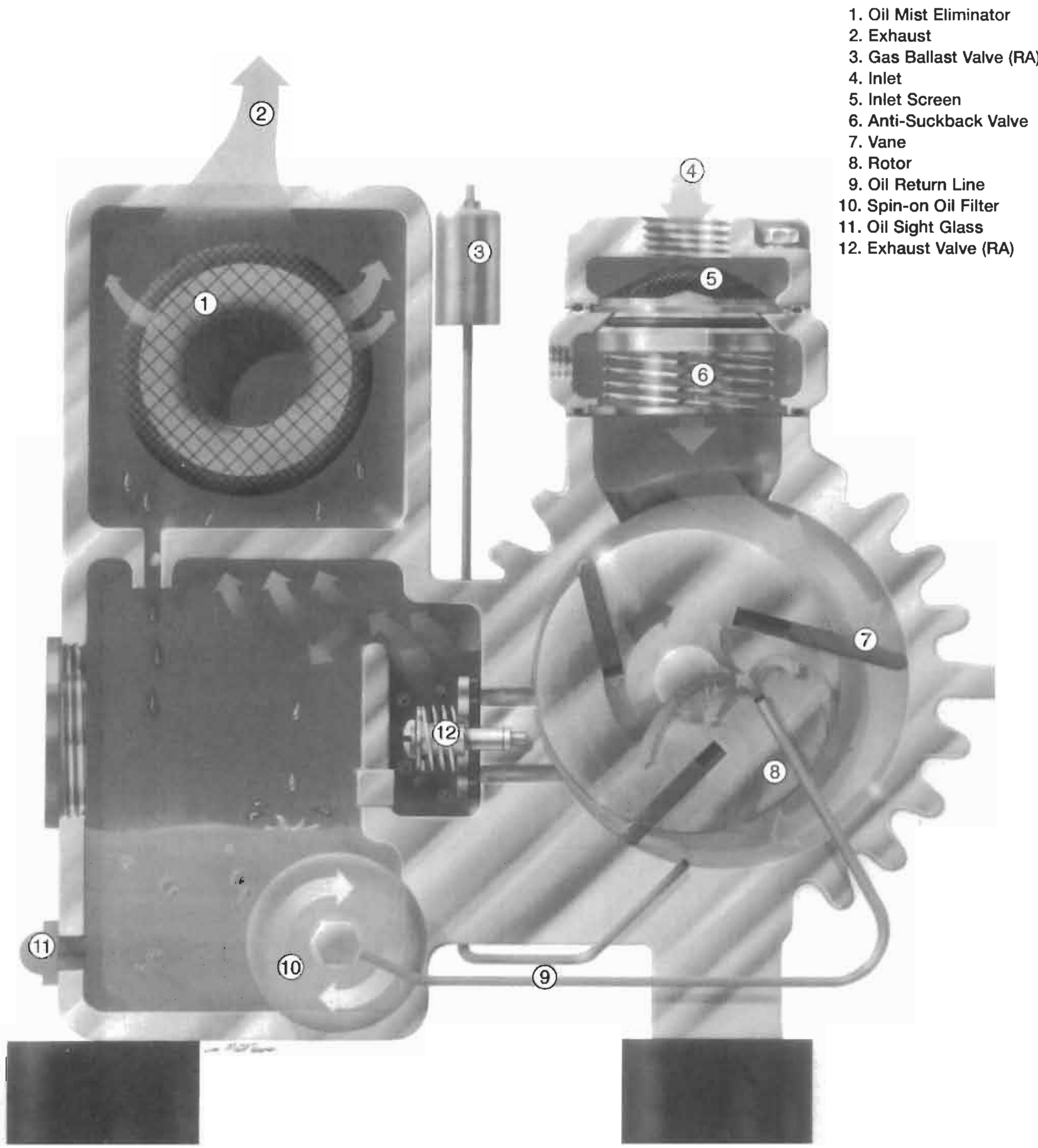
Working vacuum = 60mm Hg. abs.
Evacuating time = 3 minutes

Total volume to be extracted = 200 x f = 200 x 2.5 = 500 ft^3

Capacity to be extracted = $\frac{500}{3}$ = 167 CFM at 60mm Hg. abs;

Pump selection = V5212 (from chart p.6) at 1750 rpm.

Figure 6-38. Chart for liquid ring vacuum pump to estimate the total volume to be displaced to evacuate a closed vessel to a predetermined vacuum. By permission, Graham Manufacturing Co., Inc.



R5 Operating Principle

Rotation of the pump rotor, which is mounted eccentrically in the pump cylinder, traps entering vapor between rotor vane segments. As rotation continues, vapor is compressed then discharged into the exhaust box. Vapors then pass through four stages of internal oil and smoke eliminators to remove 99.9% of lubricating oil from the exhaust. Oil is then returned to the recirculating oil system.

The four stage exhaust box includes the oil box separator, the demister pad, the oil mist eliminator, and the synthetic oil baffle. Additional features include an automotive type spin-on oil filter, a built-in inlet anti-suckback valve that prevents oil from being drawn into the system when the pump is stopped, and a built-in gas ballast, available on the RA version, which permits pumping with high water vapor loads.

Figure 6-39. Rotary vane-type vacuum pump without external cooling jacket. By permission, Busch, Inc.

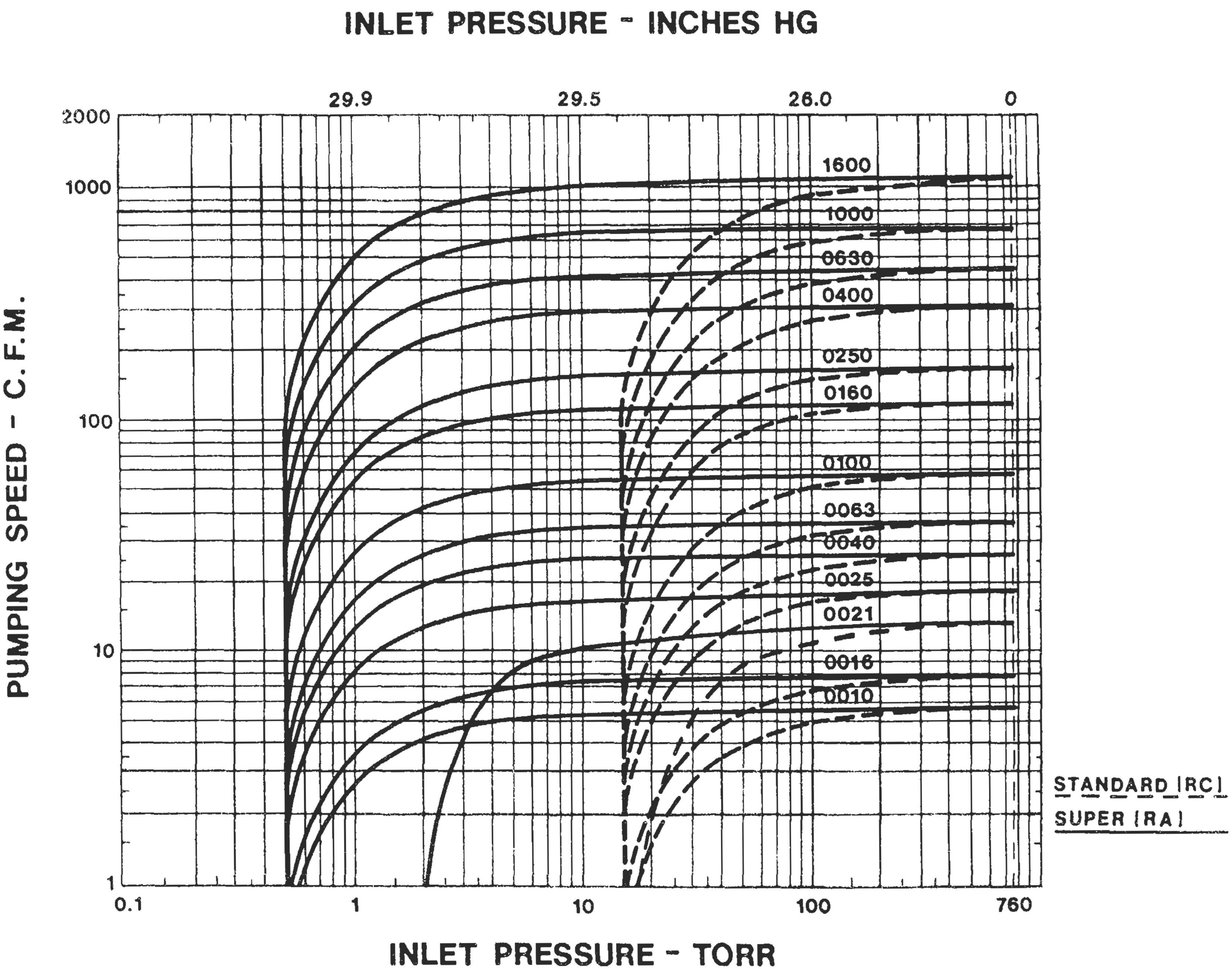


Figure 6-40. Typical performance curves for rotary vane type vacuum pump. By permission, Busch, Inc.

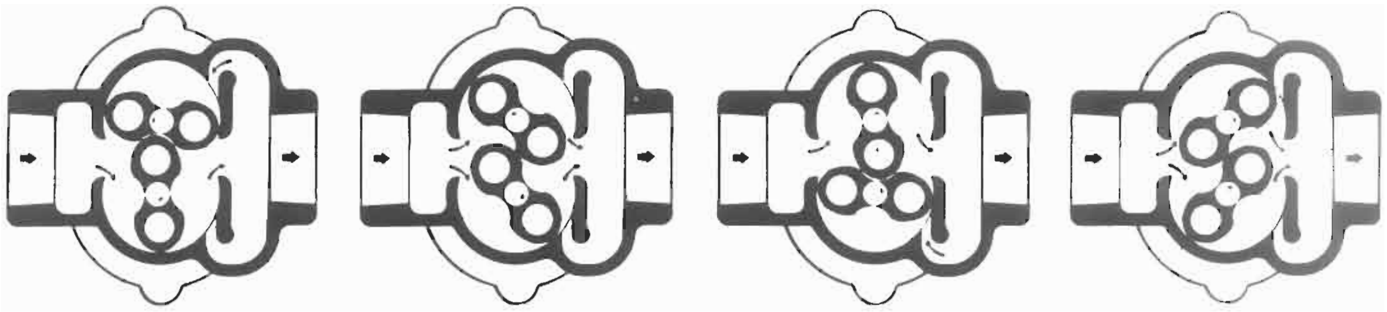


Figure 6-41. Rotating lobe vacuum blowers showing lobe rotation as gas moves through the unit. By permission, Roots Division, Dresser Industries, Inc.

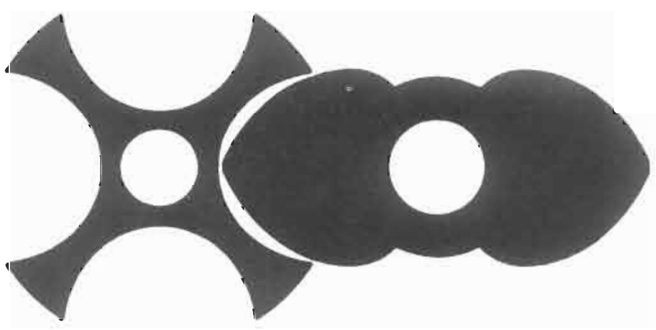


Figure 6-42. Helical four fluted gate blower rotors/two lobes main rotor intermeshing. See Figure 6-44. By permission, Gardner-Denver Industrial Machinery-Cooper Industries.

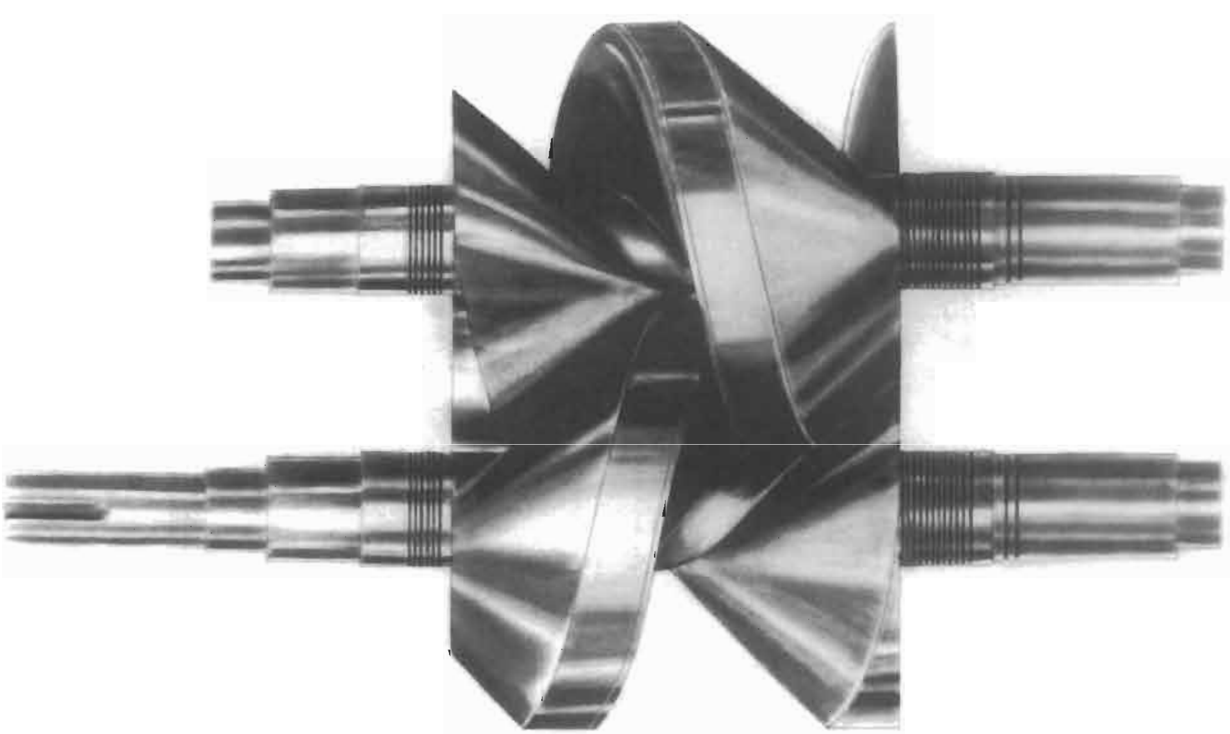


Figure 6-43. Screw-type rotors for rotary lobe blower. By permission, Roots Division, Dresser Industries, Inc.

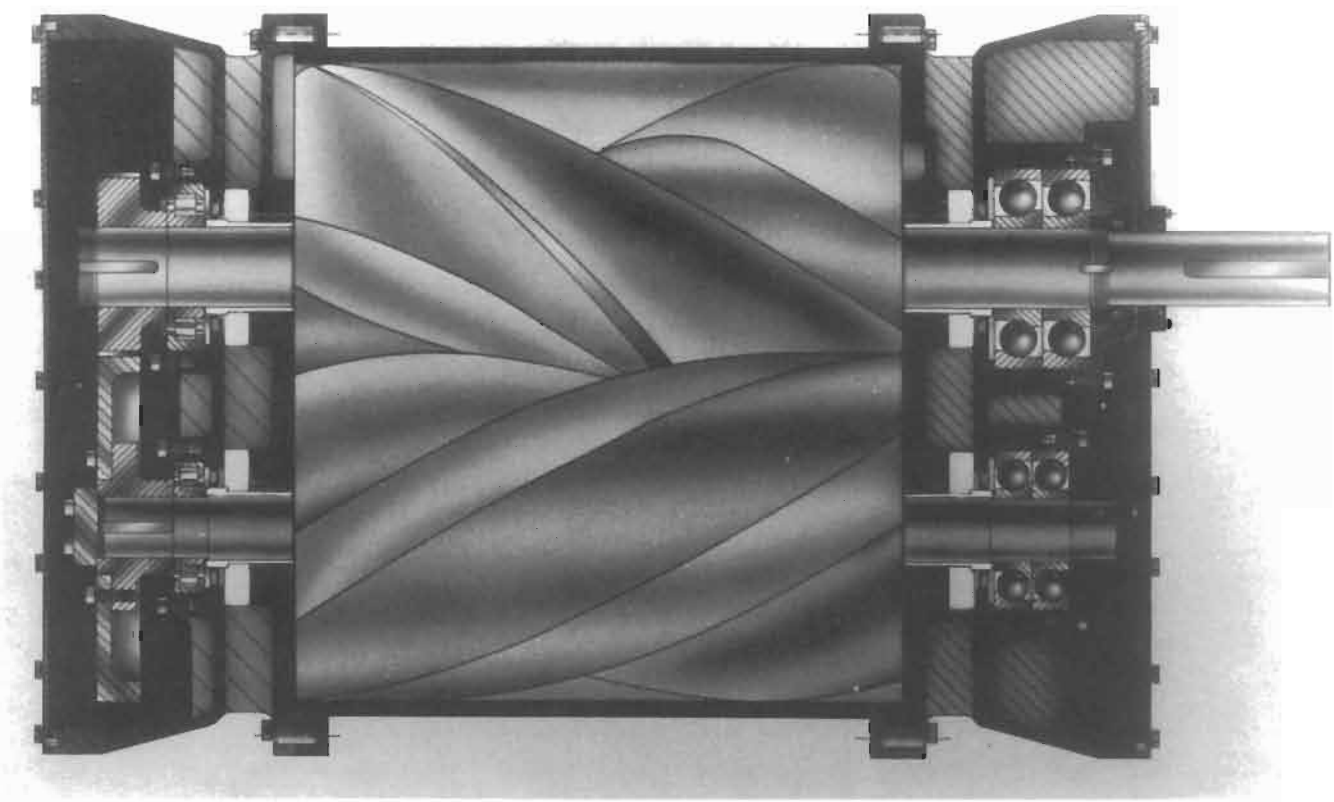


Figure 6-44. Rotary blower with screw-type rotors. By permission, Gardner-Denver Industrial Machinery-Cooper Industries.

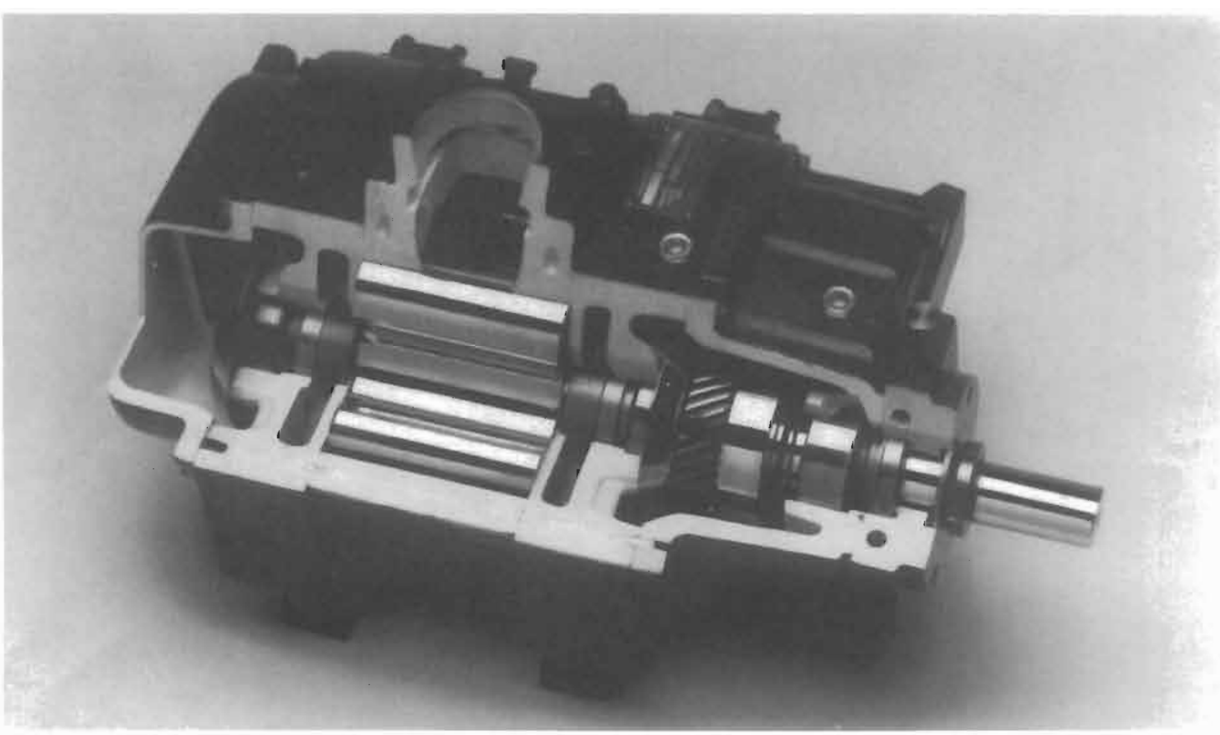


Figure 6-45. Cut-away view of internal assembly of rotary lobe vacuum pump. By permission, Tuthill Corp., M.D. Pneumatics Division.

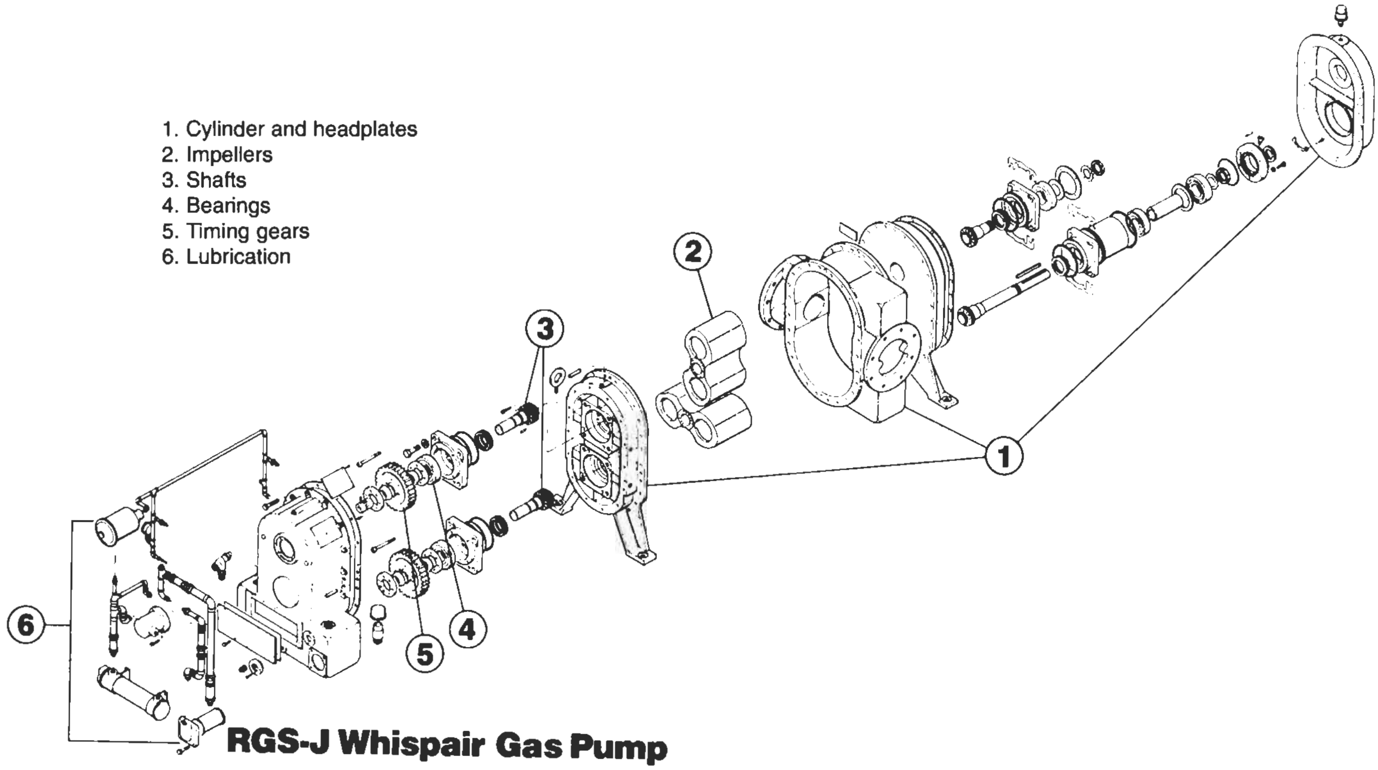


Figure 6-46A. Exploded view of rotary lobe gas pump with mechanical seals and pressure lubrication for bearings. By permission, Roots Division; Dresser Industries, Inc.

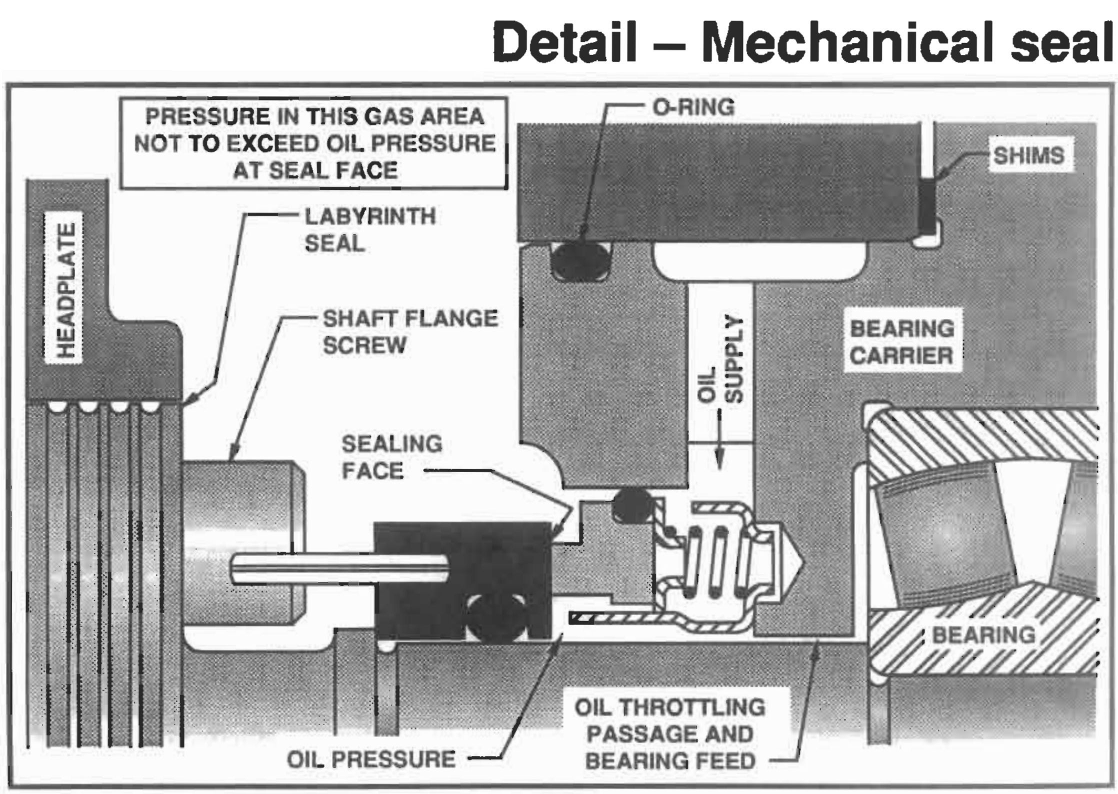
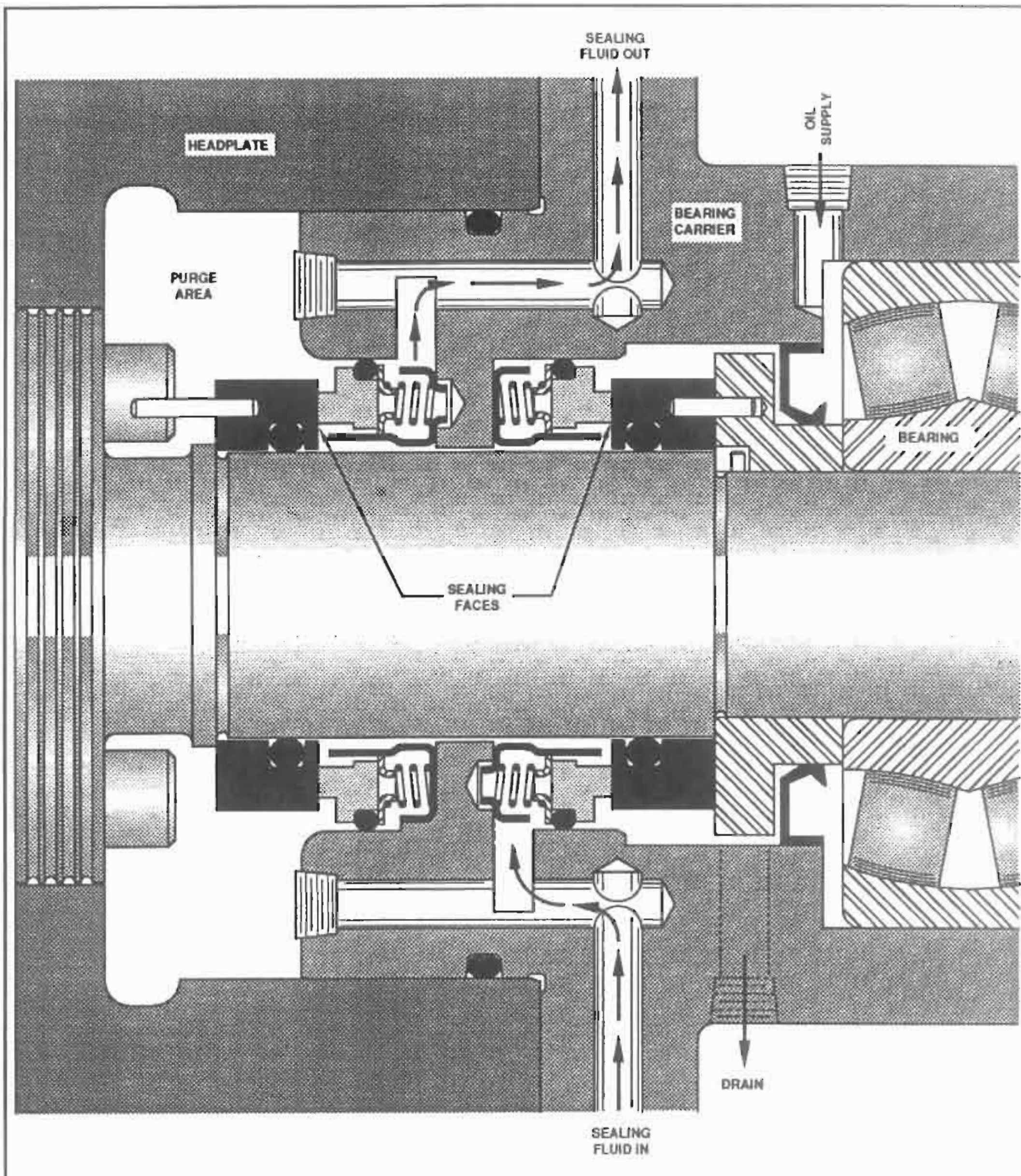


Figure 6-46B. Mechanical seal to be used with Figure 6-46A. By permission, Roots Division, Dresser Industries, Inc.



PROCESS INDUSTRY REQUIREMENT DOUBLE MECHANICAL SEAL

Figure 6-46C. Double mechanical seal used for special gas sealing requirements in Figure 6-46A, and substitutes for the single seal of Figure 6-46B. By permission, Roots Division, Dresser Industries, Inc.

(text continued from page 385)

in expected capacity as recirculated liquid (seal) inside the pump casing is varied in temperatures.

Materials of construction for this type of unit are usually modular cast iron rotors on steel shafts and cast iron casings or bodies. For special requirements in corrosive situations that cannot be remedied by changing the seal liquid, pumps can be furnished in Type 316 stainless steel or other alloys (expensive).

Performance capacity curves are based on standard dry air with 60°F water as the liquid compressant or seal liquid. The pumps operate on a displacement or volumetric basis; therefore, the CFM capacities are about the same for any particular pump for any dry gas mixture. To calculate pounds/hours of air or gas mixture, the appropriate calculation must be made.

Figure 6-37 presents a typical performance curve for this type of vacuum pump. Note that it is specific to 60°F

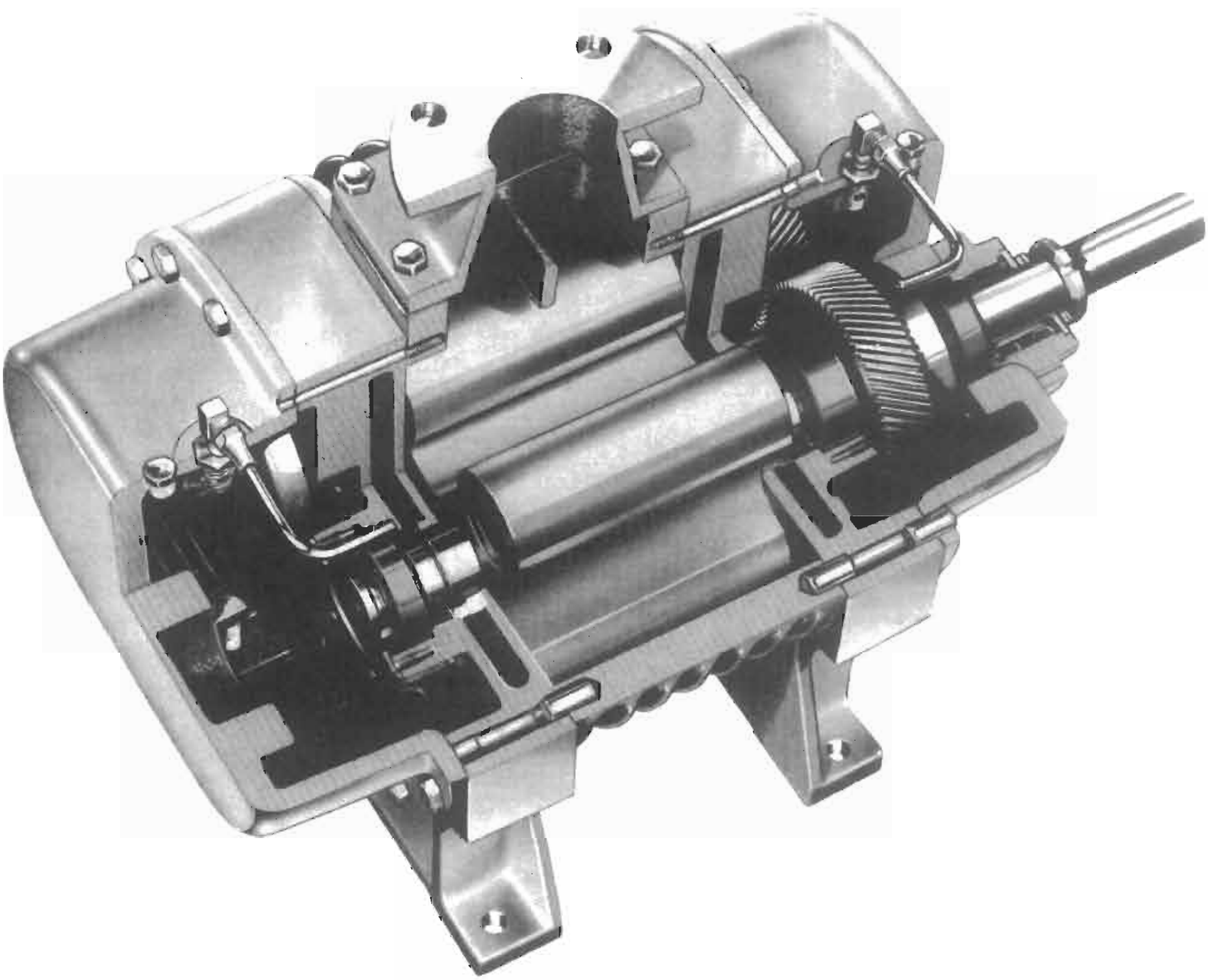


Figure 6-47. Cut-away view of internal assembly of rotary lobe vacuum pump. By permission, Tuthill Corp., M.D. Pneumatics Div., Bull. A-5/888.

seal or compressant water, and that the capability to hold a vacuum with higher seal water temperature is reduced. Typically, for a 95°F seal water, the vacuum may only be 26 to 27 inches Hg compared to the 28-inch Hg vacuum gauge shown. Also, care must be used to recognize that the curves represent inches of mercury, vacuum gauge, not absolute. To convert, use the 30-inch Hg barometer less the vacuum reading to attain absolute vacuum, inches Hg abs. The estimated pump-down capacity performance for a typical liquid-ring vacuum pump is given in Figure 6-38.

Rotary Vane Vacuum Pumps

Figure 6-39 illustrates a single-stage rotary vane vacuum pump, without external cooling jacket. The sliding vanes (No. 7 in illustration) are oiled by a closed system to aid in sealing the moving vane against the casing. The rotary shaft is off-center in the casing to provide a continuously decreasing volume from suction to discharge of the machine. Other styles of rotary vane units do not have suction or discharge valves. Note that the volume pumped is expressed as free air, which is measured at 60°F and 14.7 psia. Figure 6-40 illustrates a typical performance curve.

These pumps are relatively simple mechanical units compared to rotary-piston pumps. The "pumping compartment vanes" are spring loaded to hold them against the off-centered position in the casing/housing. Some designs do not use springs on the vanes, but rely on centrifugal force to position the sliding vanes sealing against the casing wall. These pumps are used for medium vacuum of less than 1 torr [20]. Also see Parkinson [34].

Units without an oil pump rely on once-through oiling for vacuum sealing. The oil usage is low for most units.

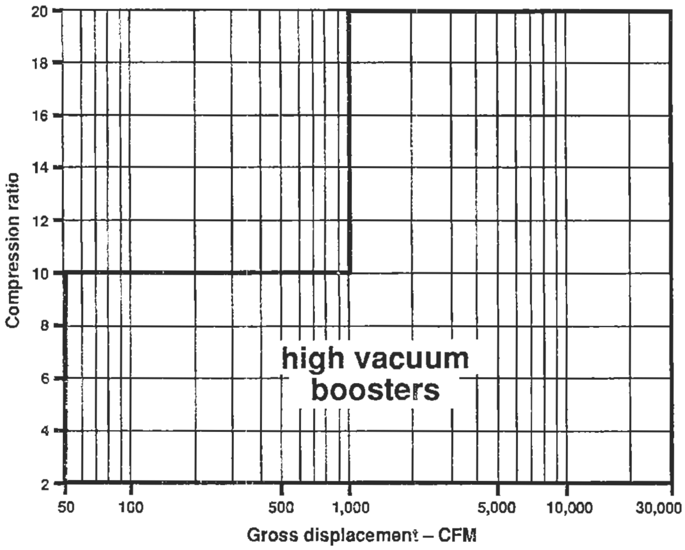


Figure 6-48A. Typical performance of high vacuum booster lobe-type high volume draw-down for evacuating vacuum systems before use of higher vacuum (lower absolute pressure) pump. By permission, Roots Division, Dresser Industries, Inc.

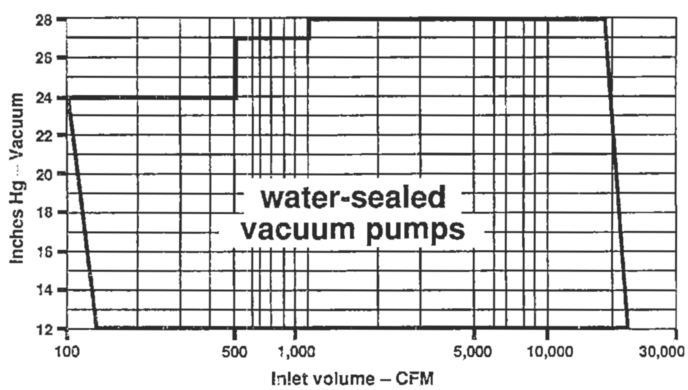


Figure 6-48B. Performance of lobe-type vacuum pump using water spray internally to reduce slip of gases from discharge to suction. By permission, Roots Division, Dresser Industries, Inc.

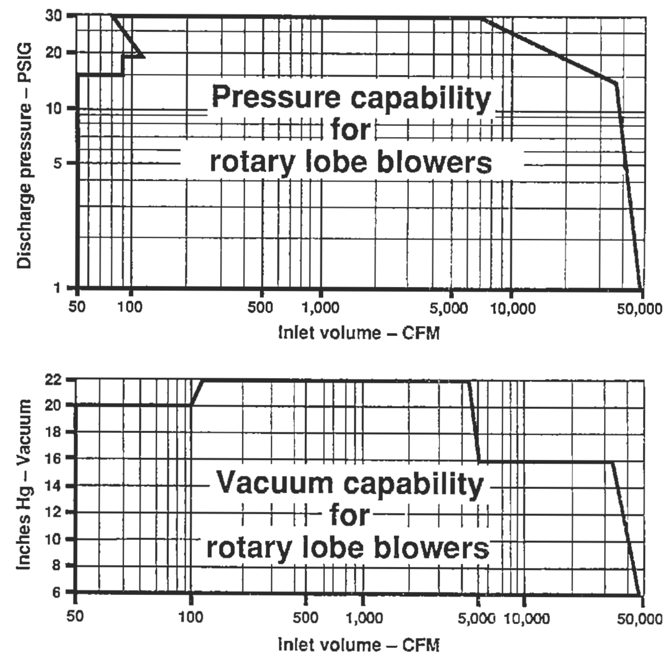


Figure 6-48C. Typical application rotary lobe vacuum blower performance. By permission, Roots Division, Dresser Industries, Inc.

These units cannot handle liquid slugs or dirt particles; hence, they require gas/vapor cleaning before entering the unit. For units with cooling jackets and automatic temperature regulation of the process gas, the temperature can range as high as 302°F [20].

By adjusting the jacket temperatures, condensation or polymerization can be avoided inside the pump. The volume handled Figure 6-40 is often expressed as "free air, CFM." This is simply the mechanical displacement of the pumping volume of the unit for the particular driven speed of the unit.

Rotary Blowers or Rotary Lobe-Type Blowers

These units are useful in both vacuum and pressure ranges and can usually develop compression ratios of 3:1 to 10:1, depending on the inlet absolute pressure. The units are positive displacement in performance.

The units use meshing balanced lobes (Figures 6-41, 6-42, 6-43, 6-44, 6-45, 6-46 A, B, C and 6-47) or screw-type rotors that are synchronized by timing gears and do not touch each other or the housing with very close tolerances. The shafts of the driving shaft and driven shaft are sealed with labyrinth type seals for a minimum leakage. Purged labyrinth or mechanical type seals are available.

The rotors do not require lubrication per se because they do not touch; however, some designs add a small amount of sealing liquid such as water or other compatible fluid to reduce the "slip" or backflow as the lobes rotate. Depending on the type of design, i.e., lobe or screw-type cycloidal or helical rotors, the discharge pressure may have some pulsation, such as −60% to 140+% of gas discharge pressure, or it may be essentially "smooth," almost pulsation free. The shafts of the rotors rotate in opposite directions by means of the drive gears. The air or process gas/vapor is drawn into the suction or inlet cavity of the intersecting and rotating "lobes" by the rotor mesh. As the rotors continue to rotate, the cavity of suction gas is sealed from the inlet by the moving and/or advancing "lobes" as they pass the fixed boundary of the inlet opening of the casing. As the rotors rotate, the gas is

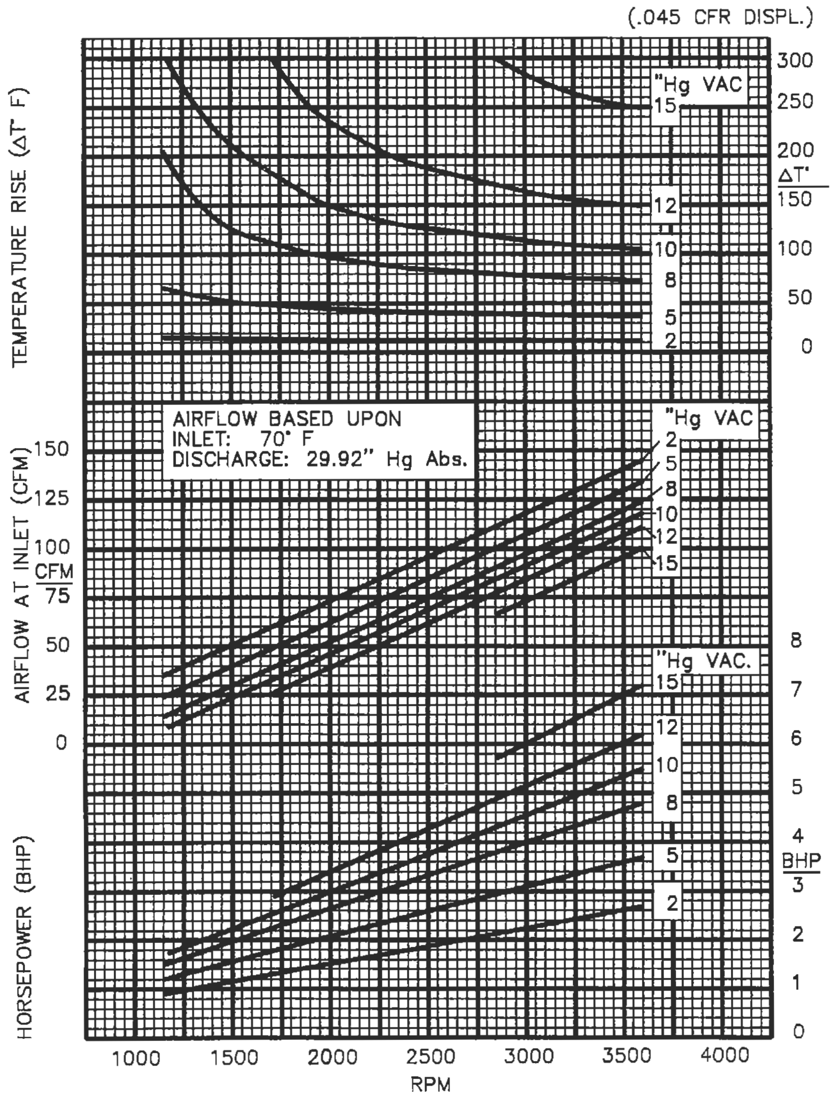


Figure 6-49. Typical performance curves for rotary lobe vacuum pump. By permission, Tuthill Corp., M.D. Pneumatics, Division.

compressed against the discharge headplate, similar in concept to a piston compressor against a piston compressor cylinder head [21].

Booster vacuum pumps are used to shorten the pump-down on evacuation (time) of a vacuum system before switching to the smaller vacuum pump to maintain the system opening vacuum and to handle the air inleakage to the system.

A typical performance range of capacities of rotary lobe vacuum pumps is shown in Figures 6-48A, 6-48B and 6-48C. Another set of curves for rotary lobe pumps, shown in Figure 6-49, provides the brake horsepower, airflow at inlet (CFM) (referred to as their standard volume at 70°F and 29.92 inch Hg abs discharge pressure—essentially atmosphere), and the temperature rise through a non-cooled (no internal or external cooling) vacuum. All data are referenced to the indicated vacuum, not absolute pressure. (See the beginning of this chapter for clarification).

Rotary Piston Pumps

These units are also positive displacement oil-sealed pumps. The pumping action is shown in Figure 6-50. The action of the rotating piston assembly draws in a set volume (depending on the pump's size/capacity) of gas, compresses it in the eccentric rotating cylinder against the inside wall of the casing, and exhausts it to atmosphere, as do most other vacuum pumping devices. As the oil-sealed piston revolves, it opens the inlet port, draws in gas, and traps it in the fixed suction space. As the piston rotates, the gas is compressed, and the discharge valve opens to discharge the gas. A single-stage pump can have

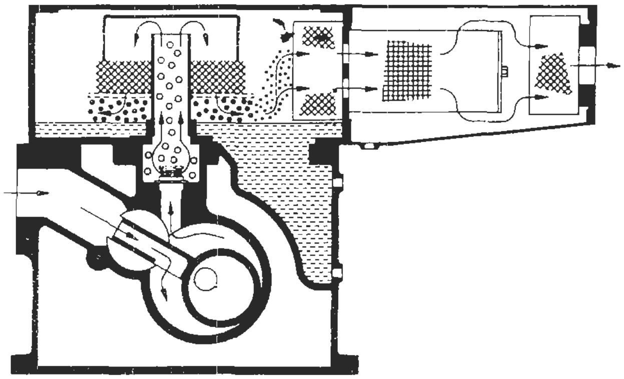


Figure 6-50. Typical rotary displacement vacuum pump, oil sealed, single-stage. By permission, Kinney Vacuum Co.

compression ratios up to 100,000:1 when discharging to atmosphere [22].

Mechanically, the pump operates in an oil bath, with the sealing oil lubricating the pump and seals against back flow from the exhaust to the intake/suction.

These pumps cannot effectively handle condensation of vapors inside the unit, because the capacity is reduced and the condensate creates lubrication problems, which in turn leads to mechanical breakdown.

To prevent/reduce the undesirable condensation in the pump, a small hole is drilled in the pump head to admit air or other process non-condensable gas (gas ballast) into the latter portion of the compression stroke. This occurs while the vapor being compressed is sealed off from the intake port by the piston. By reducing the partial pressure of the vapor's condensables, the condensation is avoided. Obviously, this can reduce the capacity of the pump, as the leakage past the seals allows the gas ballast to dilute the intake volume of base suction gas. For most process applications, the effect of this leakage is negligible, unless the vacuum system suction is below 1 torr [22].

Huff [23] found that reciprocating and rotary piston pumps were the most economical mechanical systems for their range of application. Obviously, the economic discussions are dependent on the vacuum expected and the local utility costs, plus the cost of maintenance.

Nomenclature

c_{pa} = Specific heat of air at constant pressure (0.24 approx.)
c_{ps} = Specific heat of steam at constant pressure corresponding to downstream absolute pressure (0.45 approx.)
D = Sealed diameter, inches (estimates of nominal diameter acceptable)
E = Evacuation factor, at final evacuation suction pressure, Tables 6-9 and 6-10
F = Steam pressure factor
h_L = Liquid height, ft
K = Non-condensable load factor
L = Latent heat of vaporization of steam, BTU/lb
M = Average mol weight of system vapors
M_n = Molecular weight of non-condensable gas
M_v = Molecular weight of condensable vapor
P = Total absolute pressure, lbs/sq in. absolute (or other consistent units), or system operating pressure, torr
P_a = Partial pressure of air in mixture, lbs/sq in. abs
P_c = Absolute intake pressure of pump
P_d = Piston displacement, cu ft/min
P_n = Partial pressure of non-condensable gas; pounds per square inch absolute (or other absolute units)
P_s = Static pressure, atm
P_v = Vapor pressure of condensable vapor, pounds per square inch absolute (or other absolute units)
P' = Partial pressure of a vapor in a mixture, psia
P_s' = Design suction pressure of ejector, torr
P_n'' = Final pressure in vessel or system, torr
P_o'' = Starting pressure in vessel or system, torr
P = Atmospheric pressure, mm Hg
P_1 = Intake pressure of pump, psia or, initial pressure in sys., in Hg abs. (Eq. 6-25)
P_2 = Discharge pressure of pump, psia
P_c = Intake pressure of pump with closed intake, psia
P_2 = Final pressure in system, in. Hg abs
R = Gas constant, = 1544/mol weight
R_{ps} = Pump speed, revolutions (or strokes) per second
S = Pump speed, liters/sec
S_n = Pump speed at P_n'', liters/sec
S_o = Pump speed at P_o'', liters/sec
T = Temperature, °R = 460 + °F
t = Evacuation pump downtime, min
t_s = Evacuation pump downtime, sec
t_a = Ambient air temperature, °F
t_m = Temperature of mixture at ejector suction, °F
t_s = Temperature of steam on downstream side of nozzle, °F
Δt_w = Temperature rise of water, °F
V = Volume of tank or system, cu ft
V' = Volume of vessel or system, liters
V_o' = Volume of pumping chambers, liters
W or W_a = Flow rate, lbs/hr
W_n = Weight of non-condensable gas, lbs/hr
W_m = Total pounds of mixture handled per hour
W_s = Total steam consumption, lbs/hr
W_v = Weight of condensable vapor, lbs/hr
W_t = Total weight of gas, lbs
W_{Ta} = Total calculated air inleakage, lbs/hr
or, W_T = Total calculated air inleakage, lbs/hr
W_a' = Air inleakage resulting from metal porosities and cracks along weld lines, lbs/hr
W_m' = Ejector capacity at final evacuation suction pressure, lbs/hr
W_s' = Pounds of motive steam per pound of mixture handled
W_v' = Pounds water vapor per pound air
w = Constant flow rate, lbs/min
w_a = Acceptable air leakage rate assigned to a system component, lbs/hr
w_j = Ejector capacity, 70°F dry air basis, lbs/hr

Greek Symbols

ρ = Density, lbs/cu ft
ρ_L = Specific gravity of liquid, relative to water = 1.0
θ = Specific air leakage rate, lb/hr/in.
π = Pi = 3.1418

Subscripts

a = air
1, 2, etc. = Refers to components in system, or initial and final system pressures, psia
90 = Steam pressure, psia
v = Condensable vapor
n = Non-condensable gas/vapor

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Chapter

7

Process Safety and Pressure-Relieving Devices

The subject of process safety is so broad in scope that this chapter must be limited to the application design, rating, and specifications for process over-pressure relieving devices for flammable vapors and dusts; process explosions and external fires on equipment; and the venting or flaring of emergency or excess discharge of gases to a vent flare stack. The subject of fire protection per se cannot be adequately covered because partial treatment would be "worse" than no treatment; therefore, the engineer is referred to texts dealing with the subject in a thorough manner [1, 30, 31, 32, 33, 34]. The important topic of steam boiler safety protection is not treated here for the same reason.

The possibilities for development of excess pressure exists in nearly every process plant. Due to the rapidly changing and improved data, codes, regulations, recommendations, and design methods, it is recommended that reference be made to the latest editions of the literature listed in this chapter. While attempting to be reliable in the information presented, this author cannot be responsible or liable for interpretation or the handling of the information by experienced or inexperienced engineers. This chapter's subject matter is vital to the safety of plants' personnel and facilities.

It is important to understand how the over-pressure can develop (source) and what might be the eventual results. The mere solving of a formula to obtain an orifice area is secondary to an analysis and understanding of the pressure system. Excess pressure can develop from explosion, chemical reaction, reciprocating pumps or compressors, external fire around equipment, and an endless list of related and unrelated situations. In addition to the possible injury to personnel, the loss of equipment can be serious and an economic setback.

Most states have laws specifying the requirements regarding application of pressure-relieving devices in process and steam power plants. In essentially every instance at least part of the reference includes the *A.S.M.E. Boiler and Pressure Vessel Code*, Section VIII, Division 1 (Pressure Vessels) and/or Division 2 [1]; and Section VII, *Recommended Rules for Care of Power Boilers* [2]. In addition the publications of the American Petroleum Institute are helpful in evaluation and design. These are API-RP-520 [10], *Design and Installation of Pressure-Relieving Systems in Refineries; Part I—Design; Part II—Installation;* and API-RP-521 [13], *Guide for Pressure Relief and Depressurizing Systems,* ANSI/ASME B31.1 *Power Piping;* B16.34; and NFPA [27], Sections 30, 68, and 69.

The ASME Code requires that all pressure vessels be protected by a pressure-relieving device that shall prevent the internal pressure from increasing more than 10% above the maximum allowable working pressures of the vessel (MAWP) to be discussed later. Except where multiple relieving devices are used, the pressure shall not increase more than 16% above the MAWP or, where additional pressure hazard is created by the vessel being exposed to external heat (not process related) or fire, supplemented pressure relieving devices must be installed to prevent the internal pressure from rising more than 21% above the MAWP. See Ref. [1] sections U-125 and UG-126. The best practice in industrial design recommends that (a) *all* pressure vessels of *any* pressure be designed, fabricated, tested and code stamped per the applicable ASME code [1] or American Petroleum Institute (API) Codes and Standards, Ref. [33] and (b) that pressure relieving devices be installed for pressure relief and venting per codes [1, 10, 13] [33].

Although not specifically recognized in the titles of the codes, the rupture disk as a relieving device, is, nevertheless, included in the requirements as an acceptable device.

Usual practice is to use the terms safety valve or relief valve to indicate a relieving valve for system overpressure, and this will be generally followed here. When specific types of valves are significant, they will be emphasized.

Types of Positive Pressure Relieving Devices (see manufacturers' catalogs for design details)

Relief Valve: a relief valve is an automatic spring loaded pressure-relieving device actuated by the static pressure upstream of the valve, and which opens further with increase in pressure over the opening pressure. It is used primarily for liquid service [1,10] (Figure 7-1A and 7-1B). Rated capacity is usually attained at 25 percent overpressure.

Safety Valve: this is an automatic pressure-relieving device actuated by the static pressure upstream of the valve and characterized by rapid full opening or "pop" action upon opening [1,10], but does not reseat. It is used for steam or air service (Figure 7-2). Rated capacity is reached at 3%, 10% or 20% overpressure, depending upon applicable code.

Safety-Relief Valve: this is an automatic pressure-relieving device actuated by the static pressure upstream of the valve and characterized by an adjustment to allow reclosure, either a "pop" or a "non-pop" action, and a nozzle type entrance; and it reseats as pressure drops. It is used on steam, gas, vapor and liquid (with adjustments), and is probably the most general type of valve in petrochemical and chemical plants (Figures 7-3, 7-3A, and 7-4). Rated capacity is reached at 3% or 10% overpressure, depending upon code and/or process conditions. It is suitable for use either as a safety or a relief valve [1,10]. It opens in proportion to increase in internal pressure.

Pressure Relief Valve

The term Pressure-relief valve applies to relief valves, safety valves or safety-relief valves [10].

Pilot Operated Safety Valves

When properly designed, this type of valve arrangement conforms to the ASME code. It is a pilot operated pressure relief valve in which the major relieving device is combined with and is controlled by a self-activating auxiliary pressure relief valve. See Figures 7-5A and B.

Types of Valves/Relief Devices

There are many design features and styles of safety relief valves, such as flanged ends, screwed ends, valves fitted internally for corrosive service, high temperature service, cryogenic service/low temperatures, with bonnet or without, nozzle entrance or orifice entrance, and resistance to discharge piping strains on body. Yet most of these variations have little, if anything to do with the actual performance to relieve overpressure in a system/vessel.

A few designs are important to the system arrangement and relief performance:

Conventional Safety Relief Valve

This valve design has the spring housing vented to the discharge side of the valve. The performance of the valve upon relieving overpressures is directly affected by any changes in the backpressure on the valve (opening pressure, closing pressure, relieving capacity referenced to opening pressure) [35]. See Figures 7-3, 7-6, and 7-6A. When connected to a multiple relief valve manifold, the performance of the valve can be somewhat unpredictable from a relieving capacity standpoint due to the varying backpressure in the system.

Balanced Safety Relief Valve

This valve provides an internal design (usually bellows) above/on the seating disk in the huddling chamber that minimizes the effect of backpressure on the performance of the valve (opening pressure, closing pressure and relieving capacity) [35]. See Figures 7-4, 7-6, and 7-6A.

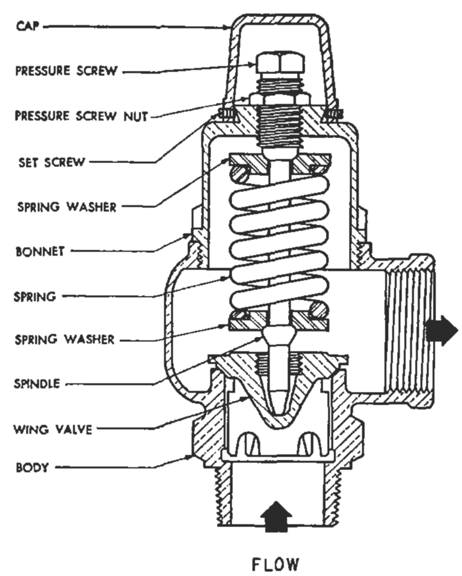


Figure 7-1A. Relief valve. Courtesy of Crosby-Ashton Valve Co.

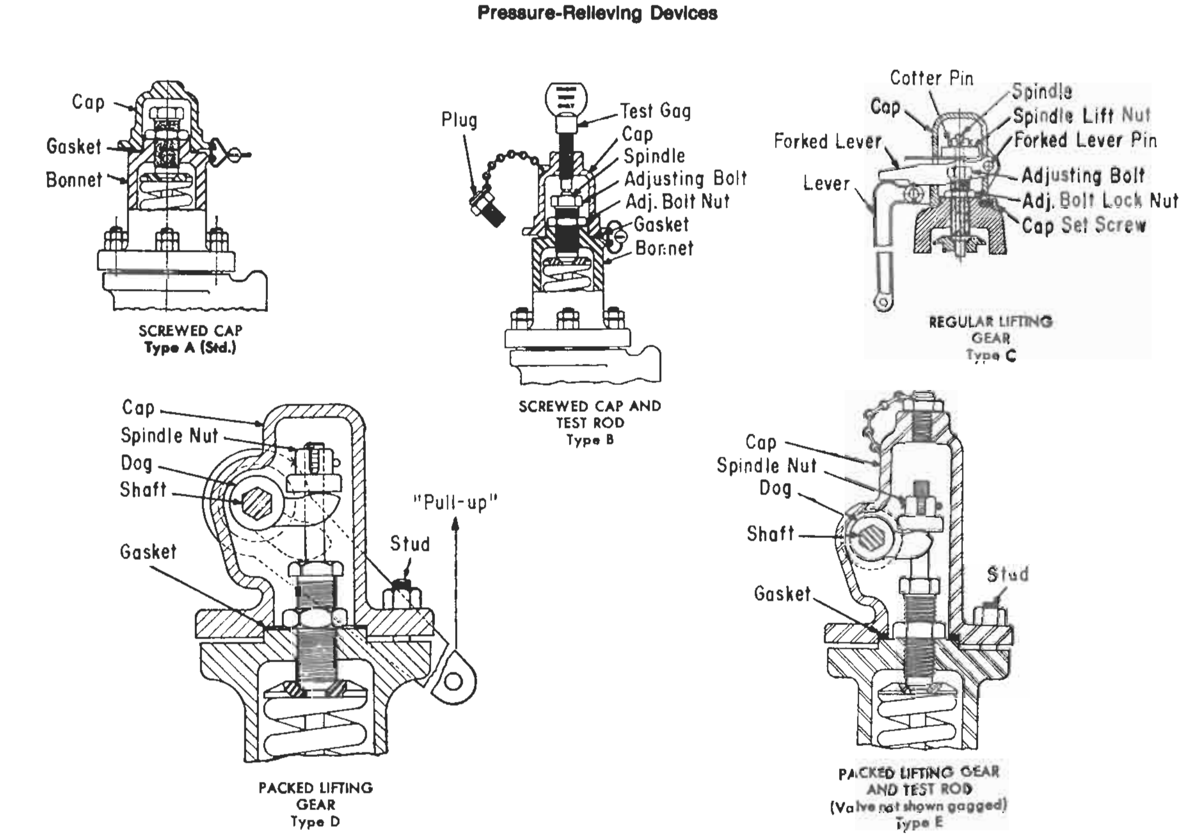


Figure 7-1B. Accessories for all types of safety relieving valves. Courtesy of Crosby-Ashton Valve Co.

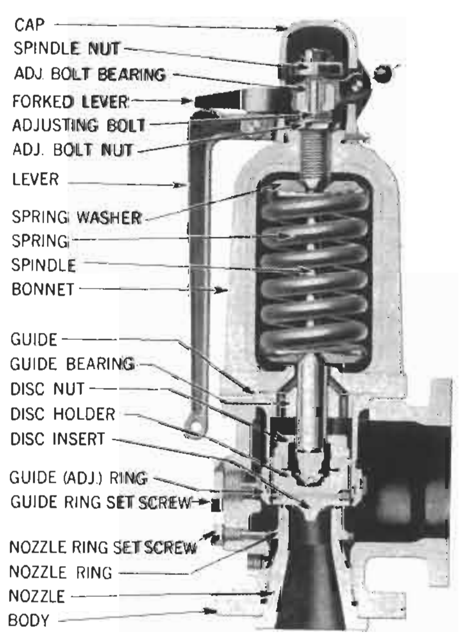


Figure 7-2. Safety valve. Courtesy of Crosby-Ashton Valve Co.

Special Valves

a. Internal spring safety relief valve
b. Power actuated pressure relief valve
c. Temperature actuated pressure relief valve

These last three are special valves from the viewpoint of chemical and petrochemical plant applications, but they can be designed by the major manufacturers and instrumentation manufacturers as these are associated with instrumentation controls. Care must be taken in the system design to make certain it meets all ASME code requirements.

Rupture Disk

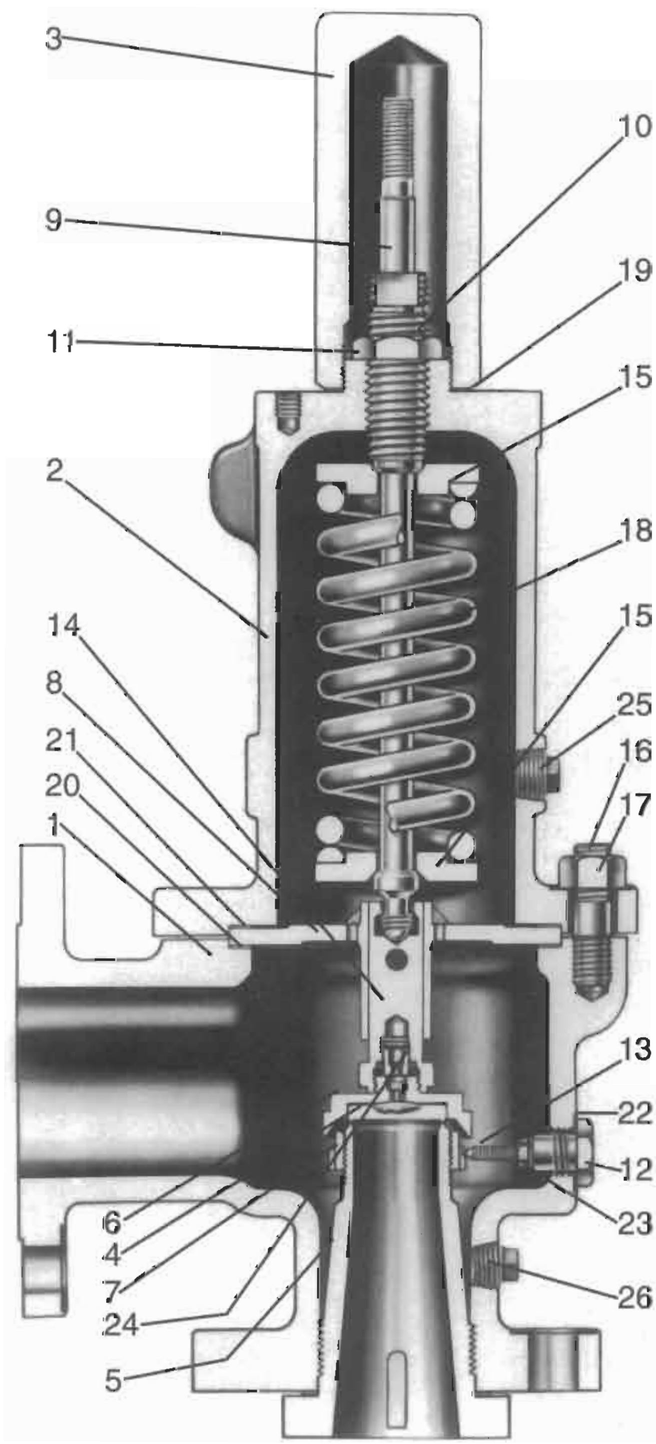
A rupture disk is a non-reclosing thin diaphragm (metal, plastic, carbon/graphite (non-metallic)) held between flanges and designed to burst at a predetermined internal pressure. Each bursting requires the installation of a new disk. It is used in corrosive service, toxic or "leak-proof" applications, and for required bursting pressures not easily accommodated by the conventional valve such as explosions. It is applicable to steam, gas, vapor, and liquid systems. See Figures 7-7B, 7-8A through R, and 7-9A through F. There are at least four basic types of styles of disks, and each requires specific design selection attention.

An explosion rupture disc is a special disc (or disk designed to rupture at high rates of pressure rise, such a run-away reactions. It requires special attention from th manufacturer [35].

Other rupture devices suitable for certain applicatio are [35]:

a. breaking pin device
b. shear pin device
c. fusible plug device

Pressure-Vacuum Valves: See page 458



Bill of Materials-Conventional

ITEM	PART NAME		MATERIAL
1	Body	26( )A10 thru 26( )A26	SA-216 GR. WCB, Carbon Steel
		26( )A32 thru 26( )A36	SA-217 GR. WC6, Alloy St. (1¼ CR—½ Moly)
2	Bonnet	26( )A10 thru 26( )A26	SA-216 GR. WCB, Carbon Steel
		26( )A32 thru 26( )A36	SA-217 GR. WC6, Alloy St. (1¼ CR—½ Moly)
3	Cap, Plain Screwed		Carbon Steel
4	Disc		Stainless Steel
5	Nozzle		316 St. St.
6	Disc Holder		300 Series St. St.
7	Blow Down Ring		300 Series St. St.
8	Sleeve Guide		300 Series St. St.
9	Stem		Stainless Steel
10	Spring Adjusting Screw		Stainless Steel
11	Jam Nut (Spr. Adj. Scr.)		Stainless Steel
12	Lock Screw (B.D.R)		Stainless Steel
13	Lock Screw Stud		Stainless Steel
14	Stem Retainer		Stainless Steel
15	Spring Button		Carbon Steel Rust Proofed
16	Body Stud		ASTM A193 Gr. B7, Alloy St.
17	Hex Nut (Body)		ASTM A194 Gr. 2H, Alloy St.
18	Spring	26( )A10 thru 26( )A16	Carbon Steel Rust Proofed
		26( )A20 thru 26( )A36	High Temp. Alloy Rust Proofed
19	Cap Gasket		Soft Iron or Steel
20	Body Gasket		Soft Iron or Steel
21	Bonnet Gasket		Soft Iron or Steel
22	Lock Screw Gasket		Soft Iron or Steel
23	Hex Nut (B.D.R.L.S.)		Stainless Steel
24	Lock Screw (D.H.)		Stainless Steel
25	Pipe Plug (Bonnet)		Steel
26	Pipe Plug (Body)		Steel

Also suitable for liquid service where ASME Code certification is not required.

Figure 7-3. Conventional or unbalanced nozzle safety relief valve. By permission, Teledyne Farris Engineering Co.

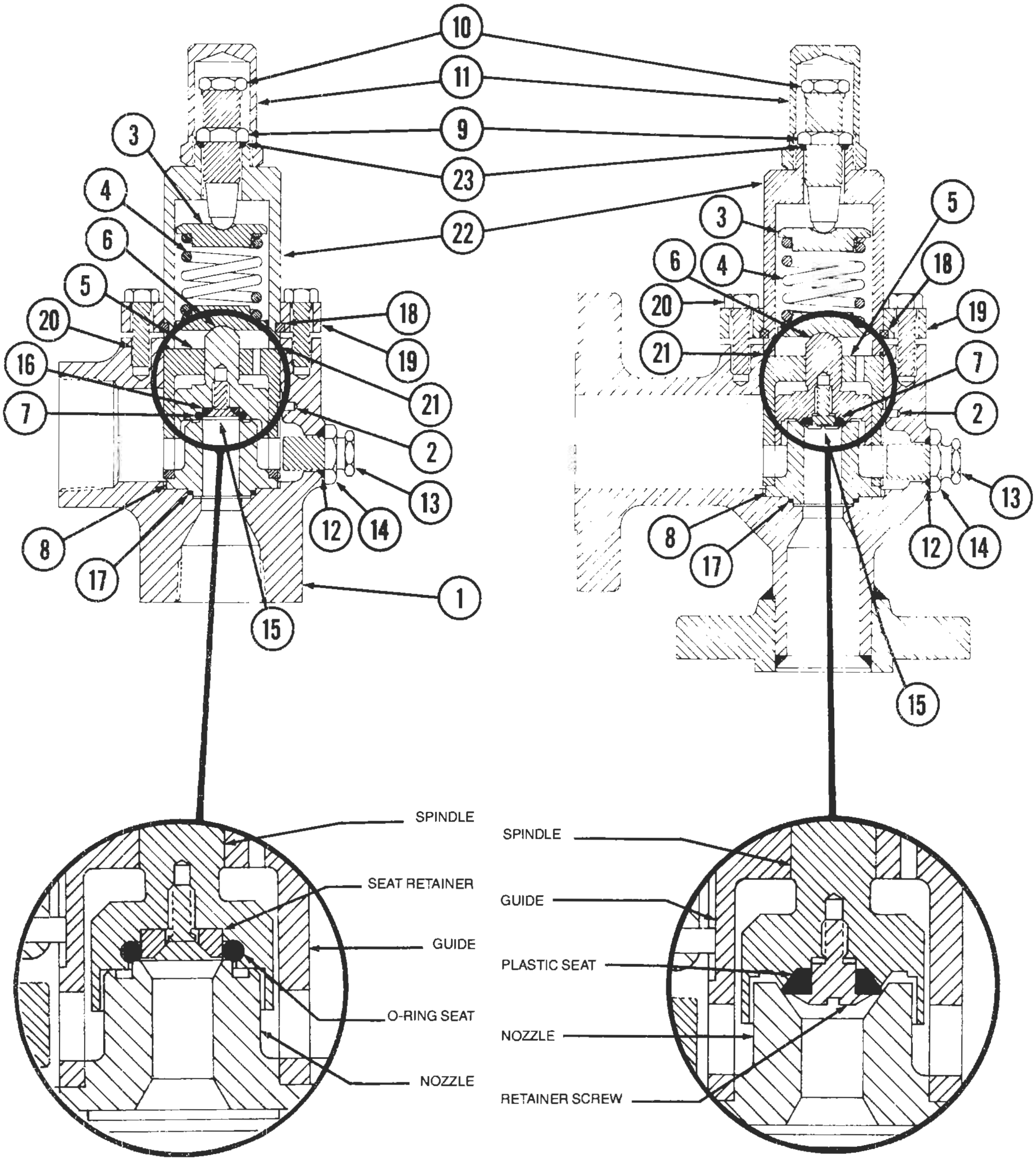


Figure 7-3A. Safety relief valve with rubber or plastic seats. By permission, Anderson, Greenwood and Co.

Definition of Pressure-Relief Terms (See Figures (7-3, 7-3A, 7-4, and Ref. [35])

Set Pressure: the set pressure, in pounds per square inch gauge, is the inlet pressure at which the safety or relief valve is adjusted to open [10,13]. This pressure is set regardless of any back pressure on the discharge of the valve, and is not to be confused with a manufacturer's spring setting.

Overpressure: pressure increase over the set pressure of the primary relieving device is overpressure. It is the same as accumulation when the relieving device is set at the maximum allowable working pressure of the vessel [10].

Accumulation: pressure increase over the maximum allowable working pressure of the vessel *during* discharge through the safety or relief valve, expressed as a percent

Figure 7-3a cont.

ITEM	PART NAME	MATERIAL	
		81C, 83C	81S, 83S
1	BODY	CARBON STEEL	316 S.S.
2	DRIVE PIN	316 S.S.	316 S.S.
3	SPRING WASHER	CARBON STEEL	316 S.S.
4	SPRING	CARBON STEEL	316 S.S.
5	GUIDE	303 S.S.	316 S.S.
6	SPINDLE	81C-17-4 S.S. 83C-303 S.S.	81S-17-4 S.S. 83S-316 S.S.
7	SEAT	81C-TEFLON, KEL-F, OR VESPEL 83C-BUNA-N	81S-TEFLON,KEL-F, OR VESPEL 83S-BUNA-N
8	NOZZLE	81C-17-4 S.S. 83C-303 S.S.	81S-17-4 S.S. 83S-316 S.S.
9	LOCK NUT	CARBON STEEL	316 S.S.
10	PRESS. ADJ.	CARBON STEEL	316 S.S.
11	CAP	CARBON STEEL	316 S.S.
12	SEAL, BLOWDOWN	TEFLON	TEFLON
13	BLOWDOWN ADJ.	CARBON STEEL	316 S.S.
14	JAM NUT	CARBON STEEL	316 S.S.
15	RETAINER SCREW	81C-17-4 S.S. 83C-CARBON STEEL	81S-17-4 S.S. 83S-316 S.S.
16	RETAINER	303 S.S.	316 S.S.
17	SEAL, NOZZLE	81C—BUNA-N* 83C—BUNA-N*	81S—TEFLON** 83S—BUNA-N*
18	SPLIT RING	CARBON STEEL	316 S.S.
19	BONNET FLANGE	CARBON STEEL	316 S.S.
20	FLANGE BOLTS	CARBON STEEL	316 S.S.
21	SEAL, BONNET	81C—BUNA-N* 83C—BUNA-N*	81S—TEFLON** 83S—BUNA-N*
22	BONNET	CARBON STEEL	316 S.S.
23	PRESSURE SEAL	TEFLON	TEFLON

*Teflon or Viton also available.
**Buna-N or Viton also available.

of that pressure, pounds per square inch, is called accumulation [10].

Blowdown: blowdown is the difference between the set pressure and the reseating pressure of a safety or relief valve, expressed as a percent of the set pressure, or pounds per square inch [10].

Back Pressure: pressure existing at the outlet or discharge connection of the pressure-relieving device, resulting from the pressure in the discharge system of the installed device [35]. This pressure may be only atmospheric if discharge is directly to atmosphere, or it may be some positive pressure due to pressure drops of flow of discharging vapors/gases (or liquids where applicable) in the pipe collection system, which in turn may be connected to a blowdown or flare system with definite back-pressure conditions during flow, psig (gauge). The pressure drop during *flow* discharge from the safety relief valve is termed "built-up Back pressure."

Burst Pressure: the inlet static pressure at which a rupture disk pressure-relieving device functions or opens to release internal pressure.

Design Pressure: the pressure used in the vessel design to establish the minimum code permissible thickness for containing the pressure.

Maximum Allowable Working Pressure (MAWP): the maximum pressure pounds per square inch gauge permissible at the top of a completed vessel in its operating position for a specific designated temperature corresponding to the MAWP pressure. This pressure is calculated in accordance with the ASME code (Par. UG-98) [1] for all parts or elements of the vessel using closest next larger to calculated value nominal thickness (closest standard for steel plate) (see Par. UG-A22) but exclusive of any corrosion allowance or other thickness allowances for loadings (see ASME Par.-UG-22) on vessels other than pressure (for example, extreme wind loadings for tall vessels). The

Bill of Materials-BalanSeal

ITEM	PART NAME		MATERIAL
1	Body	26( )B10 thru 26( )B26	SA-218 GR. WCB, Carbon Steel
		26( )B32 thru 26( )B36	SA-217 GR. WC6, Alloy St. (1¼ CR—½ Moly)
2	Bonnet	26( )B10 thru 26( )B26	SA-216 GR. WCB, Carbon Steel
		26( )B32 thru 26( )B36	SA-217 GR. WC6, Alloy St. (1¼ CR—½ Moly)
3	Cap. Plain Screwed		Carbon Steel
4	Disc		Stainless Steel
5	Nozzle		316 St. St.
6	Disc Holder		300 Series St. St.
7	Blow Down Ring		300 Series St. St.
8	Sleeve Guide		300 Series St. St.
9	Stem		Stainless Steel
10	Spring Adjusting Screw		Stainless Steel
11	Jam Nut (Spr. Adj. Scr.)		Stainless Steel
12	Lock Screw (B.D.R)		Stainless Steel
13	Lock Screw Stud		Stainless Steel
14	Stem Retainer		Stainless Steel
15	Bellows		316L St. St.
16	Bellows Gasket		Flexible Graphite
17	Spring Button		Carbon Steel Rust Proofed
18	Body Stud		ASTM A193 Gr. B7, Alloy St.
19	Hex Nut (Body)		ASTM A194 Gr. 2H, Alloy St.
20	Spring	26( )B10 thru 26( )B16	Carbon Steel Rust Proofed
		26( )B20 thru 26( )B36	High Temp. Alloy Rust Proofed
21	Cap Gasket		Soft Iron or Steel
22	Body Gasket		Soft Iron or Steel
23	Bonnet Gasket		Soft Iron or Steel
24	Lock Screw Gasket		Soft Iron or Steel
25	Hex Nut (B.D.R.L.S.)		Stainless Steel
26	Lock Screw (D.H.)		Stainless Steel
27	Pipe Plug (Body)		Steel

Also suitable for liquid service where ASME Code certification is not required.

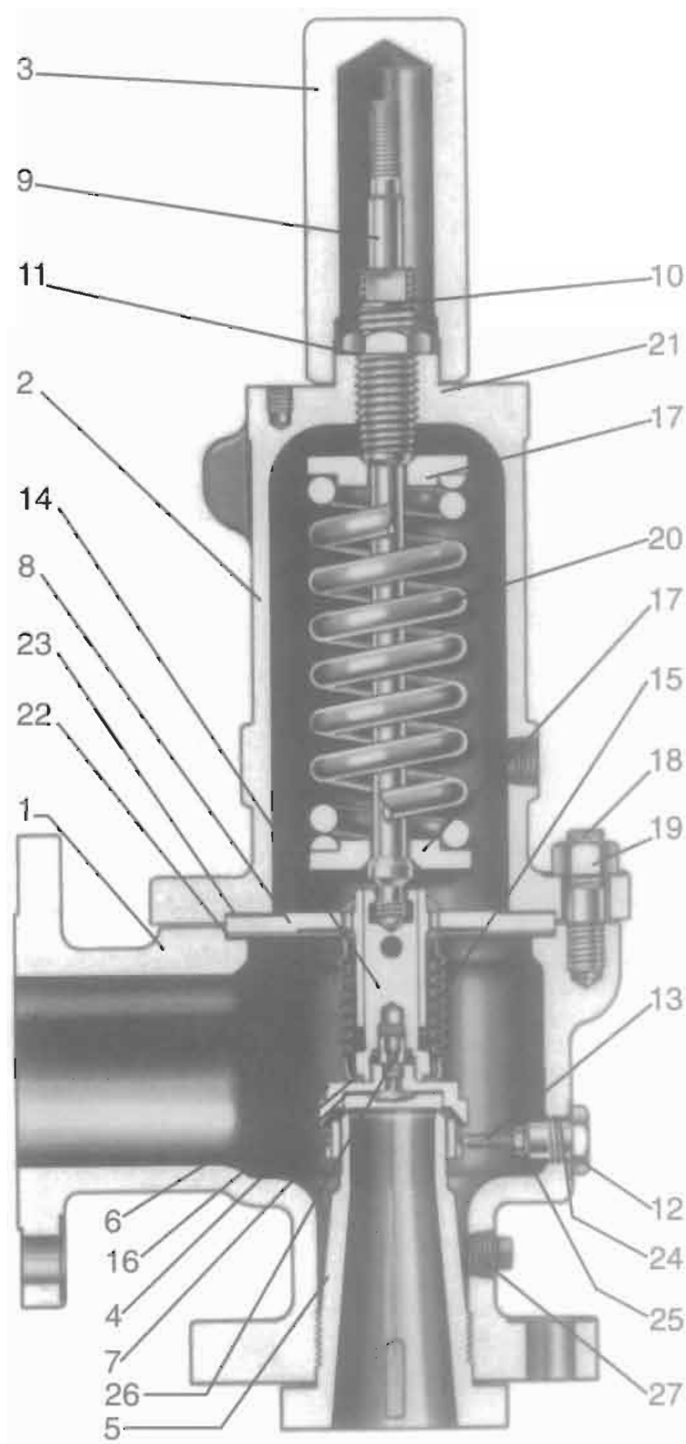
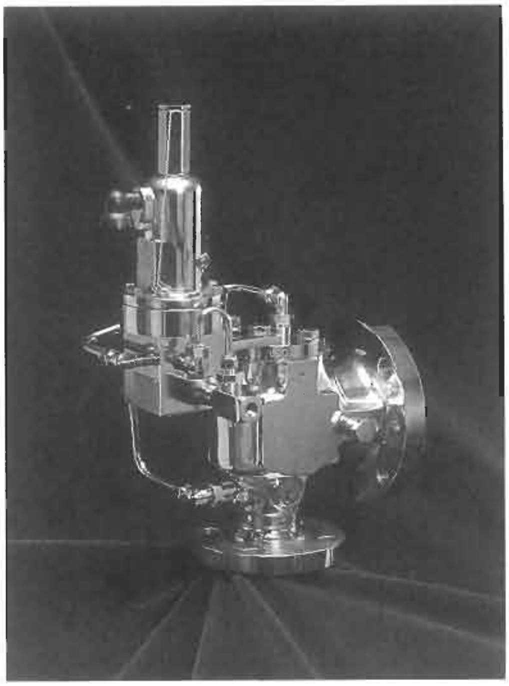
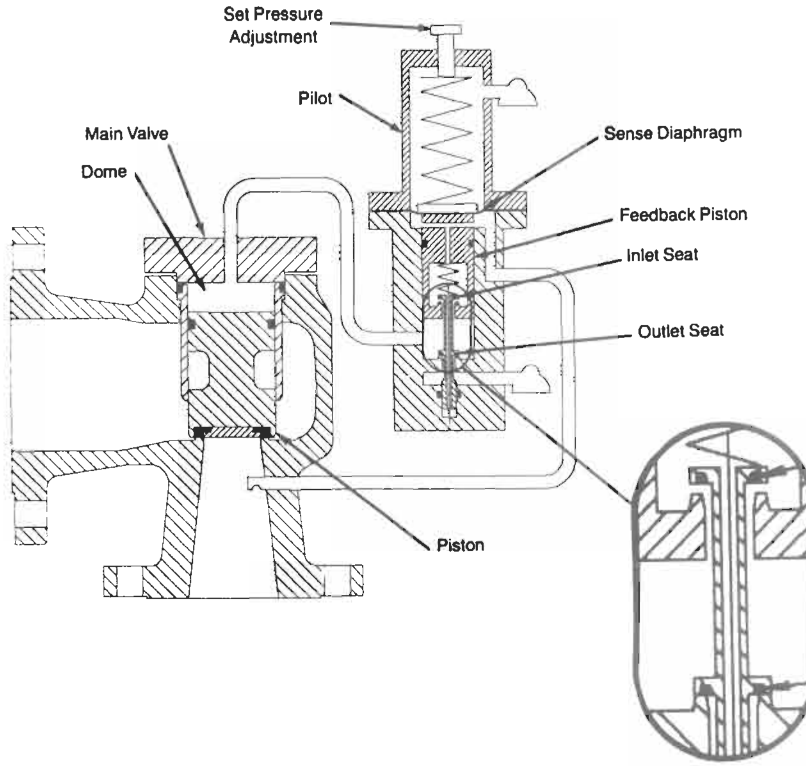


Figure 7-4. Balanced nozzle safety relief valve, Balanseal®. By permission, Teledyne Farris Engineering Co.





With no system pressure, the pilot inlet seat is open and outlet seat is closed. As pressure is admitted to the main valve inlet, it enters the pilot through a filter screen and is transmitted through passages in the feedback piston, past the inlet seat, into the main valve dome to close the main valve piston.

As system pressure increases and approaches valve set pressure, it acts upward on the sense diaphragm, with the feedback piston moving upward to close the inlet seat, thus sealing in the main valve dome pressure, as the outlet seat is also closed. A small, further increase in system pressure opens the outlet seat, venting the main valve dome pressure. This reduced dome pressure acts on the unbalanced feedback piston to reduce feedback piston lift, tending to "lock in" the dome pressure. Thus, at **any** stable inlet pressure there will be no pilot flow (i.e. zero leakage).

As inlet pressure rises above set pressure, dome pressure reduction will be such as to provide modulating action of the main valve piston proportional to the process upset. The spool/feedback piston combination will move, responding to system pressure, to alternately allow pressure in the main valve dome to increase or decrease, thus moving the main valve piston to the exact lift that will keep system pressure constant at the required flow. Full main valve lift, and therefore full capacity, is achieved with 5% overpressure. As system pressure decreases below set pressure, the feedback piston moves downward and opens the inlet seat to admit system pressure to the dome, closing the main valve.

Due to the extremely small pilot flow, the pilot on gas/vapor valves normally discharge to atmosphere through a weather and bug-proof fitting. Pilots for liquid service valves have their discharge piped to the main valve outlet.

Figure 7-5A. Pilot operated safety relief valve. By permission, Anderson, Greenwood and Co.

MAWP is calculated using nominal standard steel plates (but could be other metal—use code stresses) thickness, using maximum vessel operating temperature for metal stress determinations. See Ref [1] Par. UG-98.

Example 7-1: Hypothetical vessel design, carbon steel grade A-285, Gr C

Normal operating: 45 psig at 600°F

Design pressure: 65 psig at 700°F corres. to the 65 psig.

Assume calculated thickness per ASME code Par. UG-27: 0.43 in.

Closest standard plate thickness to fabricate vessel is 0.50 in. with − 0.01 in. and + 0.02 in. tolerances at mill.

Then

1. Using 0.50 in. − 0.01 in. (tolerance) = 0.49 in. min. thickness.

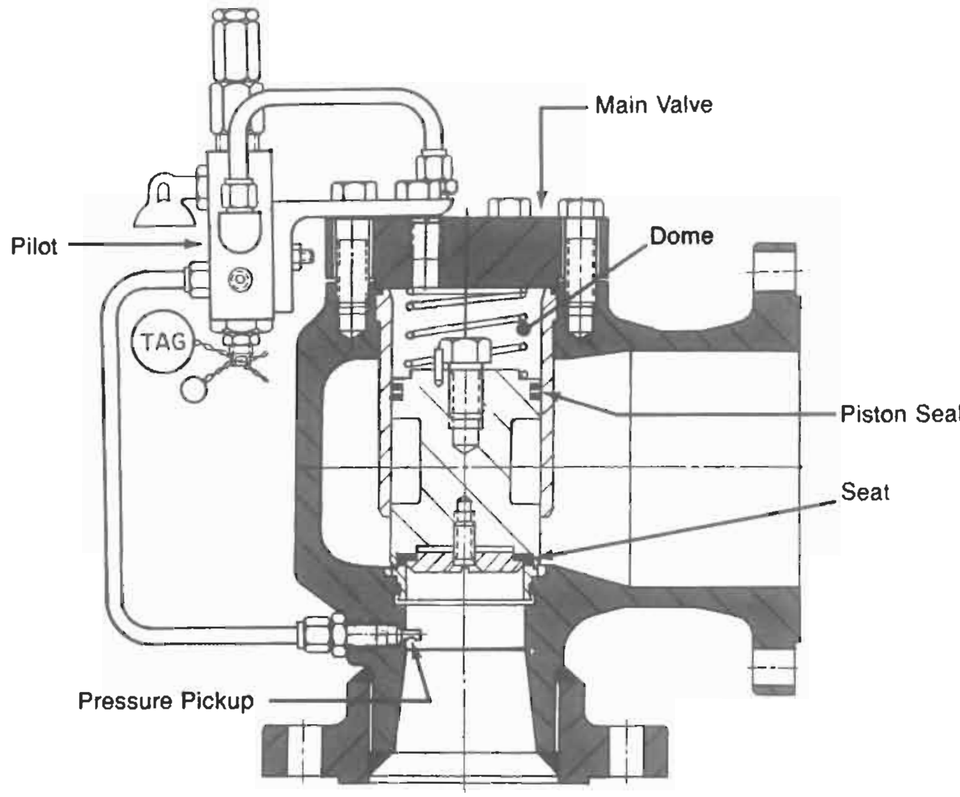


Figure 7-5B. Safety relief valve mechanism as connected to a non-flow (zero flow) pilot safety relief valve. By permission, Anderson Greenwood and Co.

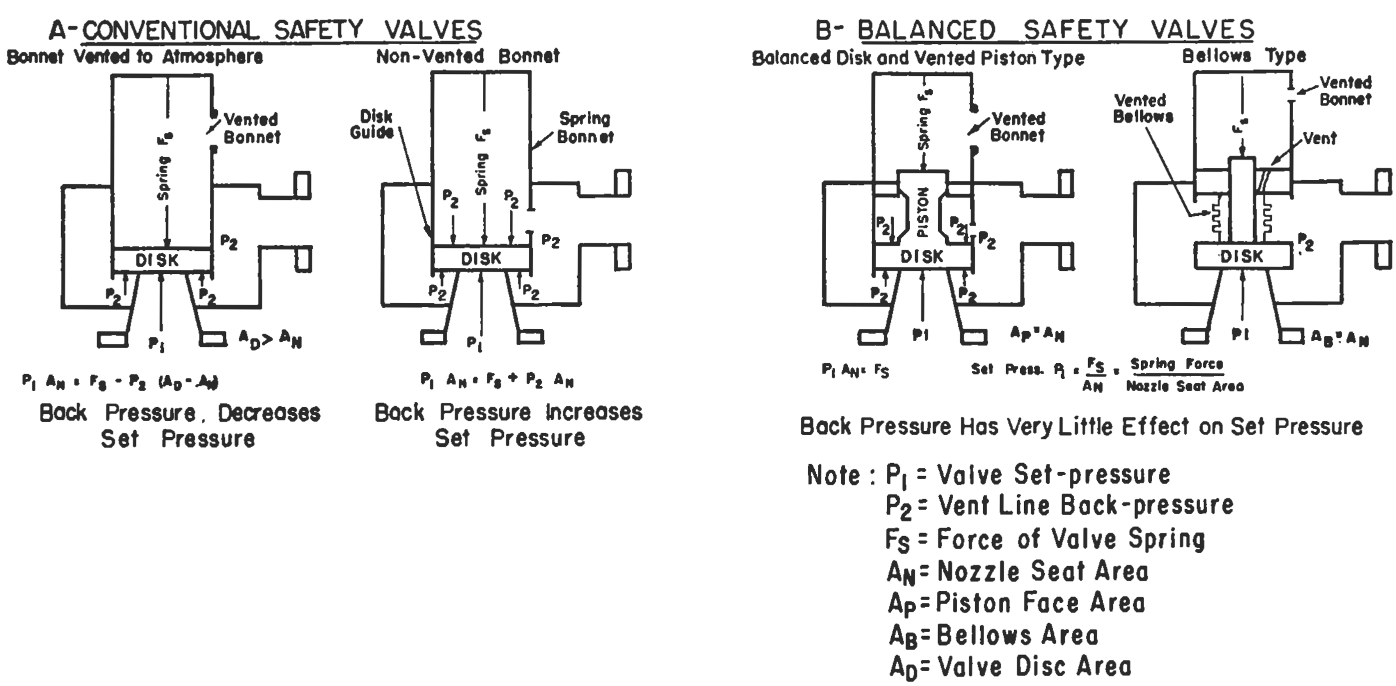


Figure 7-6. Effect of backpressure on set pressure of safety or safety relief valves. By permission, *Recommended Practice for Design and Construction of Pressure-Relieving Systems in Refineries,* API RP-520, 5th Ed. American Petroleum Institute (1990) (also see Ref. [33a]).

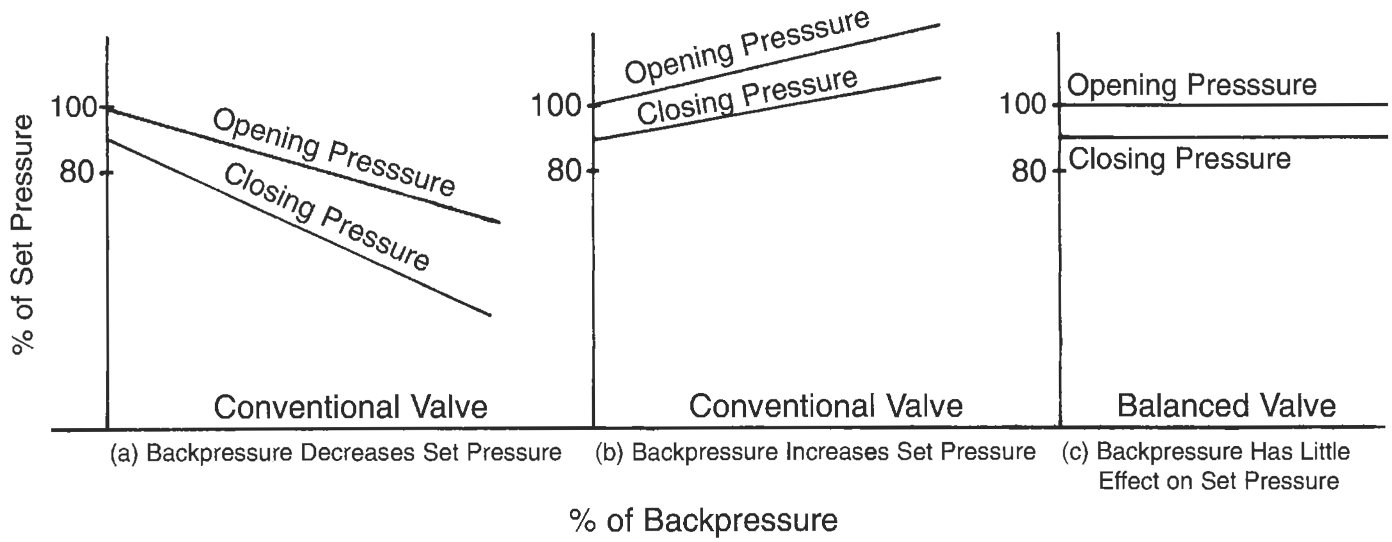


Figure 7-6A. Diagram of approximate effects of backpressure on safety relief valve operation. Adapted by permission, Teledyne Farris Engineering Co.

2. Using 0.50 in. + 0.02 in. (tolerance) = 0.52 in. max. thickness.

Generally, for design purposes, with this type of tolerance, nominal thickness = 0.50 in. can be used for calculations.

Now, using Par. UG-27, 0.50 in. thickness and ASME code stress at 750°F (estimated or extrapolated) per Par. UCS-23 at 750°F, the maximum allowable stress in tension is 12,100 psi.

Recalculate pressure (MAWP) using Par. UG-27 [1]

For cylindrical shells under *internal* pressure:

(1) Circumferential stress (longitudinal joint)

$$P_d = SEt/(R_i + 0.6t),\ psi = psig \quad (7\text{-}1)$$

$$t = PR/[SE - 0.6P] \quad (7\text{-}2)$$

where t = minimum actual plate thickness of shell, no corrosion, = 0.50″

P_d = design pressure, for this example equals the MAWP, psi

R_i = inside radius of vessel, no corrosion allowance added, in.

S = maximum allowable stress, psi, from Table UCS-23

E = joint efficiency for welded vessel joint, plate to plate to heads. See ASME Par. UW-12, nominal = 85% = 0.85

t = required thickness of shell, exclusive of corrosion allowance, inches

(2) Longitudinal stress (circumferential joints).

$$P_d = 2SEt/(R - 0.4t) \quad (7\text{-}3)$$

$$t = PR/[2SE + 0.4P] \quad (7\text{-}4)$$

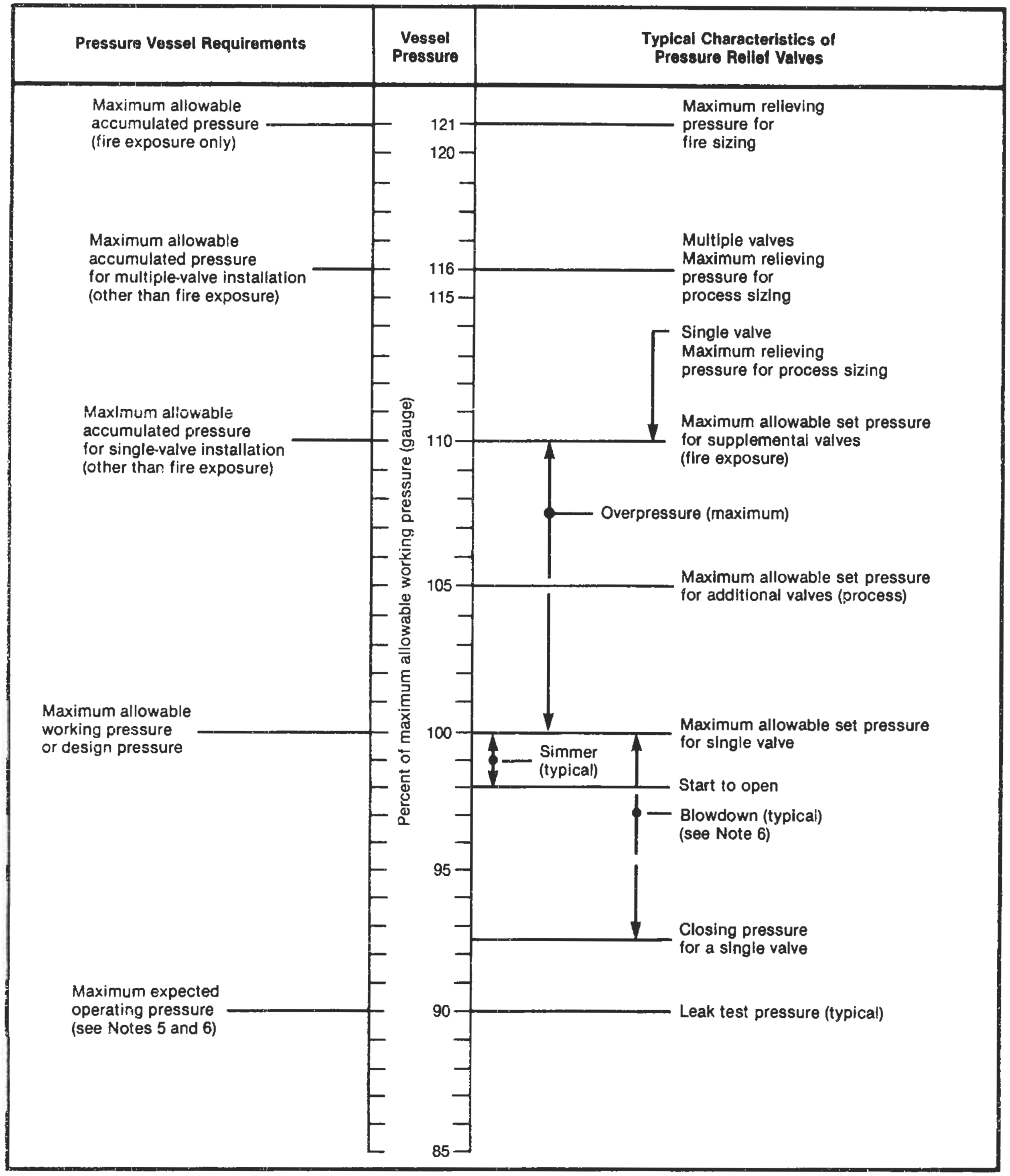
The vessel shell wall thickness shall be the greater of Equations 7-2 or 7-4, or the pressure shall be the lower of Equation 7-1 or 7-3 [1].

For above example *assume* calculated MAWP (above) = 80 psig. *This is the maximum pressure that any safety relief valve can be set to open.*

For pressure levels for pressure relief valves referenced to this MAWP, see Figures 7-7A and B.

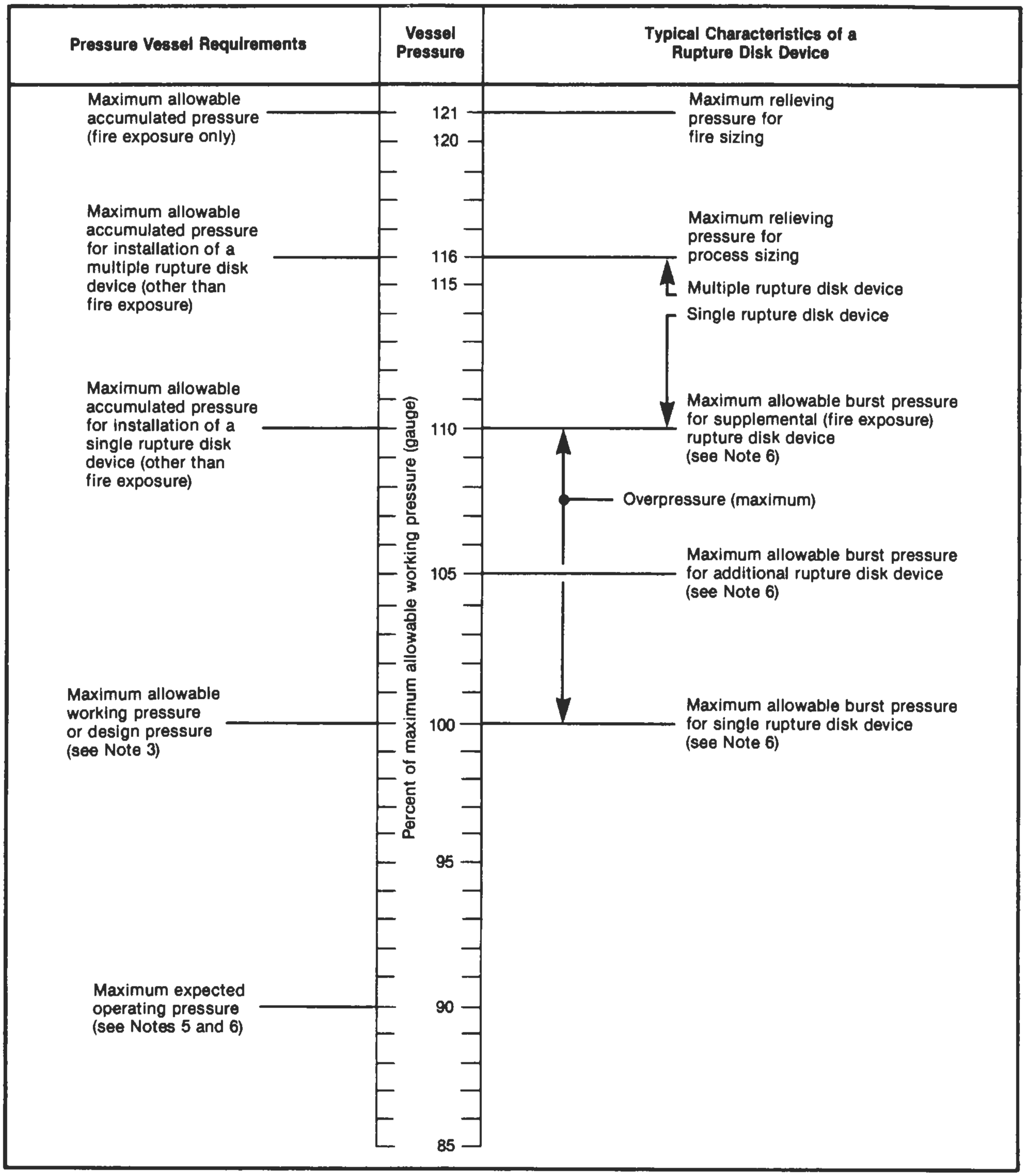
Operating Pressure: the pressure, psig, to which the vessel is expected to be subjected during *normal* or, the maximum probable pressure *during* upset operations. There is a difference between a pressure generated internally due to controlled rising vapor pressure (and corresponding temperature) and that generated due to an unexpected runaway reaction, where reliance must depend on the sudden release of pressure at a code conformance pressure/temperature. In this latter case, careful examination of the possible conditions for a runaway reaction should be made. This examination is usually without backup data or a firm basis for calculating possible maximum internal vessel pressure to establish a maximum operating pressure and from this, a design pressure.

Design Pressure of a Vessel: the pressure established as a nominal maximum *above* the expected process maximum operating pressure. This design pressure can be established by reference to the chart in Chapter 1, which is based on experience/practice and suggests a percentage increase of the vessel design pressure above the expected



Notes:
1. This figure conforms with the requirements of Section VIII of the ASME *Boiler and Pressure Vessel Code.*
2. The pressure conditions shown are for pressure relief valves installed on a pressure vessel.
3. Allowable set-pressure tolerances will be in accordance with the applicable codes.
4. The maximum allowable working pressure is equal to or greater than the design pressure for a coincident design temperature.
5. The operating pressure may be higher or lower than 90.
6. Section VIII, Division 1, Appendix M, of the ASME Code should be referred to for guidance on blowdown and pressure differentials.

Figure 7-7A. Pressure level relationship conditions for pressure relief valve installed on a pressure vessel (vapor phase). Single valves (or more) used for process or supplemental valves for external fire (see labeling on chart). Reprinted by permission, *Sizing, Selection and Installation of Pressure Relieving Devices in Refineries,* Part 1 "Sizing and Selection," API RP-520, 5th Ed., July 1990, American Petroleum Institute.



Notes:

1. This figure conforms with the requirements of Section VIII of the ASME *Boiler and Pressure Vessel Code.*
2. The pressure conditions shown are for rupture disk devices installed on a pressure vessel.
3. The margin between the maximum allowable working pressure and the operating pressure must be considered in the selection of a rupture disk.
4. The allowable burst-pressure tolerance will be in accordance with the applicable code.
5. The operating pressure may be higher or lower than 90 depending on the rupture disk design.
6. The stamped burst pressure of the rupture disk may be any pressure at or below the maximum allowable burst pressure.

Figure 7-7B. Pressure level relationships for rupture disk devices. Reprinted by permission, *Sizing, Selection and Installation of Pressure Relieving Devices in Refineries,* Part 1 "Sizing and Selection," API RP-520, 5th Ed., July 1990, American Petroleum Institute.

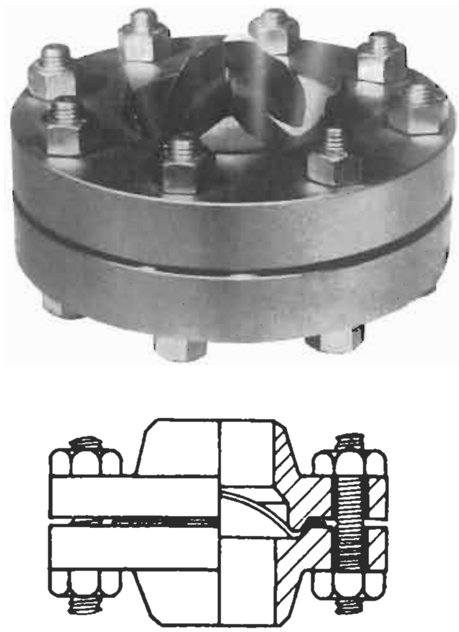


Figure 7-8A. Metal type frangible disk (above) with cross-section (below) Courtesy of Black, Sivalls and Bryson Safety Systems, Inc.

maximum process operating pressure level. There is no code requirement for establishing the design pressure. (See chart in Chapter 1.) Good judgment is important in selecting each of these pressures. See operating pressure description in above paragraph. Depending on the actual operating pressure level, the increase usually varies from a minimum of 10% higher or 25 psi, whichever is greater, to much higher increases. For instance, if the maximum expected operating pressure in a vessel is 150 psig, then experience might suggest that the design pressure be set for 187 to 200 psig. Other factors known regarding the possibility of a run-away reaction might suggest setting it at 275 psig. A good deal of thought needs to enter into this pressure level selection. (Also see section on explosions and DIERS technology this chapter [55] [67].)

Relieving Pressure: this is the pressure-relief device's *set pressure plus accumulation or overpressure.* See Figures 7-7A and 7-7B. For example, at a set pressure equal to the maximum allowable at the MAWP of the vessel of 100 psig, and for process internal vessel pressure, the pressure relief device would begin relieving at nominal 100 psig (actually begin to open at 98 psig, see figures above) and the device (valve) would be relieving at its maximum conditions at 110 psig (the 10 psig is termed accumulation pressure) for a single valve installation, or 116 psig, for a multiple valve installation on the same vessel. These are all process situations, which do not have an external fire around the vessel (See External Fire discussion later in

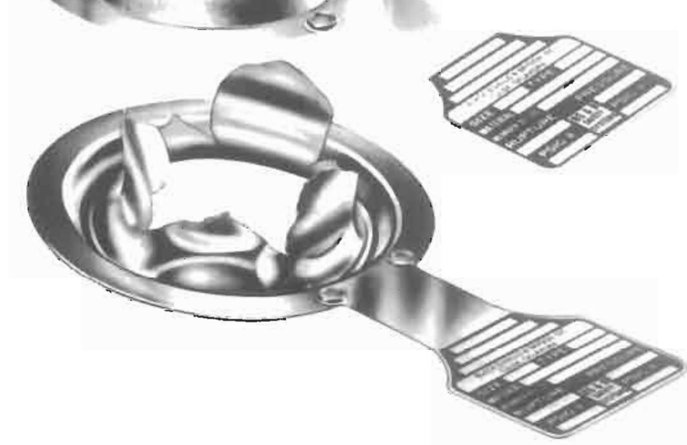


Figure 7-8B. Standard rupture disk. A prebulged rupture disk available in a broad range of sizes, pressures, and metals. Courtesy of B.S. & B. Safety Systems.

Figure 7-8C. Disk of Figure 7-8B after rupture. Note 30° angular seating in holder is standard for prebulged solid metal disk. By permission, B.S.&B. Safety Systems, Inc.

Figure 7-8D. Disk of Figure 7-8B with an attached (underside) vacuum support to prevent premature rupture in service with possible less than atmospheric pressure on underside and/or pulsation service. By permission, B.S.&B. Safety Systems, Inc.

this chapter and Figures 7-7B, 7-31A, B for these allowable pressure levels) and in no case do the figures apply to a sudden explosion internally.

Resealing Pressure: the pressure after valve opening under pressure that the internal static pressure falls to when there is no further leakage through the pressure relief valve. See Figure 7-7A.

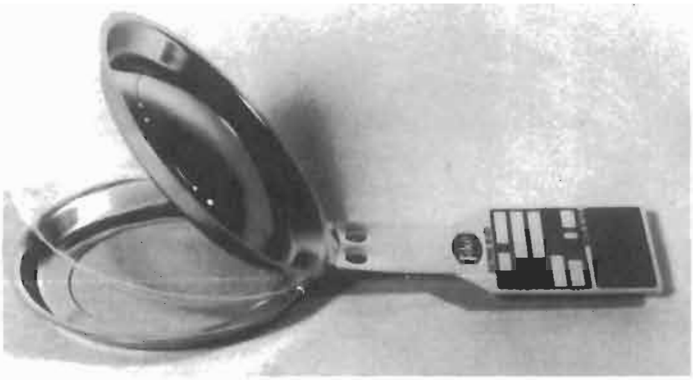


Figure 7-8E. Rupture disk (top) with Teflon® or other corrosion-resistant film/sheet seal, using an open retaining ring. For positive pressure only. By permission, Fike Metal Products Div., Fike Corporation, Blue Springs, Mo.

Figure 7-8F. Rupture disk (top), similar to Figure 7-8E, except a metal vacuum support is added (see Figure 7-8F(A)). By permission, Fike Metal Products Div., Fike Corporation, Blue Springs, Mo.

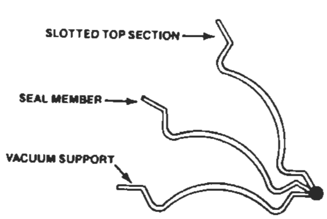


Figure 7-8F(A). Cross section of disk assembly for Figure 7-8F. By permission, Fike Metal Products Div., Fike Corporation, Blue Springs, Mo.

Closing Pressure: the pressure established as decreasing inlet pressures when the disk of the valve seats and there is no further tendency to open or close.

Simmer: the audible or visible escape of fluid between the seat/disk of a pressure-relieving valve at an inlet static pressure below the popping pressure, but at no measurable capacity of flow. For compressible fluid service.

Popping Pressure: the pressure at which the internal pressure in a vessel rises to a value that causes the inlet valve seat to begin to open and to continue in the opening direction to begin to relieve the internal overpressure greater than the set pressure of the device. For compressible fluid service.

Materials of Construction

Safety and Relief Valves; Pressure-Vacuum Relief Valves

For most process applications, the materia's of construction can be accommodated to fit both the corrosive-erosive and mechanical strength requirements. Manufacturers have established standard materials which will fit a large percentage of the applications, and often only a few parts need to be changed to adapt the valve to a corrosive service. Typical standard parts are: (See Figures 7-3, 7-3A, and 7-4)

	Option 1 (typical only)
Body	carbon steel, SA 216, gr. WCB
Nozzle	316 stainless steel
Disc/Seat	stainless steel
Blow Down Ring	300 Ser. stainless steel
Stem or Spindle	stainless steel
Spring	C.S. rust proof or high temp. alloy, rust proof
Bonnet	SA-216, Gr. WCB carbon steel
Bellows	316L stainless steel

	Option 2 (typical only)
Body	316 stainless steel
Nozzle	17-4 stainless steel or 316 stainless steel
Disc/Seat	Teflon, Kel-F, Vespel or Buna-N
Blow Down Ring	316 stainless steel
Stem or Spindle	17-4 stainless steel or 316 SS
Spring	316 stainless steel
Bonnet	316 stainless steel
Bellows	—

For pressure and temperature ratings, the manufacturers' catalogs must be consulted. In high pressure and/or temperature the materials are adjusted to the service.

For chemical service the necessary parts are available in 3.5 percent nickel steel; monel; Hastelloy C; Stainless Type 316, 304, etc.; plastic coated bellows; nickel; silver; nickel plated springs and other workable materials.

The designer must examine the specific valve selected for a service and evaluate the materials of construction in contact with the process as well as in contact or exposed to the vent or discharge system. Sometimes the corrosive

FEATURES:

- Isolates Safety Relief Valves
- No Fragmentation
- Operates up to 90% Rated Pressure
- Can Withstand Full Vacuum without Supports
- Available in Sizes 1" thru 36"
- Wide Material Availability
- U.S. Patent Number 3,294,277

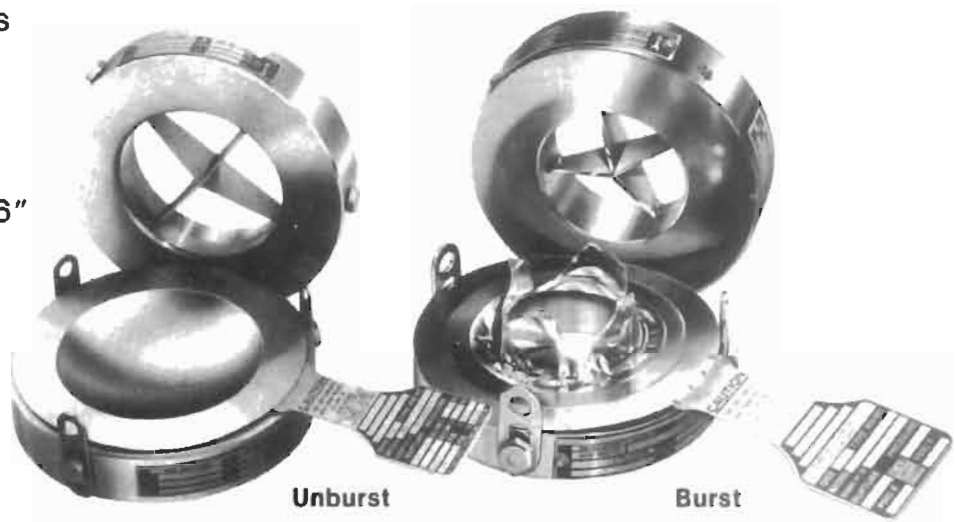
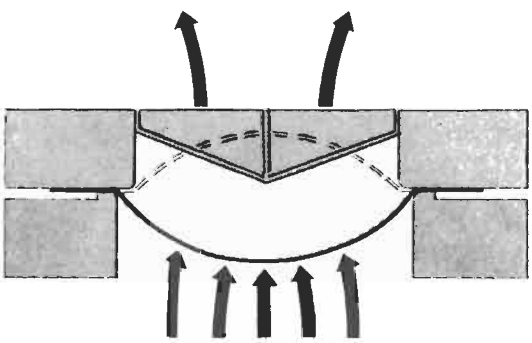


Figure 7-8G. Reverse buckling® disk, showing top holder with knife blades (underside) that cut the disk at time of rupture. By permission, B.S.&B. Safety Systems, Inc.



RB-90 reverse buckling disk. Pressure on CONVEX side of disk and patented seating design puts compression loading on disk metal.

Figure 7-8G(A). Reverse buckling® disk. Pressure on convex side of disk and patented seating design puts compression loading on disk metal. By permission, B.S.&B. Safety Systems, Inc.

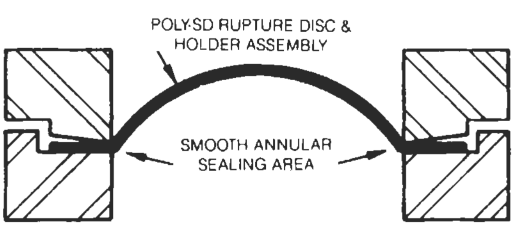


Figure 7-8H. Special metal disk holder for polymer systems using a smooth disk surface to reduce polymer adherence, and a smooth annular sealing area. Usually thick to avoid need for vacuum support and to allow for corrosion attack. By permission, Fike Metal Products Co. Div., Fike Corporation, Inc.

Figure 7-8I. Flat disk used for low pressure and for isolation of corrosive environments. Usual pressure range is 2 to 15 psig with ±1 psi tolerance. Stainless steel disk with Teflon® seal is usually standard. By permission, Fike Metal Products Div., Fike Corporation. Catalog 73877-1, p. 35.

nature of the materials in the vent system present a serious corrosion and fouling problem on the back or discharge side of the valve while it is closed.

For these special situations *properly designed* rupture disks using corrosion-resistant materials can be installed both before the valve inlet as well as on the valve discharge. For these cases, refer to both the valve manufacturer and the rupture disk manufacturers. See later discussion of code requirements for this condition.

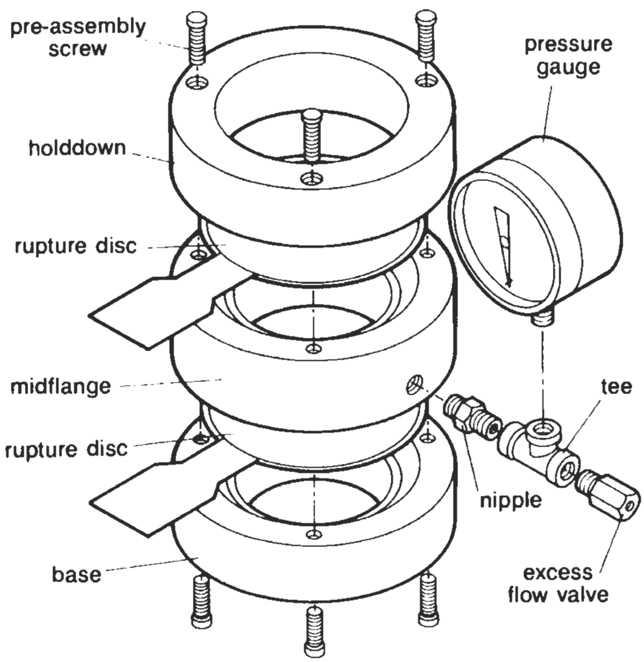


Figure 7-8J. Exploded view of double disk assembly. Usually burst pressure is same for each disk. Used for corrosive/toxic conditions to avoid premature loss of process and at remote locations. Note the use of tell-tale between disks. By permission, Fike Metal Products Div., Fike Corporation, Inc.

Figure 7-8L. Rupture disk indicator alarm strip breaks when rupture disk breaks and alarms to monitor. FM system approved. By permission, Continental Disc Corporation.

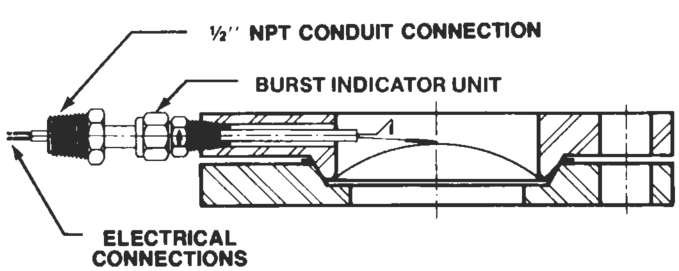


Figure 7-8K. Rupture disk with burst indicator. Several other techniques available. By permission, Fike Metal Products Div., Fike Corporation, Inc.

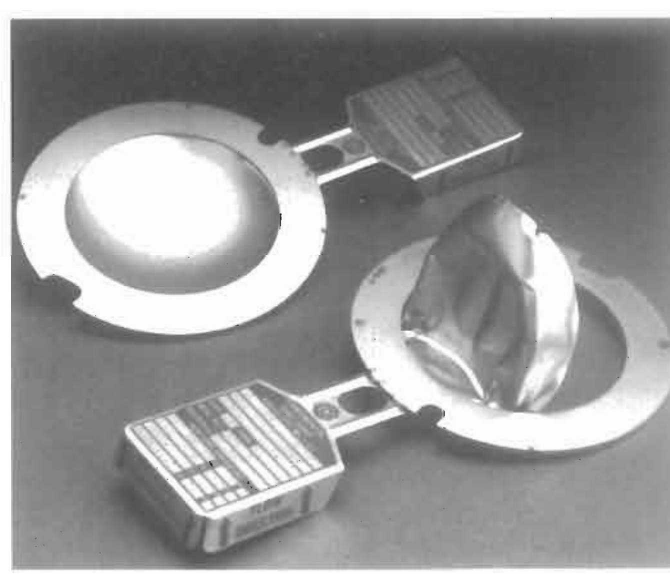


Figure 7-8M. Ultrex® Reverse Acting Rupture Disc providing instantaneous full opening, non-reclosing. By permission, Continental Disc Corporation.

Rupture Disks

Rupture disks are available in:

1. Practically all metals that can be worked into thin sheets, including lead, Monel, nickel, aluminum, silver, Inconel, 18-8 stainless steel, platinum, copper, Hastelloy and others.
2. Plastic coated metals, lead lined aluminum, lead lined copper.
3. Plastic seals of polyethylene, Kel-F®, and Teflon®
4. Graphite, impregnated graphite or carbon.

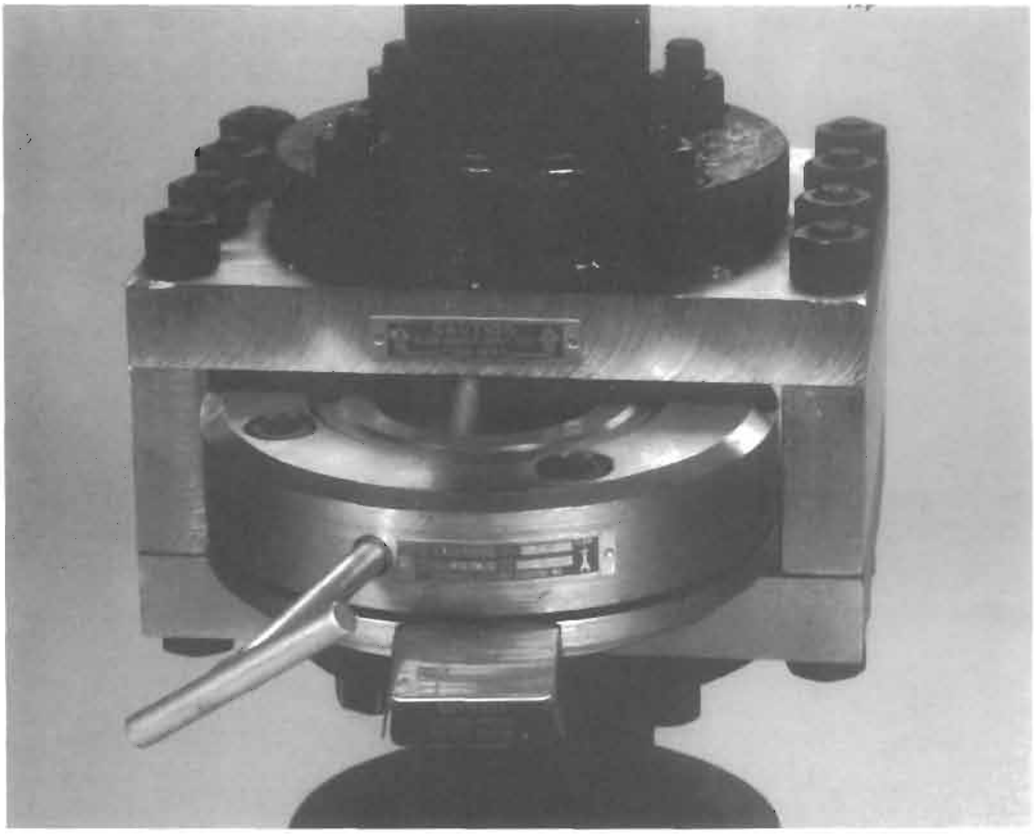


Figure 7-8N. Quick Change™ for quick rupture disk changeout. By permission, Continental Disc Corporation.

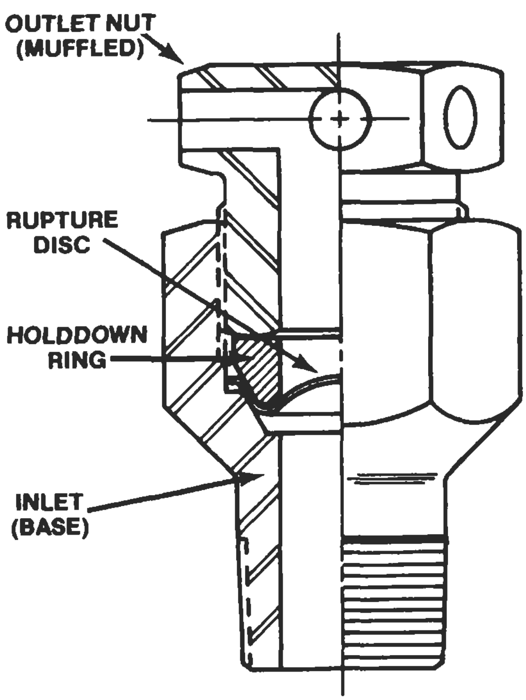


Figure 7-8O. Reusable screw type holder (30° seat) for smaller disks. By permission, Fike Metal Products Div., Fike Corporation, Inc.

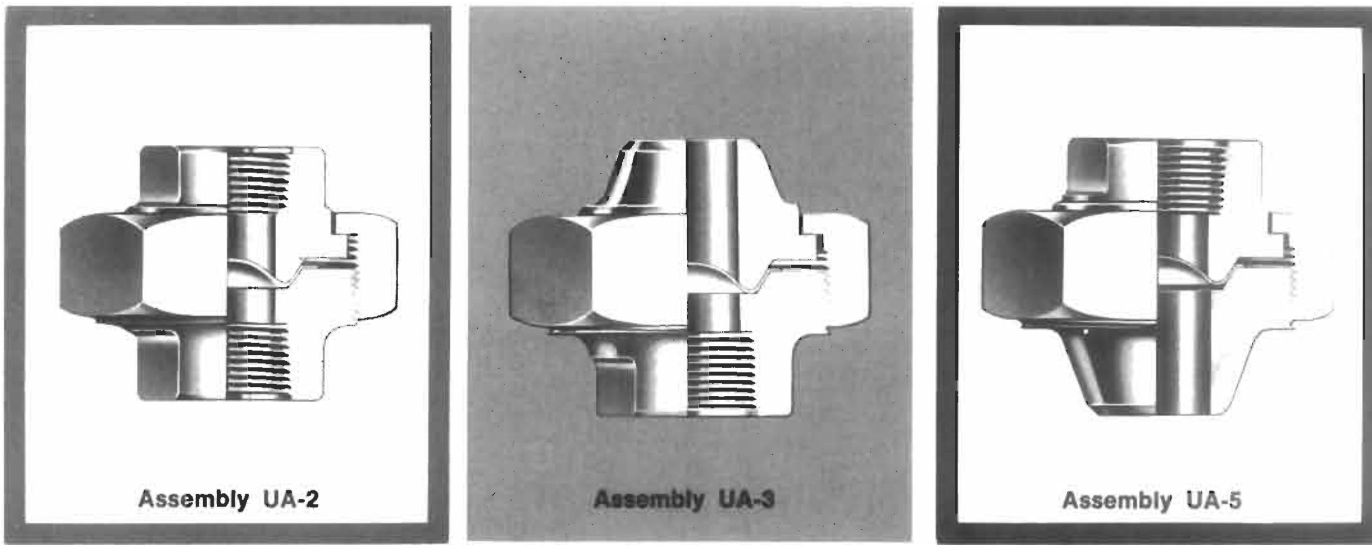
The selection of the material suitable for the service depends upon the corrosive nature of the fluid and its bursting characteristics in the pressure range under consideration. For low pressure, a single standard disk of some materials would be too thin to handle and maintain its shape, as well as give a reasonable service life from the corrosion and fatigue standpoints. See section on Selection and Application.

General Code Requirements [1]

It is essential that the ASME code requirements be understood by the designer and individual rating and specifying the installation details of the safety device. It is not sufficient to merely establish an orifice diameter, since process considerations which might cause overpressure must be thoroughly explored in order to establish the maximum relieving conditions.

An abbreviated listing of the key rating provisions is given in paragraphs UG-125 through 135 of the ASME code, Section 8, Div. 1, for unfired pressure vessels [1].

1. All pressure vessels covered by Division 1 or 2 of Section VIII are to be provided with protective over-



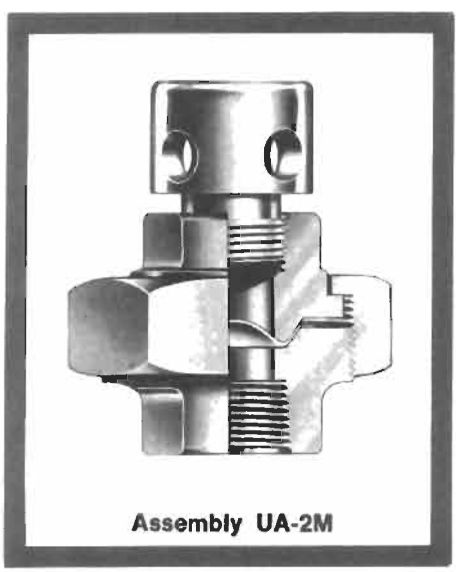


Figure 7-8P. Typical union type disk holders. They are not all available. By permission, B.S.&B. Safety Systems, Inc.

pressure devices. There are exceptions covered by paragraph U-1 of the code, and in order to omit a protective device this paragraph should be examined carefully. For example, vessels designed for above 3000 psi are not covered; also vessels with <120 gallons of water, vessels with inside diameter not over 6 inches (at any pressure), vessels having internal or external operating pressures not over 15 psig (regardless of size), and a few other conditions may not be subject to this code.

2. Unfired steam boilers must be protected.

3. Pressure relief must be adequate to prevent internal pressures from rising over 10% above the maximum allowable working pressure, except when the excess pressure is developed by external fire or other unforseen heat source. See design details in later paragraph. Papa [85] proposes an improved technique for relief sizing. Also see Figures 7-7A and 7-7B.

4. When a pressure vessel is exposed to external heat or fire, *supplemental* pressure relieving devices are required for this excessive pressure. These devices must have capacity to limit the overpressure to not more than 21% above the maximum allowable working pressure of the vessel. (See Figures 7-7A and 7-7B.) A single relieving device may be used to handle the capacities of paragraph UG-125 of the code, provided it meets the requirements of both conditions described.

5. Rupture disks may be used to satisfy the requirements of the code for conditions such as corrosion and polymer formations, which might make the safety/relief valve inoperative, or where small leakage by a safety valve cannot be tolerated. They are particularly helpful for internal explosion pressure release.

6. Liquid relief valves should be used for vessels that operate full of liquid.

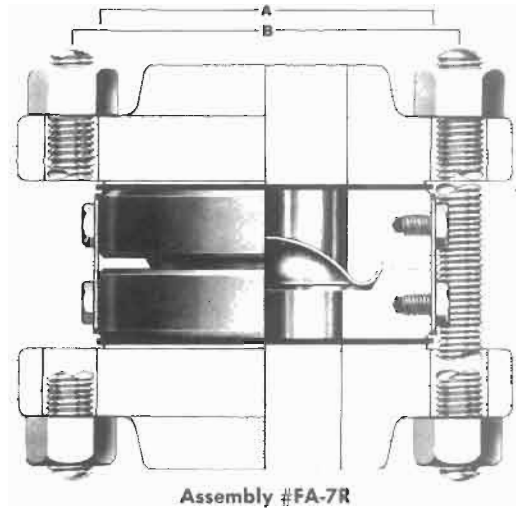
Relief Mechanisms

Reclosing Devices, Spring Loaded

Safety and relief valves must be the direct spring loaded type, and for pressure ranges noted below the code [1] requires:

Set Pressure	Max. Spring Reset Referenced to Set Pressure*
≤250 psig	±10%
≥250 psig	± 5%

*Marked on valve



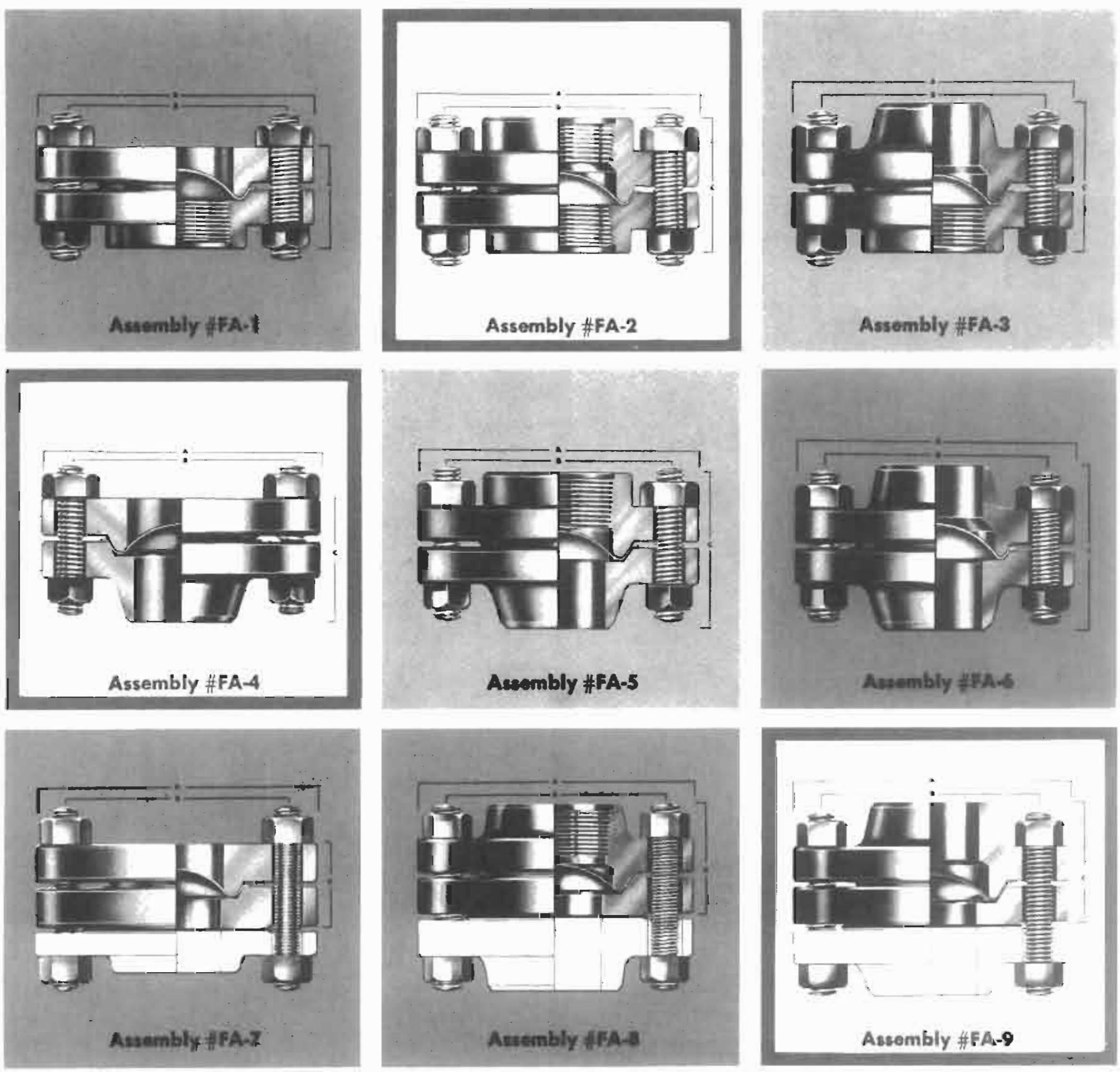
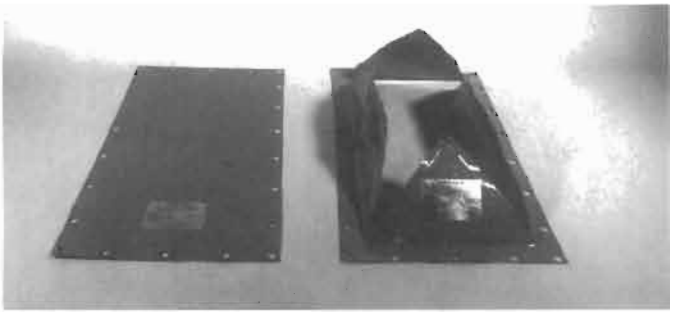
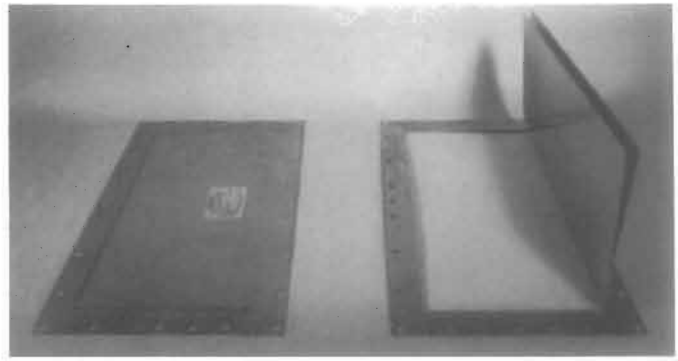


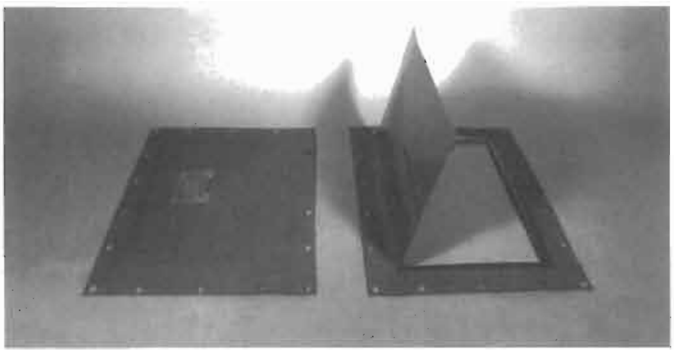
Figure 7-8Q. Typical bolted type safety head assemblies with angular seat design. By permission, B.S.&B. Safety Systems, Inc.



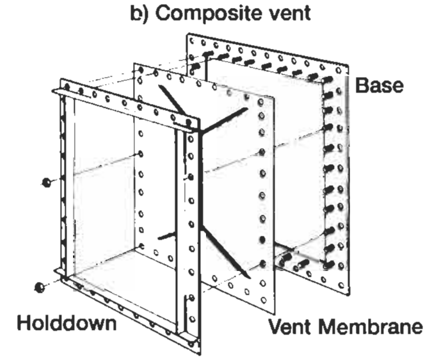
a) Epoxy vent



b) Composite vent



c) Epoxy vent with support



d) Typical explosion vent and mounting frame

Figure 7-8R. Low pressure bursting vents for explosion relief on storage silos, buildings, etc. Usually burst 1.0 to 8.0 psig depending on design. Round vents also available. By permission, Fike Metal Products Div., Fike Corporation, Inc.

Figure 7-9A. Non-metal frangible disk. Ruptured disk showing complete breakout of membrane. Courtesy of Falls Industries, Inc.

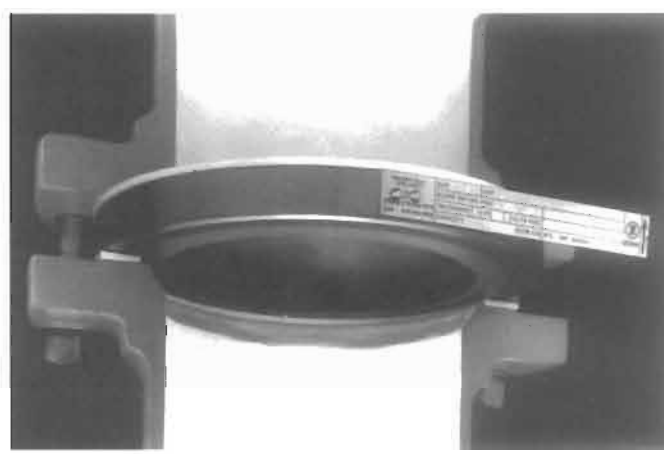


Figure 7-9B. Standard non-metal frangible disk (graphite), Teflon® coatings or linings are available on entire disk. By permission, Zook Enterprises.

The set pressure tolerances of pressure relief valves are not to exceed ±2 psi for pressures up to and including 70 psig and ±3% for pressures above 70 psig. Indirect operation of safety valves, for example, by pilot valve, is not acceptable unless the primary unloading valve will automatically open at not over the set pressure and will operate fully in accordance with design relieving capacity conditions if some essential part of the pilot or auxiliary device should fail [1].

The pilot valve is a self-actuated pressure relief valve that controls the main valve opening.

Nonreclosing Pressure Relieving Devices

Rupture disks must have a specified bursting pressure at a specified temperature. There must be complete iden-

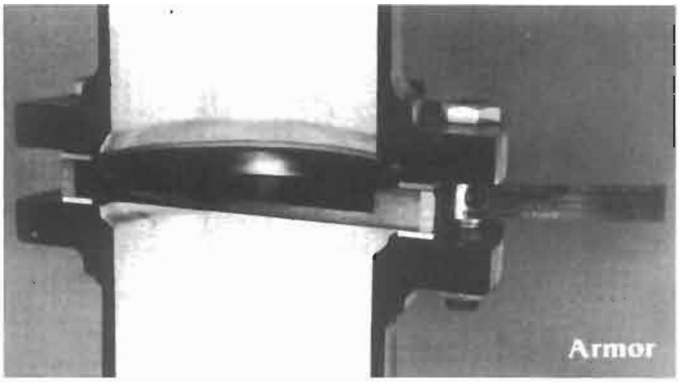
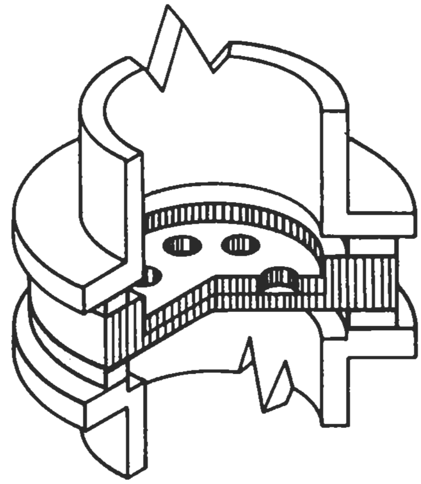


Figure 7-9C. Armored graphite disk. Note steel ring bonded to circumference of disk to increase safety in toxic or flammable services and improve reliability by preventing unequal piping stresses from reaching the pressure membrane. Teflon® coatings or linings are available on the entire disk. By permission, Zook Enterprises.



Rupture disc installation

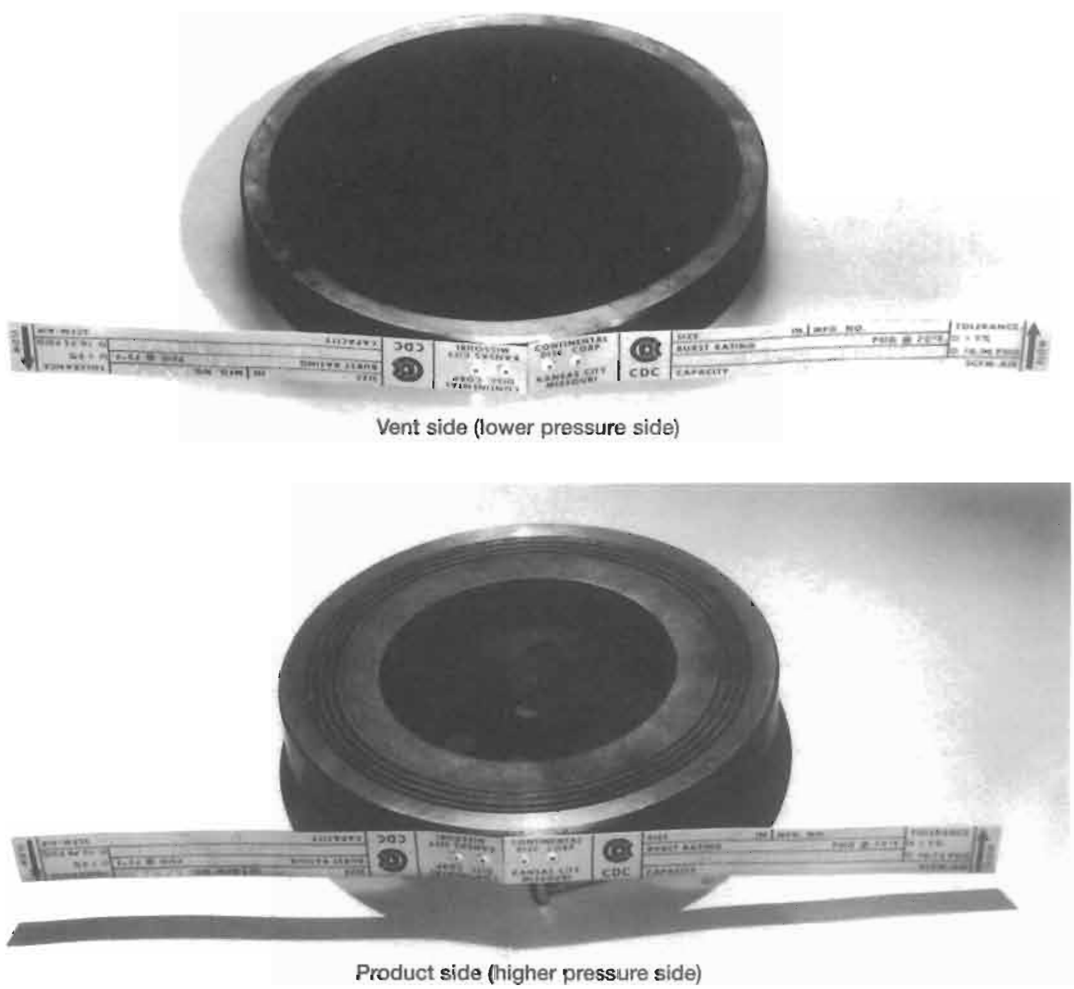


Figure 7-9D. Protection against two different pressures from opposite directions using graphite disks, such as in closed storage tanks. Particularly API-type to guard against failure of primary breathers, conservation vents, etc. These require a differential of at least 10 psig between the two burst ratings, depending on diameters of disks. By permission, Continental Disc Corporation.

tification of the metallurgy (if metal) or other properties if graphite or plastic, and the disk must be guaranteed by the manufacturer to burst within 5% (±) of the specified bursting pressure at the rated temperature.

The connection nozzle holding the disk must have a net cross-sectional area no less than that required for the design rated conditions of the disk.

The certification of disk performance is to be based on actual bursting tests of two or more disks from a lot of the same material of the exact same size as the disk to be sold by the manufacturer. The holder for the test disks must be identical to the design, dimensions, etc., for the disk being certified. (See details ASME code, Par. UG-127 [1]).

Pressure Settings and Design Basis

Unfired steam boilers, i.e., nominally termed waste heat boilers or, heat exchangers, which generate steam by heat interchange with other fluids (See ASME code [1] Par. U-

Duplex
DUPLEX Disks extend corrosion resistance to highly oxidizing agents, halogens except free fluorine, and virtually all other corrosives. A sheet of PTFE is used as a barrier on the service side of the disk. See page 12. Additionally, these disks are processed to accommodate temperatures to 392F without insulation.

***Insulated**
INSULATED Disks are available in MONO, INVERTED, and DUPLEX styles to accommodate temperatures exceeding 338F to 700F. They are furnished as an attached unit as shown because the nameplate rating of the disk must be established at the cold face temperature of the insulation.

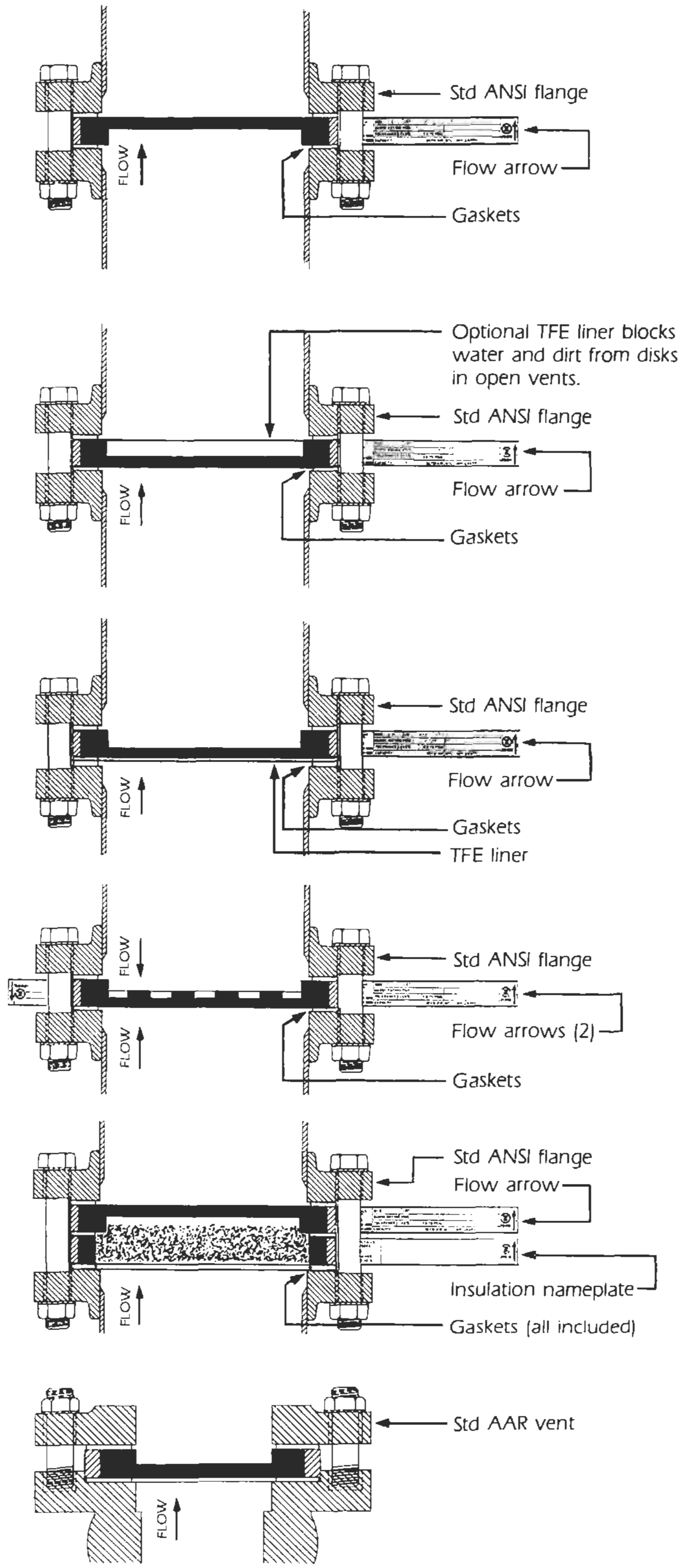


Figure 7-9E. Duplex and insulated disks. By permission, Zook Enterprises.

Figure 7-9E. Continued.

Figure 7-9F. For pressure ratings of 15 psig or lower, subject to internal vacuum conditions, a vacuum support is required that is an integral part of the rupture disk and cannot be added in the field. By permission, Falls Industries.

1('g), should be equipped with pressure relieving devices required by the ASME Code, Section I, as far as applicable; otherwise, use Par. UG-125. Vessels, which per Par. U-1 (g)), follow Par. UG-125ff are:

1. Evaporators or heat exchangers

2. "Vessels in which steam is generated by the use of heat resulting from operation of a processing system containing a number of pressure vessels such as used in the manufacturer of chemical and petrochemical products" [1]

3. Par. U-1 (h) "Pressure vessels or parts subject to direct firing from the combustion of any fuel, which are not within the scope of Sections I, III or IV, may be constructed in accordance with the rules of Section VIII, Div. I, Par. UW-2 (d) [1].

To meet code requirements, the relieving device must be directly open to the system to be relieved, see Figures 7-10, 7-11, and 7-12. For Figures 7,10, 7-11, and 7-12, the rupture disk and the relief valve must be designed to handle the *relieving capacity* at the *relieving temperature* without allowing more than a 10% pressure build-up above the maximum allowable working pressure of the unfired pressure vessel (or corresponding overpressure for other code requirements). Figure 7-11 requires that the rupture disk be designed the same as for Figure 7-10, 7-13A and 7-13B; and Figure 7-12 requires that the relief valve be the *primary* device and meet the process relief requirements; it may have additional capacity to accommodate such conditions as external fire, or this additional requirement may be installed in a separate relief valve or rupture disk as shown. Also the separate rupture disk may be in a *secondary* function not covered by the code for such conditions as runaway reactions and internal explosion. For these conditions the setting of the rupture disk is left up to the designer, and may be higher than that for the usual relief. Of course, it should be set sufficiently below the rupture condition for the vessel or component in order to avoid a hazardous condition and meet Code requirements.

Unfired Pressure Vessels Only, But Not Fired or Unfired Steam Boilers

- Non-fire exposure

Single pressure relief valve installation. Must be set to operate at a pressure not exceeding the maximum allowable working pressure of the vessel (MAWP), Ref [1] Par. UG-134, but may be *set* to operate at pressures below the

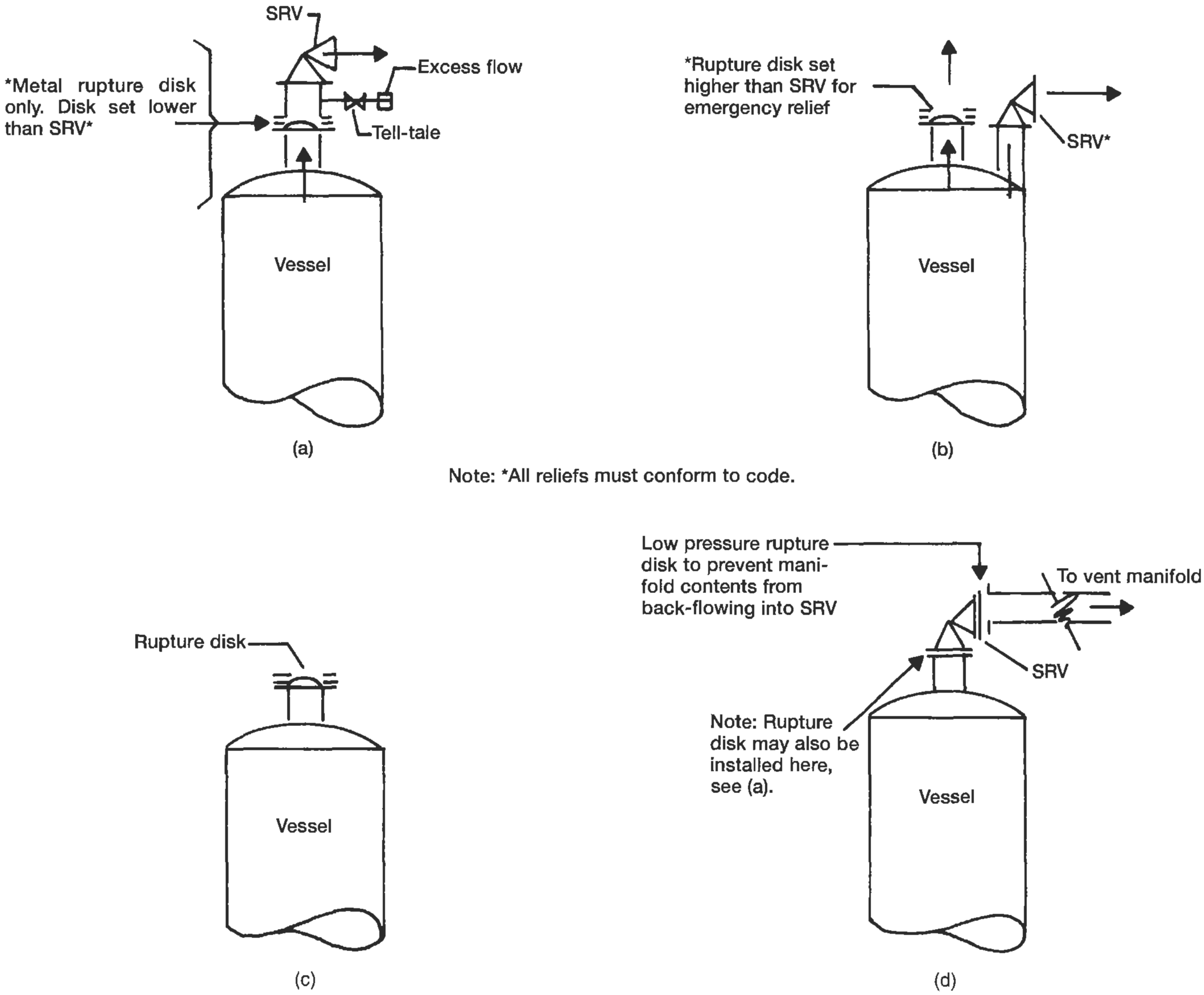


Figure 7-10. Rupture disk installations.

MAWP. The device must prevent the internal pressure from rising more than 10% above the MAWP.

Multiple pressure relief valves installation. If the required capacity is provided using *more than one* pressure relieving device, a. only one device must be *set* at or below the MAWP of the vessel, and, b. the additional device(s) may be *set* to open at higher pressures, but in no case at a pressure any higher than 105% of the MAWP. The combination of relieving valves must prevent the pressure from rising more than 16% above the MAWP. See ASME Ref [1] Par. UG-125C and C-1 and Par. UG-134a.

- External fire or heat exposure only and process relief

Valves to protect against excessive internal pressures must be *set* to operate at a pressure not in excess of 110% of the MAWP of the vessel (ASME Par. UG-134b).

When valves are used to meet the requirements of both Par. UG-125(c) and UG-125c-(2); that is, both internal process pressure and external fire/heat requirements, the valve(s) must *be set to operate not over the MAWP of the vessel. For these conditions of the additional hazard of extreme fire or heat, supplemental* pressure relieving devices must be installed to protect the vessel. The supplemental devices must be capable of preventing the pressures from rising more than 21% above the MAWP (note: this is *not* the *setting*). The same pressure relieving devices may be used to satisfy the capacity requirements of Par. UG-125c or C(1) *and* Par. UG-125c-(2) provided the pressure *set-*

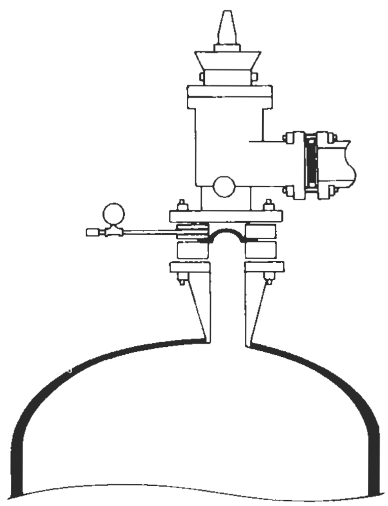


Figure 7-11. Safety valve and rupture disk installation using pressure rupturing disk on inlet to safety relief valve, and low pressure disk on valve discharge to protect against back flow/corrosion of fluid on valve discharge side, possibly discharge manifold. By permission, Fike Metal Products Div., Fike Corporation, Inc.

ting requirements of Par. UG-134(a) are met. See Par. (A) 1 and 2 above and see Figure 7-7A.

When pressure relief devices are intended primarily for protection against overpressure due to external fire or heat, have no permanent supply connection, and are used for storage at ambient temperature of non-refrigerated liquefied compressed gases, they are excluded from requirements of Par. UG-125c (1) and C (2), with specific provisions. See ASME code [1] for detailed references and conditions.

- Vessels operating completely filled with liquid must be equipped with liquid relief valves, unless otherwise protected (Par. UG-125-3(g).
- Safety and safety relief valves for *steam service* should meet the requirements of ASME Par. UG-131(b), Ref [1]. Note that the requirements for these valves are slightly different than for process type valves.

Relieving Capacity of Combinations of Safety Relief Valves and Rupture Disks or Non-Reclosure Devices (Reference ASME Code, Par. UG-127, UG-132).

- Primary Relief

  A single rupture disk can be used as the only overpressure protection on a vessel or system (Figure 7-10). The disk must be stamped by the manufacturer with the guaranteed bursting pressure at a specific temperature. The disk must rupture within ±5% of its stamped bursting pressure at its specified burst temperature of operation. The expected burst temperature may need to be determined by calculation or extrapolation to be consistent with the selected pressure.

  The set burst pressure should be selected to permit a sufficiently wide margin between it and the vessel's used or design operating pressure and temperature to avoid premature failure due to fatigue or creep of metal or plastic coatings.

Selected Portions of ASME Pressure Vessel Code, quoted by permission [1]

Section VIII, Division I Superscript = Footnote reference
July 1, 1989 Edition in Code Figure No., for this text.

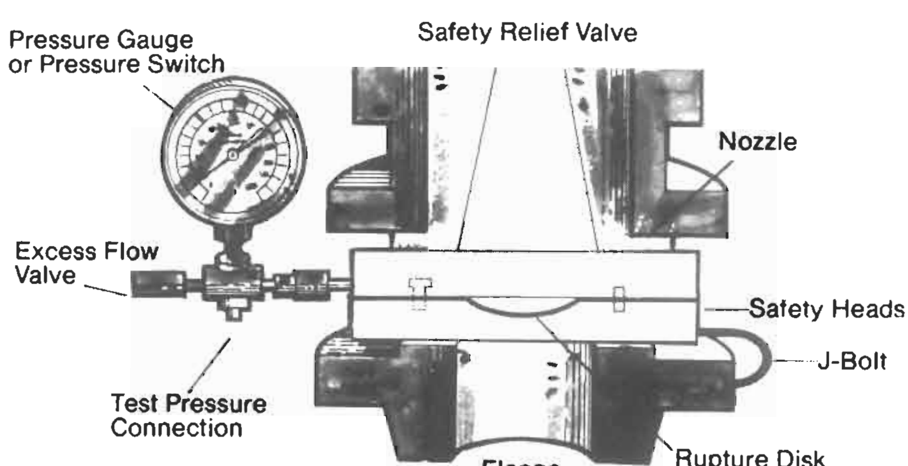


Figure 7-12. Rupture disk mounted beneath a pressure relieving spring-loaded valve. A reverse buckling® disk arrangement is often recommended here. By permission, B.S.&B. Safety Systems, Inc.

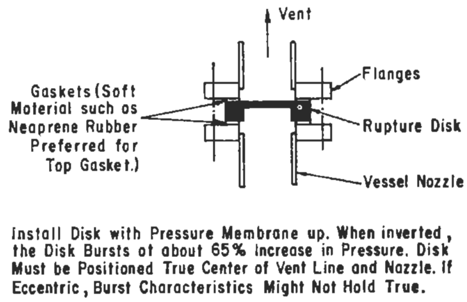
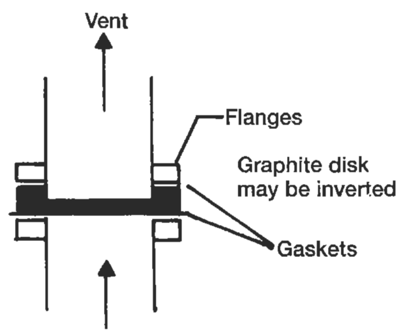


Figure 7-13A. Installation of graphite rupture disk. Adapted by permission, Falls Industries, Inc.



(b) Inverted graphite disk bursts at higher pressure than with flat surface on top.

Figure 7-13B. Inverted graphite disk bursts at higher pressure than with flat surface on top. Adapted by permission, Falls Industries, Inc.

Rupture Disk Devices,[44] Par UG-127

1. General
   a. Every rupture disk shall have a stamped bursting pressure within a manufacturing design range[35] at a specified disk temperature[36] and shall be marked with a lot number, and shall be guaranteed by its manufacturer to burst within 5% (plus or minus) of its stamped bursting pressure at the coincident disk temperature.

2. Capacity Rating
   a. The calculated capacity rating of a rupture disk device shall not exceed a value based on the applicable theoretical formulas (see Par. UG-131) for the various media multiplied by K = coefficient = 0.62. The area A (square inches) in the theoretical formula shall be the minimum net area existing after burst.[47]

3. Application of Rupture Disks
   a. A rupture disk device may be used as the sole pressure relieving device on a vessel.

      Note: When rupture disk devices are used, it is recommended that the design pressure of the vessel be sufficiently above the intended operating pressure to provide sufficient margin between operating pressure and rupture disk bursting pressure to prevent premature failure of the rupture disk due to fatigue or creep.

      Application of rupture disk devices to liquid service should be carefully evaluated to assure that the design of the rupture disk device and the dynamic energy of the system on which it is installed will result in sufficient opening of the disk.
   b. A rupture disk device may be installed between a pressure relief valve[48] and the vessel provided. (See Figure 7-10.)

      1. The combination of the spring loaded safety or safety relief valve and the rupture disk device is ample in capacity to meet the requirements of UG-133 (a) and (b).
      2. The stamped capacity of a spring loaded safety or safety relief valve (nozzle type) when installed with a rupture disk device between the inlet of the valve and the vessel shall be multiplied by a factor of 0.80 of the rated relieving capacity of the valve alone, or alternatively, the capacity of such a combination shall be established in accordance with Par. 3 below;
      3. The capacity of the combination of the rupture disk device and the spring loaded safety or safety relief valve may be established in accordance with the appropriate paragraphs of UG-132, Certification of Capacity of Safety Relief Valves in Combination with Non-reclosing Pressure Relief Devices.
      4. The space between a rupture disk device and a safety or safety relief valve shall be provided with a pressure gauge, a try cock, free vent, or suitable telltale indicator. This arrangement permits detection of disk rupture or leakage.[49]
      5. The opening[47] provided through the disk, after burst, is sufficient to permit a flow equal to the capacity of the valve (Par. 2 and 3 above) and there is no chance of interference with proper functioning of the valve; but in no case shall this area be less than 80% of the area of the inlet of the valve unless the capacity and functioning of the specific combination of rupture

disk and valve have been established by test in accordance with UG-132.

Note that in lieu of testing, Par (b) 2 and (b) 3 above allows the use of a capacity factor of 0.80 as a multiplier on the stamped capacity of the spring loaded safety relief valve (nozzle type). Some manufacturers test specific valve/rupture disk combinations and determine the actual capacity factor for the combination, and then use this for the net capacity determination. See Figures 7-10, 7-11, 7-12, 7-13A and 7-13B.

c. A rupture disk device may be installed on the outlet side[50] of a spring loaded safety relief valve which is opened by direct action of the pressure in the vessel provided. (Figure 7-12)

1. The valve is so designed that it will not fail to open at its proper pressure setting regardless of any back pressure that can accumulate between the valve disk and the rupture disk. The space between the valve disk and rupture disk shall be vented or drained to prevent accumulation of pressure due to a small amount of leakage from the valve.[51]
2. The valve is ample in capacity to meet the requirements of UG-133 (a) and (b).
3. The stamped bursting pressure of the rupture disk at the coincident disk temperature plus any pressure in the outlet piping shall not exceed the design pressure of the outlet portion of the safety or safety relief valve and any pipe or fitting between the valve and the rupture disk device. However, in no case shall the stamped bursting pressure of the rupture disk at the coincident operating temperature plus any pressure in the outlet piping exceed the maximum allowable working pressure of the safety or safety relief valve.
4. The opening provided through the rupture disk device after breakage is sufficient to permit a flow equal to the rated capacity of the attached safety or safety relief valve without exceeding the allowable overpressure.
5. Any piping beyond the rupture disk cannot be obstructed by the rupture disk or fragment.
6. The contents of the vessel are clean fluids, free from gumming or clogging matter, so that accumulation in the space between the valve inlet and the rupture disk (or in any other outlet that may be provided) will not clog the outlet.
7. The bonnet of the safety relief valve shall be vented to prevent accumulation of pressure.

Footnotes to ASME Code

47. The minimum net flow area is the calculated net area after a complete burst of the disk with appropriate allowance for any structural members which may reduce the net flow through the rupture disk device. The net flow area for sizing purposes shall not exceed the nominal pipe size area of the rupture disk device.
48. Use of a rupture disk device in combination with a safety or safety relief valve shall be carefully evaluated to ensure that the media being handled and the valve operational characteristics will result in pop action of the valve coincident with the bursting of the rupture disk.
49. Users are warned that a rupture disk will not burst at its design pressure if back pressure builds up in the space between the disk and the safety or safety relief valve which will occur should leakage develop in the rupture disk due to corrosion or other cause.
50. This use of a rupture disk device in series with the safety or safety relief valve is permitted to minimize the loss by leakage through the valve of valuable or of noxious or otherwise hazardous materials and where a rupture disk alone or disk located on the inlet side of the valve is impracticable, or to prevent corrosive gases from a common discharge line from reaching the valve internals.
51. Users are warned that an ordinary spring loaded safety relief valve will not open at its set pressure if back pressure builds up in the space between the valve and rupture disk. A specially designed valve is required, such as a diaphragm valve or a valve equipped with a bellows above the disk.

*Reprinted with ASME permission. ASME Pressure Vessel Code, Section VIII, Division I, UG-127, 1989 Edition, pp. 86–88.

Establishing Relieving or Set Pressures

The pressure at which the valve is expected to open (set pressure) is usually selected as high as possible consistent with the effect of possible high pressure on the process as well as the containing vessel. Some reactions have a rapid increase in temperature when pressure increases, and this may fix the maximum allowable process pressure. In other situations the pressure rise above operating must be kept to some differential, and the safety valve must relieve at the peak value. A set pressure at the maximum value (whether maximum allowable working pressure of vessel, or other, but insuring protection to the weakest part of the system) requires the smallest valve. Consult manufacturers for set pressure compensation (valve related) for temperatures >200°F.

Table 7-1A
Compensation Factors for Safety Relief Valves Between Atmospheric Test Temperature and Actual Operating Temperature [24]

Operating Temperature °F	Percent Increase in Set Pressure at Atmospheric Temperature
−450 to 200	None
201 to 450	2
451 to 900	3
901 to 1200	4

By permission, Teledyne Farris Engineering Corp., Cat. FE-316, p. 12.

Table 7-1B
Set Pressure Compensation for Saturated Steam Service Safety-Relief Valves Between Atmospheric Test Temperature and Actual Operating Temperature

Saturated Steam Service Set Pressure (PSIG)	% Increase in Spring Settling
10–100	2%
101–300	3%
301–1000	4%
1001–3000	5%

By permission, Teledyne-Farris Engineering, Cat. 187C

When the pressure rise in a system is gradual and not "explosive" in nature, a safety or safety relief valve is the proper device, but when it is critical to completely depressure a system or the rate of pressure increase might be expected to be rapid, then a rupture disk is the proper device. Properly designed a pilot operated valve may be selected after checking its performance with the manufacturer.

Often a system (a group of vessels not capable of being isolated from each other by block valves, or containing restriction to flow and release of pressure) may need a relief valve set reasonably close, sat +15% to 20% when system is below 1000 psig; above, typically use 7% to 15% above as set criteria related to normal operating pressure to catch any pressure upswing. Then this may have a back-up valve set higher (but within code) to handle further pressure increase. Or, the second device may be a rupture disk. It is not unusual to have two relief devices on the same equipment set at different pressures.

For situations where explosions may be involved—chemical liquid, vapor or dust—it is generally advisable to obtain rate of pressure rise data and peak explosion pressure data in order to intelligently establish the design parameters. Such data are available [7, 8, 14, 15, 16, 19, 41, 54]; however, it is important to evaluate whether the conditions are comparable between the systems when selecting the values for design. In general the lower the setting for pressure relief, the lower will be the final internal peak pressure in the vessel. It is extremely important to realize that the higher the system pressure before relief, the higher will be the peak pressure attained in the vessel. In some difficult cases it may be advisable to set relief devices at two pressures, one lower than the other. Of course, each must be designed for the conditions expected when it relieves, and one or all must satisfy code requirements or be more conservative than code.

For pulsating service, the set pressure is usually set greater than the nominal 10% or 25 psig above the average *operating* pressure of the system in order to avoid unnecessary releases caused by surging pressure peaks, but still not exceeding the MAWP of the vessel/system. Careful analysis must be made of the proper set condition.

Safety-relief valves are available for relieving or set pressures as low as 2, 10, and 20 psig, as well as higher pressures. Lower pressures are available on special order. Usually a more accurate relief is obtained from the higher pressures.

Safety relief valves are normally tested in the shop, or even on the equipment at atmospheric temperature. The set tolerances on the valves as manufactured are established by the Code as discussed earlier. In order to recognize the difference between the test temperature and the actual operating temperature at actual relief, the corrections shown in Table 7-1A and 7-1B are applied. An increase in temperature above design causes a reduction in valve set pressure due to the effects of temperature on the spring and body.

Testing of pressure relieving spring loaded valves at atmospheric temperature requires an adjustment in set pressure at ambient conditions to compensate for higher operating temperatures. For process services see Table 7-1A and for *saturated* steam, use Table 7-1B.

Safety and Safety Relief Valves for Steam Service

Pressure relieving devices in process plants for process and utility steam systems must conform to the requirements of ASME [1] Par. UG-131b. This is not necessarily satisfactory to meet the ASME Power Boiler Code for applications on power generating equipment.

Vessels or other pressure containing equipment that operates filled with liquid must be provided with liquid relief valves, unless protected otherwise [1]. Any liquid relief valve must be at least ½ in. in pipe size, [1] Par UG-128 (see [79]).

Selection and Application

Causes of System Overpressure

Figure 7-14, Operational Check Sheet [25], lists 16 possible causes of overpressure in a process system. There are many others, and each system should be reviewed for its peculiarities. System evaluation is the heart of a realistic, safe and yet economical overpressure protection installation on any single equipment or any group of equipment. Solving formulas with the wrong basis and/or data can be disastrous. The following should be reviewed:

1. The sources of possible overpressure
2. Maximum overpressure possible from all sources
3. Maximum rate of volume increase at the burst pressure, and temperature at this condition.
4. Length of duration of overpressure.

Capacity Requirements Evaluation for Process Operation (Non-Fire)

Each system and item of equipment should be examined for operational safety as set forth by specific plant area (and process fluids) requirements and the codes previously cited. The codes particularly [10, 13, 27, 33a, b, c] establish guides based on wide experience, and are sound requirements for design. Relief capacity is based on the most severe requirement of a system, including possible two-phase flow [67]. A system is generally equipment or groups of equipment which is isolated by shut-off valves. Within these isolated systems a careful examination of the probable causes of overpressure is made [6]. Figures 7-15, 7-16, and 7-17 are suggested guides [25]. Capacities are calculated for conditions of temperature and pressure at actual state of discharge. Final discharge pressure is the set pressure plus overpressure.

It *must be emphasized,* that the determination of the anticipated maximum overpressure volume at a specified pressure and temperature is vital to a proper protection of the process system. The safety relief calculations should be performed at the actual worst conditions of the system, for example, at the allowable accumulated pressure and its corresponding process temperature. These can be tedious and perhaps time-consuming calculations, but they must not be "glossed" over but developed in a manner that accounts for the seriousness of the effort. *They must be documented carefully and preserved permanently.*

The situation is just as critical, if not more so, for runaway reactions or reaction conditions that are not adequately known. They should be researched or investigated by laboratory testing for possible runaway conditions and then the kinetic and heat/pressure rise calculations should be performed, even if some assumptions must be made to establish a basis. Refer to later paragraphs and the American Institute of Chemical Engineers Design Institute for Emergency Relief [67]. At the time of a vessel or pressure/vacuum system failure, the calculations for the effected pressure-relief devices are always reviewed by plant management and the Occupational Safety and Health Administration (OSHA) inspectors. A few notes on causes of process system failures are noted below, with additional comments in API-521 [13, 33a, b, c]

Failure of Cooling Water: assume all cooling mediums fail, determine relief capacity for the total vapors entering the vessel, including recycle streams. See [3] and [10].

Reflux Failure: (a) At top of distillation column, capacity is total overhead vapor [10], (b) when source of heat is in feed stream, capacity is vapor quantity calculated in immediate feed zone [3], (c) when reboilers supply heat to system, capacity is feed plus reboil vapors [3]. Each situation must be examined carefully.

Blocked Outlets on Vessels: (a) For liquid, capacity is maximum pump-in rate. (b) For liquid-vapor system, capacity is total entering vapor plus any generated in vessel [10].

Blocked Outlets and Inlets: for systems, lines or vessels, capable of being filled with liquid and heated by the sun or process heat, require thermal relief to accommodate the liquid expansion (assuming vaporization is negligible).

Instrument Failure: assume instrument control valves freeze or fail in open position (or closed, which ever is worse), determine capacity for relief based on flows, temperatures, or pressures possible under these circumstances. The judicious selection of instrument failure sequence may eliminate or greatly reduce relief valve requirements.

Equipment Failure: pumps, tubes in heat exchangers and furnaces, turbine drivers and governor, compressor cylinder valves are examples of equipment which might fail and cause overpressure in the process. If an exchanger tube splits or develops a leak, high pressure fluid will enter the low side, overpressuring either the shell or the channels and associated system as the case may be.

SAFETY VALVE DESIGN OPERATIONAL CHECK SHEET

Date: ______________ Job No.: ___________

Checked: ______________ By: ______________

Vessel or System: Process Evaporator

Design Pressure: 75 PSIG

Allowable Pressure for Capacity Relief: 75 + (75+ 10%) =

= 82.5 PSIG

Operating Conditions: Fluid: PDC Mol Wt. 113.5

Sp.Gr. 1.16 Temp. Oper.

Latent Heat 125 BTU/Lb. Corrsive No

Physical Conditions: Vessel Dia. 5 feet x Length : 6 feet

Insulation: Yes, 2"

Fire Control Measures: No Sprinkler Sys.

Cause of Overpresure	Capacity Requiremant, Lb./Hr.
1. Failure Cool water/Elect/Mechanical	------------------------
2. Reflux and/or Condensing Failure	------------------------
3. Entrance of a Highly Volatile Fuel	------------------------
4. Vapor generation, external fire	----2,920---------------
5. Excessive Operating Heat Inputs	---22,500---------------
6. Accumulation of Non-Condensibles	------------------------
7. Closed Outlets	---22,500 or less-------
8. Failure of Automatic Controls/Instr.	---22,500 or less-------
9. Internal Explosions(Use Rupture Disk)	------------------------
10.Chemical Reaction/Run-a-Way(" ")	------------------------
11.Two Phase Flow Conditions	------------------------
12.Inadvertent opening valve into system	------------------------
13.Check Valve failure	------------------------
14.Cooling Fans failure	------------------------
15.Heat Exchanger Tube rupture/failure	------------------------
16.Circulating Pump failure	------------------------

Causes that may occur simutaneously: ------------------------

------ Any of the four considered may occur simultaneously------

Probability of occurence: --Closed outlet with a failure of automatic control resulting in excessive heat input.----------

Allowance to made:----None------------------------------------

Relief capacity used for sizing valve(s)------22,500 Lb./Hr.--

Auxiliaries	Cause of Overpressure	Capacity Requirement
1. Exchangers	Split tube(s)	1482 Lb.Hr.
	Thermal Vaporization	--------------------
2. Pumps	Discharge Restriction	--------------------
3. Length of Line	Thermal Vaporization	--------------------

Figure 7-14. Safety valve design operational check sheet. Adapted and added to by permission, N. E. Sylvander and D. L. Klatz, *Design and Construction of Pressure Relieving Systems,* Univ. of Michigan Press, Ann Arbor (1948). Six items of overpressure list above by this author and from API Rec. Practice 521 (1982).

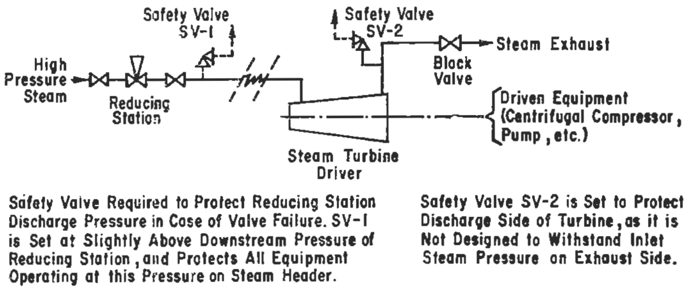


Figure 7-15. Safety valve protecting specific equipment operation.

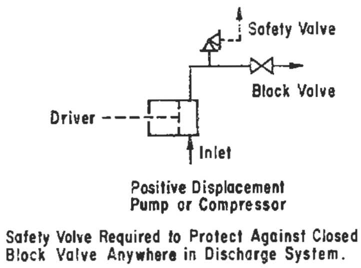


Figure 7-16. Safety valve in positive displacement system.

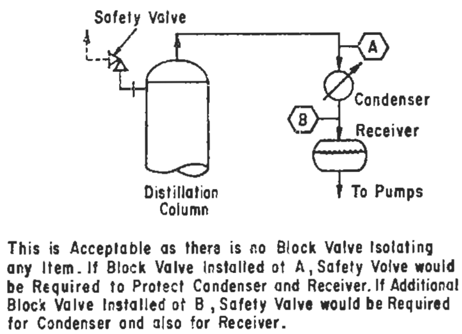


Figure 7-17. System protected by safety valve on column.

Vacuum: (a) Removal of liquid or vapor at greater rate than entering a vessel, capacity determined by volume displaced. (b) Injecting cold liquid into hot (steamed out) vessel, the condensing steam will create vacuum, and must be relieved. Capacity is equivalent to vapor condensed.

In-breathing and Out-breathing Pump In and Out: See section on Pressure-Vacuum Relief for Low Pressure Storage Tanks.

Installation

Never place a block valve on the discharge side of a pressure relief device of any kind, except see [1] Par. U-135(e).

Never place a block valve on the inlet side of a pressure relief device of any kind, unless it conforms to the code practice for rupture disks or locking devices. See [1] Par. UG-135(e) and Appendix M, ASME code.

Note that the intent of the ASME Code is to ensure that under those circumstances where a pressure relieving device can be isolated by a block valve from its pressure, or its discharge, that a responsible individual lock and unlock the block valve to the safe open position and that this individual remain at the block valve the entire time that the block valve is closed.

Safety (relief, or safety-relief) valves are used for set pressures from 10 psig to 10,000 psig and even higher. At the low pressures the sensitivity to relieving pressure is not always as good as is required for some processes, and for this reason most valve installations start at 15 to 20 psig.

Figures 7-10 and 7-18 illustrate a few typical safety valve installations. Care must be shown in designing any manifold discharge headers collecting the vents from several valves. Sharp bends are to be avoided. Often two or more

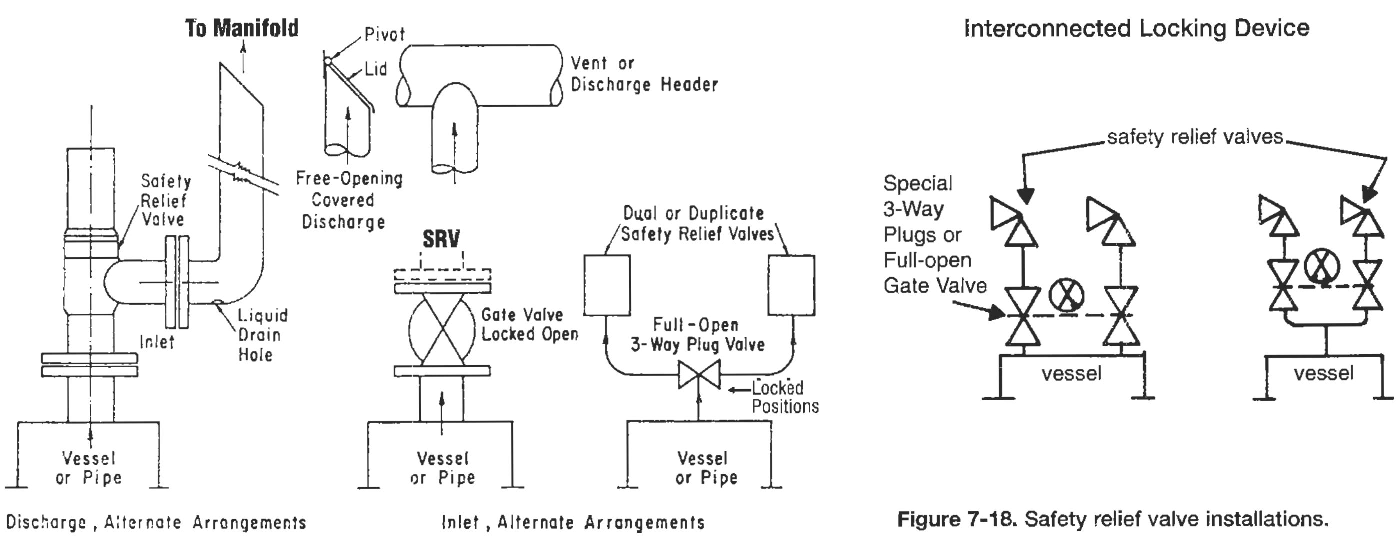


Figure 7-18. Safety relief valve installations.

collection systems are used in order to avoid discharging a high pressure valve into the same header with a low pressure valve. The simultaneous discharge of both valves might create too great a back pressure on the low pressure valve, unless adequate arrangement has been made in the valve design and selection. The balanced safety-relief valve can overcome most of the problems of this type system.

Whenever possible the individual installation of valves is preferred, and these should be connected directly to the vessel or pipe line [1, 28]. If a block-type valve is considered necessary for a single valve installation, it must be of the full open type, and locked open with the key in responsible hands, as stated earlier.

Dual installations are frequently made in continuous processes, to allow switching from one valve to another without shut down of the pressure system. A special three-way plug of full open type is installed directly on the vessel, and the safety valves are attached to it with short piping (Figure 7-18). The three-way valve insures that one side of the safety valve pair is always connected to the vessel, as this pattern valve does not have a blind point during switching (Also, see Figure 7-19). One of the important justifications for this dual arrangement is that safety-relief valves may leak on reseating after discharging. This leak may be caused by a solid particle lodged on the seat. This valve can be removed for repair and cleaning after the process has been switched to the second valve. Each valve must be capable of relieving the full process requirements. Multiple valves may also be individually installed separately on a vessel.

Figure 7-19 illustrates a newer approach at simplifying the dual safety relief valve installation, ASME Sect. VIII, Div. 1, UG-135(b) [1] and API RP-520, Part II Conformance [33]. Note that the SRV valves are mounted on top of each of one dual vertical connections and are bubble tight. Also see cross section view. The flow C_v valves for each size device are available from the manufacturer.

The operational instructions of Anderson Greenwood & Co. for Figure 7-19 are quoted by permission:

> The AGCO Safety Selector Valve body houses a uniquely designed switching mechanism. The internal rotor smoothly diverts flow to either safety relief valve. Conventional direct spring operated valves or pilot operated valves may be used. The inactive valve is totally isolated by external adjustment. To begin switchover, the retraction bushing is rotated to its stop. This separates the isolation disk from the standby valve channel and temporarily "floats" it in the main valve cavity. The index shaft is then rotated 180° to the alternate channel. The retraction bushing is then returned to its original position, securely seating the isolation disk beneath the valve taken out of service. A red pointer indicates which valve is in service and double padlocking provisions allow the safety selector valve to be locked in either safety relief valve position. The padlocks or car seals can only be

Figure 7-19. Safety selector valve for dual safety relief valve installation with switching. By permission, Anderson, Greenwood and Co. © AGGO.

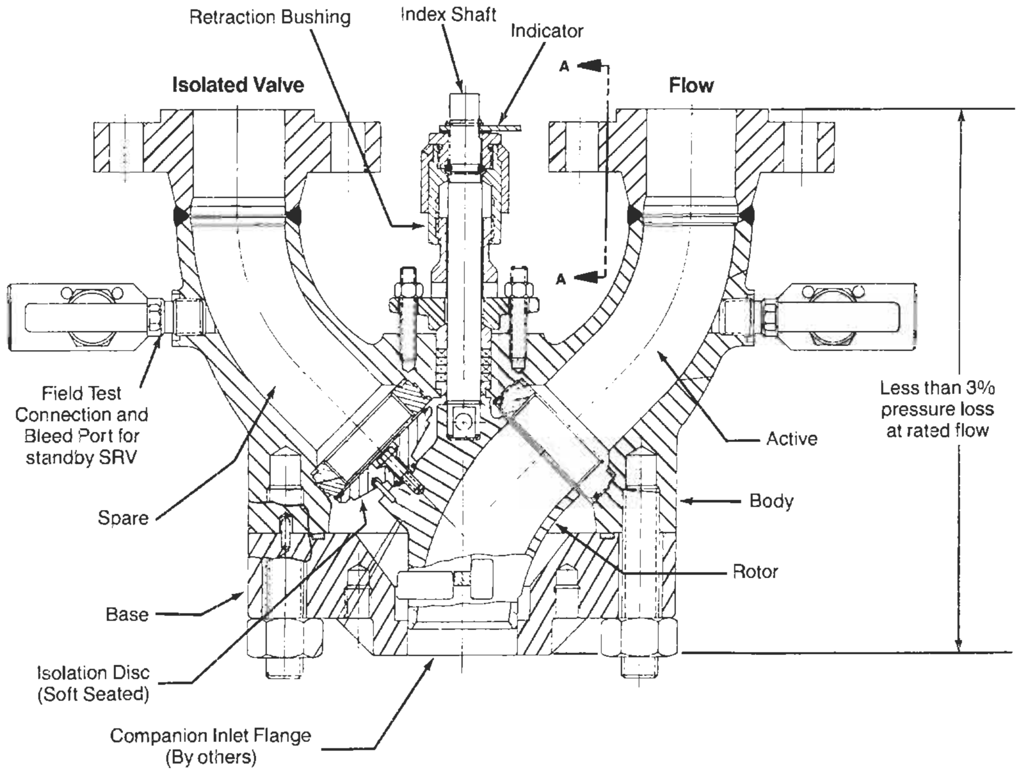


Figure 7-19 cont.

installed with the internals in the proper position. No special tools are necessary for switching.

The alternate concept which has been in use for many years is to fabricate or purchase a Tee connection upon which the two safety relief valves can be mounted on top of their full-port plug or gate valve with required locking lugs.

Rupture disks are often used in conjunction with safety valves as shown in Figures 7-10, 7-11, 7-12, and 7-18.

Inlet piping is held to a minimum, with the safety device preferably mounted directly on the equipment and with the *total system* pressure drop loss to pressure relief valve *inlet* not exceeding 3% of the set pressure in psig, of maximum relief flowing conditions [10]. To conform to code (see ASME code, Sect. VIII, Div. 1-UG-127 [1]) avoid high inlet pressure drop and possible valve chatter:

1. Never make pipe connection smaller than valve or disk inlet
2. Keep friction pressure drop very low, not over 1 to 2 percent of allowable pressure for capacity relief [1, 10, 25, 28, 33].
3. Velocity head loss should be low, not over 2% of allowable pressure for capacity relief [25].

Discharge piping must be sized for low pressure drop at *maximum flow* not only from any one valve, but for the combined flow possibilities in the discharge collection manifold all the way to the vent release point, whether it be a flare, incinerator, absorber or other arrangement [13]. See Figures 7-20 illustrations.

Conventional safety relief valves, as usually installed, produce unsatisfactory performance when variable back pressure exists [10, 33]. See Figure 7-6. The same variable back pressure forces affect the set pressure release also. At low back pressures, the valve flow falls rapidly as compared with the flow for a theoretical nozzle. See Figures 19 and 20 in Ref. [33a].

For conventional valves, pressure drop or variations in back pressure should not exceed 10% of set pressure. Because most process safety valves are sized for critical pressure conditions, the piping must accommodate the capacity required for valve relief and not have the pressure at the end of vent or manifold exceed the critical pressure. Designing for pressure 30% to 40% of critical with balanced valves, yields smaller pipes yet allows proper functioning of the valve. The discharge line size must not be smaller than the valve discharge. Check the manufacturer for valve performance under particular conditions, especially with balanced valves which can handle up to 70% to 80% of set pressure as back pressure.

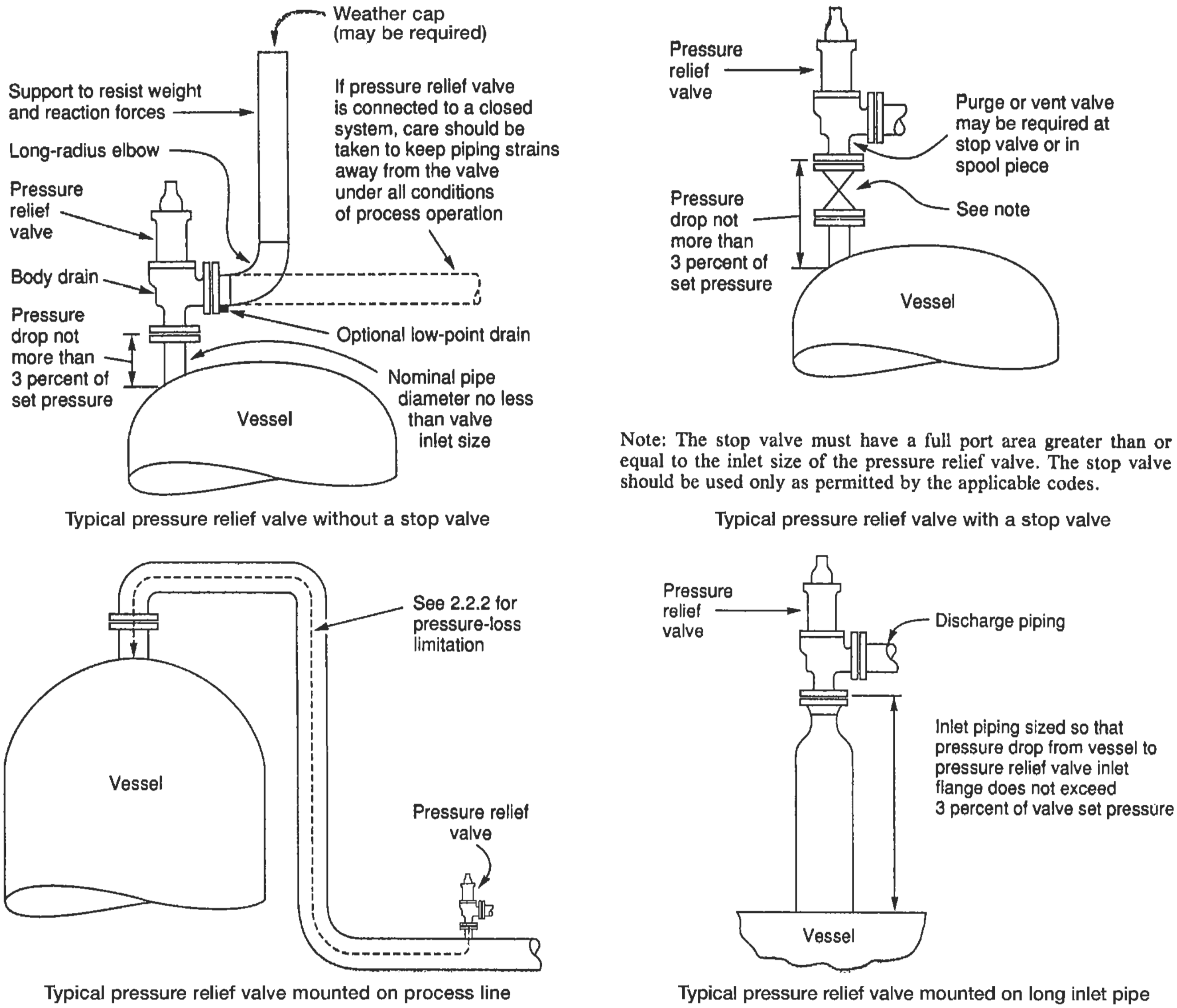


Figure 7-20. Recommended API-520 piping for safety relief valve installations. Reprinted by permission, American Petroleum Institute, *Sizing, Selection and Installation of Pressure Relieving Devices in Refineries, Part II-Installation,* API RP-520, 3rd Ed., Nov. 1988.

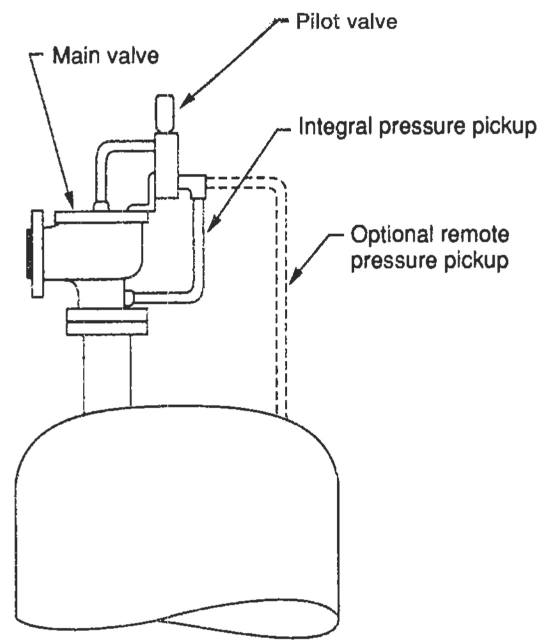
For non-critical flow the maximum back pressure must be set and pressure drop calculated by the usual friction equations.

When process conditions permit, the low pressure range is handled by bursting disks which will relieve down to 2 psig. These disks are also used up to 100,000 psig and above. The rupture pressures and manufacturing ranges of metal disks are given in Tables 7-2 and 7-3. For non-metallic materials such as graphite, bursting pressures are available from the manufacturers. From these manufacturing tolerances it can be seen that the relation of disk bursting pressure to required relieving pressure must be carefully considered. Manufacturing practice is to furnish a disk which will burst within a range of pressures and tolerances, and whose rated pressure is the result of bursting tests of representative sample disks which burst within the range specified. The engineer should specify only ASME code certified disks. It is not possible to obtain a disk for the usual process application set to burst at a given pressure, as is the relieving pressure of a safety valve. Increase in temperature above the disk rating temperature (72°F) decreases the bursting pressure to 70% to 90% depending upon the metal and temperature. See Tables 7-4A and B.

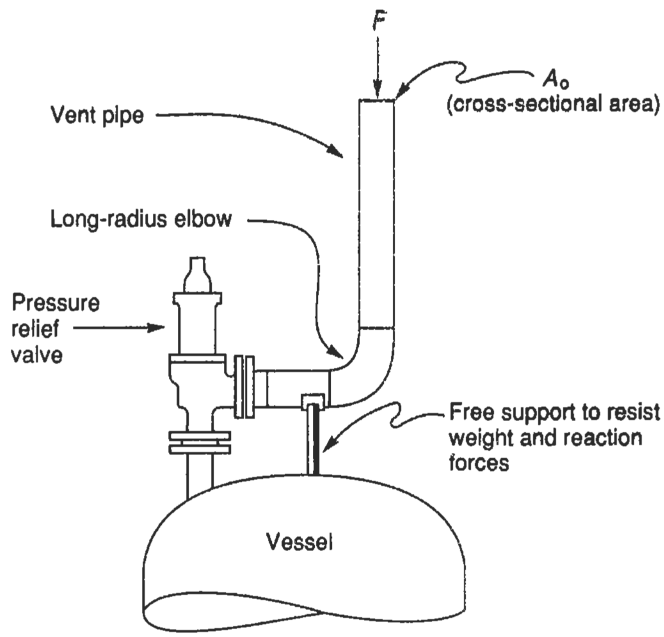
The minimum rupture pressure of disks of various metals and combinations vary so widely that individual manufacturer must be consulted.

For the usual installation, the rupture disk is installed as a single item between special flanges which hold the

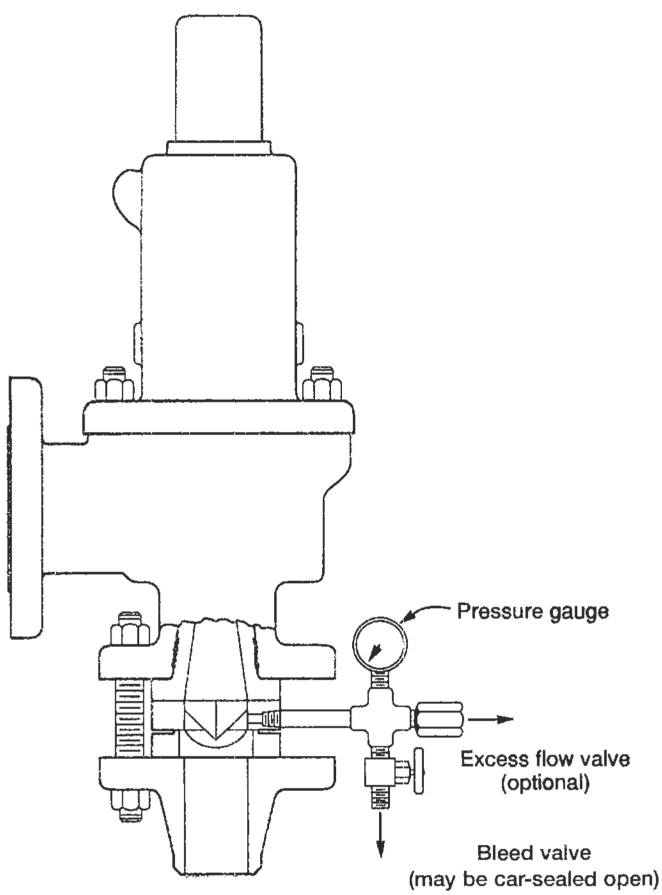
Figure 7-20 cont.



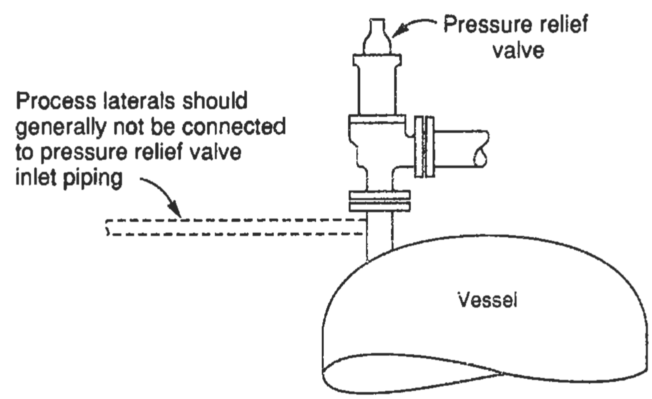
Typical pilot-operated pressure relief valve installation



Typical pressure relief valve installation with vent pipe



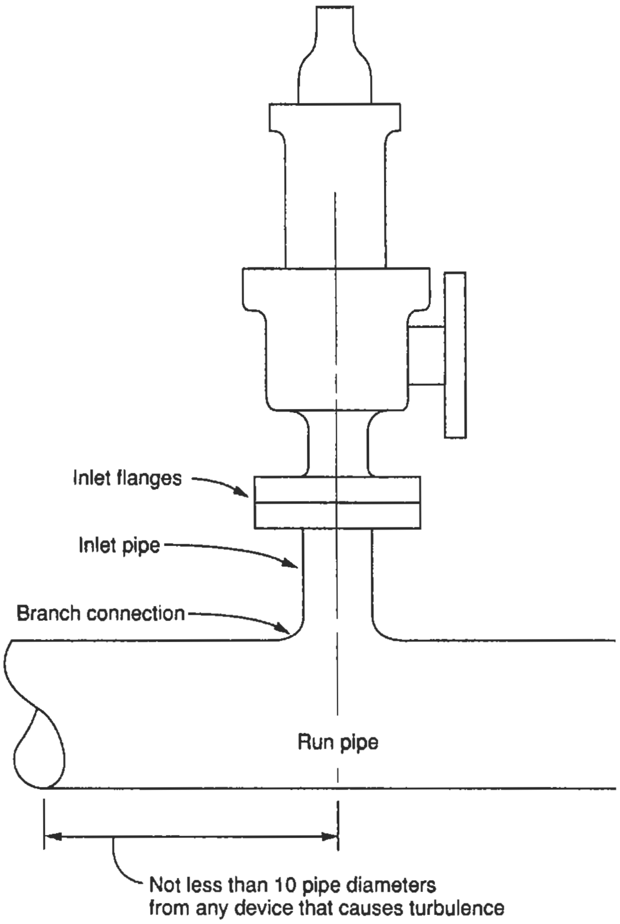
Typical rupture disk assembly installed in combination with a pressure relief valve



Typical installation avoiding process laterals connected to pressure relief valve inlet piping

edges securely and prevent pulling and leakage. If the system is subject to vacuum or pressure surges, a vacuum support must be added to prevent collapse of the sealing disk. The flanges which hold the disk may be slip-on, weld neck, etc. Disks to fit screwed and union-type connections are also available. See Figures 7-8 and 7-9.

The service life of a rupture disk is difficult to predict, since corrosion, cycling pressures, temperature and other process conditions can all affect the useful life and cause premature failure. A graphite-type disk is shown in Figure 7-9. In some processes it is safer to replace disks on a schedule after the life factor has been established, as a planned shut-down is certainly less costly than an emergency one.



Typical installation avoiding excessive turbulence at pressure relief valve inlet.

Figure 7-20 cont.

Rupture disks are often placed below a safety valve to prevent corrosive, tarring or other material from entering the valve nozzle. Only disks which do not disintegrate when they burst (Figures 7-10, 7-11, 7-12, 7-18) can be used below a safety valve, as foreign pieces which enter the valve might render it useless. This is acceptable to certain code applications [1]. These disks are also used to provide secondary relief when in parallel with safety valves set at lower pressures. They can also be installed on the discharge of a safety valve to prevent loss of hazardous vapors, but caution should be used in any serious situation.

Selection Features: Safety, Safety-Relief Valves, and Rupture Disks

Referring to the description and definitions in the introduction for this chapter, it is important to recognize that in order to accomplish the required pressure relief the proper selection and application of type of device is essential.

Table 7-2
Typical Prebulged Solid Metal Disk Manufacturing Ranges and Tolerances

Manufacturing Range/Burst Tolerance @ 72°F			
Specified Burst Pressure Rating, psig	Manufacturing Range % Under	Manufacturing Range % Over	Rated (Stamped) Burst Tolerance %
2–5	−40	+40	±25%
6–8	−40	+40	±20%
9–12	−30	+30	±15%
13–14	−10	+20	±10%
15–19	−10	+20	± 5%
20–50	−4	+14	± 5%
51–100	−4	+10	± 5%
101–500	−4	+ 7	± 5%
501–up	−3	+ 6	± 5%

Note:
1. Special reduced manufacturing ranges can be obtained for the STD prebulged metal disk. ¼, ½, and ¾ ranges are available upon request. Please consult your representative or the factory for additional information.
2. Burst tolerances are the maximum expected variation from the disk's rated (stamped) burst pressure.
3. Standard-type rupture disks comply with ASME code requirements.

Manufacturing Range
The manufacturing range is defined as the allowable pressure range within which a rupture disk is rated. It is based upon the customer specified burst pressure. The manufacturing ranges for Continental's standard rupture disk are outlined in Table 7-2.

Burst Tolerance
After the disk has been manufactured and tested, it is stamped with the rated burst pressure. The rated (stamped) burst pressure is established by bursting a minimum of two disks and averaging the pressures at which the disks burst. This average is the rated (stamped) burst pressure of the disk. Standard rupture disks above 15 psig @ 72°F are provided with a burst tolerance of ±5% of the rated (stamped) burst pressure. This is in accordance with the ASME code. Burst tolerances for disks below 15 psig @ 72°F are outlined in Table 7-2. Burst tolerance applies only to the rated (stamped) burst pressure of the disk. Burst certificates are provided with each disk lot.
By permission, Continental Disc Corporation, Catalog STD-1184.

Safety Valve: normally used for steam service, but suitable for gases or vapors. When used in steam generation and process steam service the valves conform to the *ASME Power Boiler Code* as well as the *ASME Pressure Vessel*

Table 7-3
Typical Metal Disk (Single) Bursting Pressures at 72°F Using Different Metals

Size	Disk Minimum Burst Pressure PSIG @ 72°F (Without Liners)					
	Alum	Silver	Nickel	Monel	Inconel	316SS
¼″	160	450	600	700	1120	1550
½″	65	220	300	350	560	760
1″	29	120	150	180	250	420
1-½″	22	80	100	116	160	275
2″	13	48	60	70	110	150
3″	10	35	45	50	80	117
4″	7	26	35	40	70	90
6″	5	20	25	30	47	62
8″	4	15	20	23	34	51
10″	4	—	16	17	30	43
12″	3	—	13	15	25	36
14″	3	—	11	13	21	31
16″	3	—	10	12	19	28
18″	3	—	9	11	17	24
20″	3	—	8	9	16	22
24″	3	—	—	—	—	—
30″	—	—	—	—	—	—
36″	—	—	—	—	—	—

(—) = Consult factory

*Special designs of some manufacturers may exceed 150,000 psi for small sizes. The pressures listed are generally typical but certainly not the only ones available for the size shown.

Note:
1. Maximum burst pressure depends upon disk size and application temperature. Pressures to 80,000* psig are available.
2. Other materials and sizes are available upon request.
3. Other liner materials are available upon request. Minimum burst pressures will change with change in liner material.
4. For larger sizes or sizes not shown, consult your representative, or the factory. Courtesy, Continental Disc Corp., Bul. 1184, p. 4–5.

Code, Section VIII, and are tested at capacity by the National Board of Boiler and Pressure Vessel Inspectors. This type of valve characteristically "pops" full open and remains open as long as the overpressure exists.

Relief Valve: normally selected for liquid relief service such as hydraulic systems, fire and liquid pumps, marine services, liquefied gases, and other total liquid applications. The valve characteristically opens on overpressure to relieve its rated capacity, and then reseats.

Table 7-4A
Typical Recommended Maximum Temperatures for Metals Used in Disks

	Max. Temperature, F
Aluminum	250–260
Silver	250–260
Nickel	750–800
Monel	800
Inconel	900–1000
316 Stainless steel	900

Source: Various manufacturers' technical catalogs.

Table 7-4B
Typical Recommended Maximum Temperatures for Linings and Coatings with Metals Used with Disks

Teflon® FEP Plastic	400
Polyvinylchloride	150–180
Lead	250

Source: Various manufacturers' technical catalogs.

Safety-Relief Valve: normally selected for vapors and gases as may be found in all types of industrial processes. Characteristically this valve will open only enough to allow the pressure to drop below the set pressure, and then it will reseat until additional overpressure develops. If the pressure persists or increases, then the valve will remain open or increase its opening up to the maximum design, but as the pressure falls the valve follows by closing down until it is fully reseated. Of course, as in any installation of any "safety" type valve, the valve may *not* reseat completely gas tight. In such cases it may be necessary to switch to a stand-by valve and remove the leaking valve for repair. [See Figures 7-7A and 7-7B.]

Special Valves: because of the difficult and special sealing requirements of some fluids such as chlorine and Dowtherm, special valves have been developed to handle the requirements.

Vacuum Relief and Combined Pressure-Vacuum Relief for Low Pressure Conditions: normally used for low pressures such as 1 ounce water to 1.5 psig above atmospheric by special spring or dead weight loading; and for vacuum protection such as 0.5 psi below atmospheric. Usually these conditions are encountered in large process, crude oil, ammonia, etc., storage tanks. See later section covering this topic.

Rupture Disks: used for low as well as high pressure protection of vessels and pipelines where sudden and total release of overpressure is required. Once the disk has ruptured, the process system is exposed to the environment of the backpressure of the discharge system, whether atmospheric or other. The process system is depressurized and the disk must be replaced before the process can be restarted. Typically, the types of disks available are

1. Solid Metal Rupture Disk (Figure 7-8). This is the original type of rupture device, available in various metals and non-metals. It should normally not be used for operating pressures greater than 70% of rupture pressure in a non-corrosive environment. The metal disks use a domed or hemispherical shape, with pressure on the *concave* side. As the pressure internally increases, the metal wall thins as the metal stretches to achieve a smaller radius of curvature. After the wall has thinned sufficiently, it will burst to relieve the pressure and tension loading on the metal. The accuracy of metal disks is ±5%, except for the Reverse Buckling Disk Assembly, which is ±2%. The usual recommended maximum operating temperatures for metal rupture disks is given in Tables 7-4A and B. Also see Figures 7-8G and 8N.

   (a) *Solid metal disk with vacuum support:* When vacuum can occur internally in the system, or when external pressure on the *convex* side of the disk can be greater than the pressure on the concave side of the disk, a vacuum support is necessary to prevent reversal of the disk, Figure 7-8D.

   (b) *Solid metal disk with rating near minimum for size:* With a rating pressure near the minimum available for the size and material of construction a special thin disk is attached to the under and possibly the upper sides of the rupture disk to ensure freedom from deformity caused by the condition of the disk holding surfaces, Figure 7-8E. There are several versions of what to include under such conditions, therefore it is advisable to clearly explain the installation conditions and application to the manufacturer.

   (c) *Composite rupture disk:* This type consists of a metal disk (not necessarily solid, it may have slots) protected by an inner and/or an outer membrane seal, Figure 7-8E. There are several possible arrangements, including vacuum support, as for the styles of paragraph (b) above. This general class has the same use-rating limitation as for the solid disk.

   (d) *Reverse acting or buckling disk assembly:* This design allows the disk to be operated in a system at up to 90% of its rated burst pressure. The pressure is operating on the *convex* side of the disk and when bursting pressure is reached, the disk being in compression reverses with a snap action at which time the four knife edges, Figure 7-8G cut the metal and it clearly folds back without fragmentation. There is another version of the same concept of reverse buckling, but it uses a pre-scored disk and thereby omits the knife blades. These types of disks do not need vacuum supports, unless there is unusually high differential pressure across the disk.

2. Graphite Rupture Disk (Figure 7-9). There are special designs of disks and disk assemblies for specific applications, and the manufacturer should be consulted for his recommendation. Disks are available for pressure service, pressure-vacuum applications, high temperature conditions, and close tolerance bursting conditions. The bursting accuracy of most designs is ±5% for rated pressures above 15 psig and ±0.75 psig for rated pressures 14 psig and below. It should be noted that these ratings are not affected by temperature up to 300°F.

A new concept in graphite disks includes addition of a fluorocarbon film barrier between the process and the disk, and is termed a duplex disk. These disks are suitable for temperatures to 392°F, with accuracies as just mentioned.

Graphite disks are normally used in corrosive services and/or high temperature situations where metal wall thickness and corrosion rates make the metal units impractical because of unpredictable life cycles. The disks are available down to 1 psig ±0.75 psi, and are not affected by fatigue cracking. An interesting feature is the use of standard ASA (ANSI) flanges rather than special flanges. See Figure 7-9C.

It is important to recognize here also that once the disk bursts the system is depressured, and there will be fragments of graphite blown out with the venting system. Special discharge designs are often used to prevent plugging of discharge pipe and fragments from being sprayed into the surrounding environment.

Calculations of Relieving Areas: Safety and Relief Valves

References to the ASME Code [1] and the API Code [10] [33] are recommended in order that the design engineer may be thoroughly aware of the many details and special situations that must be recognized in the final sizing and selection of a pressure relieving device. All details of these codes cannot be repeated here; however, the usually important requirements are included for the typical chemical and petrochemical application for the guidance of the engineer.

Before performing any calculations, a thorough examination of the possible causes and flow conditions of temperature and pressure should be evaluated. From this list, select the most probable and perhaps the worst case possibility and establish it as a design basis, Figure 7-14. See [80].

When the possibilities of internal explosion or runaway chemical reaction exists, or are even suspected, they must also be rigorously examined and calculations performed to establish the magnitude of the flow, pressure, and temperature problems. Select the worst condition and plan to provide for its proper release to prevent rupture of equipment. This latter situation can only be handled by application of rupture disks and/or remote sensing and predetermined rupture of the disks (see Figures 7-5A, 7-8K and 7-8L) or remote sensing and application of quenching of the reaction/developing explosive condition by automatic process action and/or commercial application of quenching medium. See later discussion under Explosions.

Standard Pressure Relief Valves Relief Area Discharge Openings

The "orifice" area of these devices (see illustrations) is at the outlet end of the SRV nozzle through which the discharging vapor/gases/liquids must pass. These values are identified in industry as: (valve body inlet size in.) × (orifice letter) × (valve body outlet size, in.). For example, a valve would be designated 3E4.

The standard orifice area designations are (also refer to mechanical illustrations of valves, previously shown this chapter):

Orifice letter	D	E	F	G	H	J
Area, sq. in.	0.11	0.196	0.307	0.503	0.785	1.287
Orifice letter	K	L	M	N	P	Q
Area, sq. in.	1.838	2.853	3.600	4.340	6.380	11.05
Orifice letter	R	T	V*	W	W2*	X*
Area, sq. in.	16.0	26.0	42.19	57.26 / 60.75	93.6	101.8
Orifice letter	Y*	Z*	Z_2*	AA	BB	BB2
Area, sq. in.	128.8 / 82.68	159.0 / 90.95	— / 108.86	— / 136.69	— / 168.74	— / 185.00

*Note: These letters and orifice areas are not consistent for these large orifices between various manufacturers. Some sizes go to 185 sq in., which is a very large valve. When two values are shown, they represent two different published values by manufacturers.

Sizing Safety Relief Type Devices for Required Flow Area at Time of Relief

> Before initiating any calculations, it is necessary to establish the general category of the pressure relief valve being considered. This section covers conventional and balanced spring-loaded types.
>
> Given the rate of fluid flow to be relieved, the usual procedure is to first calculate the minimum area required in the valve orifice for the conditions contained in one of the following equations. In the case of steam, air or water, the selection of an orifice may be made directly from the capacity tables if so desired.
>
> In either case, the second step is to select the specific type of valve that meets the pressure and temperature requirements.
>
> General equations are given first to identify the basic terms which correlate with ASME Pressure Vessel Code, Section VIII.
>
> It is recommended that computations of relieving loads avoid cascading of safety factors or multiple contingencies beyond the reasonable flow required to protect the pressure vessel.*

*Extracted by permission from Teledyne-Farris Engineering Catalog.

Effects of Two-Phase Vapor-Liquid Mixture on Relief Valve Capacity

Many process systems when at conditions for safety relief valve discharge also are not single phase of all liquid (through the valve) or all vapor, but a mixture either inside the "containing" vessel or quite often as the fluid passes through the valve orifice and the liquid flashes to partial vapor, or the flashing starts just ahead of the orifice. Here a mixture attempts to pass through the orifice, and the size must be sufficient or a restriction will exist and pressure will build up in the vessel due to inadequate relief. This problem was of considerable concern to the Design Institute for Emergency Relief of the American Institute of Chemical Engineers during their studies [67]. As a result, considerable research was performed leading to design techniques to handle this problem. The details are more than can be presented here; therefore, the designer is referred to the references in the bibliography of this chapter. Also see Leung [77] for detailed procedure and additional references.

The API-RP-520 [10] recommends calculating the amount of vapor flashed and the amount of residual liquid (unflashed) and then sizing valve orifices for each condition. Select a valve(s) area that has a total area at least equal to the sum of these two areas. Before settling for this approach, this author recommends examining the details of the DIERS work noted in the above paragraph and at least comparing the results. Keep in mind that the problem of two-phase flow under the relieving conditions cannot be ignored.

Sizing for Gases or Vapors or Liquids for Conventional Valves with Constant Backpressure Only

This type of valve may be used when the variations in backpressure on the valve discharge connection do not exceed 10% of the valve *set* pressure, and provided this backpressure variation does not adversely affect the set pressure.

Procedure

1. For a new installation, establish pressure vessel normal maximum operating pressure, and temperature, and then the safe increment above this for *vessel design* conditions and determine the maximum allowable working pressure (MAWP) of the new vessel. (Have qualified fabricator or designer establish this. See previous discussion of topic.)
2. Establish the maximum *set* pressure for the pressure relieving valves as the MAWP, or lower, but *never* higher.
3. Establish actual *relieving* pressure (and corresponding temperature) from Figure 7-7A (at 110% of set pressure for non-fire and non-explosive conditions). Explosive conditions may require total separate evaluation of the set pressure (*never above the MAWP*), which should be lower or staged; or, most likely, will not be satisfied by a standard SRV due to the extreme rapid response needed.

   The capacity for flow through the valve is established by the conditions of *this* paragraph.

4. For *existing vessel* and re-evaluation of pressure relieving requirements, start with the known MAWP for the vessel, recorded on the vessel drawings and on its ASME certification papers. Then follow step 2 and 3 above.

Establish critical flow for gases and vapors

Critical or sonic flow will usually exist for most (compressible) gases or vapors discharging through the nozzle orifice of a pressure relieving valve. The rate of discharge of a gas from a nozzle will increase for a decrease in the *absolute* pressure ratio P_2/P_1 (exit/inlet) until the linear velocity in the throat of the nozzle reaches the speed of sound in the gas at that location. Thus, the critical or sonic velocity or critical pressures are those conditions that exist when the gas velocity reaches the speed of sound. At that condition, the actual pressure in the throat will not fall below P_1/r_c even if a much lower pressure exists downstream [36]. The maximum velocity at outlet end (or restriction) in a pipe or nozzle is sonic or critical velocity. This is expressed [9]:

$$v_s = \sqrt{kgRT} = \sqrt{kg\,(144)P'\overline{V}} \qquad (7\text{-}5)$$

where
- k = ratio of specific heats at constant pressure/constant volume, c_p/c_v see Table 7-5.
- v_s = sonic velocity of gas, ft/sec
- g = acceleration of gravity, 32 ft/sec/sec
- R = individual gas constant = (MR/M) = 1544/M
- MR = universal gas constant = 1544
- M = mol weight
- T = upstream absolute temperature, °R
- $\overline{V}$ = specific volume of fluid, cu ft/lb
- $P_1 = P'$ = upstream pressure, psi abs
- d = pipe inside diameter, in.
- W = gas rate, lb/hr
- Z = gas compressibility factor
- $P_c = P_{crit}$ = critical pressure, psia

The critical pressure at a pipe outlet is [33c]:

$$P_{crit} = [W/(408\,d^2)] - (Z\,T/M)^{1/2},\ \text{psiabs} \qquad (7\text{-}6)$$

The velocity v_s will occur at the outlet end or in a restricted area [9] when the pressure drop is sufficiently high. The condition of temperature, pressure and specific volume are those occurring at the point in question.

Critical pressure will normally be found between 53% and 60% of the upstream pressure, P′, at time of relief from overpressure, including accumulation pressure in psia. That is, P′ represents the actual pressure at which the relief device is "blowing" or relieving, which is normally above the set pressure by the amount of the accumulation pressure, (see Figure 7-7A).

Thus, if the downstream or backpressure on the valve is less than 53%–60% (should be calculated) of the values of P′, note above, critical (sonic) flow will usually exist. If the downstream pressure is over approximately 50% of the relief pressure, P′, the actual critical pressure should be calculated to determine the proper condition. Calculation of critical pressure [29]:

$$P_c = P_1\,[2/(k+1)]^{k/(k-1)} \qquad (7\text{-}7)$$

$$P_c/P_1 = r_c = [2/(k+1)]^{k/(k-1)} \qquad (7\text{-}8)$$

For critical flow conditions $@\beta \leqq 0.2$.

This equation is conventionally solved by Figure 7-21.

Table 7-5
Properties of Gases and Vapors

Gases and Vapors	Hydrocarbons Reference Symbols	Chemical Formula	Mol. Wt.	R = 1545 / Mol. Wt.	Critical Conditions: Pressure (Psia)	Critical Conditions: Temperature (°R)	Boiling Point (F) @ 14.7 Psia	Specific Volume Cu ft/lb @ 14.7 Psia & 60F (Z Factor Accounted For)	Latent Heat of Vaporization (Btu/lb @ 14.7 Psia)	Specific heat Constant Pressure (C_p @ 60F)	Specific heat Constant Volume (C_v @ 60F)	Specific heat ratio $K = C_p/C_v$
1. Acetylene	$C_2^{\equiv}$	C_2H_2	26.04	59.5	905	557	-118.7	14.37	356.0	.397	.320	1.24
2. Air		N_2+O_2	28.97	53.3	547	239	-317.7	13.09	91.8	.240	.171	1.40
3. Ammonia		NH_3	17.03	90.8	1657	731	- 28.1	22.10	590.0	.523	.399	1.31
4. Argon		A	39.94	38.7	705	272	- 30.3	9.50	71.7	.125	.075	1.66
5. Benzene		C_6H_6	78.11	19.8	714	1013	176.2	*	169.3	.240	.215	1.12
6. Iso-Butane	iC_4	C_4H_{10}	58.12	26.6	529	735	10.9	6.26	157.8	.387	.352	1.10
7. n - Butane	nC_4	C_4H_{10}	58.12	26.6	551	766	31.1	6.25	165.9	.397	.363	1.09
8. Iso-Butylene	$iC_4^=$	C_4H_8	56.10	27.5	580	753	19.6	6.54	169.5	.368	.333	1.10
9. Butylene	$nC_4^=$	C_4H_8	56.10	27.5	583	756	20.7	6.54	167.9	.327	.292	1.11
10. Carbon Dioxide		CO_2	44.01	35.1	1073	548	-109.3	8.53	248.8(1)	.199	.153	1.30
11. Carbon Monoxide		CO	28.01	55.1	514	242	-313.6	13.55	91.0	.248	.177	1.40
12. Carbureted Water Gas (3)		-	19.48	79.5	454	235	-	19.60	-	.281	.208	1.35
13. Chlorine		Cl_2	70.91	21.8	1119	751	- 29.6	5.25	123.8	.115	.084	1.36
14. Coke Oven Gas (3)		-	11.16	138.5	407	197	-	34.10	-	.679	.514	1.32
15. n - Decane	nC_{10}	$C_{10}H_{22}$	142.28	10.9	312	1115	345.2	*	120.0	.401	.387	1.03
16. Ethane	C_2	C_2H_6	30.07	51.5	708	550	-127.5	12.52	210.7	.410	.343	1.19
17. Ethyl Alcohol		C_2H_5OH	46.07	33.5	927	930	172.9	*	368.0	.370	.328	1.13
18. Ethyl Chloride		C_2H_4Cl	64.52	23.9	764	829	54.4	5.59	168.5	.274	.230	1.19
19. Ethylene	$C_2^=$	C_2H_4	28.05	55.1	749	510	-154.7	13.40	207.6	.361	.291	1.24
20. Flue Gas (2)		-	30.00	51.5	563	264	-	12.63	-	.240	.174	1.38
21. Helium		He	4.00	386.0	33	9	-450.0	94.91	9.9	1.24	.748	1.66
22. n - Heptane	nC_7	C_7H_{16}	100.20	15.4	397	973	209.2	*	136.2	.399	.379	1.05
23. n - Hexane	nC_6	C_6H_{14}	86.17	17.9	434	915	155.7	*	144.8	.398	.375	1.06
24. Hydrogen		H_2	2.02	765.0	188	60	-423.0	187.80	194.0	3.41	2.42	1.41
25. Hydrogen Sulphide		H_2S	34.08	45.3	1306	673	- 76.5	11.00	236.0	.254	.192	1.32
26. Methane	C	CH_4	16.04	96.4	673	344	-258.8	23.50	219.7	.526	.402	1.31
27. Methyl Alcohol		CH_3OH	32.04	48.3	1157	924	148.1	*	473.0	.330	.275	1.20
28. Methyl Chloride		CH_3Cl	50.49	30.6	968	750	- 10.8	6.26	184.2	.200	.167	1.20
29. Natural Gas (3)		-	18.82	82.1	675	379	-	20.00	-	.485	.382	1.27
30. Nitrogen		N_2	28.02	55.1	492	228	-320.0	13.53	85.8	.248	.177	1.40
31. n - Nonane	nC_9	C_9H_{20}	128.25	12.0	335	1073	303.4	*	125.7	.400	.385	1.04
32. Iso-Pentane	iC_5	C_5H_{12}	72.15	21.4	483	830	82.1	*	145.7	.388	.361	1.08
33. n - Pentane	nC_5	C_5H_{12}	72.15	21.4	485	847	96.9	*	153.8	.397	.370	1.07
34. Pentylene	$C_5^=$	C_5H_{10}	70.13	22.0	586	854	86.0	*	149.0	.382	.353	1.08
35. n - Octane	nC_8	C_8H_{18}	114.22	13.5	362	1025	258.2	*	131.7	.400	.382	1.05
36. Oxygen		O_2	32.00	48.3	730	278	-297.4	11.85	92.0	.219	.156	1.40
37. Propane	C_3	C_3H_8	44.09	35.1	617	666	- 43.7	8.45	183.5	.388	.342	1.13
38. Propylene	$C_3^=$	C_3H_6	42.08	36.7	668	658	- 53.9	8.86	188.2	.354	.307	1.15
39. Refinery Gas (High Paraffin) (4)		-	28.83	53.6	674	515	-	13.20	-	.395	. 33	1.20
40. Refinery Gas (High Olefin) (4)		-	26.26	58.8	639	456	-	14.40	-	.397	. 33	1.20
41. Sulphur Dioxide		SO_2	64.06	24.1	1142	775	14.0	5.80	168	.147	.118	1.24
42. Water Vapor		H_2O	18.02	85.8	3208	1166	212.0	*	970.3	.445	.332	1.33

* These substances are not in a vapor state at 14.7 psia and 60 F and therefore sp. vol. values are not listed.

NOTES: Most values taken from Natural Gasoline Supply Men's Association Engineering Data Book, 1951 - Sixth Edition.

1 - Heat of Sublimation.

2 - Flue gas - Approximate values based on 80.5% N_2, 16% CO_2, 3.5% O_2. Actual properties depend on exact composition. Reference: Mark's Engineering Handbook

3 - Carbureted Water Gas, Coke Oven Gas and Natural Gas. Based on average compositions. Actual properties will differ depending on exact composition. Reference: Perry's Handbook (3rd Edition)

4 - Refinery gas (High Paraffin) - Has a greater mol. percent of saturated hydrocarbons (example C_2H_6)
Refinery gas (High Olefins) - Has a greater mol. percent of unsaturated hydrocarbons (example C_2H_4)
Reference: Perry's Handbook (3rd Edition).

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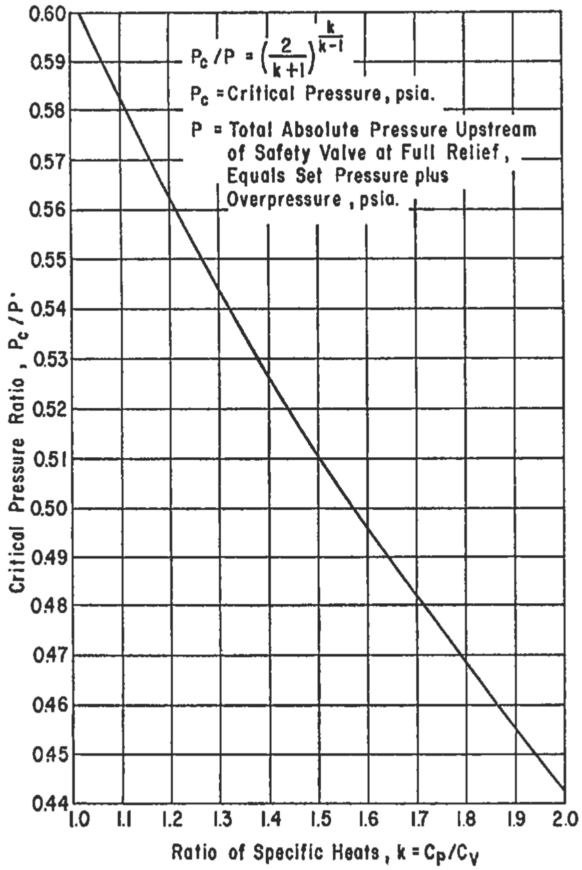


Figure 7-21. Critical backpressure ratio for vapors and gases.

At critical conditions, the maximum flow through the nozzle or orifice is: [29]

$$W_{max\ (critical\ flow)} = C_o A P_o \sqrt{\frac{k g_c M}{R_g T_o}\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}} \qquad (7\text{-}9)$$

where
- M = mol wt of vapor or gas
- T_o = temp of service, °R
- R_g = ideal universal gas constant = 1544, ft lb/lb-mol-°R, also = MR
- C_o = discharge coefficient for sharp-edged orifice = 0.61 for Reynolds Number > 30,000 and not sonic
- C_o = 1.0 for sonic flow, C_o increases from 0.61 to 1.0. Use 1.0 to be conservative [29] [36], 5th Ed.]
- A = area of opening, orifice, or hole, or nozzle, sq. ft.
- $P_1 = P_o$ = upstream pressure, lb/sq foot, abs, (psfa)
- g = 32.12 ft/sec/sec
- M = mol. wt. of vapor or gas
- β = ratio of orifice diameter/pipe diameter (or nozzle inlet diameter)
- W_{max} = maximum mass flow at critical or choked conditions, lb/sec
- $P_c = P_{crit}$ = critical flow throat pressure, psia = sonic = choked pressure = maximum downstream pressure producing maximum flow

When/if the downstream pressure exceeds the critical flow pressure, then sub-critical pressure will occur and the equations for sub-critical flow should be used.

When the downstream pressure is less than (or below) the critical or choked pressure, the velocity and fluid flow rate at a restriction or throat will not/cannot be increased by lowering the downstream pressure further, and the fluid velocity at the restriction or throat is the velocity of sound at the conditions [29].

The critical or sonic ratio is conveniently shown on Figure 7-21, but this does not eliminate the need for calculating the P_c/P_1 ratio for a more accurate result.

Example 7-2: Flow through Sharp Edged Vent Orifice (adapted after Ref. [29])

A small hole has been deliberately placed in a vessel near the top to provide a controlled vent for a nitrogen purge/blanket. The hole is 0.2-inch diameter with the vessel operating at 150 psig at 100°F. Determine the flow through this vent hole. Assume it acts as a sharp edged orifice.

k (for nitrogen) = 1.4

P_c/P_1 = See Equation 7-7 $[2/(k + 1)]^{k/k-1}$

$P_c = (150 + 14.7)\ (2/1.4 + 1)^{1.4/1.4 - 1} = 86.9$ psia, critical pressure

Hole area = $A = \pi d^2/4 = \pi\ (0.2)^2/4 = 0.0314$ sq in. = 0.0002181 sq ft

Discharge coef. C_o = assumed = 1.0 (Note, could calculate Re to verify.)

Inside pressure = 150 psig + 14.7 = 164.7 psia

T_o = 100 + 460 = 560°R

$$W_{max} = [1.0\ (0.0002181)(164.7)(144\ \text{sq in./sq ft})]\left[\sqrt{\left[\frac{(1.4)(32.12)(28)}{(1545)(560)}\right]\left(\frac{2}{1.4+1}\right)^{(1.4+1)/(1.4-1)}}\ \right]$$

critical flow rate, W_{max} = 0.002518 lb/sec

Orifice Area Calculations [68]

Calculations of Orifice Flow Area for Conventional Pressure Relieving Valves, and Flow is Critical (sonic) Through Part of Relieving System, i.e., backpressure is less than 55% of the absolute relieving pressure (including set pressure plus accumulation). See Figure 7-7A, use

$K_b = 1.0$ (Figure 7-26), constant backpressure with *variation* not to exceed 10% of set pressure.

a. for vapors and gases, in lb/hr; $K_b = 1.0$; "C" from Figure 7-25, P is relieving pressure absolute, psia

$$A = \frac{W\sqrt{TZ}}{CK_d\ PK_b\ \sqrt{M}},\ \text{sq in.} \quad \text{(Effective net discharge area)} \qquad (7\text{-}10)$$

b. For vapors and gases, in SCFM, $K_b = 1.0$

$$A = \frac{V\sqrt{GTZ}}{1.175\ CK_d\ PK_b},\ \text{sq in.} \qquad (7\text{-}11)$$

c. For steam, in lb/hr; $K_b = 1.0$ and $K_{sh} = 1.0$ for saturated steam when backpressure is below 55% of absolute relieving pressure

$$A = \frac{W_s}{51.5\ K_d\ PK_b\ K_{sh}\ K_n},\ \text{sq in.} \qquad (7\text{-}12)$$

d. For air, in SCFM; $K_b = 1.0$, when backpressure is below 55% of absolute relieving pressure

$$A = \frac{V_a\ \sqrt{T}}{418\ K_d\ PK_b} \qquad (7\text{-}13)$$

e. For liquids, GPM;
$K_p = 1.0$ @ 10% overpressure
$K_u = 1.0$ at normal viscosities
$\Delta P = P_1 - P_2$ = upstream pressure, psig (set + overpressure) − total backpressure, psig

ASME Code valves: Board Certified for liquids only.

$$A = \frac{V_L\ \sqrt{G}}{38\ K_d\ K_w\ K_u\ \sqrt{P_1 - P_2}} \qquad (7\text{-}14)$$

f. Non-ASME Code Liquid Valves [33a] non-board certified for liquids, but code acceptable for other services. K_p from Figure 7-22, $K_d = 0.62$, and 25% overpressure.

$$A = \frac{V_L\sqrt{G}}{38\ K_d\ K_w\ K_p\ K_u\ \sqrt{1.25\ P_1 - P_2}} \qquad (7\text{-}15)$$

To apply the viscosity correction K_u, a preliminary or trial calculation should be made for the areas required using the equation of paragraph (e) above or the modified equation (still ASME conformance [33] but not capacity certified). A simplified equation based on the ASME Pressure Vessel Code equations, Section VIII, Div. 1, Mandatory Appendix XI uses K coefficient of discharge in the equations, where K is defined as 90% of the average K_d of certified tests with compressible or incompressible fluids, see reference [68], pg 40.

For first trial, assume K_u for viscosity = 1.0

For final calculation use K_u from Figures 7-23 or 7-24 and substitute in the above equation. Determine the needed Reynolds number, Re, using the next size larger orifice. Area is determined from that made in the first trial calculation [33].

$$Re = V_L\ \frac{(2800\ G)}{\mu\sqrt{A}},\ \text{or} \qquad (7\text{-}16)$$

$$Re = 12{,}700\ V_L/\left(U\sqrt{A}\right),\quad \text{(Do not use when } U < 100 \text{ SUS)} \qquad (7\text{-}17)$$

Re = Reynolds number
μ = absolute viscosity @ flowing temperature, centipoise
P_1 = set pressure, psig
U = viscosity @ flowing temperature, Saybolt Universal Seconds (See Appendix A-12 and A-13)
P_2 = total backpressure, psig

Calculations of Orifice Flow Area using Pressure Relieving Balanced Bellows Valves, with Variable or Constant Back Pressure. Must be used when backpressure variation exceeds 10% of the *set* pressure of the valve. Flow may be critical or non-critical for balanced valves. All orifice areas, A, in sq in. [68]. The sizing procedure is the same as for conventional valves listed above (Equations 7-10 ff), but uses equations given below incorporating the correction factors K_v and K_w. With variable backpressure, use maximum value for P_2 [33a, 68].

a. For vapors or gases, lb/hr

$$A = \frac{W\sqrt{TZ}}{CK_d\ PK_v\ \sqrt{M}},\ \text{sq in.} \qquad (7\text{-}18)$$

b. For vapors or gases, SCFM

$$A = \frac{V\sqrt{GTZ}}{1.175\ CK_d\ PK_v},\ \text{sq in.} \qquad (7\text{-}19)$$

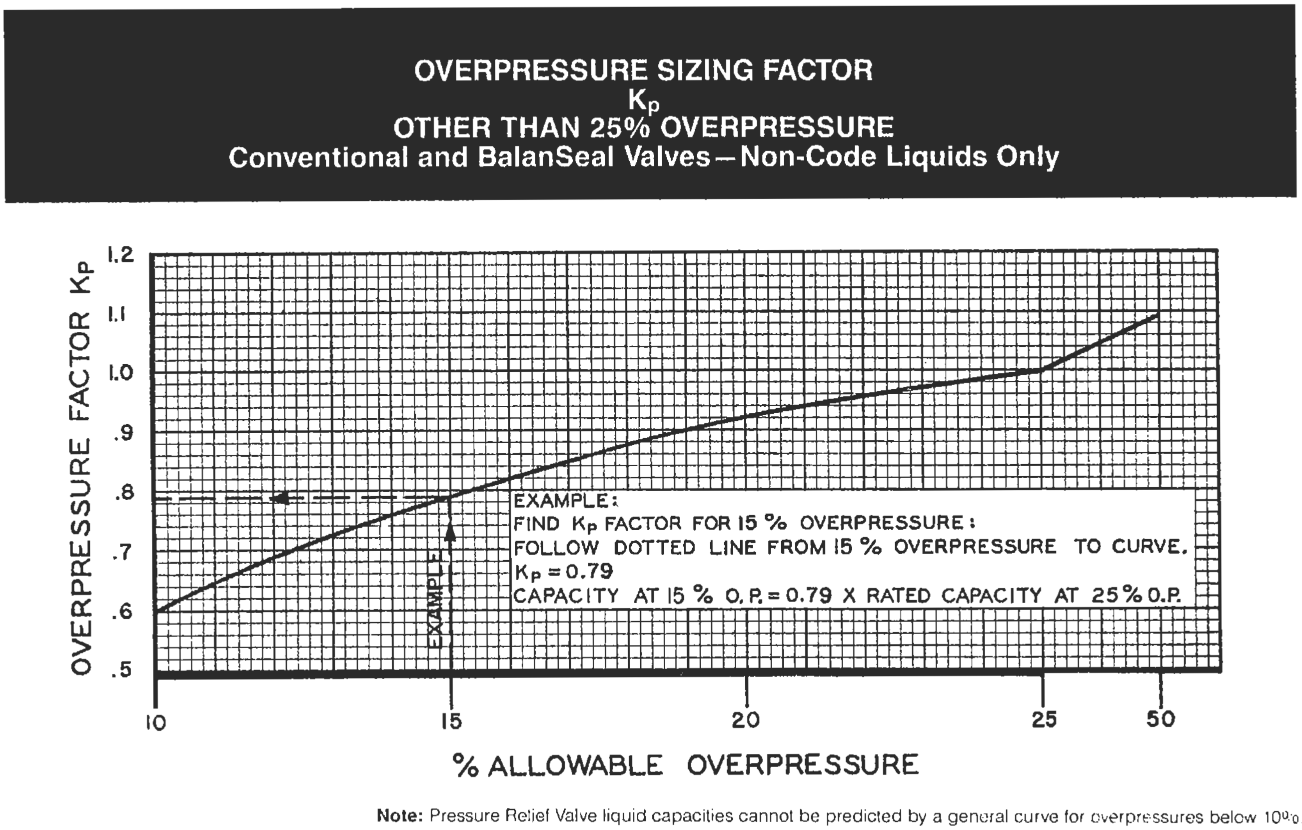


Figure 7-22. Liquids overpressure sizing factor, K_p, for other than 25% overpressure. Applies to Non-code liquids only using conventional and balanced valves. By permission, Teledyne Farris Engineering Co.

c. For steam, lb/hr

$$A = \frac{W_s}{51.5\ K_d K_v K_{sh} K_n P}, \text{ sq in.} \tag{7-20}$$

d. For air, SCFM

$$A = \frac{V_a \sqrt{T}}{418\ K_d P K_v}, \text{ sq in.} \tag{7-21}$$

e. For liquids, GPM; ASME Code valve

$$A = \frac{V_L \sqrt{G}}{38.0\ K_d\ K_w\ K_u\ \sqrt{\Delta P}}, \text{ sq in.} \tag{7-22A}$$

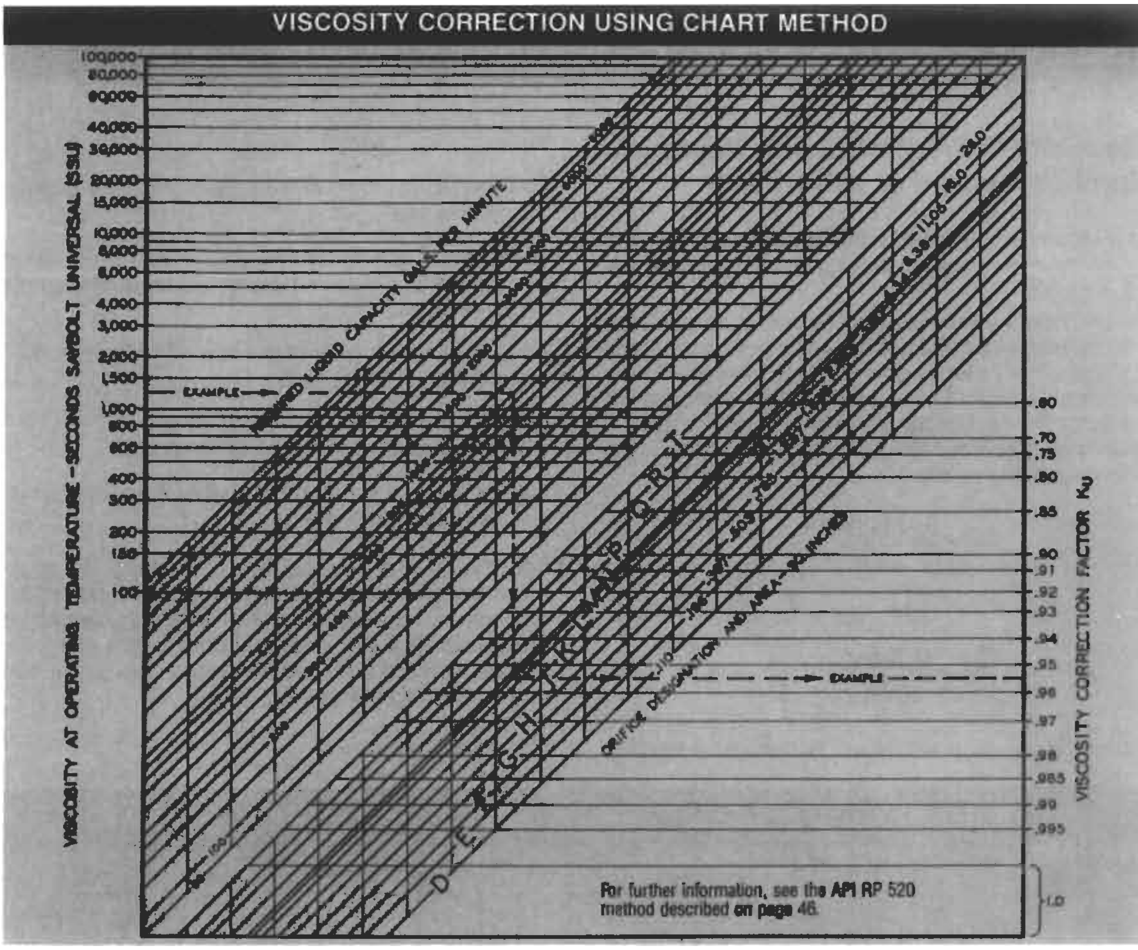
f. For liquids, GPM, non-ASME Code valve

$$A = \frac{V_L \sqrt{G}}{38.0\ K_d\ K_p\ K_w\ K_u \sqrt{(1.25\ P_1) - P_2}}, \text{ sq in.} \tag{7-22B}$$

When the backpressure, P_2, is variable, use the maximum value.

where (Courtesy of Teledyne Farris Engineering Co. [68]):

- A = required orifice area in square inches. This is as defined in the ASME Code and ANSI/API Std 526.
- W = required vapor capacity in lb/hr
- W_s = required steam capacity in lb/hr
- V = required gas capacity in SCFM
- V_a = required air capacity in SCFM
- V_L = required liquid capacity, gal/min (gpm)
- G = specific gravity of gas (air = 1.0) or specific gravity of liquid (water = 1.0) at actual discharge temperature. A specific gravity at any lower temperature will obtain a safe valve size.
- M = average molecular weight of vapor
- P = relieving pressure in lbs per square inch abs. = [set pressure, psig + overpressure, psig + 14.7,] psia. Minimum overpressure = 3 psi.
- P_1 = set pressure at inlet, psig
- P_2 = back pressure at outlet, psig



The viscosity of the liquid may reduce the velocity and capacity enough to require a larger orifice size than the usual liquid capacity equation would indicate. This simplified viscosity chart and the "K_u" viscosity correction factors obtainable from it are for use in properly sizing relief valves intended for viscous liquid service. Equations and graphs used in preparing this chart reflect conservative engineering data on the subject.

For viscous liquid service, it is advisable to allow 25% overpressure where permissable, to size conservatively, and to consider the use of the bellows and/or steam jacketed bodies (see page 63) for the purpose of isolating the moving parts and to prevent freezing of the lading fluid.

Sizing Method

Since the viscosity correction factor is dependent upon the actual orifice area, direct solution is not possible and a trial orifice size must be found before the "K_u" can be determined accurately.

***Example:**

Viscosity—Secs. Saybolt Universal	1250 SSU @ 100° F.
Capacity Required	800 GPM
Set Pressure	100 PSIG
Constant Back Pressure	10 PSIG
Differential Pressure (1.25 P_1-P_2)	115 PSIG
Allowable Overpressure	25%
Specific Gravity	0.98@100°F.
Relieving Temperature	100°F.

Step 1— Calculate Trial Orifice: Calculate the trial required orifice area from the liquid equation on page 40 (Alternate used in this example).

$$A = \frac{V_l}{24.3\sqrt{1.25\,P_1 - P_2}\;K_p\,K_q\,K_u} = \frac{800}{24.3\sqrt{125-10}\;(1)(1.010)(1)} = 3.04\text{ sq. in.}$$

If BalanSeal valve construction is used and variable back pressure conditions exist, use the maximum back pressure to determine P_d in equation and correct K_w factor (see graph page 44).

Use the following equation.

$$A = \frac{V_L}{24.3\sqrt{1.25\,P_1 - P_2}\;K_p\,K_q\,K_u\,K_w}$$

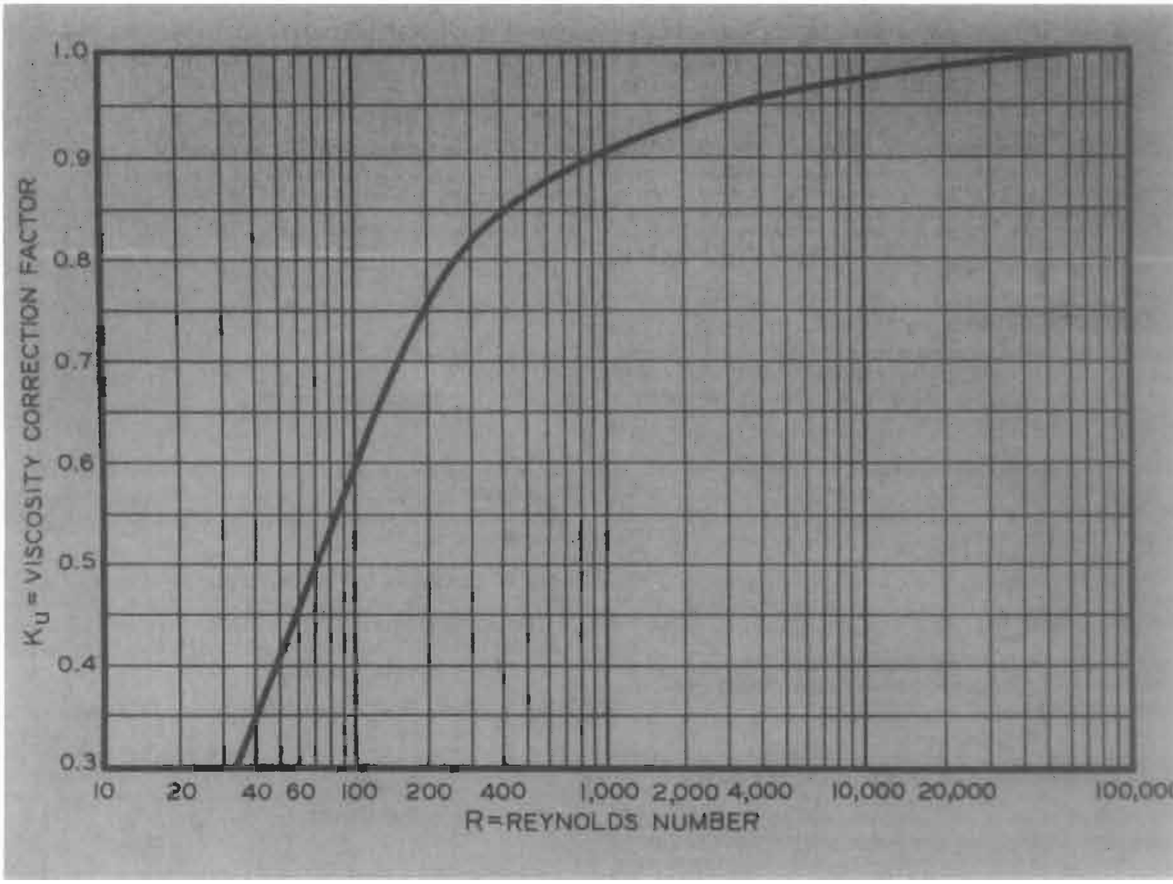
Select the next larger orifice size, or an "M" orifice with 3.60 sq. in. orifice area. (This should be about 20% greater than the calculated area to allow for reduction of capacity due to the viscosity correction factor "K_u").

Step 2—Use Chart To Find "K_u": Enter the Viscosity Correction Chart from the left, reading 1250 SSU. Follow the example line horizontally to the required 800 GPM. Drop vertically to the selected trial orifice "M" and proceed horizontally to the right to the K_u scale, reading K_u = 0.955.

Step 3—Verify Orifice Selection: This chart is designed to minimize the trial and error required for solution. Note that the exit from the chart is from the orifice line to the K_u scale. By inspection vertically, the next larger or smaller orifice show alternate values of the "A" term and the corresponding "K_u" term, without repeating all the steps.

*Example given is for a non ASME Code liquid application. If ASME Code is required, substitute appropriate equation.

Figure 7-23. Liquids viscosity correction using chart method for K_u. By permission, Teledyne Farris Engineering Co.



Viscosity Correction Using Reynold's Number Method of API RP520

As an alternate to Figure 7-23, you may wish to use the method given in API RP 520 for sizing viscous liquids.

When a relief valve is sized for viscous liquid service, it is suggested that it be sized first as for nonviscous type application in order to obtain a preliminary required discharge area, A. From manufacturer's standard orifice sizes, the next larger orifice size should be used in determining the Reynold's number R, from either of the following relationships:

$$R = \frac{V_L(2{,}800\ G)}{\mu\sqrt{A}}$$

or

$$^{*}R = \frac{12{,}700\ V_L}{U\sqrt{A}}$$

Where:

V_L = flow rate at the flowing temperature in U.S. gallons per minute.

G = specific gravity of the liquid at the following temperature referred to water = 1.00 at 70 degrees Fahrenheit.

μ = absolute viscosity at the flowing temperature, in centipoises.

A = effective discharge area, in square inches (from manufacturers' standard orifice areas).

U = viscosity at the flowing temperature, in Saybolt Universal seconds.

After the value of R is determined, the factor K_V† is obtained from the graph. Factor K_V is applied to correct the "preliminary required discharge area." If the corrected area exceeds the "chosen standard orifice area", the above calculations should be repeated using the next larger standard orifice size.

*Use of this equation is not recommended for viscosities less than 100 SSU

†K_v of API = K_u of Teledyne Farris

Figure 7-24. Viscous liquid valve sizing using the method of API RP-520. Reprinted by permission, Teledyne Farris Engineering Co. and *Sizing, Selection and Installation of Pressure-Relieving Devices in Refineries,* Part I "Sizing and Selection," API RP-520, 5th Ed., July 1990, American Petroleum Institute.

ΔP = Set pressure + overpressure, psig − back pressure, psig. At 10% overpressure delta P equals 1.1 $P_1 - P_2$. Below 30 psig set pressure, delta P = $P_1 + 3 - P_2$.

T = inlet temperature °F absolute = (°F plus 460)

Z = compressibility factor corresponding to T and P. If this factor is not available, compressibility correction can be safely ignored by using a value of Z = 1.0.

C = gas or vapor flow constant, see Figure 7-25.

k = ratio of specific heats, C_p/C_v. If a value is not known the use of k = 1.001, C = 315 will result in a safe valve size. Isentropic coefficient, n, may be used instead of k.

K_p = liquid capacity correction factor for overpressures lower than 25% for non-code liquids equations only, (see Figure 7-22).

K_b = vapor or gas flow correction factor for constant back pressures above critical pressure (see Figure 7-26).

K_v = vapor or gas flow factor for variable back pressures for balanced seal valves only (see Figure 7-27A and Figure 7-27B).

K_w = liquid flow factor for variable back pressures for balanced seal valves only (see Figure 7-28). For atmos., K_w = 1.0.

K_u = liquid viscosity correction factor (see Figures 7-23 or Figure 7-24).

K_{sh} = steam superheat correction factor (see Table 7-7).

K_n = Napier steam correction factor for set pressures between 1500 and 2900 psig (see Table 7-6).

K_d = coefficient of discharge [68]:**

0.953 for air, steam, vapors and gases**

0.724 for ASME Code liquids

0.64 for non-ASME Code liquids

0.62 for rupture disks and non-reclosing spring loaded devices; ASME [1], Par. UG-127.

**0.975 per API RP-520, balanced valves

Where the pressure relief valve is used in series with a rupture disk, a combination capacity of 0.8 must be applied to the denominator of the referenced equations. Refer to a later section this text or to specific manufacturers.

**Some manufacturers' National Board Certified Tests will have different values for some of their valves. Be sure to obtain the manufacturer's certified coefficient for the valve you select.

Sizing Valve for Liquid Expansion (Hydraulic Expansion of Liquid Filled Systems/Equipment/Piping)

The API Code RP-520 [33a] suggests the following to determine the liquid expansion rate to protect liquid-filled (full) systems or locations where liquid could be trapped in parts of a system or an area could be subject to blockage by process or operational accident. When thermal input from any source can/could cause thermal expansion of the enclosed liquid:

$$GPM = BH/(500\ G\ C_h) \tag{7-23}$$

This relation can be converted to solve for the required orifice area at 25% overpressure for non-viscous liquids discharging to atmosphere [24]

Table 7-6
Napier Steam Correction Factors, K_n, for Set Pressures between 1500 and 2900 psig

SIZING FACTORS FOR STEAM

K_n NAPIER CORRECTION FACTOR FOR SET PRESSURES BETWEEN 1500 AND 2900 PSIG AT 10% OVERPRESSURE

Equation:

$$K_n = \frac{0.1906P-1000}{0.2292P-1061}$$

Where P = relieving pressure, psia

Set Press. psig	K_n	Set Press. psig	K_n	Set Press. psig	K_n	Set Press. psig	K_n	Set Press. psig	K_n	Set Press. psig	K_n	Set Press. psig	K_n	Set Press. psig	K_n	Set Press. psig	K_n	Set Press. psig	K_n
1500	1.005	1640	1.014	1780	1.025	1920	1.037	2060	1.050	2200	1.066	2340	1.083	2480	1.104	2620	1.128	2760	1.157
1510	1.005	1650	1.015	1790	1.026	1930	1.038	2070	1.051	2210	1.067	2350	1.085	2490	1.105	2630	1.130	2770	1.159
1520	1.006	1660	1.016	1800	1.026	1940	1.039	2080	1.052	2220	1.068	2360	1.086	2500	1.107	2640	1.132	2780	1.161
1530	1.007	1670	1.016	1810	1.027	1950	1.040	2090	1.053	2230	1.069	2370	1.087	2510	1.109	2650	1.134	2790	1.164
1540	1.007	1680	1.017	1820	1.028	1960	1.040	2100	1.054	2240	1.070	2380	1.089	2520	1.110	2660	1.136	2800	1.166
1550	1.008	1690	1.018	1830	1.029	1970	1.041	2110	1.055	2250	1.072	2390	1.090	2530	1.112	2670	1.138	2810	1.169
1560	1.009	1700	1.019	1840	1.030	1980	1.042	2120	1.057	2260	1.073	2400	1.092	2540	1.114	2680	1.140	2820	1.171
1570	1.009	1710	1.019	1850	1.031	1990	1.043	2130	1.058	2270	1.074	2410	1.093	2550	1.115	2690	1.142	2830	1.174
1580	1.010	1720	1.020	1860	1.031	2000	1.044	2140	1.059	2280	1.075	2420	1.095	2560	1.117	2700	1.144	2840	1.176
1590	1.011	1730	1.021	1870	1.032	2010	1.045	2150	1.060	2290	1.077	2430	1.096	2570	1.119	2710	1.146	2850	1.179
1600	1.011	1740	1.022	1880	1.033	2020	1.046	2160	1.061	2300	1.078	2440	1.098	2580	1.121	2720	1.148	2860	1.181
1610	1.012	1750	1.023	1890	1.034	2030	1.047	2170	1.062	2310	1.079	2450	1.099	2590	1.122	2730	1.150	2870	1.184
1620	1.013	1760	1.023	1900	1.035	2040	1.048	2180	1.063	2320	1.081	2460	1.101	2600	1.124	2740	1.152	2880	1.187
1630	1.014	1770	1.024	1910	1.036	2050	1.049	2190	1.064	2330	1.082	2470	1.102	2610	1.126	2750	1.155	2890	1.189
																		2900	1.192

Courtesy Teledyne-Farris Engineering Co., Cat. 187C.

Table 7-7
Steam Superheat Correction Factors, K_{sh}

K_{sh} SUPERHEAT CORRECTION FACTOR

Set Press. p.s.i.g.	Saturated Steam Temp. °F.	TOTAL TEMPERATURE IN DEGREES FAHRENHEIT																																				
		280	300	320	340	360	380	400	420	440	460	480	500	520	540	560	580	600	620	640	660	680	700	720	740	760	780	800	820	840	860	880	900	920	940	960	980	1000
15	250	100	100	100	99	99	96	98	97	96	95	94	93	92	91	90	89	88	87	86	86	85	84	83	83	82	81	81	80	79	79	78	76	77	76	76	75	75
20	259	100	100	100	99	99	96	98	97	96	95	94	93	92	91	90	89	88	87	86	86	85	84	83	83	82	81	81	80	79	79	78	76	77	77	76	75	75
40	287	–	100	100	100	99	99	98	97	96	95	94	93	92	91	90	89	88	87	87	86	85	84	84	83	82	82	81	80	79	79	78	78	77	77	76	75	75
60	308	–	–	100	100	99	99	98	97	96	95	94	93	92	91	90	89	88	87	87	86	85	84	84	83	82	82	81	80	80	79	78	78	77	77	76	76	75
80	324	–	–	–	100	100	99	99	98	97	96	94	93	92	91	90	89	89	88	87	86	85	84	84	83	82	82	81	80	80	79	78	78	77	77	76	76	75
100	338	–	–	–	–	100	100	99	98	97	96	95	94	93	92	91	90	89	88	87	86	85	85	84	83	82	82	81	80	80	79	78	78	77	77	76	76	75
120	350	–	–	–	–	100	100	99	98	97	96	95	94	93	92	91	90	89	88	87	86	85	85	84	83	82	82	81	80	80	79	78	78	77	77	76	76	75
140	361	–	–	–	–	–	100	100	99	98	96	95	94	93	92	91	90	89	88	87	86	85	85	84	83	82	82	81	80	80	79	78	78	77	77	76	76	75
160	371	–	–	–	–	–	100	100	99	98	97	95	94	93	92	91	90	89	88	87	86	86	85	84	83	82	82	81	80	80	79	79	78	77	77	76	76	75
180	380	–	–	–	–	–	–	100	99	98	97	96	95	93	92	91	90	89	88	87	86	86	85	84	83	82	82	81	80	80	79	79	78	77	77	76	76	75
200	388	–	–	–	–	–	–	100	99	99	97	96	95	93	92	91	90	89	88	87	86	86	85	84	83	83	82	81	81	80	79	79	78	77	77	76	76	75
220	395	–	–	–	–	–	–	100	100	99	98	96	95	94	93	92	91	90	89	88	87	86	85	84	84	83	82	81	81	80	79	79	78	78	77	76	76	75
240	403	–	–	–	–	–	–	–	100	99	98	97	95	94	93	92	91	90	89	88	87	86	85	84	84	83	82	81	81	80	79	79	78	78	77	76	76	75
260	409	–	–	–	–	–	–	–	100	99	98	97	96	94	93	92	91	90	89	89	87	86	85	85	84	83	82	81	81	80	79	79	78	78	77	76	76	75
280	416	–	–	–	–	–	–	–	100	100	99	97	96	95	93	92	91	90	89	88	87	86	85	85	84	83	82	82	81	80	80	79	78	78	77	77	76	75
300	422	–	–	–	–	–	–	–	–	100	99	98	96	95	93	92	91	90	89	88	87	86	86	85	84	83	82	82	81	80	80	79	78	78	77	77	76	75
350	436	–	–	–	–	–	–	–	–	100	100	99	97	96	94	93	92	91	90	89	88	87	86	85	84	83	83	82	81	81	80	79	78	78	77	77	76	76
400	448	–	–	–	–	–	–	–	–	–	100	99	98	96	95	93	92	91	90	89	88	87	86	85	84	84	83	82	81	81	80	79	79	78	77	77	76	76
450	460	–	–	–	–	–	–	–	–	–	–	100	99	97	96	94	93	92	91	89	88	87	86	86	85	84	83	82	82	81	80	79	79	78	78	77	76	76
500	470	–	–	–	–	–	–	–	–	–	–	100	99	95	96	94	93	92	91	90	89	88	87	86	85	84	83	82	82	81	80	79	79	78	78	77	76	76
550	480	–	–	–	–	–	–	–	–	–	–	–	100	99	97	95	94	92	91	90	89	89	87	86	85	84	83	82	82	81	80	80	79	78	78	77	76	76
600	489	–	–	–	–	–	–	–	–	–	–	–	100	99	98	96	94	93	92	90	89	88	87	86	85	84	84	83	82	81	80	80	79	78	78	77	76	76
650	497	–	–	–	–	–	–	–	–	–	–	–	–	100	99	97	95	94	92	91	90	89	87	86	86	85	84	83	82	81	81	80	79	78	78	77	77	76
700	506	–	–	–	–	–	–	–	–	–	–	–	–	100	99	97	96	94	93	91	90	89	88	87	86	85	84	83	82	81	81	80	79	79	78	77	77	76
750	513	–	–	–	–	–	–	–	–	–	–	–	–	100	100	98	96	95	93	92	90	89	88	87	86	85	84	83	83	82	81	80	79	79	78	77	77	76
800	520	–	–	–	–	–	–	–	–	–	–	–	–	–	100	99	97	95	94	92	91	90	88	87	86	85	84	84	83	82	81	80	80	79	78	78	77	76
850	527	–	–	–	–	–	–	–	–	–	–	–	–	–	100	99	98	96	94	93	92	90	89	88	87	86	85	84	83	82	81	81	80	79	78	78	77	76
900	533	–	–	–	–	–	–	–	–	–	–	–	–	–	100	100	99	97	95	93	92	90	89	88	87	86	85	84	83	82	81	81	80	79	79	78	77	77
950	540	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	99	97	95	94	92	91	89	88	87	86	85	84	83	82	82	81	80	79	79	78	77	77
1000	546	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	99	98	96	94	93	91	90	89	87	86	85	84	83	83	82	81	80	79	79	78	77	77
1050	552	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	100	99	97	95	93	92	90	89	88	87	86	85	84	83	82	81	80	80	79	78	77	77
1100	558	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	99	98	95	94	92	91	89	88	87	86	85	84	83	82	81	81	80	79	78	78	77
1150	563	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	99	98	96	94	92	91	90	88	87	86	85	84	83	82	81	81	80	79	78	78	77
1200	569	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	99	98	97	95	93	91	90	89	87	86	85	84	83	82	81	81	80	79	78	78	77
1250	574	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	99	97	95	93	92	90	89	88	87	85	85	84	83	82	81	80	79	79	78	77
1300	579	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	99	98	96	94	92	91	89	88	87	86	85	84	83	82	81	80	79	79	78	77
1350	584	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	99	98	96	94	93	91	90	88	87	86	85	84	83	82	81	80	80	79	78	78
1400	588	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	100	99	97	95	93	92	90	89	88	86	85	84	83	82	81	81	80	79	78	78
1450	593	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	99	97	95	94	92	90	89	88	86	85	84	83	82	81	81	80	79	78	78
1500	597	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	99	97	96	94	92	91	89	88	87	86	84	84	83	82	81	80	79	78	78
1600	606	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	98	97	96	94	93	91	89	88	86	85	84	83	82	81	80	79	78	78	77
1700	615	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	99	97	96	95	93	91	89	88	86	85	84	83	82	81	79	79	78	77	77
1800	622	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	99	98	97	95	93	91	89	88	86	85	84	82	81	80	79	79	78	77	76
1900	630	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	98	97	96	93	91	89	88	86	85	84	82	81	80	79	78	78	77	76
2000	636	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	98	96	93	91	89	87	86	84	83	82	81	80	79	78	77	76	76
2100	644	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	98	95	93	91	89	87	86	84	83	82	81	79	78	77	77	76	75
2200	650	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	98	95	93	91	89	87	85	84	82	81	80	79	78	77	76	75	74
2300	658	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	98	95	93	91	89	87	85	84	82	81	79	78	77	76	75	75	74
2400	663	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	98	95	93	91	89	87	85	83	82	80	79	78	77	76	75	74	73
2500	669	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	98	95	93	90	88	86	84	83	81	80	78	77	76	75	74	73	72
2600	675	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	96	93	90	88	85	84	82	80	79	77	76	75	74	73	72	71
2700	680	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	95	90	88	85	83	81	79	78	76	75	74	73	72	71	70
2800	686	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	95	90	88	85	82	80	78	76	75	74	73	72	71	70	69
2900	691	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	90	87	84	82	80	78	76	75	73	72	71	70	69	68

Courtesy Teledyne-Farris Engineering Co., Cat. 187C.

$$A = B\ H/(13600)\ (P_1G)^{0.5} \qquad (7\text{-}24)$$

where GPM = flow rate at the flowing temperature, gal/min

B = cubical expansion coefficient per degree F for the liquid at the expected temperature. For values see specific liquid data or see table below for typical values.

H = total heat transfer rate, BTU/hr. For heat exchangers, Ref [33a] recommends that this value be taken as the maximum exchanger duty during operation.

G = specific gravity referred to water. Ignore fluid compressibility.

C_h = specific heat of the trapped fluid, BTU/lb/°F

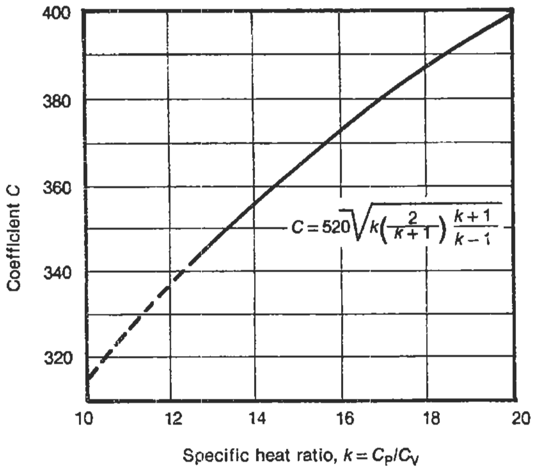


Figure 7-25. Constant "C" for gas or vapor related to specific heats. By permission, *Sizing, Selection and Installation of Pressure-Relieving Devices in Refineries,* Part I "Sizing and Selection," API RP-520, 5th Ed., July 1990.

Values of Coefficient *C*

k	*C*	*k*	*C*	*k*	*C*	*k*	*C*
1.01	317[a]	1.31	348	1.61	373	1.91	395
1.02	318	1.32	349	1.62	374	1.92	395
1.03	319	1.33	350	1.63	375	1.93	396
1.04	320	1.34	351	1.64	376	1.94	397
1.05	321	1.35	352	1.65	376	1.95	397
1.06	322	1.36	353	1.66	377	1.96	398
1.07	323	1.37	353	1.67	378	1.97	398
1.08	325	1.38	354	1.68	379	1.98	399
1.09	326	1.39	355	1.69	379	1.99	400
1.10	327	1.40	356	1.70	380	2.00	400
1.11	328	1.41	357	1.71	381	—	—
1.12	329	1.42	358	1.72	382	—	—
1.13	330	1.43	359	1.73	382	—	—
1.14	331	1.44	360	1.74	383	—	—
1.15	332	1.45	360	1.75	384	—	—
1.16	333	1.46	361	1.76	384	—	—
1.17	334	1.47	362	1.77	385	—	—
1.18	335	1.48	363	1.78	386	—	—
1.19	336	1.49	364	1.79	386	—	—
1.20	337	1.50	365	1.80	387	—	—
1.21	338	1.51	365	1.81	388	—	—
1.22	339	1.52	366	1.82	389	—	—
1.23	340	1.53	367	1.83	389	—	—
1.24	341	1.54	368	1.84	390	—	—
1.25	342	1.55	369	1.85	391	—	—
1.26	343	1.56	369	1.86	391	—	—
1.27	344	1.57	370	1.87	392	—	—
1.28	345	1.58	371	1.88	393	—	—
1.29	346	1.59	372	1.89	393	—	—
1.30	347	1.60	373	1.90	394	—	—

[a] Interpolated value, since *C* becomes indeterminate as *k* approaches 1.00.

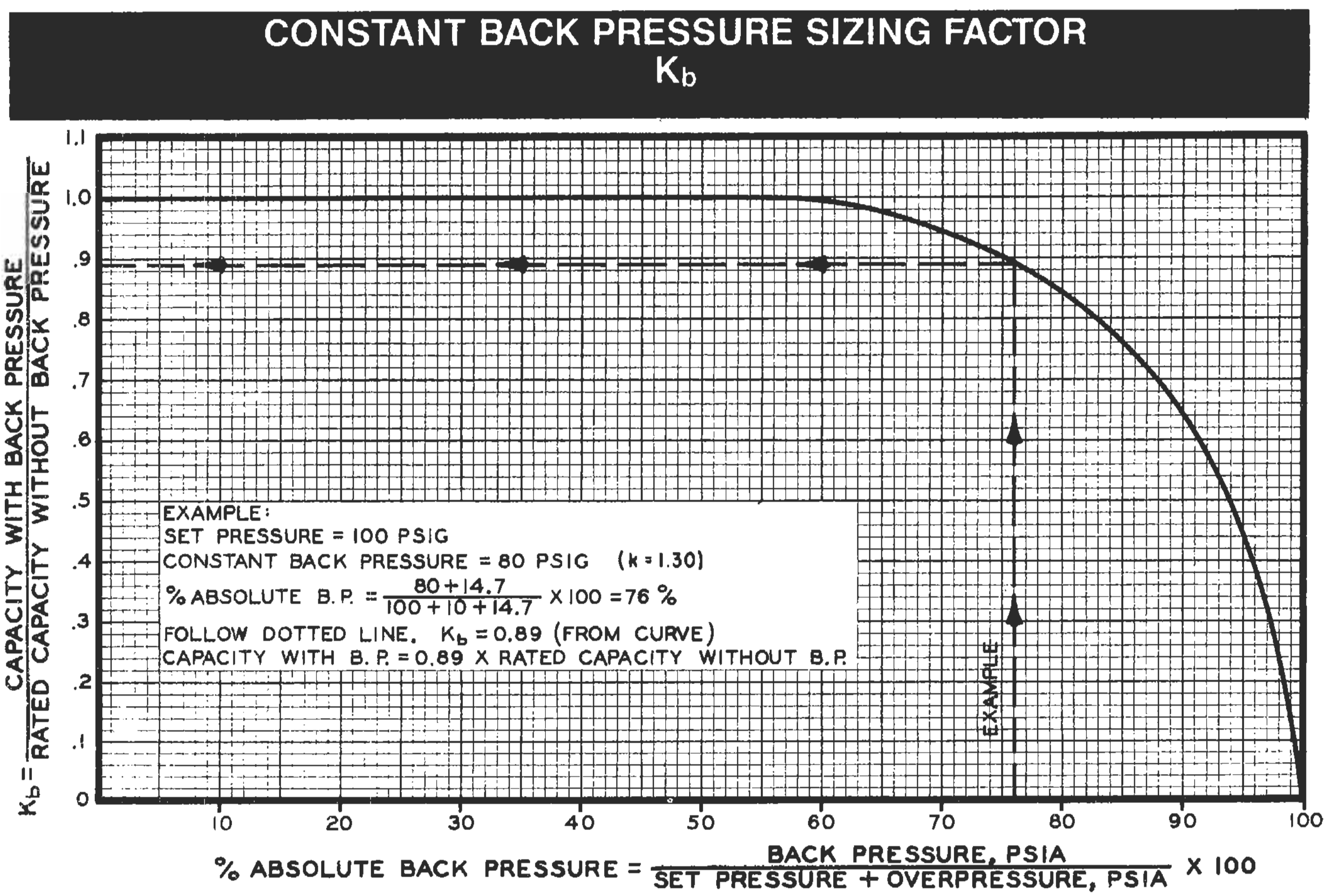


Figure 7-26. Constant backpressure sizing factor, K_b, conventional valves-vapors and gases. By permission, Teledyne Farris Engineering Co.

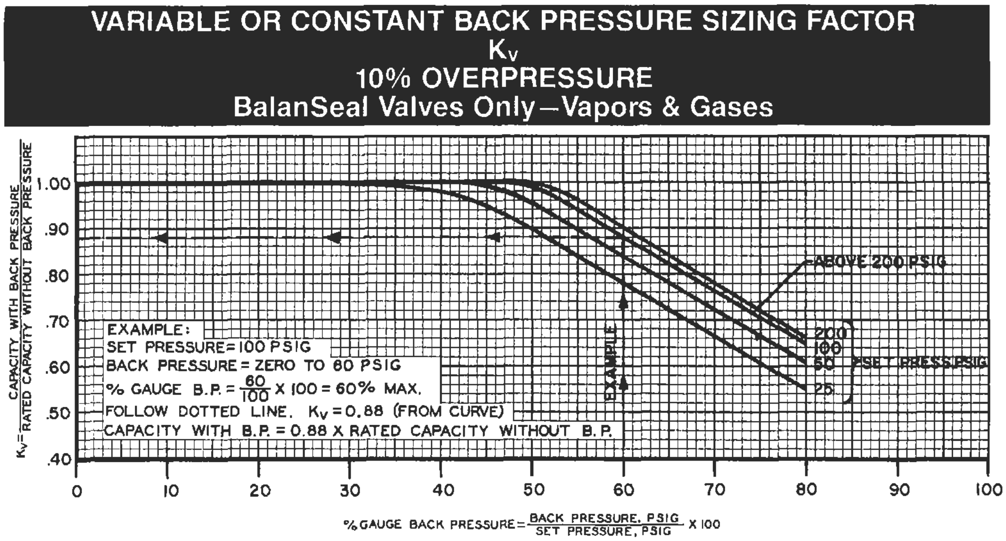


Figure 7-27A. Variable or constant backpressure sizing factor, K_v, at 10% overpressure, BalanSeal® valves only: vapors and gases. By permission, Teledyne Farris Engineering Co.

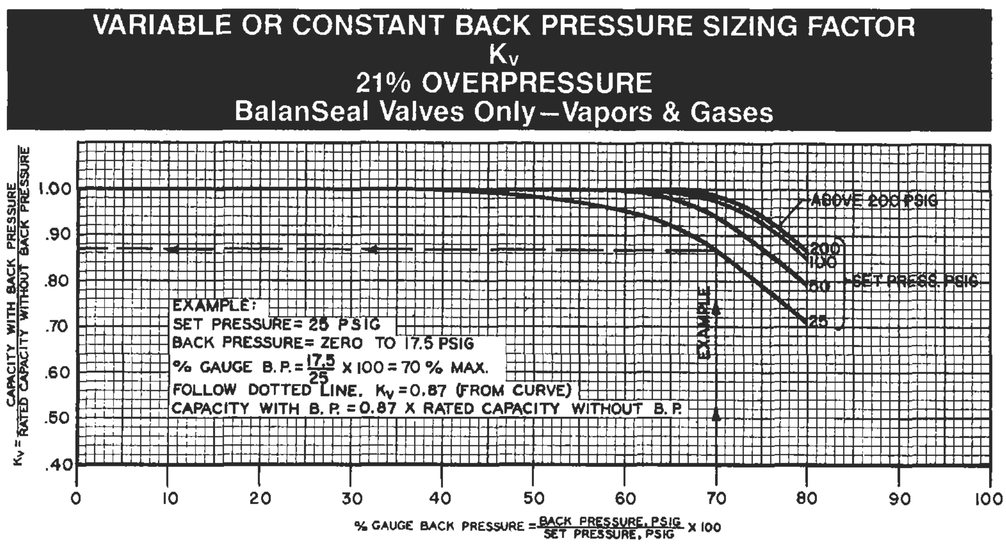


Figure 7-27B. Variable or constant backpressure sizing factor, K_v, at 21% overpressure, BalanSeal® valves only: vapors and gases. By permission, Teledyne Farris Engineering Co.

A = required orifice area, sq in.
P_1 = set pressure of valve, psig

Ref [33a] in Appendix C cautions that if the vapor pressure of the fluid at the temperature is greater than the relief set/design pressure that the valve must be capable of handling the rate of vapor generation. Other situations should be examined as the thermal relief by itself may be insufficient for total relief.

Typical Values for Cubical Expansion Coefficient*

Liquid	Value
3–34.9 degree API gravity	0.0004
35–50.9 degree API gravity	0.0005
51–63.9 degree API gravity	0.0006
64–78.9 degree API gravity	0.0007
79–88.9 degree API gravity	0.0008
89–93.9 degree API gravity	0.00085
94–100 degree API gravity and lighter	0.0009
Water	0.0001

*By permission API, Ref [33a]

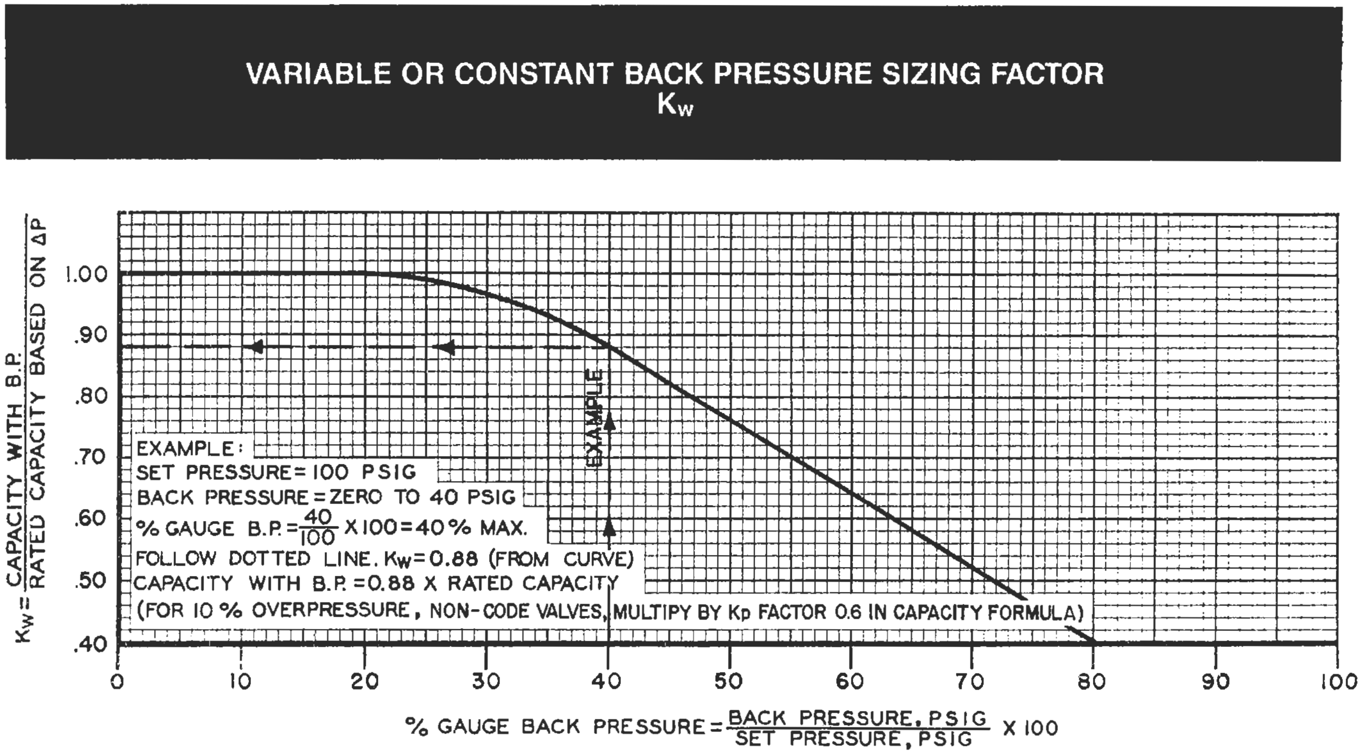


Figure 7-28. Variable or constant backpressure sizing factor, K_w, for liquids only, BalanSeal® valves. Use this factor as a divisor to results of constant backpressure equations or tables. By permission, Teledyne Farris Engineering Co.

Note: Reference of the API gravity values to refinery and petrochemical plant fluids will show that they correspond to many common hydrocarbons.

Sizing Valves for Subcritical Flow: Gas or Vapor, but not Steam [33A].

If the ratio of backpressure to inlet pressure to valve exceeds the critical pressure ratio, P_c/P_1,

$$\left(\frac{P_c}{P_1}\right) = \left[\frac{2}{(k+1)}\right]^{k/(k-1)} \qquad (7\text{-}7)$$

the flow through the valve is subcritical. The required area (net, free unobstructed) is calculated for a conventional relief valve, including sizing a pilot-operated relief valve [33A, Par 4.3.3]:

$$A = \frac{W}{735\,F_2 K_d}\sqrt{\frac{ZT}{MP\,(P - P_2)}}, \text{ sq in.} \qquad (7\text{-}25)$$

$$\text{or, } A = \frac{V}{4645.2\,F_2 K_d}\sqrt{\frac{ZTM}{P(P - P_2)}}, \text{ sq in.} \qquad (7\text{-}26)$$

$$\text{or, } A = \frac{V}{863.63\,F_2 K_d}\sqrt{\frac{ZT(\text{Sp Gr})}{P(P - P_2)}}, \text{ sq in.} \qquad (7\text{-}27)$$

Note: K_d = effective coefficient of discharge for valve = 0.975 for equations above and Equations 7-25, 26, 27

When using a balanced/bellows relief valve in the subcritical, use Equations 7-18 through 7-22; however, the backpressure correction factor for this condition should be supplied by the valve manufacturer [33A]. For sub-critical, conventional valve:

$$F_2 = \sqrt{\left(\frac{k}{k-1}\right)(r)^{2/k}\left[\frac{1-(r)^{(k-1)/k}}{1-r}\right]} \qquad (7\text{-}28)$$

where:
- A = required effective discharge area, sq in.
- F_2 = coefficient of subcritical flow, see Figure 7-29
- T = relieving temperature of inlet gas or vapor, °R
- P = upstream relieving pressure, psia, = set pressure + allowable overpressure + atmospheric pressure, usually 14.7 psia), psia
- P_2 = backpressure on valve, psia
- W = required flow through valve, lbs/hr
- V = vapor flow required through valve, standard cu ft/min at 14.7 psia and 60°F

SpGr = specific gravity referred to air = 1.0.
r = ratio of backpressure to upstream relieving pressure, P_2/P

Emergency Pressure Relief: Fires and Explosions Rupture Disks

Process systems can develop pressure conditions that cannot *timely* or adequately be relieved by pressure-relieving valves as described earlier. These conditions are primarily considered to be (1) internal process explosions due to runaway reactions (see DIERS [67]) in pressure vessels or similar containers such as an atmospheric grain storage silo (dust explosion typically) or storage bin; (2) external fires developed around, under, or encompassing a single process vessel or a system of process equipment, or an entire plant; and (3) other conditions in which rapid/instantaneous release of developed pressure and large volumes of vapor/liquid mixture is vital to preserve the integrity of the equipment. For these conditions, a rupture disk may perform a vital safety relief function. Sometimes the combination of a rupture disk and pressure-relieving valve will satisfy a prescribed situation, but the valve cannot be relied on for instantaneous release (response time lag of usually a few seconds).

The ASME Pressure Vessel Code [1] and the API codes or recommended procedures [10, 13, 33] recognize and set regulations and procedures for capacity design, manufacture and installation of rupture disks, once the user has established the basis of capacity requirements.

External Fires

There have been at least six different formulas proposed and used to determine the proper and adequate size of rupture disk openings for a specific relieving condition. The earlier studies of Sylvander and Katz [25] led to the development of the ASME and API recommendations. This approach assumes that a fire exists under or around the various vessels in a process. This fire may have

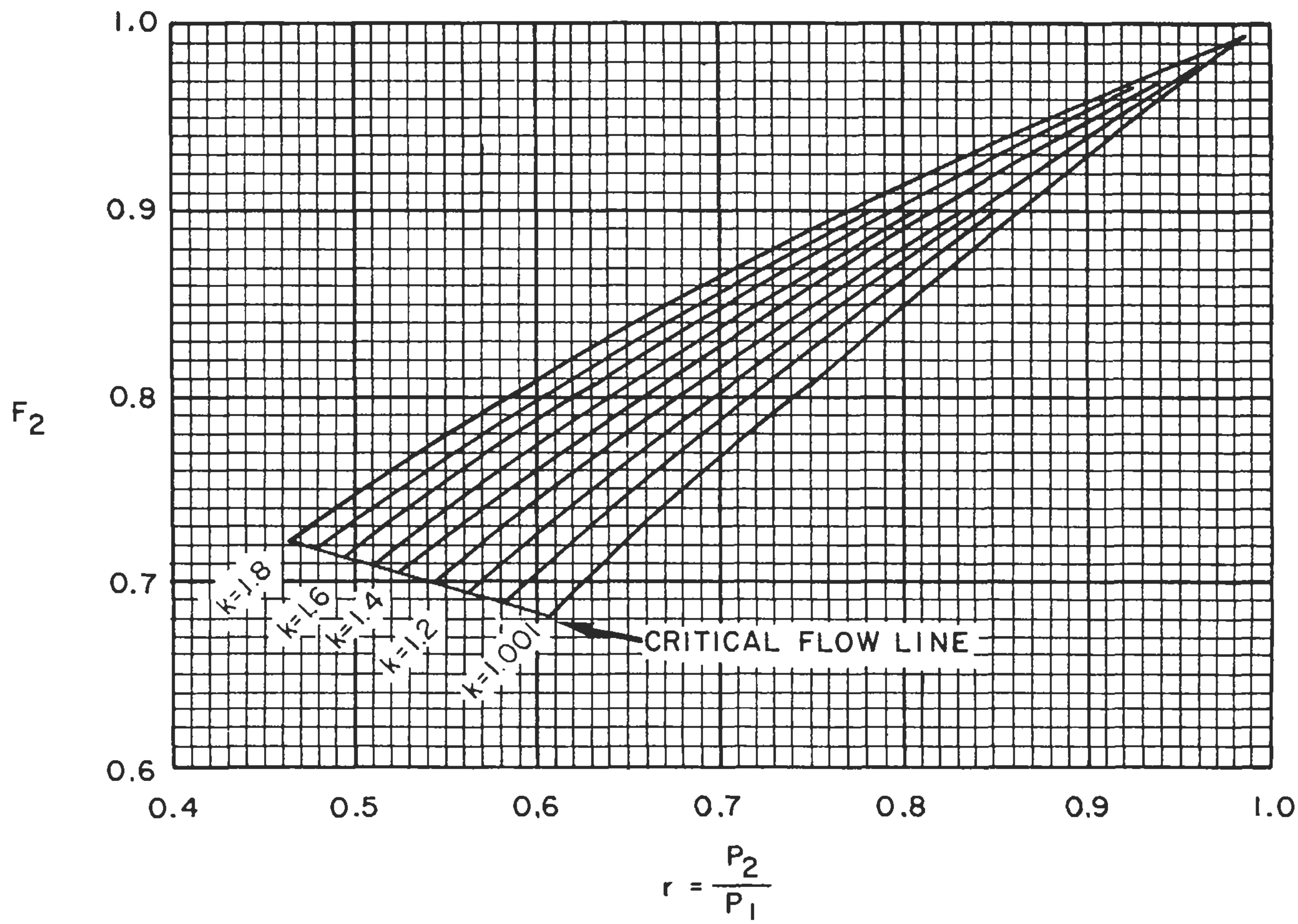


Figure 7-29. Values of F_2 for subcritical flow of gases and vapors. Reprinted by permission, *Sizing, Selection and Installation of Pressure-Relieving Devices in Refineries,* Part I "Sizing and Selection," API RP-520, 5th Ed. July 1990 American Petroleum Institute.

started from static discharge around flammable vapors, flammable liquids released into an area drainage ditch, combustible gas/vapors released through flange leaks or ruptures, over pressure, or many other potential hazards. The codes [13] suggest typical lists of potential hazards and some approach to determine the types of process conditions that can cause a fire. A suggested extended list (Figure 7-14) is presented earlier in this chapter and in the following paragraphs. There are no "pat" formulas to establish a code capacity for the volume of vapors to be released under any one of the possible plant "failure" conditions. Therefore, the codes assume, based on evaluation of test data, that fire is under or around the various vessels and that absorbed heat vaporizes the contained fluid. The information presented is taken from API-RP-520/521 latest editions [10, 13, 33] and the ASME code [1]. The designer should be familiar with the details in these codes.

Set Pressures for External Fires

The maximum allowable working pressure (MAWP) (discussed earlier) for the vessel or each vessel should be the *maximum* set pressure for the rupture disk. Furthermore, estimated flame temperatures of usually 2500°–3500°F should be used to establish the reduced vessel metal wall temperatures (recognizing the benefits of code recommended fireproof insulation if properly applied to prevent dislodging by fire water hose pressures impacting on the insulation). The MAWP should then be re-established by calculation using the metal wall Code allowable stresses at the new estimated reduced metal temperature. This should be the maximum set pressure for the rupture disk provided the new lower value does not cause it to be below or too close to the usual expected process operating temperature. In that case, this author suggests that the set pressure be 25% above the operating condition, exclusive of fire, not exceeding the MAWP values.

When a rupture disk relieves/blows/ruptures, it creates a rapid depressuring of the process system and a likely discharge of some or all vapor/liquid in the vessel(s), discharging to the properly designed disposal system. Therefore, great care should be given to setting the rupture disk pressure because it does not have an accumulation factor, but bursts at the prescribed pressure of the disk, taking into account the Code allowed manufacturing tolerances. Two-phase flow will most likely occur when the disc (or safety-relief valve) blows (see later references to explosions and DIERS work [51, 67].

For unexpected runaway or process overpressure *not subject to external fire,* the rupture disk set pressure, which is the bursting pressure, should be sufficiently higher than the expected "under acceptable control" conditions for the operation to avoid the frequent burst and shut down of the process. Usually this is found to be about 20%–30% above the maximum expected peak operation pressure. Again, recognize the requirements for the relieving device set pressure not to exceed the actual vessel MAWP at the expected relieving temperature (by calculation or pilot plant test data).

Heat Absorbed

The amount of heat absorbed by a vessel exposed to an open fire is markedly affected by the size and character of the installation and by the environment. These conditions are evaluated by the following equivalent formulas, in which the effect of size on the heat input is shown by the exponent of A_w, the vessel *wetted area,* and the effect of other conditions, including vessel external insulation is included in a factor F [33]:

$$q = 21{,}000\ FA_w^{-0.18} \tag{7-29}$$

$$Q = 21{,}000\ FA_w^{+0.82} \tag{7-30}$$

where q = average unit heat absorption, in BTU per hour per square foot of *wetted surface.*

Q = total heat absorption (input) to the wetted surface, in BTU per hour.

A_w = total wetted surface, in square feet.

The expression $A_w^{-0.18}$ is the area exposure factor or ratio. This recognizes that large vessels are less likely to be completely exposed to the flame of an open fire than small vessels. It is recommended that the total wetted surface (A in the foregoing formulas) be limited to that wetted surface included within a height of 25 feet above "grade" or, in the case of spheres and spheroids, to the elevation of the maximum horizontal diameter or a height of 25 feet, whichever is greater. (A more conservative approach is recommended.) The term "grade" usually refers to ground grade, but may be any level at which a sizable area of exposed flammable liquid could be present [10, 33].

F = environment factor, values of which are shown in Table 7-8 for various types of installation.

Surface areas of vessel elliptical heads can be estimated by 1.15 × cross-sectional area of vessel.

These are the basic formulas for the usual installation, with good drainage and available fire-fighting equipment. These formulas are plotted on Figure 7-30 showing curves for Q for various values of factor F. The approximate amount of insulation corresponding to the factors is indicated.

Referring to the wetted surface, A_w, the surface areas of ASME flanged and dished head, ASME elliptical heads, hemispherical heads, etc., are often the end assemblies on

Table 7-8
Environment Factor, F

Type of Installation		Factor*
Bare vessel		1.0
Insulated vessels† (These arbitrary insulation condance values are shown as examples and are in British Thermal Units per hour per square foot per °F)		
(a) 4.0 Btu/hr/sq ft/°F	(1 in thick)	0.3
(b) 2.0 Btu/hr/sq ft/°F	(2 in thick)	0.15
(c) 1.0 Btu/hr/sq ft/°F	(4 in thick)	0.075
(d) 0.67 Btu/hr/sq ft/°F		0.05
(e) 0.5 Btu/hr/sq ft/°F		0.0376
(f) 0.4 Btu/hr/sq ft/°F		0.03
(g) 0.32 Btu/hr/sq ft/°F		0.026
Water-application facilities, on bare vessel**		1.0
Depressurizing and emptying facilities††		1.0
Underground storage		0.0
Earth-covered storage above grade		0.03

*These are suggested values for the conditions assumed in code [33] Par D 5.21. When these conditions do not exist, engineering judgment should be exercised either in selecting a higher factor or in providing means of protecting vessels from fire exposure as suggested in [33], par. D. 8.

†Insulation shall resist dislodgement by fire hose streams. For the examples, a temperature difference of 1600°F was used. These conductance values are based on insulation having thermal conductivity of 48TS/hr-ft-°F per inch at 1600°F and correspond to various thicknesses of insulation between 1 and 12 inches.

**See code for recommendations regarding water application and insulation.

††Depressurizing will provide a lower factor if done promptly, but no credit is to be taken when safety valves are being sized for fire exposure. See [33], Part 1, par. D. 8.2.

By permission, API-RP-520, American Petroleum Institute, Div. of Refining (1967) and adapted for this current edition by this author from later editions of the code (1976) and (1990). Items d, e, and f above from API-RP-520, 5th Ed. (1990). For complete reference, see the latest code cited in its entirety.

a cylindrical vessel. If a formula is not available to accurately estimate the wetted surface, or the blank diameters used for fabrication (see Appendix), which would give a close approximation of the inside surface of the head, use *an estimated area* for the dished or elliptical heads as 1.2 × cross-section area of the vessel based on its diameter.

Surface Area Exposed to Fire

The surface area of a vessel exposed to fire which is effective in generating vapor is that area wetted by its internal liquid contents. The liquid contents under variable level conditions should ordinarily be taken at the average inventory, for example: See note below.

1. *Liquid-full vessels* (such as treaters) operate liquid full. Therefore, the wetted surface would be the total vessel surface within the height limitation.

2. *Surge drums* (vessels) usually operate about half full. Therefore, the wetted surface would be calculated at 50% of the total vessel surface, but higher if design is based on greater figure.

3. *Knockout drums* (vessels) usually operate with only a small amount of liquid. Therefore, the wetted surface would be in proportion, but to maximum design liquid level.

4. *Fractionating columns* usually operate with a normal liquid level in the bottom of the column and a level of liquid on each tray. It is reasonable to assume that the wetted surface be based on the total liquid within the height limitation—both on the trays and in the bottom.

5. *Working storage tanks'* wetted surface is usually calculated on the average inventory, but at least 25 ft height, unless liquid level can reasonably be established as higher, then use higher value. This should be satisfactory not only because it conforms with a probability, but also because it provides a factor of safety in the time needed to raise the usually large volume of the liquid's sensible heat to its boiling point.

It is recommended that the wetted area be at least to the height as defined in the definition of area, A_w.

Note: This author's suggested determination of A_w values may be more conservative and not conform exactly to Code [33a.33c] recommendations. The Code [33a, Part 1, Sect D, Par. D.4] reads, "to determine vapor generation, only that portion of the vessel that is wetted by its internal liquid and is equal to or less than 25 feet above the source of flame needs to be recognized."

6. *Based upon this author's experience* in investigating many industrial fires and explosions, it is suggested that the height limit of 25 feet above "grade" or fire source level is too low for many process plants, and therefore, the effect of a large external fire around equipment can reach to 100 feet with 75 feet perhaps being acceptably conservative. This author would never use the 25-foot limit, for example, for a horizontal butane storage "bullet" tank, 15 feet diameter and raised 15 feet off grade to its bottom. The fact that any fire will engulf the entire vessel, should be considered and the wetted surface should be the entire vessel. The same concern applies to a vertical distillation column over 25 feet high. It is this author's opinion that the wetted surface should be at least 80% of the vessel height, recognizing that the tray liquid will wet

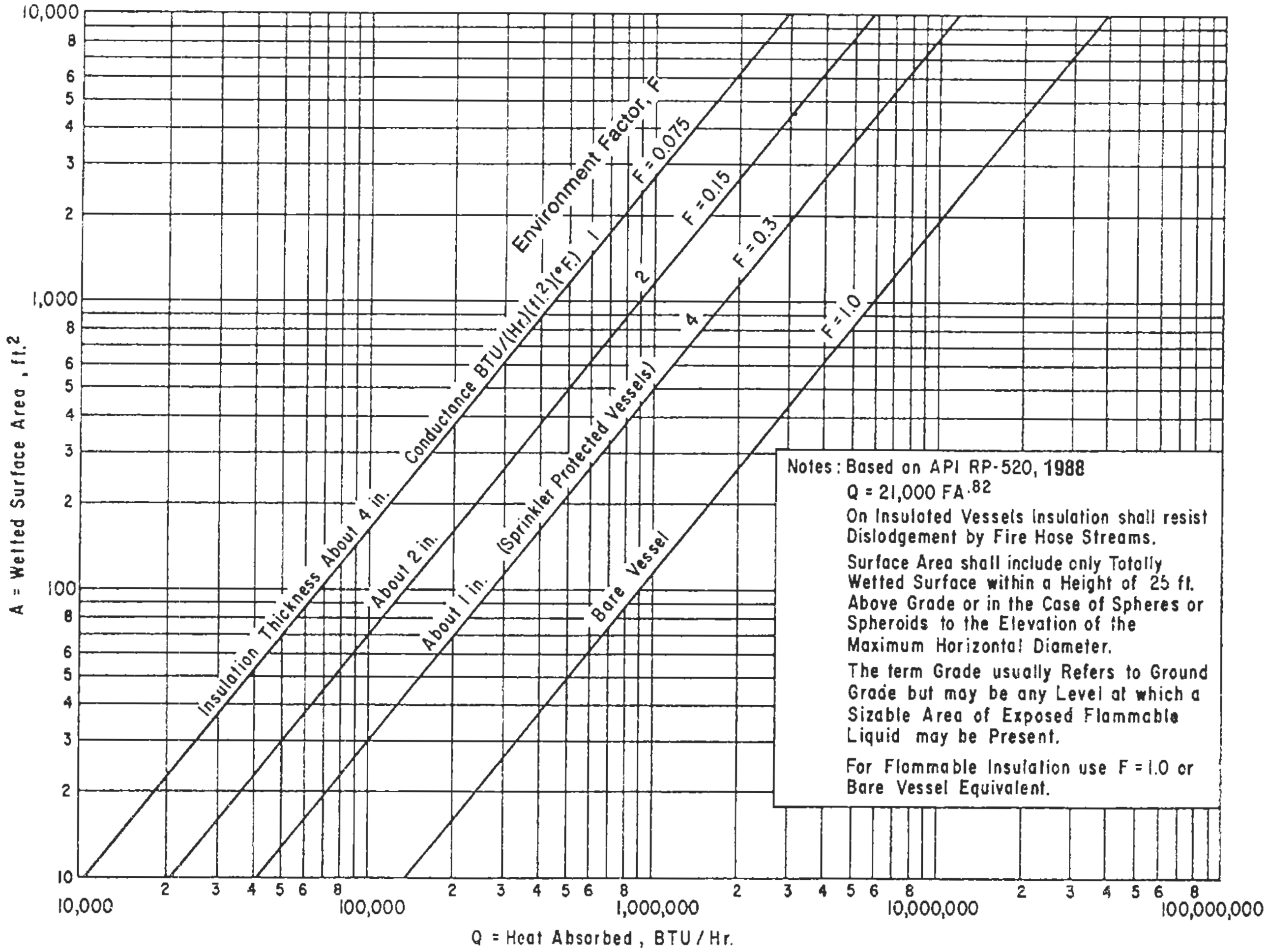


Figure 7-30. API formula for heat absorbed from fire on wetted surface of pressure vessel. Prepared by permission, *Sizing, Selection and Installation of Pressure Relieving Devices in Refineries,* Part I "Sizing and Selection," API RP-520, 5th Ed. July 1990, American Petroleum Institute.

the walls and be evaporated only as long as there is liquid to drain off the trays, but for a conservative approach, assume that there are always wetted walls in the column.

For packed columns, the wetted walls are at least to the top of the packing, with some entrainment height above that. Therefore, a vertical packed column of 60 feet of packing above the liquid level in the sump, which has a sump of 10 feet and a skirt of 10 feet, would be considered 70 feet of vertical height of wetted perimeter, not counting the skirt. If the wetted area reached a total height of 80 feet from grade, minus the unwetted area of the skirt height, because no liquid is in the skirt space, the 80 feet would not be used to establish the vertical height for fire exposure, but use 60 ft + 10 ft, or 70 feet wetted height. This is more conservative than code [33a and c], but is based on experience in investigating the damage levels from fires.

Each situation must be evaluated on its own merits or conditions and operating situation, and even its environment with respect to the plant flammable processing equipment.

Relief Capacity for Fire Exposure

In calculating the relief capacity to take care of external fire the following equation is used:

$$W = Q/L \tag{7-31}$$

where W = weight rate of flow of vapors, pounds per hour
L = latent heat at allowable pressure, BTU/lb
Q = total heat absorption from external fire, BTU/hr

Code Requirements for External Fire Conditions

Paragraph UG-125 (3) of the ASME code [1] requires that supplemental relieving capacity be available for an unfired pressure vessel subject to external accidental fire or other unexpected source of heat. For this condition, relieving devices must be installed to prevent the pressure from rising more than 21% [13] above the maximum allowable working pressure of the vessel. The set pressure should not exceed the vessel MAWP. A single relief device may be used for this capacity as long as it also meets the normal overpressure design for other possible causes of 10 percent. If desirable, multiple separate devices can be installed to satisfy both potential overpressure situations.

For this condition, the API-RP-521 Code [13] (Figure 7-7A) shows an allowable 16% maximum accumulation relieving pressure above the set pressure. For external fire conditions on a vessel, the maximum allowable accumulation pressure is 21% above the set pressure [13] for both single or multiple relieving devices (Figure 7-7A).

Design Procedure

The usual procedure for determining relief area requirements is:

1. Determine the external surface area exposed to fire, as set forth by

$$Q = 21{,}000\ FA_w^{+0.82} \tag{7-30}$$

and Table 7-8 and the paragraph on Surface Area Exposed to Fire.

2. Determine the heat absorbed, Q, from Figure 7-30.
3. Calculate the rate of vaporization of liquid from

$$W = \frac{Q}{L} \tag{7-31}$$

4. Verify critical pressure from Equation 7-7 and establish actual back pressure for relieving device.
5. Calculate relieving area by applicable equation for critical or non-critical flow, using the flow rate determined in (3) above. (See Equation 7-10 and following). The area actually selected for orifice of safety type valve must have orifice equal to or greater than calculated requirements. For a rupture disk application, the full free open cross-sectional area of pipe connections in inlet and exit sides must be equal to or be greater than the calculated area.
6. Select a valve or rupture disk to accommodate the service application.
7. To provide some external protection against the damage that an external fire can do to a pressure relief *valve* or rupture disk, this author recommends that these devices be insulated after installation in such a manner as not to restrict their action but to provide some measure of reliable performance, even if the vessel is not insulated.

Pressure Relief Valve Orifice Areas on Vessels Containing Only Gas, Unwetted Surface

Due to gas expansion from external fire, the API code [10] provides for calculation of the pressure relief valve orifice area for a gas containing vessel exposed to external fire on the unwetted surface:

$$A = F' A_3 / \sqrt{P_1}\ \text{, sq in.} \tag{7-32}$$

Based on air and perfect gas laws, vessel uninsulated, and vessel will not reach rupture conditions. Review for specific design situations [33a]

where A = effective discharge area of valve, sq in.

F' = operating environment factor, min value recommended = 0.01 when the minimum value is unknown, use $F' = 0.045$. Can be calculated by [33c]

$$F' = \frac{0.1406}{CK_d}\left[\frac{(T_\omega - T_1)^{1.25}}{T_1^{0.6506}}\right] \tag{7-33}$$

A_3 = exposed surface area of vessel, sq ft

P_1 = upstream relieving pressure, psiabs. This is the set pressure plus the allowable overpressure plus the atmospheric pressure, psiabs.

where M = Molecular weight of gas/vapor.

A_3 = exposed surface area of the vessel, in square feet.

P_1 = upstream relieving pressure, in pounds per square inch absolute. This is the set pressure plus the allowable overpressure plus the atmospheric pressure.

C = coefficient determined by the ratio of the specific heats of the gas at standard conditions. This can be obtained from Equation 2 [33a] in 4.3.2.1 of API Recommended Practice 520, Part I, or Figure 7-25.

K_d = coefficient of discharge (obtainable from the valve manufacturer). K_d is equal to 0.975 for sizing relief valves.

T_ω = vessel wall temperature, in degrees Rankine.

T_1 = gas temperature, absolute, in degrees Rankine, at the upstream pressure, determined from the following relationship:

$$T_1 = \frac{P_1}{P_\eta} T_\eta$$

P_η = normal operating gas pressure, in pounds per square inch absolute.

T_η = normal operating gas temperature, in degrees Rankine.

The recommended maximum vessel wall temperature for the usual carbon steel plate materials is 1100°F. Where vessels are fabricated from alloy materials, the value for T_ω should be changed to a more appropriate recommended maximum [33a].

The relief load can be calculated directly, in pounds per hour [33a]:

$$W = 0.1406\sqrt{MP_1}\left(A_3\ \frac{(T\omega - T_1)^{1.25}}{T_1^{\ 1.1506}}\right),\ \text{lbs/hr} \quad (7\text{-}34)$$

Rupture Disk Sizing Design and Specification

Rupture or burst pressure of the metal disks must be specified at least 25% to 40% greater than the normal nonpulsing operating pressure of the vessel or system being protected. For low pressures less than 5 to 10 psig operating, the differences between operating pressure and set pressure of a valve or disk may need to be greater than that just cited.

For mild pulsations, the disk bursting pressure should be 1.75 times the operating pressure; and for strong pulsations, use two times the operating pressure [18]. Nonmetallic impregnated graphite disks may be used to burst at 1.34 times operating pressure as these are less subject to fatigue. The bursting pressure must never be greater than the maximum allowable working pressure of the vessel, and proper allowance must be made for the possible pressure variations, plus and minus, due to the manufacturer's rupture pressure range. See Table 7-2 and specific manufacturers' literature. The ASME Code [1] Par. UG-127 requires disks to burst within 5% ± of the stamped bursting pressure at a specified disk temperature at time of burst.

Specifications to Manufacturer

When ordering rupture disks, the following information and specifications should be given to the manufacturer.

1. Net inside diameter of opening leading to the flange or holding arrangement for the disk, inches; or the cubic feet of vapor at stated conditions of burst pressure, or both.
2. Preferred material of construction, if known, otherwise state service to obtain recommendation.
3. Type of hold-down arrangement: flanged (slip on, weld neck, screwed, stud) union, screwed, or special.
4. Material of construction for hold-down (flange, screwed) arrangement. Usually forged carbon steel is satisfactory, although aluminum or other material may be required to match vessel and/or atmosphere surrounding the disk assembly.
5. Temperature for (a) continuous operation and (b) at burst pressure.
6. Required relief or burst pressure in vessel *and* the back pressure on the disk, if any.
7. Disks to be ASME Code certified.

When the flow capacity for relief can be given to the disk manufacturer, together with the conditions at bursting pressure (including temperature), the manufacturer can check against a selected size and verify the ability of the disk to relieve the required flow.

Size Selection

Rupture disks are used for the same purpose as safety valves and, in addition, serve to relieve internal explosions in many applications. If the pressure rise can be anticipated, then the volume change corresponding to this change can be calculated by simple gas laws, and the capacity of the disk at the relieving pressure is known. The system must be examined and the possible causes of overpressure and their respective relief capacities identified before a reliable size can be determined. See Figure 7-14.

Calculation of Relieving Areas: Rupture Disks for Non-Explosive Service

The vessel nozzle diameter (inside) or net free area for relief of vapors through a rupture disk for the usual process applications is calculated in the same manner as for a safety relief valve, except that the nozzle coefficient is 0.62 for vapors and liquids. Most applications in this category are derived from predictable situations where the flow rates, pressures and temperatures can be established with a reasonable degree of certainty.

For rupture disk sizing the downstream pressure is assumed to reach the critical flow pressure although the down stream pressure initially may be much lower. Under these conditions the flow through the "orifice" that the disk produces on rupture is considered to be at critical flow. The assumptions of critical pressure do not apply where a fixed downstream side pressure into which the disk must relieve is greater than the critical pressure.

The coefficient of discharge, K_D, is the actual flow divided by the theoretical flow and must be determined by tests for each type or style and size of rupture disk as well as pressure-relieving valve. For rupture disks, the minimum *net flow area* is the calculated net area after a complete burst of the disk, making allowance for any structural members that could reduce the net flow area of the disk. For sizing, the net flow area must not exceed the nominal pipe size area of the rupture disk assembly [1].

The bursting pressure, P_b, of the conventional tension-loaded disk is a function of the material of which the disk is fabricated, as well as its thickness and diameter, and lastly but not least in any manner, is the temperature at which the disk is expected to burst, and not just the temperature corresponding to the disk set pressure [37]. This type disk is best suited to be set on P_b at least 30% above the system operating pressure. The reverse-buckling disk with knives to aid the bursting are compression loaded because the dome of the disk faces the internal vessel pressure. The bursting pressure, P_b, of this disk is dependent on the dome's geometrical shape and the characteristics of the knife blades, but it is essentially independent of thickness and generally does not need a vacuum support [37]. It is often used when the operating pressure is as high as 0.9 P_b. There are some potential problems or even hazards with this design if the knives fail, come loose, or corrode, and the use must be examined carefully. This disk like any other disk, should never be installed upside down from its original design position. It should not be used in partial or total liquid service.

The reverse-buckling disk, without knives but with a pre-scored disk surface, offers some features that do not depend on the knives being in place because the thickness of the metal disk dome along the score line determines the bursting pressure of the disk.

The Manufacturing Range (MR)

The ASME code [1] requires that a ruptured disk must be stamped with a bursting pressure that falls within the manufacturing range. This range identifies the allowable range of variation from a specified burst pressure to the actual burst pressure provided by the manufacturer and as agreed upon with the disk user. The stamped burst pressure of a lot of rupture disks is the average burst pressure of all the destructive tests performed per code requirements. The average of the tests must fall within the manufacturing range (see Table 7-2).

The thickness of one material for manufacture of a disk, along with the specific disk type, is the key factor in establishing at what pressure range a disk of a specified bursting diameter will actually burst on test and then in actual service.

To specify a rupture disk:

- identify the desired bursting pressure, P_b
- list the required MR

For example, a system requiring a bursting pressure, P_b, of 150 psig at 400°F, would have an MR range of −4% to +7% at the operating temperature with a burst pressure tolerance of ±5%. The disk supplied by a specific manufacturer (MR varies with manufacturer and pressure ranges) could have a bursting pressure as low as −(0.04) (150) = −6 psi, or 144 psig; or as high as (+0.07) (150) = +10.5 or 160.5 psig. If the disk is stamped at the operating temperature at 144 psig it could burst at ±5% or 7.2 psi or 136.8 psig or 151.2 psig. On the other hand, if the disk were stamped at operating temperature to burst at 160.5 psig (its highest) then it could actually burst ±5% of this, or ±8.03 psi, giving an actual burst pressure of 168.5 psig to 152.5 psig. Adapted from [37] by permission.

The Code requires that the disks be burst on test by one of three methods using four sample disks, but not less than 5% from each lot. Figure 7-32 illustrates test results for burst pressure versus temperature of a disk design, all fabricated from the same material, and of the same diameter.

It is critical that such an examination be made to be certain that the bursting or set pressure at this temperature does not exceed the MAWP of the vessel at the operating temperature per the ASME code [1] (see Figures 7-31A and 31B).

As allowed by code [1], the average of the manufacturer's disks burst tests could be stamped, for example, (144 + 160.5)/2 = 152.3 psig with an actual ±5% of 152.3 psig allowed for actual burst pressure of any disk *at the operating temperature.*

Selection of Burst Pressure for Disk, P_b (Table 7-3)

It is essential to select the type or style of rupture disk before making the final determination of the final burst pressure, and even this selection must recognize the pressure relationships between the disk's manufacturing range and the vessel's maximum allowable working pressure. (Also see Figures 7-31A and 31B.)

Table 7-9 summarizes the usual recommended relationship between the operating pressure of a process (should be maximum expected upper range level) and the set pressure of the rupture disk. Recognize that the set pressure of the disk must not exceed the MAWP of the vessel. (See Figures 7-31A and 31B.) The burst pressure, P_b, can now be defined. The use of the manufacturing range

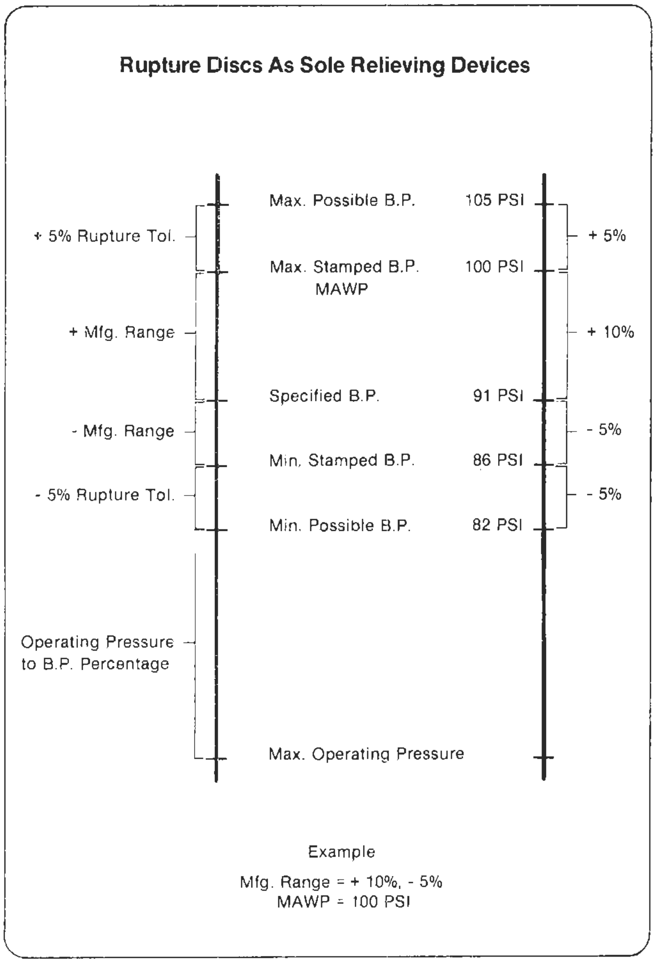


Figure 7-31A. Rupture disks as sole relieving devices. By permission, Fike Metal Products Div., Fike Corporation.

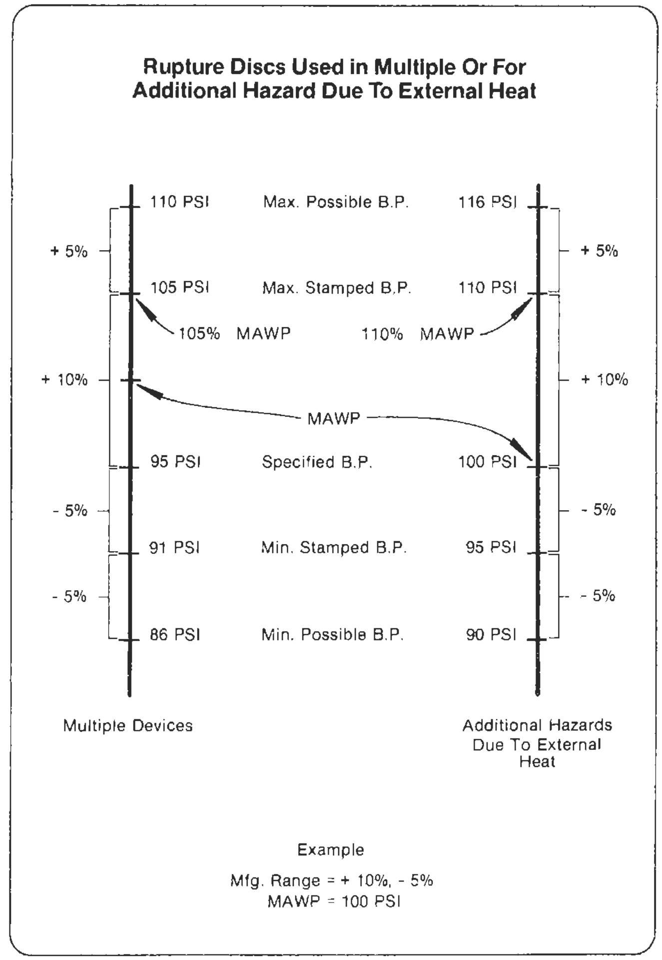


Figure 7-31B. Rupture disks used in multiple or for additional hazard due to external heat. By permission, Fike Metal Products Div., Fike Corporation.

(MR) discussed earlier applied to the burst pressure will establish the most probable maximum P_b.

The burst pressure maximum cannot exceed the MAWP of the vessel. Depending on the situation, it may be necessary to work backward to the operating pressure maximum to see if this is usable. Table 7-9 summarizes typical rupture disk characteristics noting that the maximum *normal* operating pressure of the system is shown as a function of the rupture disk bursting pressure, P_b.

Example 7-3: Rupture Disk Selection

Examination of the temperature control ranges of a process reactor reveals that the normal controls are to maintain a pressure of the reacting mixture of 80 psig, while the upper extreme could be 105 psig, which would be defined as the normal maximum operating pressure.

Select a conventional rupture disk, then from Table 7-9; $P_{max\ op} = 0.7\ P_b$

thus: $105 = 0.7\ P_b$

$P_{b-min} = 105/0.7 = 150$ psig *min* rupture disk burst pressure

From manufacturing range table for this type of disk, MR $= +10/-5\%$ @ 150 psig rupture pressure minimum. The maximum rupture pressure $= 150 + 10\% = 165$ psig, plus/minus the disk tolerance of 5%, allowing a final maximum burst pressure of 173.2 psig.

The *minimum* stamped bursting pressure of the disk would be 150 psig + 5% *tolerance* = 157.5 psig.

With a -5% MR, the *specified burst pressure* would be $157.5 + 5\% = 165.4$ psig.

With a $+10\%$ upper MR, the *maximum stamped* burst pressure of the disk could be $165.4 + 10\% = 181.9$ psig.

Using the code allowed tolerance for burst, the disk could burst at $181.9 + 5\% = 190.9$ psig. *The maximum allowable working pressure* for the vessel cannot be less than 181.9 psig.

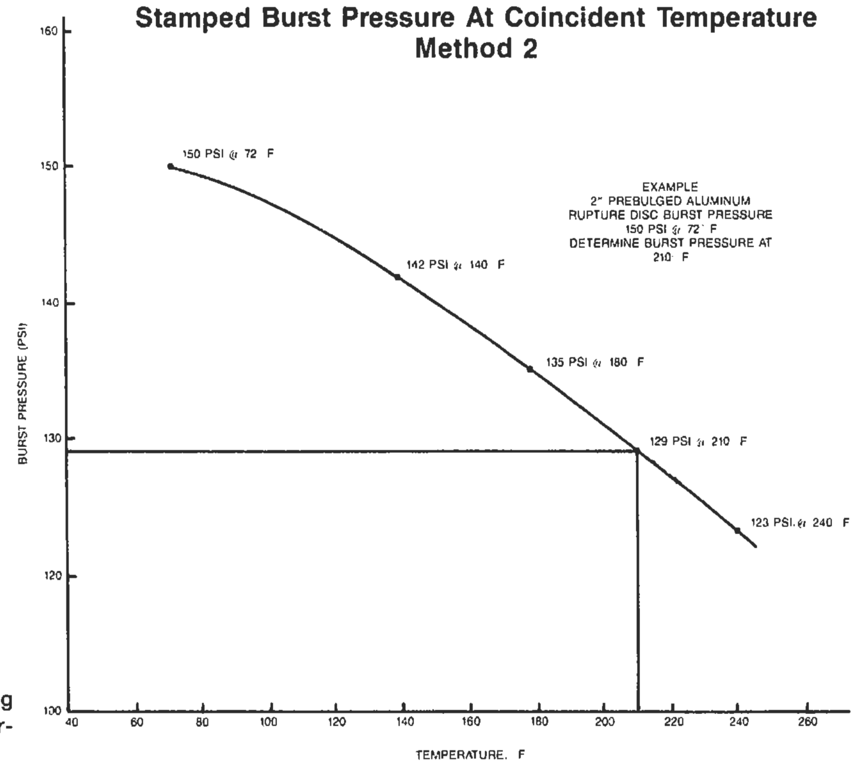


Figure 7-32. Establishing stamped rupture disk bursting pressure at coincident temperature, Method 2. By permission, Fike Metal Products Div., Fike Corporation.

Table 7-9
Summary of Rupture-Disk Characteristics

Type of disk	Vacuum support required?	Fragment upon rupture?	Gas service?	Liquid or partially-liquid service?	Maximum normal operating pressure
Conventional	Sometimes	*Yes/No*	Yes	Yes	$0.7\ P_b$
Pre-scored tension-loaded	No	No	Yes	Yes	$0.85\ P_b$
Composite	Sometimes	Yes	Yes	Yes	$0.8\ P_b$
Reverse-buckling with knife blades	No	No	Yes	No	$0.9\ P_b$
Pre-scored reverse-buckling	No	No	Yes	No	$0.9\ P_b$

*Depends on manufacturer's specifications (this author).
By permission, Nazario, F. N., *Chem. Eng.*, June 20, 1988, p. 86 [37].

From this guide to determine the bursting pressure of the rupture disk, it is apparent that some thought must be given to the process and the equipment design and ultimate MAWP. It is not an arbitrary selection. When changing services on a vessel, the MAWP and the operating pressures of the process must be established and then the bursting specifications of the disk determined. When reordering rupture disks for a specific service to repeat a performance of a ruptured disk, it is important to set the bursting specifications and the MR exactly the same as the original order, otherwise the maximum stamped P_b could be too high for the vessel and the system not properly protected against overpressure.

Effects of Temperature on Disk

The temperature at the burst pressure must be specified to the manufacturer, as this is essential in specifying the metal or composite temperature stresses for the disk finally supplied. Higher temperatures reduce the allowable working stress of the disk materials. Reference [37] shows that temperature has an effect on metals in decreasing order, with the least effect on the lowest listed metal:

aluminum
stainless steel (changes after 400°F)
nickel
Inconel

When specifying the material at the disk temperature, the heat loss at the disk/disk holder as well as in a flowing pipe must be recognized and the assembly may need to be insulated. This is important as it relates to the actual temperature at the bursting pressure. Establishing this burst temperature is an *essential* part of the system safety and must not be guessed at or taken lightly.

Table 7-10 presents a temperature conversion table for various metals from one manufacturer for conventional pre-bulged, tension-loaded disks with pressure on concave side (not prescored) as an illustration of the effect of lower or elevated temperatures referenced to 72°F on the burst pressure of a stamped disk. For other types of disk designs and from other manufacturers, the specific data for the style disk must be used to make the appropriate temperature correction.

Rupture Disk Assembly Pressure Drop

The ruptured or burst disk on a vessel or pipe system presents a pressure drop to flow at that point, and it can be estimated by assuming the disk is a flat plate orifice [37] with a discharge coefficient, K_d, of 0.62. As an alternate, the disk assembly can be assumed to be the equivalent of a section of pipe equal to 75 nominal disk diameters in length.

Gases and Vapors: Rupture Disks [33a, Par.4.8]

The sizing is based on use of the ASME Code [17] flow coefficient:

$K_d = 0.62$ [1] (Par. UG-127) for standard metal disks, but use $K_d = 0.888$ for graphite disks [70]
K_d = actual flow/theoretical flow = coefficient of discharge

To select the proper sizing equation, determine whether the flowing conditions are sonic or subsonic from the equations. When the absolute pressure downstream or exit of the throat is less than or equal to the critical flow pressure, P_c, then the flow is critical and the designated equations apply [33a]. When the downstream pressure exceeds the critical flow pressure, P_c, then subcritical flow will occur, and the appropriate equations should be used [33a].

When P_1 is increased, the flow through an open disk increases and the pressure ratio, P_2/P_1, decreases when P_2 does not change, until a value of P_1 is reached, and there is no further increase in mass flow through the disk. The value of P_1 becomes equal to P_c, and the ratio is the critical pressure ratio, and the flow velocity is sonic (equals the speed of sound).

Maximum velocity (sonic) of a compressible fluid in pipes is [9]:

$$v_s = \sqrt{kgRT} \tag{7-35}$$

$$\text{or, } v_s = \sqrt{kgP'\overline{V}(144)} \tag{7-36}$$

$$\text{or, } v_s = 68.1\sqrt{KP'\overline{V}} \tag{7-37}$$

where
v_s = sonic or critical velocity of a gas, ft/sec
k = ratio of specific heats, c_p/c_v
g = acceleration of gravity = 32.2 ft/sec/sec
R = individual gas constant = MR/M = 1544/M
M = molecular weight
MR = universal gas constant = 1544
T = absolute temperature, °R = (460 + t°F)
t = temperature, °F
P' = pressure, psia, at outlet end or a restricted location in pipe when pressure drop is sufficiently high
$\overline{V}$ = specific volume of fluid, cu ft/lb

For sonic flow [33a]:

Actual pressure ratio, P_2/P_1, less than critical pressure ratio, flow is sonic or critical.

$$P_c/P_1 = [2/(k+1)]^{k/(k-1)}, \text{ critical pressure ratio} \tag{7-7}$$

where
P_c = critical flow throat pressure, psia
P_b = stamped bursting pressure, psia = burst pressure + overpressure allowance (ASME Code of 10%) plus atmospheric pressure, psia

Important note: when actual system ratio, P_2/P_1, is less than critical pressure ratio calculated above by Equation 7-40, flow is sonic. When actual P_2/P_1 ratio is greater than critical pressure ratio, flow is subsonic.

Table 7-10
Temperature Conversion Table for Conventional Rupture Disks Only

TEMPERATURE CORRECTION FACTOR IN % FROM RUPTURE PRESSURE AT 72°F.

DISK TEMP. °F	RUPTURE DISK METALS					
	ALUM.	SILVER	NICKEL	MONEL	INCONEL	316 S.S.
−423	170	164	165	155	132	200
−320	152	152	144	140	126	181
−225	140	141	126	129	120	165
−200	136	138	122	126	118	160
−150	129	130	116	123	115	150
−130	127	126	116	121	114	145
−110	122	123	115	120	113	141
−100	120	122	115	119	112	139
−90	120	121	114	118	112	136
−80	120	120	114	117	111	134
−70	119	120	113	116	110	132
−60	119	119	112	115	110	130
−50	119	118	112	114	109	128
−40	118	117	111	113	108	125
−30	117	115	110	112	108	123
−20	116	112	109	111	107	121
−10	115	110	108	110	106	118
0	114	108	107	109	105	116
10	113	107	106	108	105	114
20	111	105	105	106	104	112
30	110	104	104	105	103	110
40	108	103	103	104	102	107
50	106	102	102	103	102	105
60	103	101	101	101	101	103
72	100	100	100	100	100	100
80	100	100	100	99	100	99
90	99	99	99	98	99	98
100	98	99	99	97	99	96
110	97	98	98	96	99	95
120	97	98	98	95	98	94
130	96	97	97	95	98	93
140	95	96	97	94	98	92
150	94	95	96	93	97	91
160	93	94	96	93	97	90
170	92	93	96	92	97	90
180	90	92	95	92	96	89
190	89	91	95	91	96	89
200	88	90	95	91	95	88
210	87	89	94	90	95	88
220	85	87	94	90	95	87
230	84	86	94	89	95	87
240	82	85	94	89	95	86
250	81	84	93	89	95	86
260			93	88	94	86
270			93	88	94	85
280			93	88	94	85
290			93	87	94	84
300			93	87	94	84

DISK TEMP. °F	RUPTURE DISK METALS			
	NICKEL	MONEL	INCONEL	316 S.S.
300	93	87	94	84
310	92	87	94	84
320	92	86	94	83
330	92	86	94	83
340	92	86	94	83
350	91	85	93	82
360	91	85	93	82
370	91	85	93	82
380	91	85	93	82
390	90	84	93	81
400	90	84	93	81
410	90	84	93	81
420	90	84	93	81
430	89	84	93	81
440	89	83	93	80
450	89	83	93	80
460	88	83	93	80
470	88	83	93	80
480	87	83	93	80
490	87	82	94	80
500	86	82	94	79
520	85	82	94	79
540	84	82	94	79
560	83	81	94	79
580	82	81	94	78
600	81	81	94	78
620	79	80	94	77
640	78	80	94	77
660	77	79	93	77
680	76	79	93	76
700	75	78	93	76
720	73	77	93	76
740	72	77	93	76
760	—	76	93	75
780	—	76	93	75
800	—	75	92	75
820	—	—	92	75
840	—	—	92	75
860	—	—	92	75
880	—	—	91	74
900	—	—	91	74

NOTE. This conversion table does not apply to Type D or reverse buckling disks.

By permission, B. S. and B. Safety Systems, Inc.

Table 7-10
continued

Example

What is rupture pressure at 500°F of a nickel disk rated 300 psi at 72°F?

1. Consult temperature conversion table. Correction factor for nickel disk at 500°F is 86%.
2. Multiply disk rating at 72°F by correction factor: 300 × 0.86 = 258.

   Rupture pressure of a nickel disk rated 300 psi at 72°F is therefore 258 psi at 500°F.

If you require a disk for a specific pressure at elevated or cold temperature and want to determine if it is a standard disk, convert the required pressure at elevated or cold temperature to pressure at 72°F.

P_2 = backpressure or exit pressure, psia
P_1 = upstream relieving pressure, psia

For sonic flow conditions [69]:

$$A = \frac{W}{CK_d P_b K_b} (ZT/M)^{0.5}, \text{ sq in.} \qquad (7-10)$$

where A = minimum net required flow discharge area after complete burst of disk, sq in.
C = sonic flow constant for gas or vapor based on ratio of specific heats, k, Figure 7-25, when k is not known use k = 1.001, or C = 315
W = required flow, lb/hr
M = molecular weight
K_d = coefficient of discharge, K = 0.62 for rupture disks, except some coefficients are different. For example, the Zook graphite standard ASME disks when tested mono-style, Figures 7-9B and 7-13A have a K_D of 0.888, and when inverted, Figure 7-13B have a K_D of 0.779. Consult manufacturer for special disks.
T = flowing relieving temperature, °F + 460 = °R absolute
M = molecular weight of flowing fluid
Z = compressibility factor for deviation from perfect gas if known, otherwise use Z = 1.0 for pressures below 250 psia, at inlet conditions.
P_b = stamped bursting pressure plus overpressure allowance (ASME 10% or 3 psi whichever is greater) plus atmospheric pressure (14.7), psia

Volumetric flow: SCFM standard conditions (14.7 psia and 60°F)

$$A = \frac{Q_s (MTZ)^{1/2}}{6.326\ CK_d P_b}, \text{ sq in.} \qquad (7-38)$$

Q_s = required flow, cu ft/min at standard conditions of 14.7 psia and 60°F, SCFM

Actual flowing conditions, ACFM

$$A = \frac{5.596\ Q_A}{CK_d} \sqrt{\frac{M}{TZ}} \qquad (7-39)$$

Q_A = required flow, cu ft/min at actual conditions, ACFM

Steam: Rupture disk sonic flow; critical pressure = 0.55 and P_2/P_1 is less than critical pressure ratio of 0.55.

API reference [33a] [69] dry and saturated steam, pressure up to 1500 psig:

$$A = \frac{W}{51.5\ K_d P_b K_n K_{sh}}, \text{ sq in.} \qquad (7-40)$$

where W = flow, lbs/hr
K_d = coefficient of discharge = 0.62.
K_n = correction for Napier equation = 1.0 when $P_1 \leqq$ 1515 psig = $(0.1906P_1 - 1000)/(0.2292P_1 - 1061)$ where $P_1 > 1515$ psia and ≤ 3215 psia, Table 7-6.
K_{sh} = superheat correction factor, see Table 7-7. For saturated steam at any pressure, K_{sh} = 1.0
P_b = stamped bursting pressure, psia

API for subsonic flow: gas or vapor (not steam)

For rupture disks, pressure ratio is greater than critical pressure; mass flow: pounds/hr

$P_c/P >$ critical pressure ratio $[2/(k+1)]^{k/k-1}$

$$A = \frac{W}{735\ C_2 K_d} \sqrt{\frac{ZT}{MP_1 (P_1 - P_2)}} \qquad (7-41)$$

Where C_2 = subsonic flow conditions based on ratio of specific heats (See Table 7-11 and equation for C_2)

Volumetric flow, SCFM conditions

$$A = \frac{Q_s}{4645.2\ C_2 K_d} \sqrt{\frac{ZTM}{P_1 (P_1 - P_2)}} \qquad (7-42)$$

Actual flowing conditions, ACFM

$$A = \frac{Q_A}{131.43\ C_2 K_d} \sqrt{\frac{P_1 M}{ZT (P_1 - P_2)}} \qquad (7-43)$$

Table 7-11
Constant, C_2, for Gas or Vapor for Subsonic Flow Conditions

SUBSONIC FLOW CONDITIONS

k \ P_2/P_1	0.95	0.90	0.85	0.80	0.75	0.70	0.65	0.60	0.55	0.50	0.45
1.001	158.1	214.7	251.9	277.8	295.7	307.4	313.7				
1.05	158.4	215.5	253.3	280.0	298.7	311.2	318.4	321.0			
1.10	158.7	216.3	254.7	282.0	301.5	314.8	322.9	326.4			
1.15	158.9	216.9	255.9	283.9	304.1	318.2	327.1	331.4			
1.20	159.2	217.6	257.0	285.6	306.5	321.3	331.0	336.0			
1.25	159.4	218.1	258.1	287.2	308.7	324.2	334.5	340.4			
1.30	159.6	218.7	259.1	288.7	310.8	326.9	337.9	344.4	346.8		
1.35	159.7	219.2	260.0	290.1	312.7	329.4	341.1	348.2	351.3		
1.40	159.9	219.6	260.8	291.4	314.5	331.7	344.0	351.8	355.5		
1.45	160.0	220.1	261.6	292.6	316.2	334.0	346.8	355.2	359.5		
1.50	160.2	220.5	262.3	293.7	317.8	336.0	349.4	358.3	363.3		
1.55	160.3	220.8	263.0	294.8	319.3	338.0	351.8	361.3	366.8		
1.60	160.4	221.2	263.7	295.8	320.7	339.8	354.1	364.2	370.2	372.3	
1.65	160.6	221.5	264.3	296.7	322.0	341.6	356.3	366.8	373.4	376.1	
1.70	160.7	221.8	264.8	297.6	323.2	343.2	358.4	369.4	376.4	379.6	
1.75	160.8	222.1	265.4	293.5	324.4	344.8	360.4	371.8	379.3	383.0	
1.80	160.9	222.4	265.9	299.3	325.5	346.2	362.3	374.1	382.1	386.3	
1.90	161.0	222.9	266.9	300.7	327.6	349.0	365.7	378.4	387.2	392.3	
2.00	161.2	223.4	267.7	302.1	329.5	351.5	368.9	382.3	391.8	397.8	400.1
2.10	161.4	223.8	268.5	303.3	331.2	353.7	371.8	385.8	396.1	402.8	405.9
2.20	161.5	224.2	269.2	304.4	332.8	355.8	374.4	389.1	400.1	407.5	411.4
2.30	161.6	224.5	269.9	305.4	334.2	357.7	376.9	392.1	403.7	411.8	416.4

$$C_2 = 735\sqrt{\frac{k}{k-1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{k}} - \left(\frac{P_2}{P_1}\right)^{\frac{k+1}{k}}\right]}$$

By permission Continental Disc Corp., Cat. 1-1110, p. 4.

Converting actual flow conditions to standard conditions of 60°F and 14.7 psia [73]:

$$Q_s = [\{(520/14.7)\}\,(P_{act}/T_{act})]\,(Q_{act}) = \text{SCFM, 60°F and 14.7 psia} \qquad (7\text{-}44)$$

where P_{act} = pressure actual, psia
T_{act} = temperature actual, °F + 460°F = °R
Q_{act} = flow at actual conditions: ACFM, at actual flowing conditions
Q_s = required flow, cu ft/min at standard conditions (14.7 psia at 60°F), (60°F + 460 = 520°R)

Liquids: Rupture disk

The test for critical or non-critical does not apply.

These equations apply to single-phase (at inlet) liquids, non-flashing to vapor on venting, fluid viscosity is less than or equal to water [69].

ASME mass flow :

$$A = \frac{W}{2407\, K_d \sqrt{(P_b - P_2)\rho_1}}, \text{ sq in.} \qquad (7\text{-}45)$$

where W = flow, lb/hr
ρ = Fluid density, lb/cu ft
K_d = coefficient of discharge = 0.62

P_b = stamped bursting disk pressure plus accumulation of 10% plus atmospheric pressure, psia
P_2 = pressure on outlet side of rupture disk, psia

$$\text{Volumetric flow}: A = \frac{W_L \sqrt{Sp\ Gr}}{38\ K_d \sqrt{P_b - P_2}}, \text{ sq in.} \quad (7\text{-}46)$$

where Sp Gr = fluid specific gravity relative to water = 1.0 @ 60°F
W_L = liquid flow, gallons per minute

Sizing for Combination of Rupture Disk and Pressure Relief Valve in Series Combination

When the rupture disk is installed on the inlet side of the pressure relief valve (see Figures 7-10, 7-11 and 7-12), the ASME code requires that for untested disk-valve combinations that the relieving capacity of the combination be reduced to 80% of the rated relieving capacity of the pressure relief valve [1].

For flow tested combinations, see a few typical data in Table 7-12. Note, for example, that using a Continental disk reverse acting knife blade rupture disc with a Crosby JOS/JBS pressure relief valve that the combined effect is to multiply the rated capacity of the Crosby valve by a multiplier of 0.985 for a set pressure in the 60–74 psig range using a 1½-inch disc with Monel metal. Other disk and valve manufacturers have their own combination data, which when available, avoids the requirement of derating the capacity to 80% of the rated capacity of the pressure relief valve. Disks (metal or graphite) that fragment *should never* be used because these may become potential problems for the safety valve performance. Therefore, a non-fragmenting disk should be selected, such as a reverse acting/reverse buckling preferably pre-scored design, but knife blades are a viable alternate.

Example 7-4: Safety Relief Valve for Process Overpressure

The conditions set forth on the Operational Checklist, Figure 7-14, are used in the example specified on the specification form (Figure 7-33).

Example 7-5: Rupture Disk External Fire Condition

An uninsulated 12-ft × 36-ft horizontal storage tank containing CCl_4 is to be protected from overpressure due to external fire by means of a rupture disk. The tank does not have a sprinkler system. Storage pressure is 5 psig and should not exceed 10 psig. Tank is assumed to be full. k =

Table 7-12
Rupture Disk/Relief Valve Combination Capacity Factors

DISC TYPE	DISC SIZE	DISC MATERIAL	SET PRESSURE PSIG	TELEDYNE FERRIS 2600 & 4500	DRESSER 1900 1900-30 1900-35	CROSBY JOS/JBS	CROSBY JB	CROSBY JO	KUNKLE 5000 THRU 5999	LONERGAN D & DB
ZAP (contd)	1.5	Monel	60-74	0.982	—	0.985	0.983	0.980	—	—
	1.5	Monel	75 PLUS	0.982	—	0.985	0.990	0.986	—	—
	1.5	Nickel	30-49	—	—	0.965	—	0.975	—	—
	1.5	Nickel	50-59	0.989	0.984	0.965	—	0.975	0.988	0.966
	1.5	Nickel	60 PLUS	0.985	0.984	0.992	—	0.994	0.988	0.996
ZAP	3.0	Stainless Steel	15-29	0.963	0.966	—	—	—	—	—
	3.0	Stainless Steel	30-34	0.993	0.966	0.955	—	—	—	—
	3.0	Stainless Steel	35-49	0.993	0.966	0.970	—	—	—	—
	3.0	Stainless Steel	50-59	0.993	0.966	0.970	0.976	0.981	—	—

Extracted by permission: Continental Disc Corp., Bul. #1-1111, pg. 4. Only portion of original tables presented for illustration.
Note: Zap is a reverse-acting rupture disk using replaceable knife blades. Patented. Other disk and valve manufacturers have tested their own combinations to obtain combination capacity factors.

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Spec. Dwg. No.	A-
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Unit Price	
No. Units	
Item No.	

SAFETY VALVE SPECIFICATIONS

DESCRIPTION

Make ______ Model ______ Standard Type-Back Press. Stamdard

Size (Inlet x Orifice No. x Outlet) 4 x L x 6 Phase Vapor

Set Press[1] 75 (PSIG) ( ) @ 347 °F Full Flow Back Press. 0 PSIG

Req'd Orifice Area 2.02 Sq. In.: Selected Orifice Area 2.853 Sq. In.

Accessories[2] Screwed Cap, No Test Gag, No Lifting Gear

Inlet Nozzle: Press. Class 250#-4" Facing Raised

Outlet Nozzle: Press. Class 125 #- 6" Facing Raised

Mfgr's. Rating 250# PSIG (Max) @ 450 °F: 125#(outlet) PSIG (Min) @ 300 °F

[1] Refers to Initial Vessel relief pressure. [2] Refers to cap type, lifting gear, gag, etc.

MATERIALS

Body and Bonnet Cast Iron Trim Bronze

Nozzle and Disc Bronze Bellows None

Spring Carbon Steel, cadium plated, or equal

REMARKS

Use three-way valve (Yes) (No) No

PROCESS RATING DATA

Location On Evaporator Shell

Fluid PDC Flow Based On Operations Set pressure Based On Vessel Max.

System Oper. Press. 60 PSIG @ 347 °F: MW* 113.5

Req'd Cap. 22,500 Lbs/Hr. @ 347 °F* Phase: ______ Liquid Dens.* ______ Lbs/Cu. Ft.

Liq. Viscosity* ______ SSU: ~~Accumulation~~ 10 %: Back Press. Corr. Factor ______
overpressure

* Refers to properties at Set Pressure

CALCULATIONS

Use: $A = W(Z\ T/M)^{\frac{1}{2}} / C\ K_d P\ K_b$

Since data for c_p/c_v not readily available, use k = 1.001 and C = 315, Z = 1.0, K_b=1.0

P (at relieving condition) = (75)(1.1) + 14.7 = 97.2 psia

T = 347 + 460 = 807 R

$$A = 22{,}500\left[(1)(807)/113.5\right]^{\frac{1}{2}} / (315)(0.953)(97.2)(1.0)$$

A = 2.055 Sq. In.

Note that critical flow conditions apply, since relieving pressure of 97.2 psia is over twice the backpressure of 14.7 psia.

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Figure 7-33. Safety relief valve specification for process overpressure example.

1.13. Disk burst pressure to be 10 psig. MAWP of vessel is 50 psig. Discharge line back pressure is 1 psig.

Heat Input

Using ASME flanged and dished heads (F&D) from Appendix Tables of Blanks, the circle size is 152 in. for a 12-ft diameter tank. Then add 3-in. straight flange which becomes 158 in. which is 158/12 = 13.166 ft diameter. Area of this diameter for surface area of head = 136.14 sq ft equivalent surface area of one head. For a horizontal vessel there are two heads possibly exposed to fire.

External Surface Area: cylindrical area + surface area heads (2)

$$= \pi (12)(36) + \left(\frac{158}{12}\right)^2 \left(\frac{\pi}{4}\right) 2$$
$$1357 + 274 = 1629 \text{ sq ft (approximate)}$$

Total Heat Input (From Figure 7-30)

$Q = 9.0 \times 10^6$ BTU/hr. at area of 1629 sq ft

Solving equation: $Q = 21{,}000 \text{ F } A^{0.82}$

$$Q = 21{,}000\ (1.0)\ (1629)^{0.82} = 9{,}036{,}300 \text{ BTU/hr}$$

Quantity of Vapor Released

Latent Heat CCl_4 = 85 BTU/lb

$$W = \frac{9{,}000{,}000}{85} = 106{,}000 \text{ lb/hr (rounded)}$$

Critical Flow Pressure

$$P_c = P\left(\frac{2}{k+1}\right)^{k/(k-1)} = 24.7\left(\frac{2}{1.13+1}\right)^{1.13/(1.13-1)}$$

$$P_c = 27.7\ (0.573) = 15.8 \text{ psia, or}$$

$$\frac{P_2}{P_1} \text{ actual} = \left(\frac{14.7 + 1.0}{(10)(1.1) + 14.7}\right) = 0.610$$

$$\frac{P_c}{P_1} = \left[\frac{2}{(1.13+1)}\right]\frac{1.3}{(1.3-1)} = 0.573$$

Disk Area

$$\text{Flow is subcritical, since} \left(\frac{P_2}{P_1}\right) > \frac{P_c}{P_1}$$

$$A = \frac{W}{735\ F_2 K_d}\sqrt{\frac{ZT}{MP_b\ (P_b - P_2)}}$$

W = 106,000 lbs/hr
K_D = 0.62
P_b = 10 + 14.7 = 24.7 psia (10) (1.10 see Fig. 7-11A) + 14.7 = 25.7 psia
P_2 = 15.7 psia
M = 154
T = 460 + 202 = 662° R (B.P. of CCl_4 @ 10 psig)
k = 1.13
F_2 = 0.691 interpolated from Figure 7-29

Area calculated substituting in above relation:

A = 43.52 sq in.

8-inch Sch. 40 pipe has a cross-sectional area of 50.0 sq in. so an 8-inch frangible rupture disk will be satisfactory. Disk material to be lead or lead covered aluminum.

Example 7-6: Rupture Disk for Vapors or Gases; Non-Fire Condition

Determine the rupture disk size required to relieve the pressure in a process vessel with the following conditions: k = 1.4.

Vessel: MAWP = 85 psig; also = disk set pressure
Vapor flow to relieve: 12,000 std. cu ft/min @ 60°F and 14.7 psia
Flowing temperature: 385°F
Vapor mol. wt: 28
Backpressure on discharge of disk: 30 psig

Determine if conditions on rupture are critical or non-critical, based on 10% overpressure for primary relief.

$$\frac{P_2}{P_1} = \frac{(30 + 14.7)}{[(85)(1.10) + 14.7]} = 44.7/108.2 = 0.413$$

$$\text{Critical pressure ratio} := P_c / P_1 = \left[\frac{2}{1.4+1}\right]^{\frac{1.4}{1.4-1}} = 0.528$$

Since actual P_2/P_1 < critical ratio 0.528, the flow is sonic.

Critical pressure = P_{cr} = 108.2 (0.528) = 57.12 psia

Flow area required to relieve: [33a]

$$A = \frac{Q_s\,(MTZ)^{0.50}}{6.32\,CK_d P_b} \qquad (7\text{-}47)$$

where Q_s = 12,000 SCFM
M = 28
C = 356, Figure 7-25
K_d = 0.62 (rupture disk)
P_b = 108.2 psia
Z = 1.0
T = 385 + 460 = 845°R

$$A = \frac{12000\,[(28)(845)(1.0)]^{0.5}}{6.32\,(356)(0.62)(108.2)}$$

A = 12.22 sq in.

Area of I.D. of 4 in. std. sch. 40 pipe = 12.7 sq in. This should be adequate.

Example 7-7: Liquids Rupture Disk

Determine the rupture disk size required to relieve the following operating condition:

Back pressure: 0 psig
Toluene flow: 1800 gpm, SpGr = 0.90
Pressure vessel: MAWP = 25 psig
Relieving pressure: set pressure; use 25 psig
Actual relief pressure = 25 + 10% = 27.5 psig

$$A = \frac{W_L}{38\,K_d}\sqrt{\frac{(SpGr)}{(P_1 - P_2)}}$$

SpGr = 0.90
Fluid: Toluene, flow − 1800 gpm

$$A = \frac{gpm\,(SpGr)^{0.5}}{38.0\,K\,(P - P_d)^{0.5}}$$

K_d = 0.62 per API [33a], or from manufacturer
P_1 = 27.5 + 14.7 = 42.2 psia
P_2 = 0 + 14.7 = 14.7 psia

$$A = \frac{1800}{38\,(0.62)}\sqrt{\frac{0.90}{(42.2 - 14.7)}}$$

A = 76.40 (0.180) = 13.75 sq in. required

Use Std. pipe size: 5 in. (cross sect. area = 26.01 sq in) disk, or check manufacturer. Use inlet and discharge pipe size = 6 in. std.

Example 7-8: Liquid Overpressure, Figure 7-34

A check of possible overpressure on a heat exchanger handling 95% aqua ammonia in the tubes indicates that tube failure is probably the condition requiring maximum relieving capacity. The aqua is being pumped by a positive displacement pump. Thirty psia steam on shell side heats (not vaporizes) the aqua. In case of tube failure the aqua would flow into the shell and soon keep the steam from entering. Relief must prevent the shell from failing. The shell is designed for a working pressure of 210 psig.

The calculations are shown on the specification sheet, Figure 7-34.

Pressure—Vacuum Relief for Low Pressure Storage Tanks

In order to accommodate "breathing" of tanks and other equipment due to temperature changes, pumping in and out, internal vapor condensation and other situations, adequate safety vacuum relief must be provided. In many cases both pressure and vacuum relief are needed (see Figure 7-35). For the average product storage tank the API Guide For Tank Venting RP 2000 [26] serves to set *minimum* venting quantities for various tank capacities. In addition to these tabulated values, calculations are made to satisfy each condition of operation to insure that there is no situation requiring more than this vent capacity. Emergency vent capacity is also required to supplement the normal requirement in case of external fire or other unusual condition.

The normal venting to be provided must not allow pressure or vacuum conditions to develop which could cause physical damage to the tanks [26].

Basic Venting For Low Pressure Storage Vessels

Usually reference to low pressure venting is associated with large storage tanks of several thousand gallons capacity (ranging from a few thousand to a million); however, small low pressure tanks can be handled in the same manner. The usual operating pressure range for the typical tank is 0.5 oz./in.^{2} to about 1.5 psig. Since low pressure vessels have pressure ratings expressed in various units, Table 7-13 can be useful for conversion.

Typical large storage vessels are illustrated in Figure 7-36.

Operations associated with storage tanks should be carefully analyzed, since there are several factors which

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SAFETY VALVE SPECIFICATIONS

DESCRIPTION

Make ______ Model ______ Standard Type-Back Press. Standard

Size (Inlet x Orifice No. x Outlet) ______ X ______ X ______ Phase Liquid

Set Press [1] 157 (PSIG) (XXX) @ 100 °F Full Flow Back Press. 0 PSIG

Req'd Orifice Area 0.0951 Sq. In.: Selected Orifice Area 0.110 Sq. In

Accessories[2] No lift gear, No finned bonnet, No test gag

Inlet Nozzle: Press. Class $1\frac{1}{2}$", 150# Facing Raised

Outlet Nozzle: Press. Class 2", 150# Facing Raised

Mfgr's. Rating 230 PSIG (Max) @ 100 °F; 160 PSIG (Min) @ 450 °F

[1] Refers to Initial Vessel relief pressure. [2] Refers to cap type, lifting gear, gag, etc.

MATERIALS

Body and Bonnet Cast carbon steel Trim Stainless Steel

Nozzle and Disc orged stainless steel Bellows None

Spring Carbon steel

REMARKS

Use three-way valve (Yes) (No) No

Plant standards do not accept 1" flanged connections, therefore use $1\frac{1}{2}$"

PROCESS RATING DATA

Location Shell side of heater

Fluid 95% Aqua Ammonia Flow Based On Tube failure and ~~pump capacity~~ Set pressure Based On Max Working Pre

System Oper. Press. 30(shell) PSIG @ 275 °F; MW* 17

Req'd Cap. 40 gpm Lbs/Hr. @ 100 °F* Phase: Liquid Liquid Dens.* 36 Lbs/Cu. Ft

Liq. Viscosity* 0.05 cp ~~SSU~~ XXX: ~~Accumulation~~ 25 %: overpressure Back Press. Corr. Factor None

* Refers to properties at Set Pressure

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CALCULATIONS

Max Working Pressure (Shell Side) = 210 psig

* For 25% accumulation, set pressure = (0.75)(210) = 157 psig

*Relieving pressure @ 10% overpressure = [157 + 10%] + (14.7) = 187.4 psia

For Liquid Relief: per ASME Code

$A = GPM(\sqrt{SpGr}) / 38.0\ K_d\sqrt{\Delta P}\ (K_u)$

$K_d = 0.64$; $K_u = 1.0$ @ normal viscosity; GPM = 40

ΔP = 157 + 10%(157) - 0 = 172.7 psig

Sp Gr = 36/62.3 = 0.578

$A = (40)(\sqrt{0.578}) / 38(0.62)(\sqrt{172.7}) = 0.0981$ Sq. In.

Select 1" std. pipe size with cross-sectional area = 0.864 sq. in.

By	Chk'd	App.	Rev.	Rev.	Rev.
Date					

Figure 7-34. Safety relief valve specification for liquid overpressure example.

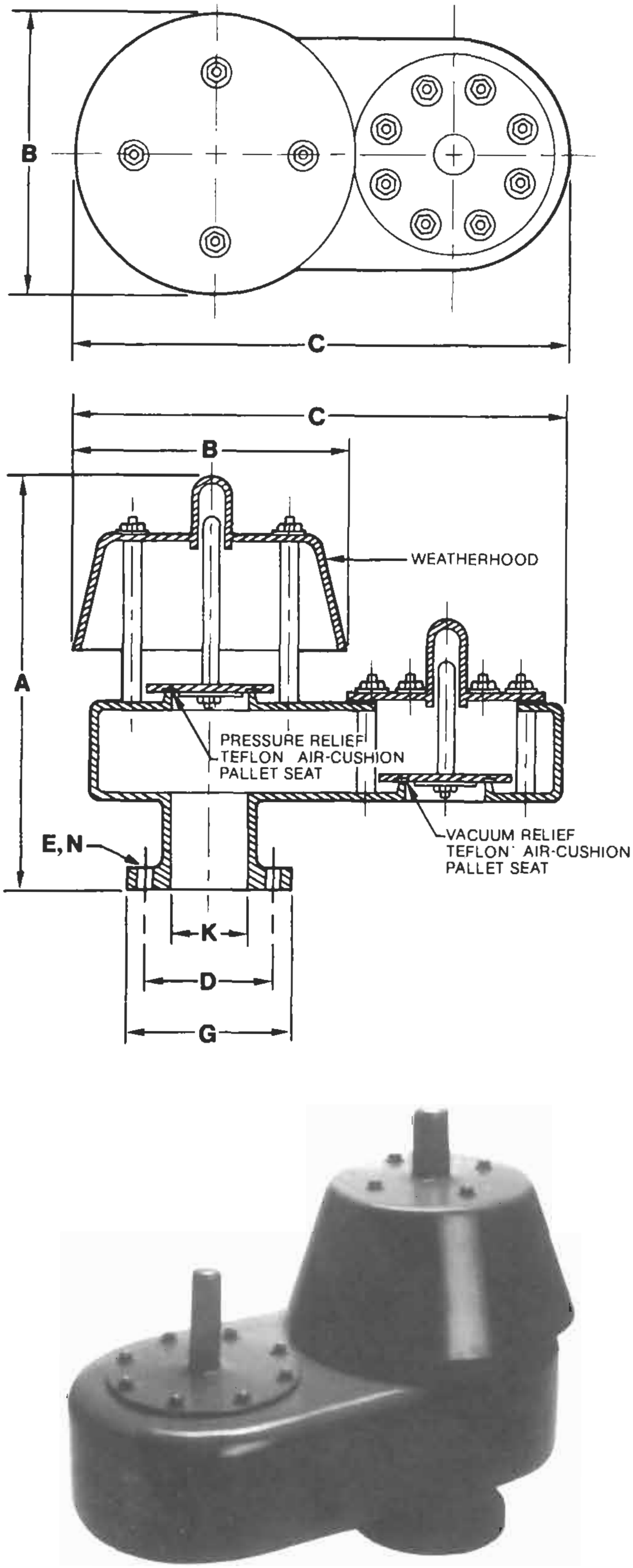

Figure 7-35. Dead weight type pressure and vacuum relief valve for low pressure storage vessels. By permission, The Protectoseal Co.

Table 7-13
Convenient Pressure Conversions

Oz./in.^{2}	Lb./in.^{2}	In. Hg (0°C)	In. H_2O (4°C)
1	0.06250	0.1272513	1.730042
16	1	2.036021	27.68068
7.85846	0.4911541	1	13.59548
0.57802	0.0361262	0.07355387	1

can significantly influence the safety relieving requirements. Usually, these are [26]

1. Normal operation
   (a) Outbreathing or pressure relief
   (b) Inbreathing or vacuum relief
2. Emergency conditions
   (a) Pressure venting
3. For tank design per API Standard 650 with weak roof to shell designs (roof lifts up) the venting requirements of API-Std-2000 do not apply for emergency venting to atmosphere or elsewhere.

Nonrefrigerated Above Ground Tanks; API-Std. 2000

Normal operations are to be established within the design parameters for the tank, thereby avoiding conditions that would damage it. Normal venting capacity should be at least the sum of venting required for oil/fluids movement and thermal effect. Required capacity can be reduced when liquid volatility is such that vapor generation or condensation in the allowable operating range of vessel pressure will provide all or part of the venting requirements.

Outbreathing conditions are usually established when (a) the tank is being filled and the vapor space is being displaced with liquid, (b) thermal expansion and evaporation of the liquid, and (c) external fire on the vessel creating additional heat input to the contents.

The standard [26] specifies venting capacity of:

1. Twelve hundred cubic foot of *free* air per hour for every 100 barrels (4200 gal) per hour of maximum filling rate, for liquids with flash points below 100°F.
2. Six hundred cubic foot of *free* air per hour for each 100 barrels (4200 gal) per hour of maximum filling rate, for liquids with flash points 100° F. and above.
3. Thermal outbreathing or venting requirements, including thermal evaporation for a fluid (the code refers to oil) with a flash point of 100°F or below, use at least the figures of Column 4 in Table 7-14.
4. Thermal outbreathing or venting requirements, including thermal evaporation for a fluid (the code

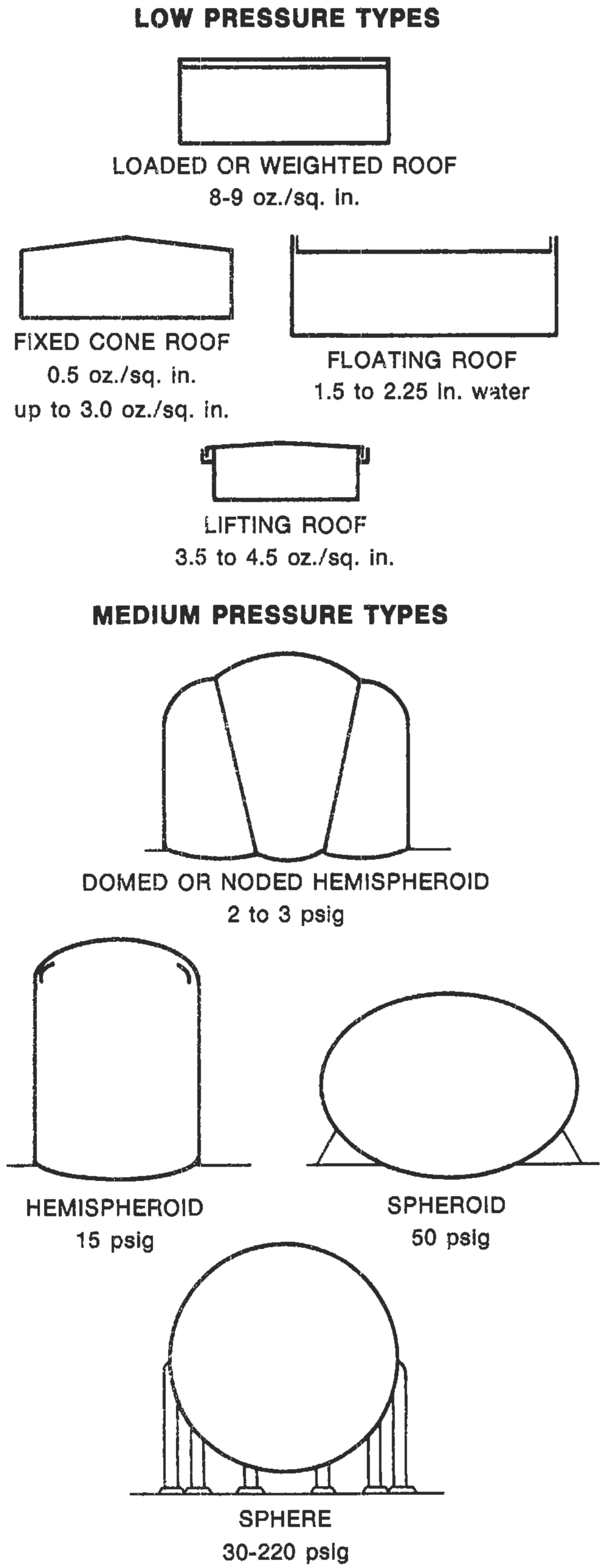

Figure 7-36. Representative configurations of large storage tanks. Usual operating/design pressures (max.) shown below illustration.

uses oil) with a flash point of 100°F or above, use at least the figures of column 3 in Table 7-14.

5. To attain the total venting (outbreathing) requirements for a tank, refer to the appropriate flash point column and add the outbreathing plus the thermal venting flows.

Corrections to Express Miscellaneous Liquids Venting in Terms of Free Air (14.7 psia and 60°F)

Tank vent equipment ratings are expressed as free air capacity at 14.7 psia and 60°F, and in order to handle vapors from liquids of the chemical and petrochemical industry, corrections must be made. Likewise corrections are required to recognize temperatures other than 60°F, refer to Table 7-16.

For any specific liquid vapors emergency venting:

$$V' = V_c \frac{(1,337)}{L_V \sqrt{M}} \sqrt{\frac{T}{520}} \qquad (7\text{-}48)$$

V_c is from Table 7-17

V' = venting requirement, cf/hr free air at 14.7 psia and 60°F

$$V' = 1107\ A_w^{0.82} \qquad (7\text{-}49)$$

A_w = Wetted area, sq ft, See Table 7-17

Correcting for effect of temperature at flowing conditions of vapor is included.

Note, most manufacturers' tables or charts give SCFH capacities at 14.7 psia and 60°F, and these must be corrected by the gas laws to the *actual* volume at flowing conditions in order to represent the actual performance of the system. The tables or charts of the manufacturers *read* in SCFH for selected *relief device setting and for tank pressure,* expressed as air *at SCFH* (see Figures 7-37A and 37B).

The following are convenient forms to express the capacities, using Table 7-14. Note "free" does not mean actual.

$$\text{Free } \textit{air} \text{ capacity of valve} = \frac{\text{(free gas capacity) (SpGr factor)}}{\text{temperature correction factor}} \qquad (7\text{-}50)$$

$$\text{Free gas (or vapor) capacity of valve} = \frac{\text{(free air capacity) (temp. corr. factor)}}{\text{SpGr factor}} \qquad (7\text{-}51)$$

At the same pressure and temperature, the free gas capacity of a valve varies inversely as the square root of the specific gravity of the vapor, with air = 1.0.

Table 7-14
Requirements for Thermal Venting Capacity

Column 1		Thermal Venting Capacity (cubic feet of free air[a] per hour)		
			Outbreathing (Pressure)	
		Column 2[b]	Column 3[c]	Column 4[d]
Tank Capacity		Inbreathing (Vacuum)	Flash Point ≥ 100 F (37.78 C)	Flash Point < 100 F (37.78 C)
Barrels	Gallons			
60	2,500	60	40	60
100	4,200	100	60	100
500	21,000	500	300	500
1,000	42,000	1,000	600	1,000
2,000	84,000	2,000	1,200	2,000
3,000	126,000	3,000	1,800	3,000
4,000	168,000	4,000	2,400	4,000
5,000	210,000	5,000	3,000	5,000
10,000	420,000	10,000	6,000	10,000
15,000	630,000	15,000	9,000	15,000
20,000	840,000	20,000	12,000	20,000
25,000	1,050,000	24,000	15,000	24,000
30,000	1,260,000	28,000	17,000	28,000
35,000	1,470,000	31,000	19,000	31,000
40,000	1,680,000	34,000	21,000	34,000
45,000	1,890,000	37,000	23,000	37,000
50,000	2,100,000	40,000	24,000	40,000
60,000	2,520,000	44,000	27,000	44,000
70,000	2,940,000	48,000	29,000	48,000
80,000	3,360,000	52,000	31,000	52,000
90,000	3,780,000	56,000	34,000	56,000
100,000	4,200,000	60,000	36,000	60,000
120,000	5,040,000	68,000	41,000	68,000
140,000	5,880,000	75,000	45,000	75,000
160,000	6,720,000	82,000	50,000	82,000
180,000	7,560,000	90,000	54,000	90,000

NOTE: Interpolate for intermediate tank sizes. Tanks with a capacity of more than 180,000 barrels require individual study. Refer to Appendix A for additional information about the basis of this table.
[a]At 14.7 pounds per square inch absolute (1.014 bar) and 60 F (15.56 C).
[b]For tanks with a capacity of 20,000 barrels or more, the requirements for the vacuum condition are very close to the theoretically computed value of 2 cubic feet of air per hour per square foot of total shell and roof area. For tanks with a capacity of less than 20,000 barrels, the requirements for the vacuum condition have been based on 1 cubic foot of free air per hour for each barrel of tank capacity. This is substantially equivalent to a mean rate of vapor-space–temperature change of 100 F per hour.
[c]For stocks with a flash point of 100 F or above, the outbreathing requirement has been assumed to be 60 percent of the inbreathing requirement. The tank roof and shell temperatures cannot rise as rapidly under any condition as they can drop, for example, during a sudden cold rain.
[d]For stocks with a flash point below 100 F, the outbreathing requirement has been assumed to be equal to the inbreathing requirement to allow for vaporization at the liquid surface and for the higher specific gravity of the tank vapors.

By permission, API Std. 2000, 3rd Ed., Jan. 1982, reaffirmed Dec. 1987, American Petroleum Institute [26].

The free air capacity of a valve varies directly as the square root of the absolute standard temperature, expressed as 460°F + 60°F, divided by the square root of the valve absolute inlet temperature in °Rankine.

The correction factors are noted for convenience in Table 7-16. The factors are determined as follows: If molecular weight of vapor in tank is 26.1, then the SpGr of gas = 26.1/29, referenced to air, = 0.90; so the SpGr correction factor = (0.90)½ = 0.9486.

If the temperature at the valve inlet is expected to be 50°F, then the temperature correction factor

$$= \left(\frac{460+60}{460+50}\right)^{1/2} = \left(\frac{520}{510}\right)^{1/2} = (1.0196)^{1/2} = 1.00975$$

Example 7-9: Converting Valve Capacities

The capacity of a valve as read from a manufacturer's table or chart is 45,000 cubic feet per hour of *free air* (14.7 psia and 60°F). What is the capacity of the valve in terms of the vapors expected to pass through the valve under the rated conditions at the *same* setting? If methanol is in the tank at 55°F

$$\text{SpGr} = \frac{32.04 \text{ M.W. methanol}}{29} = 1.104$$

The SpGr correction factor = 1.0518 (interpolated) and the temperature correction at 55°F = 1.0048; both are from Table 7-16.

(text continued on page 474)

Table 7-15
Convenient Constants for Venting Selected Chemicals

Chemical	$L\sqrt{M}$	Molecular Weight	Heat of Vaporization Btu per Lb. at Boiling Point
Acetaldehyde	1673	44.05	252
Acetic acid	1350	60.05	174
Acetic anhydride	1792	102.09	177
Acetone	1708	58.08	224
Acetonitrile	2000	41.05	312
Acrylonitrile	1930	53.05	265
n-Amyl alcohol	2025	88.15	216
iso-Amyl alcohol	1990	88.15	212
Aniline	1795	93.12	186
Benzene	1493	78.11	169
n-Butyl acetate	1432	116.16	133
n-Butyl alcohol	2185	74.12	254
iso-Butyl alcohol	2135	74.12	248
Carbon disulfide	1310	76.13	150
Chlorobenzene	1422	112.56	134
Cyclohexane	1414	84.16	154
Cyclohexanol	1953	100.16	195
Cyclohexanone	1625	98.14	164
o-Dichlorobenzene	1455	147.01	120
cis-Dichloroethylene	1350	96.95	137
Diethyl amine	1403	73.14	164
Dimethyl acetamide	1997	87.12	214
Dimethyl amine	1676	45.08	250
Dimethyl formamide	2120	73.09	248
Dioxane (diethylene ether)	1665	88.10	177
Ethyl acetate	1477	88.10	157
Ethyl alcohol	2500	46.07	368
Ethyl chloride	1340	64.52	167
Ethylene dichloride	1363	98.97	137
Ethyl ether	1310	74.12	152
Furan	1362	68.07	165
Furfural	1962	96.08	200
Gasoline	1370-1470	96.0	140-150
n-Heptane	1383	100.20	138
n-Hexane	1337	86.17	144
Hydrogen cyanide	2290	27.03	430
Methyl alcohol	2680	32.04	474
Methyl ethyl ketone	1623	72.10	191
Methyl methacrylate	1432	100.14	143
n-Octane	1412	114.22	132
n-Pentane	1300	72.15	153
n-Propyl acetate	1468	102.13	145
n-Propyl alcohol	2295	60.09	296
iso-Propyl alcohol	2225	60.09	287
Tetrahydro furan	1428	72.10	168
Toluene	1500	92.13	156
Vinyl acetate	1532	86.09	165
o-Xylene	1538	106.16	149

Note: For data on other chemicals, see chemistry handbook.

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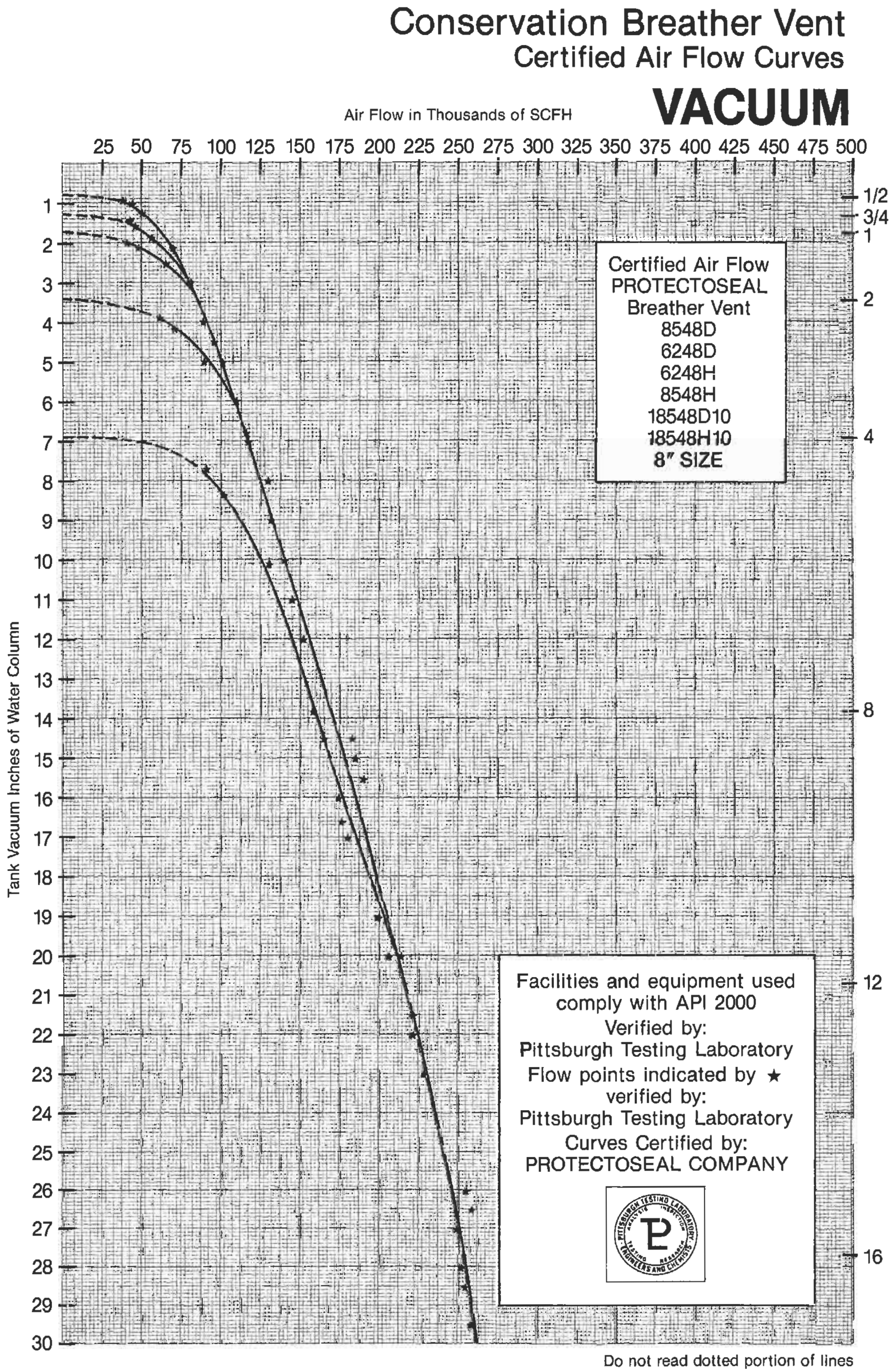


Figure 7-37A. Performance curve for conservation breather vent, dead weight type for vacuum, 8 inch size. By permission, The Protectoseal Co.

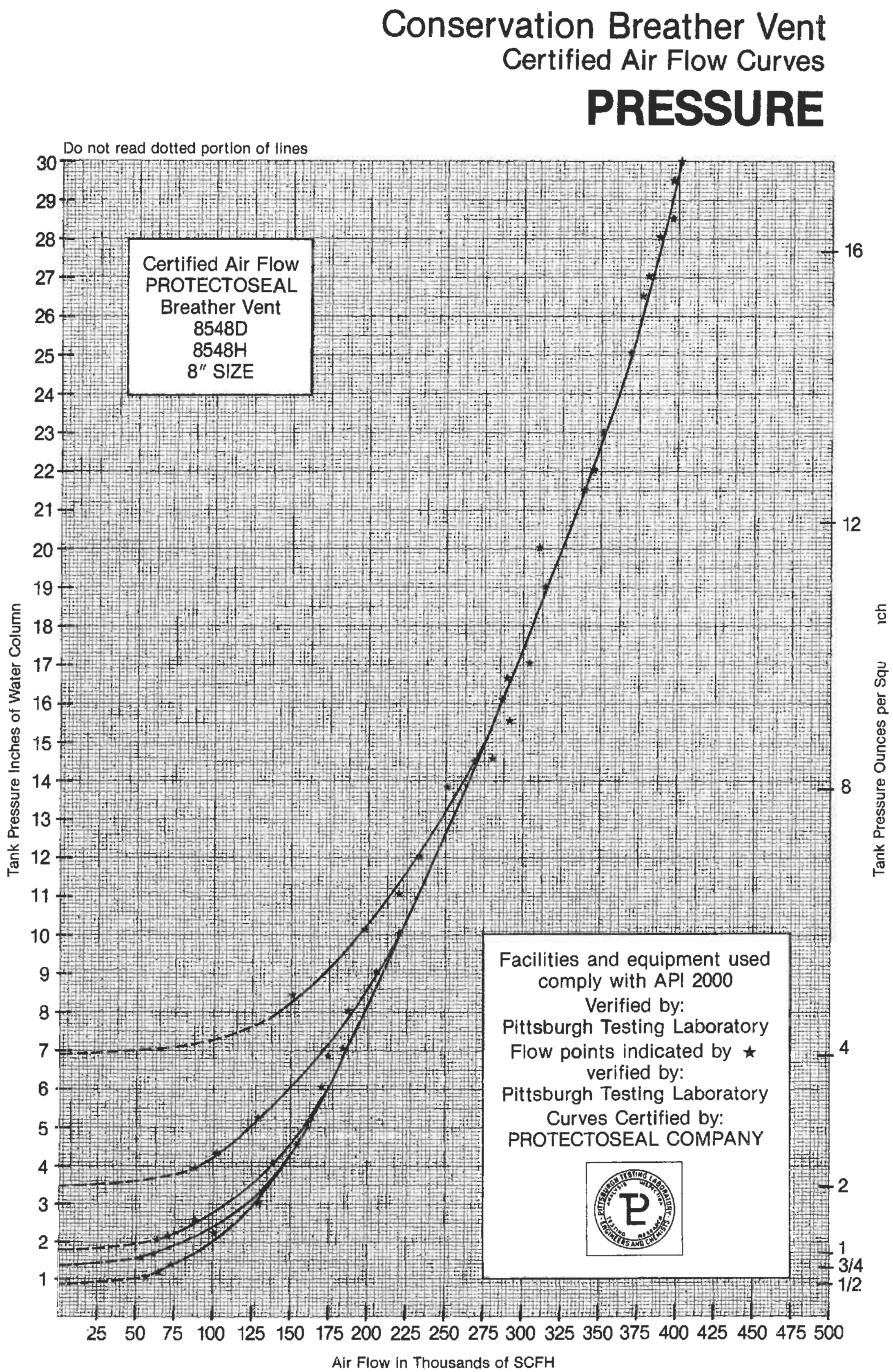


Figure 7-37B. Performance curve for conservation vent, dead weight type for pressure, 8 inch size. By permission, The Protectoseal Co.

Table 7-16
Gravity and Temperature Correction Factors for Low Pressure Venting Calculations for Vapors

Sp. Gr. of Gas. AIR = 1.00	Sp. Gr. Factors	Sp. Gr. of Gas AIR = 1.00	Sp. Gr. Factors	Temp. (°F)	Factor	Temp. (°F)	Factor	Temp. (°F)	Factor
.20	.447	1.10	1.050	5	1.0575	70	.9905	200	.8932
.30	.548	1.20	1.095	10	1.0518	80	.9813	220	.8745
.40	.632	1.30	1.141	15	1.0463	90	.9723	240	.8619
.50	.707	1.40	1.185	20	1.0408	100	.9638	260	.8498
.60	.775	1.50	1.223	25	1.0355	110	.9551	280	.8383
.65	.806	1.60	1.265	30	1.0302	120	.9469	300	.8272
.70	.837	1.70	1.305	35	1.0249	130	.9388	320	.8165
.75	.866	1.80	1.340	40	1.0198	140	.9309	340	.8063
.80	.894	1.90	1.380	45	1.0147	150	.9233	360	.7963
.85	.922	2.00	1.412	50	1.0098	160	.9158	380	.7868
.90	.949	2.50	1.581	55	1.0048	170	.9084	400	.7776
.95	.975	3.00	1.731	60	1.0000	180	.9014	420	.7687
1.00	1.000	3.50	1.870						
1.05	1.025	4.00	2.000						

By permission, Groth Equipment Corp., Tank Equipment Division.

(text continued from page 470)

$$\text{Free gas capacity of valve (at 14.7 psia and 60° F} = \frac{\text{(free air capacity) (temp. corr. factor)}}{\text{(SpGr correction factor)}} \quad (7\text{-}52)$$

$$\text{Free air capacity of valve} = \frac{\text{(free gas capacity) (SpGr corr factor)}}{\text{temperature corr. factor}} \quad (7\text{-}53)$$

$$\text{Free gas capacity of valve} = \frac{(45{,}000)\ (1.0)}{1.0518}$$

$$= 42{,}783 \text{ SCFH at } 60° \text{ F and } 14.7 \text{ psia}$$

If this volume were needed to be expressed at 55°F and 1.5 psig, then using the gas laws:

Actual flowing volume at 55°F and 1.5 psig

$$= (42{,}783 \text{ SCFH}) \frac{(460+55)}{(460+60)} \frac{(14.7)}{(14.7+1.5)}$$

$$= 38{,}448 \text{ cu ft/hr}$$

Example 7-10: Converting Required Free Air Capacity

The capacity of a relief valve for a tank has been calculated to be 50,000 cu ft/hr at 110°F and 1.0 psig, with a benzene vapor. Determine the required free air capacity.

For benzene: MW = 78.11

$$\text{SpGr} = \frac{78.11}{29} = 2.69$$

SpGr correction $= (2.69)^{1/2} = 1.64$
Temperature correction = 0.9551 from Table 7-16
The volume at standard conditions of 14.7 psia and 60°F:

$$= (50{,}000) \frac{(460+60)}{(460+110)} \frac{(15.7)}{(14.7)} = 48{,}717 \text{ cu ft/hr}$$

Free air capacity of valve at same setting:

$$= \frac{(48{,}717)\ (1.64)}{0.9551} = 83{,}651 \text{ SCFH}$$

Example 7-11: Storing Benzene in Cone Roof Tank

A 26,000 gallon outdoor vertical storage tank contains benzene. The tank is not insulated, but has a dike around it with a volume equal to one-and-a-half times the volume of the tank. Size 15′ diameter × 20′. Tank does not have weak roof. Temp. = 60°F, pressure = 14.7 psia.

Data: Flash point of benzene: < 100°F
Latent heat of vaporization: 169 BTU/lb
Tank size (nominal): 15′ diameter and 20′ high
Operating pressure: 1.5 oz/in.^{2}
Maximum allowable tank pressure: 3.5 oz/in.^{2}
Maximum allowable vacuum: 1.0 oz/in.^{2}
Relief valve settings:
(1) For pressure: 2.0 oz/in.^{2}
(2) For vacuum: 0.5 oz/in.^{2}
Filling rate: 350 gpm (max)
Draw-out rate (emptying): 250 gpm

A. Normal venting requirements
1. Pressure: filling
Rate = (350 gpm) (60) = 21,000 gal/hr

Code requires: 1200 cu ft of free air/hour for each 4200 gallons/hour maximum filling rate

$$\text{SCFH free air required} = \frac{(1200)}{(4200)}(21,000) = 6,000$$

2. Thermal outbreathing:
   Using Table 7-14, for 26,000 gallon capacity, required vent = 371 SCFH free air at 14.7 psia and 60°F. Interpolating, vent required for 26,000 gallon capacity.

3. Total pressure relief requirement

$$= 6000 + 371 = 6371 \text{ SCFH free air}$$

4. Total corrected pressure relief requirement

$$V_c' = \frac{V_c\,(1337)}{L\,M^{1/2}} = \frac{6371\,(1,337)}{(169)\,(78.11)^{1/2}}$$

= 5709 SCFH free air at 14.7 psia and 60°F when *benzene* in tank.

5. Read manufacturer's capacity tables or charts to select model and size of pressure relief, reading table to closest capacity, but toward larger size if in between. Consult manufacturer for final selection.

B. Vacuum (Inbreathing)

1. The draw-out on emptying is given as 250 gpm. Using the code requirements, free air:

[(560 cu ft air/4,200 gal/hr)] [(250) (60)] = 2000 cu ft free air/hr at 14.7 psia and 60°F

2. Thermal inbreathing:
   From code table (using vacuum), Table 7-14.

For tanks of 26,000 gallon capacity:

At 21,000 gal., SCFH air = 500
At 42,000 gal., SCFH air = 1,000

$$= 500 + \frac{(26,000 - 21,000)}{(42,000 - 21,000)}(1000 - 500)$$

= 619 SCFH free air @ 14.7 psia and 60°F

Total vacuum relief required:

= 2000 + 619 = 2619 SCFH free air

Select vacuum relief valve by referring to manufacturer's tables at vacuum relief setting of 0.5 oz/in.^{2} and 1.0 oz/in.^{2} maximum allowable vacuum for tank. Note, no correction is required as is shown for section A above, since air is flowing in, and not benzene.

C. Emergency venting requirements (bare tank)
Total external surface exposed to fire:

$$= \pi D\,(\text{height}) = \pi(15)\,(20) = 942.5 \text{ ft.}^2$$

Use entire shell area, since height < 30 ft

Wetted area: Entire 20 feet above grade. From Table 7-17 at 942.5 ft.^{2}, CFH free air relief:

900 ft.^{2} = 493,000 SCFH
1,000 ft.^{2} = 524,000 SCFH

For 942.5 ft.^{2}:

$$\text{Volume} = 493,000 + \frac{(942.5 - 900)}{(1000 - 900)}(542,000 - 493,000)$$

Venting Rate = 513,825 SCFH free air at 14.7 psia and 60°F

Corrected venting rate for benzene in tank:

$$V_c' = \frac{513,825\,(1,337)}{(169)\,(78.11)^{1/2}} = 459,945 \text{ SCFH}$$

For relief valve sizing, refer to manufacturer's tables at 2.0 oz/in.^{2} pressure setting and 3.5 oz/in.^{2} maximum allowable tank pressure.

Inbreathing requires vacuum relief for the tank. The usual conditions arise from liquid flow out of the tank or from condensation or contraction of the vapors by reduction in temperature of the tank contents caused by atmospheric changes (not by system mechanical refrigeration).

Table 7-17
Total Rate of Emergency Venting Required for Fire Exposure Vs. Wetted Surface Area
(wetted area vs. ft.^{3} of free air/hr, 14.7 psia, 60°F)

Wetted Area[a] (square feet)	Venting Requirement (cubic feet of free air[b] per hour)	Wetted Area[a] (square feet)	Venting Requirement (cubic feet of free air[b] per hour)
20	21,100	350	288,000
30	31,600	400	312,000
40	42,100	500	354,000
50	52,700	600	392,000
60	63,200	700	428,000
70	73,700	800	462,000
80	84,200	900	493,000
90	94,800	1000	524,000
100	105,000	1200	557,000
120	126,000	1400	587,000
140	147,000	1600	614,000
160	168,000	1800	639,000
180	190,000	2000	662,000
200	211,000	2400	704,000
250	239,000	2800	742,000
300	265,000	>2800[c]	—

NOTE: Interpolate for intermediate values. The total surface area does not include the area of ground plates but does include roof areas less than 30 feet above grade.

[a] The wetted area of the tank or storage vessel shall be calculated as follows: For spheres and spheroids, the wetted area is equal to 55 percent of the total surface area or the surface area to a height of 30 feet (9.14 meters), whichever is greater. For horizontal tanks, the wetted area is equal to 75 percent of the total surface area. For vertical tanks, the wetted area is equal to the total surface area of the shell within a maximum height of 30 feet (9.14 meters) above grade.

[b] At 14.7 pounds per square inch absolute (1.014 bar) and 60 F (15.56 C).

[c] For wetted surfaces larger than 2800 square feet (260.1 square meters), see 1.3.2.1, 1.3.2.2, and 1.3.2.4.

By permission: API-Std-2000, 3rd Ed., 1982, reaffirmed Dec. 1987, American Petroleum Institute [26].

In accordance with code [26] the inbreathing capacity is to be determined by the following:

1. Use maximum liquid flow out of tank (considered as oil by code) as equivalent to 560 cubic feet of free air per hour for each 100 barrels (4,200 gallons) per hour of maximum emptying rate. This applies to oils of any flash point. Also includes gravity flow conditions.
2. For thermal inbreathing use at least the cubic feet of free air per hour given in Table 7-14, column heading number 2. This also applies to oils or fluids of any flash point.

Convert the free air rates to the proper product in the tanks using the corrections outlined in a previous paragraph. Keep in mind that the manufacturer's rating tables are in free air; however, the actual process calculations provide flows in terms of the actual liquids at actual temperatures and pressures. It is important that the manufacturer be given the actual fluid conditions to ensure proper capacity rating.

Emergency venting of large tanks is usually associated with external fire conditions around the tanks. Under such conditions the venting requirements may be increased above the normal design levels previously reviewed. Emergency venting may be by [26]:

1. Weak roof-to-shell attachment for fixed roof tanks, as per API-Std-650 for *Welded Steel Tanks for Oil Storage.* The joint fails and excess pressure can be relieved (above the "normal" design provided). Such tanks do not require additional emergency vent equipment; however, it can be provided in order to prevent the roof seam failure with its attendant replacement/maintenance requirements. This type can *only* be used outside of a building, not confined.
2. For fixed roof tanks without the weak roof design, the required *total* venting at the time of emergency is determined as below, since normal and "thermal" venting and inbreathing can be ignored. The capacity of normal outbreathing equipment can be counted toward emergency requirements.
   (a) Tanks designed for 1 psig or below are to have the emergency venting rate determined from Table 7-17. For more than 2,800 ft.^{2} of exposed wetted internal surface, no increase in venting is required. All the code recommendations assume

that the stored liquid has the physical and thermal characteristics of hexane.

(b) Tanks designed for pressures over 1 psig (and up to 15 psig maximum as covered by API-Std-2000) are to have the emergency venting rate determined from Table 7-17; however, when the exposed surface area is above 2,800 ft.², the total rate of venting is to be calculated by (or see Figure 7-38):

$$V_c = 1{,}107\ A_w^{0.82} \tag{7-49}$$

where V_c = venting requirement, cubic feet of *free air* per hour, at 14.7 psia and 60°F.

A_w = wetted surface area, sq ft, see Table 7-17.

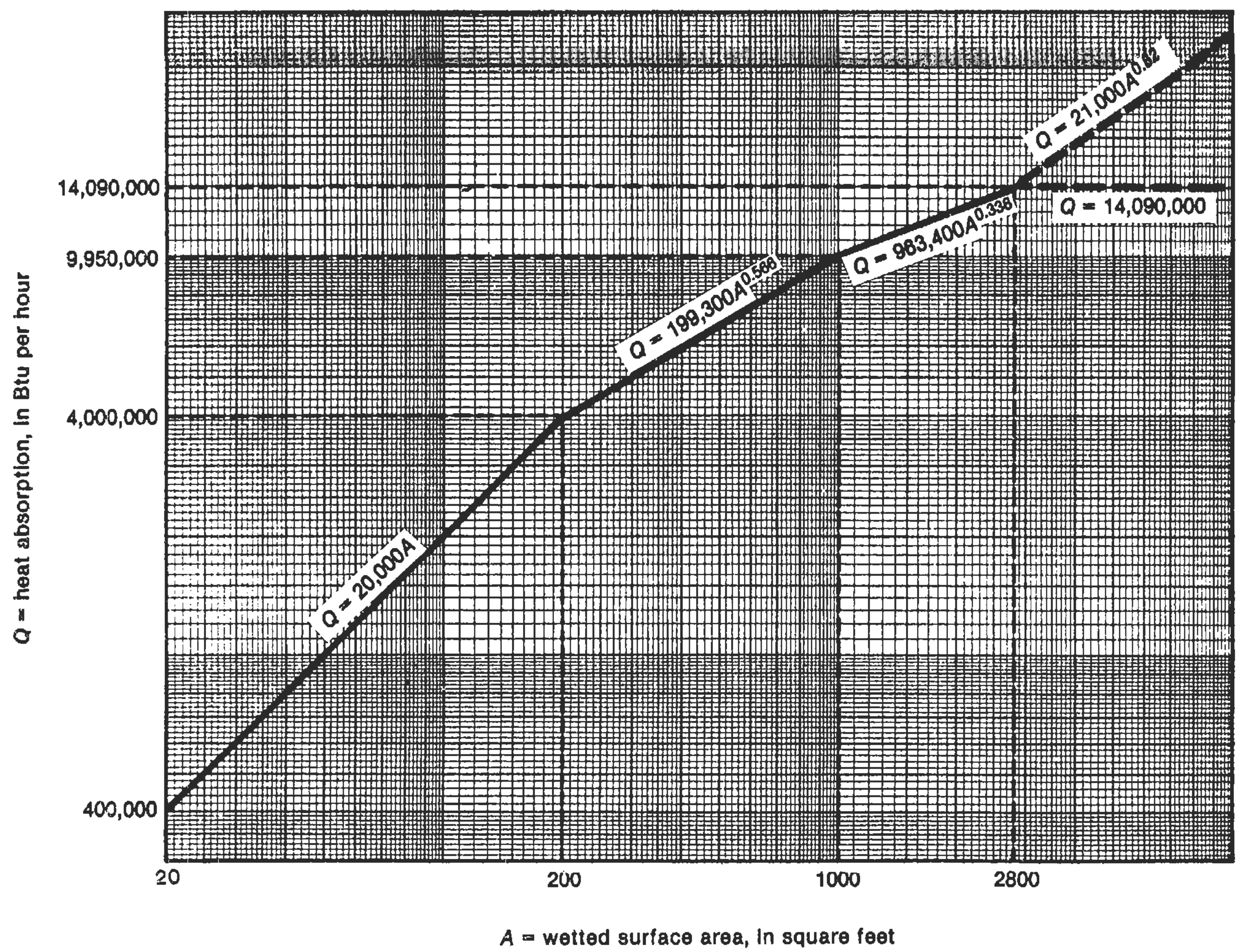
Note that the above formula is based on the one in API-RP-520 [10]:

$$Q = 21{,}000\ FA^{0.82} \tag{7-30}$$

where Q = total heat input, BTU/hr

This is again based on using hexane as the reference liquid. For *any* stored liquid; the cubic feet of free air for equipment rating is (with greater accuracy) [26]: (Also see Table 7-14)

$$V_c{}' = V_c \frac{(1{,}337)}{(L\sqrt{M})}\left(\sqrt{\frac{T}{520}}\right) \tag{7-48}$$



NOTE: Above 2800 square feet of wetted surface area, the total heat absorption is considered to remain constant for nonrefrigerated tanks below 1 pound per square inch gage. For nonrefrigerated tanks above 1 pound per square inch gage and for all refrigerated tanks, the total heat absorption continues to increase with wetted surface area. This is the reason why the curve splits above 2800 square feet.

Figure 7-38. Curve for determining requirements for emergency venting during fire exposure. Reprinted by permission, The American Petroleum Institute, API Std.-2000, 3rd Ed. 1987, *Venting Atmospheric and Low Pressure Storage Tanks.*

where V_c = cubic feet of free air/hr from Table 7-17 or Equation 7-49 for any fluid.
L = latent heat of vaporization of liquid, BTU/lb
M = molecular weight of liquid
T = temperature of the relief vapor, °R

The vent size may be determined based on the pressure that the tank can safely withstand [26].

The code [26] allows a credit or reduction of required emergency venting for specific conditions. The *final total* emergency venting requirements may be determined as:

Condition	*To Obtain Final Value, Multiply Previous V_c by
Drainage is away from tank	0.5
Tank has 1″ external insulation	0.3
Tank has 2″ external insulation	0.15
Tank has 4″ external insulation	0.075
Tank has water spray†	1.0

*Insulation values based on conductance of 4 BTU/hr-ft^2-°F/in. and insulation must be capable of withstanding fire hose streams without dislodging, and must be non-combustible.
†Water spray is not considered reliable, so no reduction is allowed.

Due to the importance of complying with all standards and regulations, and the necessary reduction in detail in such a summary as presented here, it is recommended that the ASME, API and NFPA standards and codes be consulted for final design detail and installation conformity.

Refer to API-2000 [26] for recommendation regarding installation requirements. Tank venting equipment capacities are expressed as free air/hour. For handling other fluids besides hexane or gasoline, use equation for V'_c (Equation 7-48) to convert to equivalent free air flow.

Emergency Vent Equipment

There are several types available as illustrated in Figures 7-39A, B, and C. See Figure 7-40 for specifications worksheet.

Refrigerated Above Ground and Below Ground Tanks [26]

The presence of low temperature conditions creates additional problems in design and selection of relieving devices, primarily because of the possibility of ice formation.

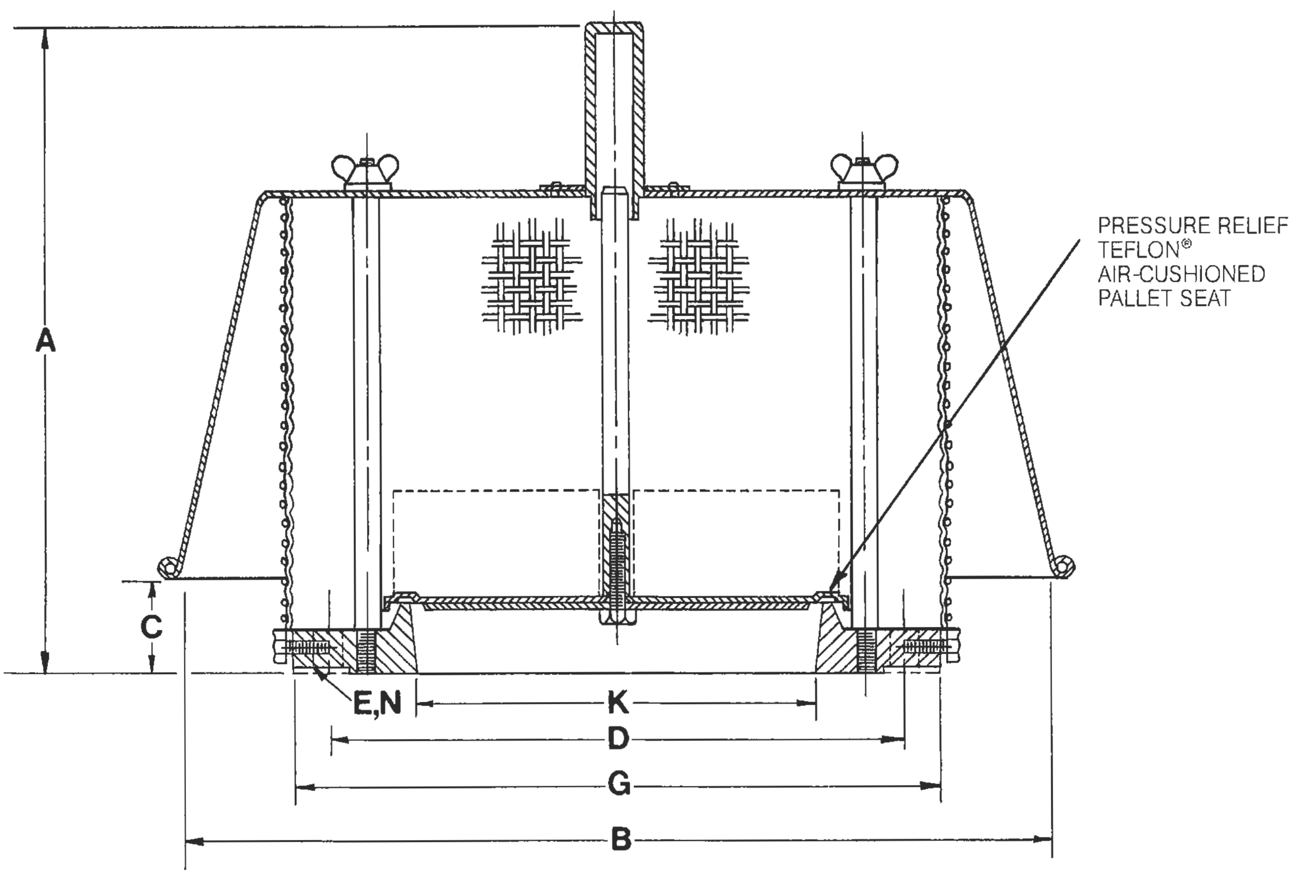


Figure 7-39A. Emergency pressure relief valve vent using guided weight. By permission, The Protectoseal Co.

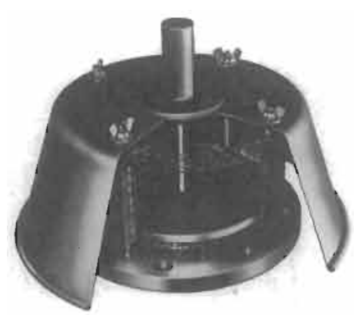


Figure 7-39A. Continued.

The capacity of pressure-vacuum devices for a tank are to be determined at 110% of their "start to discharge" pressure. Except for emergency pressure relieving because of fire, the capacity may be determined at 120% [26].

Normal conditions

At least the following conditions or some combination must be considered in establishing the normal pressure-vacuum relief capacity requirements [26]:

1. Loss of refrigeration
2. Liquid overfilling
3. Control valve failure
4. Vapor displacement during filling
5. Withdrawal of stored liquid at maximum rate (refer to section on non-refrigerated tanks)
6. Withdrawal of vapor at maximum compressor suction rate.
7. Drop of barometric pressure.

Under some circumstances it may not be appropriate to allow air to enter (inbreathe) into the tank, then the use of some other inert gas, such as nitrogen or natural gas, is acceptable on a pressure control basis; however, this cannot take the place of a vacuum relieving device to allow air to enter under a final emergency condition.

Emergency Venting for Fire Exposure

For refrigerated tanks the total venting requirement is the value determined from Table 7-17, multiplied by the environment factor F from Table 7-18. For tanks greater than 2,800 ft.^{2} exposed wetted surface, use the venting formula for V_c (Equation 7-49). Do not apply the factors from Table 7-18.

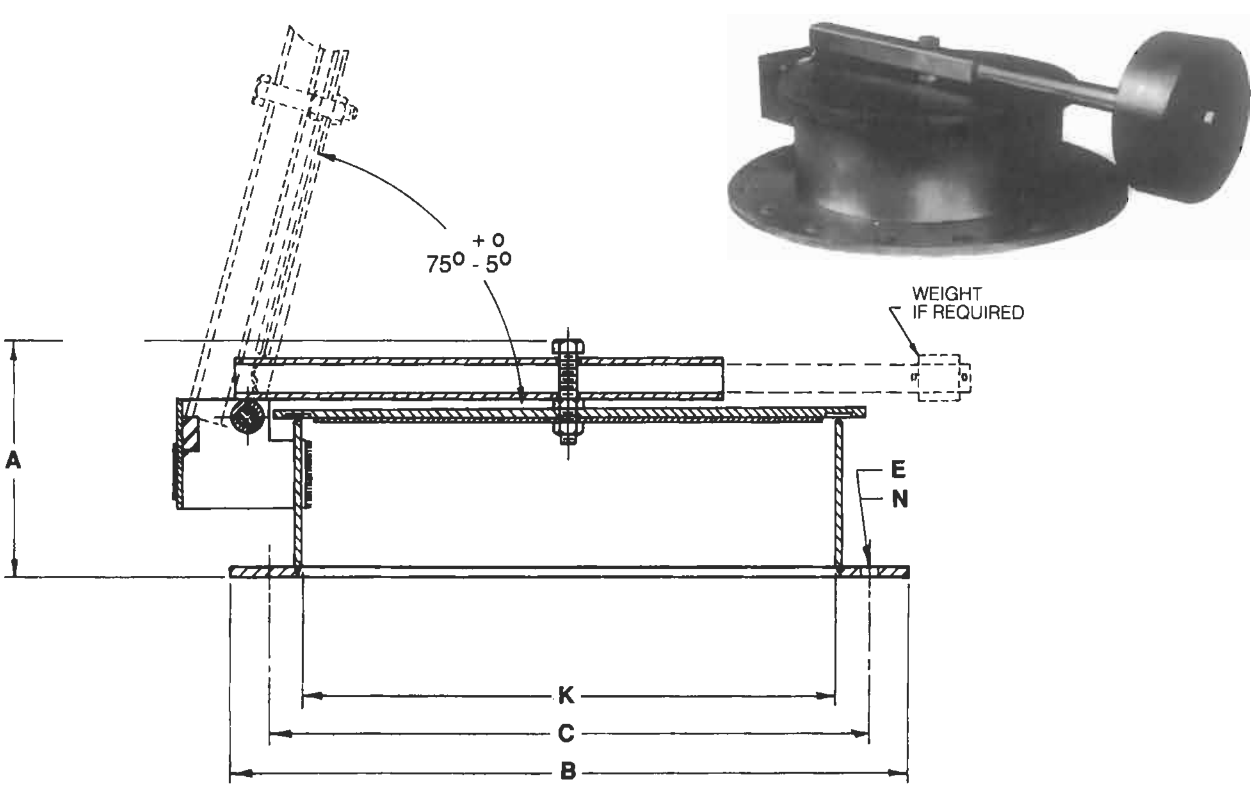


Figure 7-39B. Hinged emergency pressure manhole cover vent. By permission, The Protectoseal Co.

SIDE VIEW

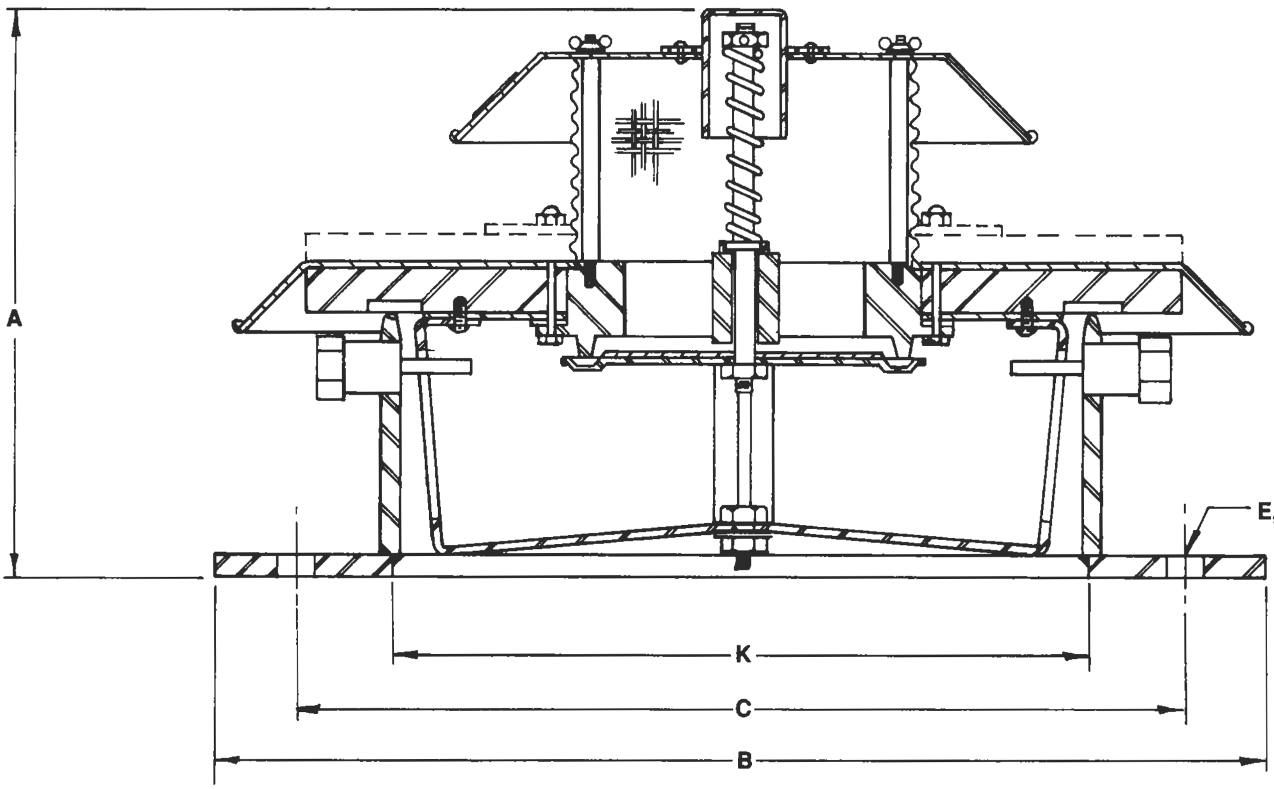


Figure 7-39C. Combination pressure/vacuum manhole cover vent. By permission, The Protectoseal Co.

Figure 7-39. Continued.

Example 7-12: Venting and Breathing in Oil Storage Tank

A 675 barrel tank (15.5 ft. diameter × 20 ft. tall) is used for oil storage. The rate for pumping oil in or out is 978 gpm (maximum) or 1400 bbl hr. Flash point of oil above 100°F.

Using API Guide for Tank Venting [26]

A. For pressure or normal outbreathing:

Required movement = 600 cu ft/hr/100 bbl of filling rate for oils with flash point above 100°F.

Required capacity = (600) (1400/100) = 8400 cu ft/hr

*Plus thermal outbreathing =

.60 (675 bbl) = 405

Total = 8805 cu ft/hr

Flame Arrestors

Flame arrestors, Figure 7-41, are valuable and necessary when the danger of flammable fumes exists and a tank is venting directly to atmosphere, with no conservation vent. Here a flame arrestor should be mounted on the open vent of the tank to guard against flashback into the tank of flames, lightning ignition, etc. of the exiting fumes.

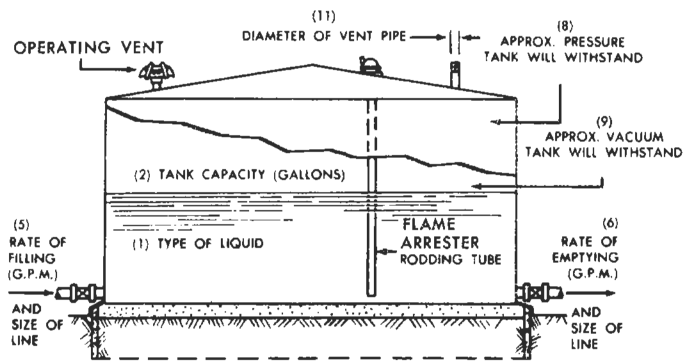
Some conservation vents (pressure-vacuum) have built-in flame arrestors on a single compact unit mounted on the tank vent. Refer to API Safety Data 2210, *Flame Arrestors for Tank Vents.*

A special study [74] was commissioned by the American Petroleum Institute (API) entitled "Mitigation of Explosion Hazards of Marine Vapor Control Systems." The report examines the effects of deflagrations and detonations in pipes in the region of detonation flame arrestors. The primary objective was "to resolve potential

OPERATING VENTS

FILL IN THIS INFORMATION FOR NEW VENT INSTALLATIONS:

1. Type of Liquid____________________
   Specific Gravity__________Flash Point______°F.
2. Tank Capacity in Gallons____________________
3. Tank is Above Ground ☐ or Below Ground ☐
4. Tank is Vertical ☐ or Horizontal ☐ Tank Dia.__________ft.
5. Rate of Filling (G.P.M.)__________Size Line__________
6. Rate of Emptying (G.P.M.)__________Size Line__________
7. Approximate operating pressure of tank__________
8. Approximate pressure tank will withstand__________
9. Approximate vacuum tank will withstand__________
10. Is tank inerted?__________If yes, approximate
    blanketing pressure____________________

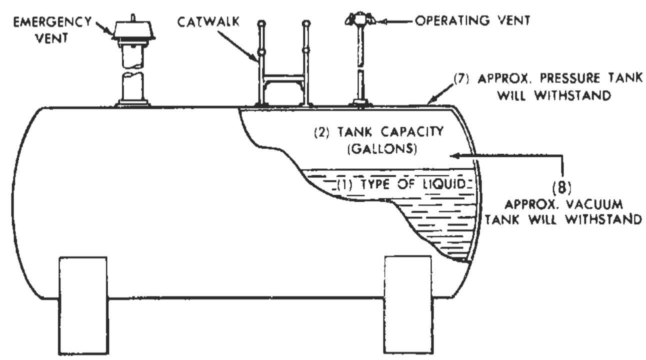


NOTES:
(Specify any temperature conditions above or below ambient and/or special materials of construction required)

EMERGENCY VENTS

FILL IN THIS INFORMATION FOR NEW VENT INSTALLATION:

1. Type of Liquid__________
   Specific Gravity__________ Flash Point__________°F
   Molecular Weight__________ Latent Heat of
   $L\sqrt{M}$ Value__________ Vaporization ________BTU/lb.
2. Tank Capacity in: Gallons__________or Barrels__________
3. Tank is Above Ground ☐ or Below Ground ☐
4. Tank is Vertical ☐ or Horizontal ☐
   Tank Diameter________Height________or Length________
5. Approximate operating pressure of tank__________
6. Valve setting on Operating vent__________
7. Approximate pressure tank will withstand__________
8. Approximate vacuum tank will withstand__________
9. Is tank inerted?__________If yes, what is
   approximate blanketing pressure?__________



MODIFICATIONS (check, where applicable):

	YES*	NO*
10. Tank drains to a remote area	☐	☐
11. Tank equipped with water spray	☐	☐
12. Tank is insulated	☐	☐

NOTES:
(Specify any temperature conditions above or below ambient and/or special materials of construction required)

Figure 7-40. Storage and process tank specifications work sheet. By permission, The Protectoseal Co.

operational hazards that relate to the use of detonation flame arrestors in marine vapor control systems." The work examined deflagrations, deflagrations transitioning to detonations, effects of overpressure (explosion pressure versus the initial pressure at ignition), stable detonations (with relatively stable speed, peak and side-on overpressure), and overdriven detonations (unstable, occur during transition from deflagration to detonation). In propane, typical stable detonation speed is 1810 meters/second (5950 ft/sec), and a typical peak side-on overpressure is about 18.3 times the initial pressure. The side-on overpressure is measured in the side wall of

Table 7-18
Environmental Factors for Refrigerated Tanks

Datum	F Factor
Bare metal vessel	1.0
Insulation thickness[a]	
6 inches (152 millimeters)	0.05[b]
8 inches (203 millimeters)	0.037[b]
10 inches (254 millimeters)	0.03[b]
12 inches (305 millimeters) or more[c]	0.025[b]
Concrete thickness	[d]
Water-application facilities[e]	1.0
Depressuring and emptying facilities[f]	1.0
Underground storage	0
Earth-covered storage above grade	0.03

[a]To take credit for reduced heat input, the insulation shall resist dislodgment by a fire-hose stream, shall be noncombustible, and shall not decompose at temperatures up to 1000 F. If the insulation does not meet these criteria, the F factor for a bare vessel shall be used.
[b]These F factors are based on an arbitrary thermal conductivity of 4 British thermal units per hour per square foot per (degree F per inch of thickness) and a temperature differential of 1600 F when using a heat input value of 21,000 British thermal units per hour per square foot in accordance with the conditions assumed in API Recommended Practice 520. When these conditions do not exist, engineering judgment should be exercised either in selecting a higher F factor or in providing other means of protecting the tank from fire exposure.
[c]The insulation credit is arbitrarily limited to the F factor shown for 12 inches of insulation, even though greater thicknesses may be used. More credit, if taken, would result in a relieving device that would be impractically small but that might be used if warranted by design considerations.
[d]Twice the F factor for an equivalent thickness of insulation.
[e]Under ideal conditions, water films covering the metal surfaces can absorb substantially all incident radiation. However, the reliability of effective water application depends on many factors. Freezing temperatures, high winds, system clogging, unreliability of the water supply, and adverse tank surface conditions are a few factors that may prevent adequate or uniform water coverage. Because of these uncertainties, the use of an F factor other than 1.0 for water application is generally discouraged.
[f]Depressuring devices may be used, but no credit for their use shall be allowed in sizing safety valves for fire exposure.

F = environmental factor from Table 4.
A = wetted surface area, in square feet (see Table 3, Footnote a).

NOTE: The formula above is based on

$$Q = 21{,}000\, A^{0.82}$$

as given in API Recommended Practice 520. The total heat absorbed, Q, is in British thermal units per hour. The constant 1107 is derived by converting the heat input value of 21,000 British thermal units per hour per square foot to standard cubic feet of free air by using the latent heat of vaporization at 60 F and the molecular weight of hexane. When the molecular weight, latent heat of vaporization, and temperature of relief conditions for refrigerated hydrocarbons are substituted in the formula based on hexane, the venting requirements are about equal to the values for hexane. Hexane has therefore been used as a basis for simplification and standardization (see Appendix B for additional information about the derivation of the formula).

By permission: API Std-2000, 3rd Ed. (1982), reaffirmed Dec. 1987[26].

straight pipe flush with the pipe wall. Reflected side-on pressure is measured perpendicular, i.e., facing the oncoming flow.

Pilot Operated Vent Values

The Code [26] allows pilot operated vent control valves provided the vent valve can operate automatically in case the pilot valves/system failed.

Explosions

The three basic types of explosions to be concerned about in the chemical and petrochemical environment are combustion explosions (deflagrations), detonation explosions, and BLEVEs or boiling-liquid expanding vapor explosions [38].

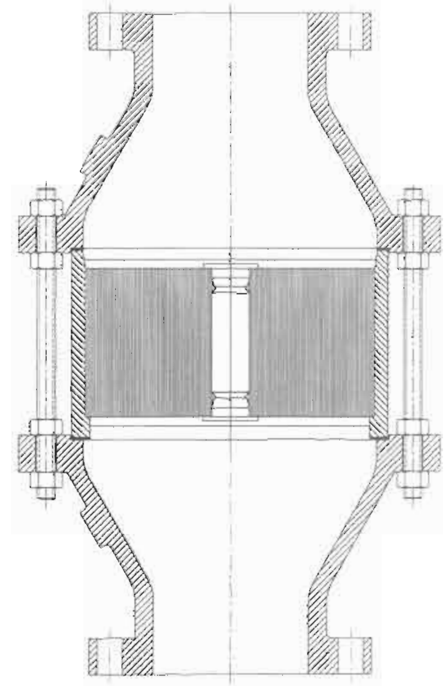
Other than reactive metals explosions, which do not truly fall in the types noted above, the two main categories of explosions are flammable gases, liquids/vapors, and dusts. Because their sources are different, they cannot be treated in the same manner for discussion.

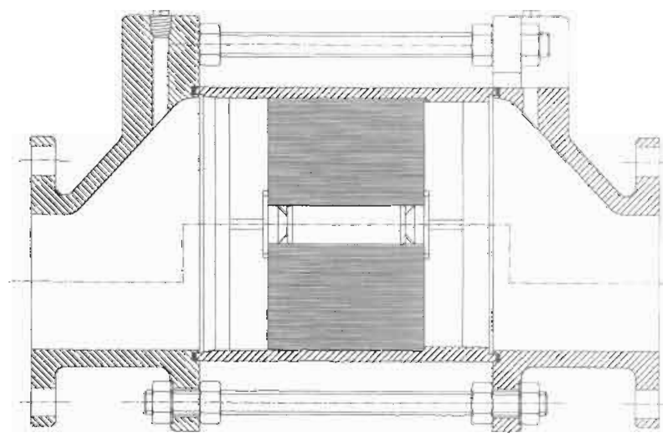
Confined Explosions

A confined explosion occurs in a contained vessel, building piping network, or other confined situation. A confined explosion has different characteristics than an unconfined explosion [40]. These explosions may be deflagrations or detonations with the detonation being much more destructive due to the higher and more rapidly moving pressure wave generated. Schwab [34] states unequivocally that a vessel containing flammable vapor as a mixture when ignited with the resultant pressure buildup, will explode. If the vessel does not rupture, but contains the deflagration or detonation, there is no explosion because the requirement for mechanical work has not been met.

Combustion explosions are explosions resulting from the uncontrolled rapid mixing and reaction of a flammable vapor from a flammable liquid with air (or oxygen) ignited from an ignition source such as flame, heat, electric spark, or static discharge. The combustion is extremely rapid with a flame propagation rate of about 7 feet per second, with the evolution of heat, light, and an increase in pressure [38]. The violence of the explosion depends on the rate at which the energy is released.

A deflagration is a slow burning exothermic reaction similar to the combustion explosion, but which propagates from the burning gases into the unreacted material at a velocity that is less than the speed of sound in the unreacted material. Most (not all) explosions are deflagrations.





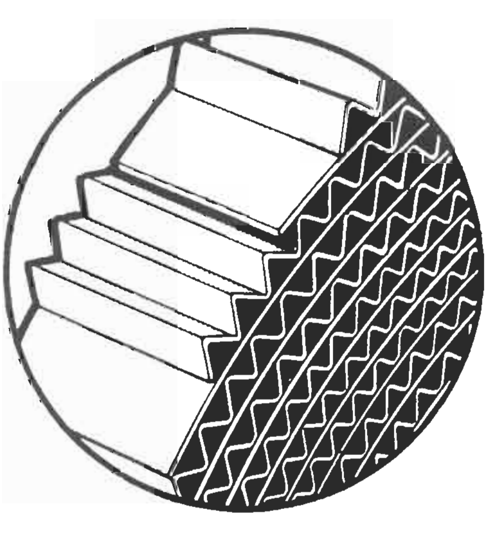


Figure 7-41. Typical individual flame arresters for tank or pipe mounting where conservation breathing vents may or may not be required, sizes 2 inch–60 inch. By permission, Groth Equipment Co., Tank Protection Division.

Detonation explosions are similar to combustion explosions and are exothermic reactions that proceed into the unreacted material at a velocity much greater than the speed of sound in an unreacted material and are accompanied by a flame front shock wave in the material followed closely by a combustion wave that releases the energy and sustains the shock wave at extremely high pressure [39] [40]. In hydrocarbons, the velocity can reach 6,000–9,000 ft/sec.

A detonation generates much greater pressure and is therefore more destructive than a deflagration. For example, in a closed vessel, a hydrocarbon mixture detonation with air may generate 2.5 times (294 psi) a similar mixture in a deflagration. A deflagration may turn into a detonation, especially if traveling down a long pipe. A shock wave is generated by the expansion of gases created by the reaction of one flammable gas/liquid with an oxidant, such as air or pure oxygen, or an oxidizing material, or by pure thermal effects [29, 38, 40, 41]. Table 7-19 and Figure 7-42A illustrate the velocities of travel of a detonating shock wave. Similar data can/has been developed for many other industrial compounds (see, for example, Ref. [32, 34, 36, 41, 43, 44]). Figure 7-42B compares selected flammable and detonability limits in air. The lean end/rich end of the flammability data does not support a detonation. Note the extreme differences in shock wave propagation velocities in Table 7-19.

FLAMMABILITY

There are several characteristics of physical materials that describe or define the extent or even the possibility of a material being flammable or whether it will support combustion.

There are probably two known organized examination and evaluation guides and/or procedures for potential fires and explosions. These are (1) Dow's Fire and Explosion Index, Hazard Classification Guide [66] (This is too involved and extensive to present here, but every serious safety design and researcher is urged to study this guide.), and (2) Design Institute for Emergency Relief Systems, American Institute of Chemical Engineers [51, 67]. This industry supported program is targeted to understanding the complexities of venting relief devices, particularly from runaway reactors.

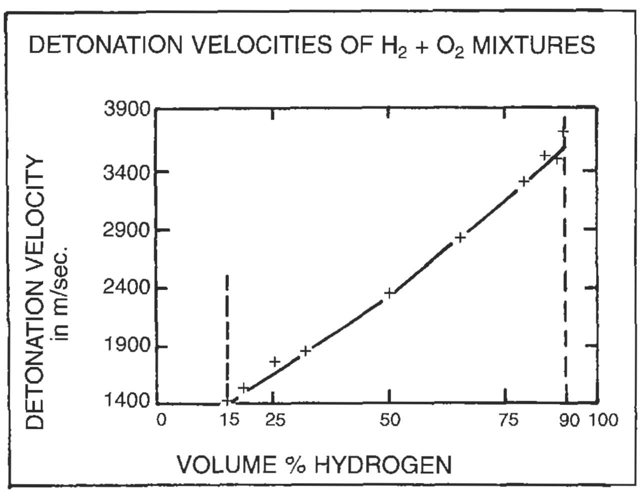


Figure 7-42A. Detonation velocities for hydrogen/oxygen mixtures. Note detonation range compared to flammability range of 4% to 95%. By permission, Ref. [41]., Stull, The Dow Chemical Co. and The American Institute of Chemical Engineers Monograph No. 10, Vol. 73 (1977).

Terminology

Flash point of a flammable liquid: The lowest temperature at which the liquid gives off enough vapors to form a flammable mixture with air (or pure oxygen, a special case) at or near the surface of the liquid or within its confined container. Some hazardous liquids have flash points at or below ordinary room temperatures and normally are covered by a layer of flammable vapors that will ignite immediately if a source of ignition is brought in contact [32]. Flash points are measured by "open cup" and "closed cup" methods. The open cup data is applicable to liquid in open containers and in open pools and usually somewhat higher temperatures than the closed cup. Refer to

Table 7-19
Comparison Data for Selected Hydrocarbon-Air Mixtures for Deflagrations and Detonations

	Deflagration Limits %	Deflagration Velocity M/Sec	Detonation Limits %	Detonation Velocity M/Sec
CH_4 - Air	5.3–15	0.37	?	1540
C_3H_8 - Air	2.2–9.5	0.40	3–7	1730
C_2H_2 - Air	2.5–80	1.31	4.2–50	1870

By permission, Stull [41] Dow Chemical Co. and American Institute of Chemical Engineers, *Monograph Series No. 10,* V. 73 (1977).

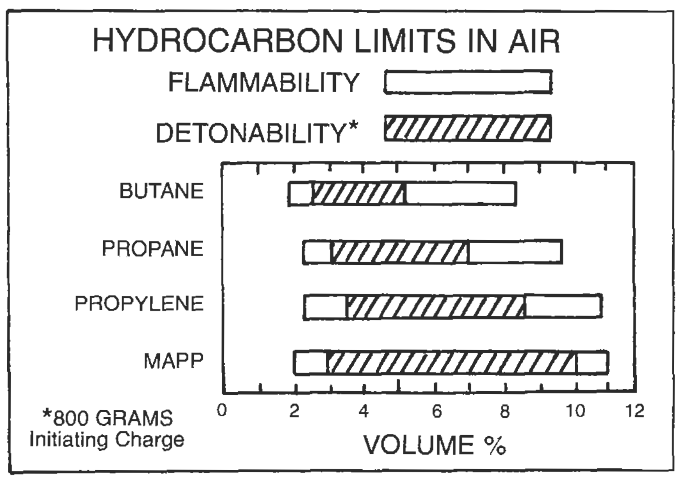


Figure 7-42B. Comparison of flammability and detonation range limits in air for selected hydrocarbons and mixtures. By permission, Ref. [41], Stull, The Dow Chemical Co. and The American Institute of Chemical Engineers, Monograph No. 10, Vol. 73 (1977).

American Society for Testing Materials (ASTM) E-502, D-56, D-92, D-93, D-1310, D-3278.

The flammable liquid itself does not burn; only the vapors emitted from the liquid burn. The vaporization of a liquid depends on its temperature and corresponding vapor pressure and increases as the temperature of the liquid increases. Thus, the warmer the liquid, the more potentially hazardous it becomes.

Fire point: The lowest temperature at which a liquid in an open container will give off enough vapors to continue to burn when once ignited [32]. This temperature is generally somewhat above the open-cup flash point.

Ignition temperature: The minimum temperature to which a material must be heated for it to ignite. Once an ignition has occurred it will continue to burn until all the available fuel or oxidant has been consumed or until the flame is extinguished by cooling or by some other means [34].

Autoignition temperature: The lowest temperature of a material required to initiate or cause self-sustained combustion in the absence of a spark or flame. This temperature can vary, depending on the substance, its size, and the shape of the igniting surface or container and other factors [32].

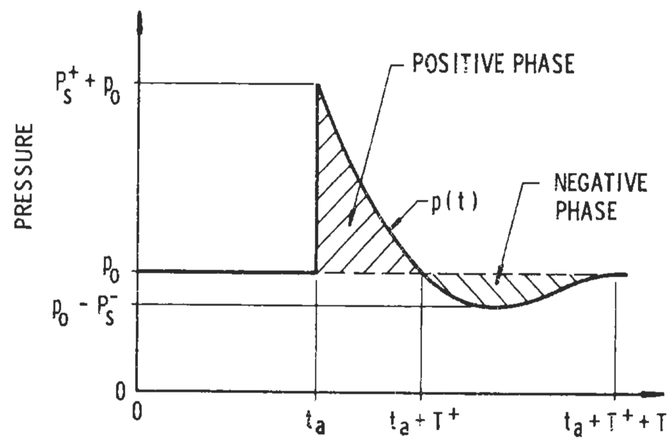
Spontaneous heating: Some flammable liquids combine readily with oxygen in the air at ordinary temperatures and give off heat. When the heat is generated faster than it can be dissipated, the temperature rises and ignition of the mixture may occur, as with liquids on waste or rubbish or other materials [32].

Flammability limits or explosive range: The entire range of concentrations of a mixture of flammable vapor or gas in air (expressed as volume percent) over which a flash will occur or a flame will travel if the mixture is ignited. Gases and vapors in air have both deflagration and detonation limits and are often the same as flammability limits. The limits of detonability can be different and are dependent on the system conditions. Under some circumstances or some mixtures the deflagration pressure developed by the shock waves can be eight times the system pressure from stoichiometric fuel-air mixtures. For fuel-oxygen mixtures, the pressure increases may be as much as 20 times. The "side-on" pressures (peak pressures) from the blast (shock) wave rise almost instantaneously on surfaces orientated parallel to the direction of the wave (see Figure 7-43, Baker [42], and Figure 7-44). For gas detonations, the pressures are about two times those for deflagrations. The reflected pressure can be another factor of 2 or greater. Therefore, for a detonating fuel/oxygen mixture, the pressure rise may be a 40-fold increase [34]. There will be no flame (except possibly a "cold flame," not examined here) or explosion when the concentrations are outside these flammability limits [32]. (See Sample listings of flammability data Tables 7-20 and 7-21. Also see in Figures 7-45, 7-46, and 7-47. The flammability limits are designated.

1. *Lower Flammability (Explosive) Limits (LEL or LFL):* The lowest percentage concentration at which a flash or flame can develop and propagate from the source of ignition when in contact with a source of ignition in a combustible material.

2. *Upper Flammability (Explosive) Limits. (UEL or UFL:* The highest percentage concentration at which a flash or flame can develop and propagate flame away from the source of ignition when in contact with a source of ignition in a combustible material. See Tables 7-20 and 7-21 [34] for common flammable compounds.

Figure 7-46 illustrates a typical relationship of limits of flammability and ignitibility for a methane air mixture. Note that energy required to ignite a flammable mixture (within its LEL and UEL) varies with the composition, and that a 0.2 millijoule (mj) spark is inadequate to ignite even a stoichiometric mixture at atmospheric pressure at 26°C, while 1-mj spark can ignite any



The gauge records ambient pressure p_0. At arrival time t_a, the pressure rises quite abruptly (discontinuously, in an ideal wave) to a peak value $P_s^+ + p_0$. The pressure then decays to ambient in total time $t_a + T^+$, drops to a partial vacuum of amplitude P_s^-, and eventually returns to p_0 in total time $t_a + T^+ + T^-$. The quantity P_s^+ is usually termed the peak side-on overpressure, or merely the peak overpressure. The portion of the time history above initial ambient pressure is called the positive phase, of duration T^+. That portion below p_0, of amplitude P_s^- and duration T^-, is called the negative phase.

Figure 7-43. Ideal blast wave from gaseous explosion in air. By permission, Wilfred Baker Engineering, Inc., *Explosions in Air,* 2nd printing (1983), Wilfred E. Baker, San Antonio, Texas, USA [42].

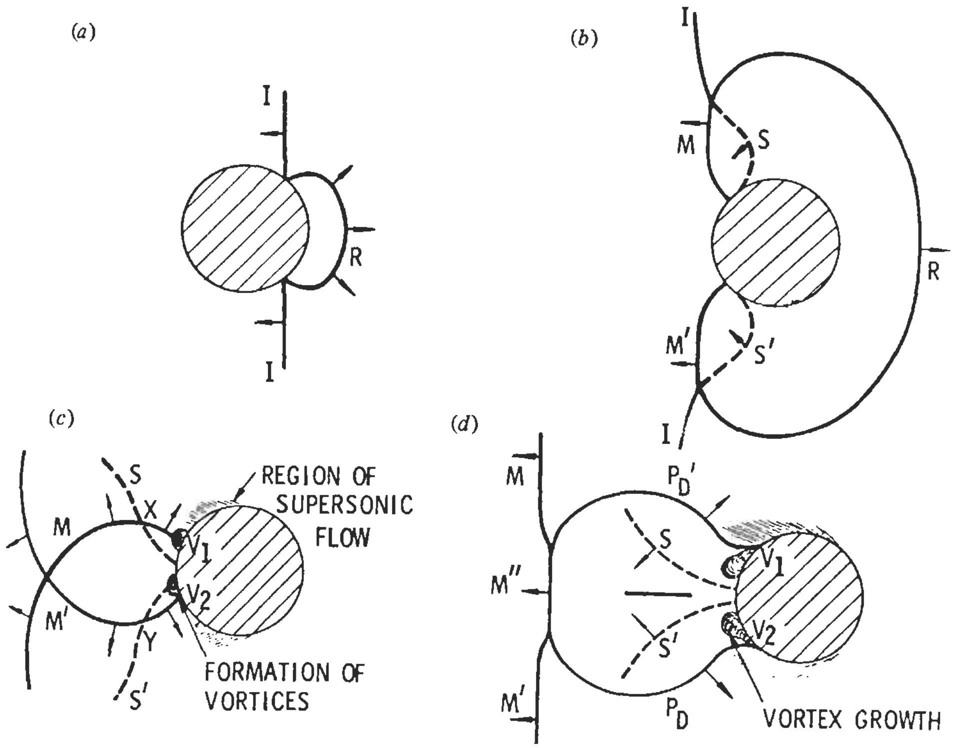


Figure 7-44 shows the sequence of events involved in diffraction of a blast wave about a circular cylinder (Bishop and Rowe 1967). In these figures the shock fronts are shown as thick lines and their direction of movement by arrows normal to the shock front. In Figure 1.13*a*, the incident shock *I* and reflected shock *R* are joined to the cylinder surface by a Mach stem *M*. *R* is now much weaker and is omitted in succeeding figures.

In this shock configuration a slipstream *S* has been formed. This slipstream is a line dividing flows of differing densities but of the same pressure. When a Mach stem is formed on a plane surface, the slipstream extends upstream, slanting down to meet the surface. In the present case, however, the increased flow near the cylinder surface has caused the foot of the slipstream to move nearer to the foot of *M*. The slipstream therefore presents a curved appearance. In Figure 1.13*c* the feet of the Mach stems have passed though each other and are moving on a second circuit of the cylinder. The slipstreams have been swept nearer the rear of the cylinder and intersect with the diffracted parts of the Mach stems *x* and *y*. The commencement of two vortices is indicated at V_1 and V_2. These are probably induced by the back pressure behind the shocks *x* and *y* interacting with the boundary layer flow at the surface of the cylinder. The shaded portion is due to a localized region of supersonic flow. In Figure 1.13*d* the Mach stems *M* and *M'* have moved some way downstream of the cylinder. A Mach stem *M''* joins the free-air parts of these Mach stems with the diffracted parts P_D and P'_D, which terminate on the cylinder surface. The growth of the vortices is apparent in this figure. In Figure 1.14*a–b*, the foot of P_D has moved further around the cylinder upstream. Notice that the point of flow separation has followed this shock.

Figure 7-44. Interaction of a shock wave with a cylinder. (Source, Bishop and Rowe, 1967). By permission, Wilfred Baker Engineering, Inc., *Explosions in Air,* 2nd printing (1983), Wilfred E. Baker, San Antonio, Texas, USA [42].

mixture between 6 and 11.5 vol percent methane. These limits are known as limits of ignitibility, indicating the igniting ability of the energy source. Limit mixtures that are essentially independent of the ignition source strength and give a measure of the flame to propagate away from the ignition source are defined as limits of flammability [43]. Considerably more spark energy is required to establish limits of flammability, and more energy is usually required to establish the upper rather than the lower limit.

Mixtures of Flammable Gases

Composite Flammability Mixtures

Le Chatelier's Rule allows the calculation of the lower flammability (explosibility) limits for flammable mixtures:

$$\text{Mixture composite LEL} = \frac{100\ \text{volume/volume}}{v_1/L_1 + v_2/L_2 + v_3/L_3 + \dots} \qquad (7-54)$$

(text continued on page 491)

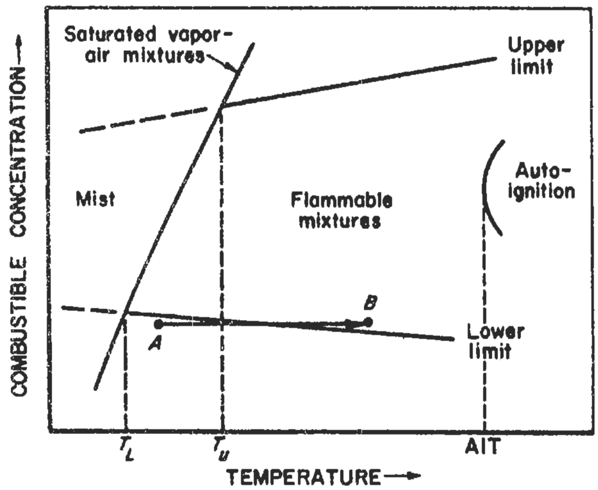


Figure 7-45. Effect of temperature on limits of flammability of a combustible vapor in air at constant initial pressure. By permission, U.S. Bureau of Mines, Bulletin 627 [43].

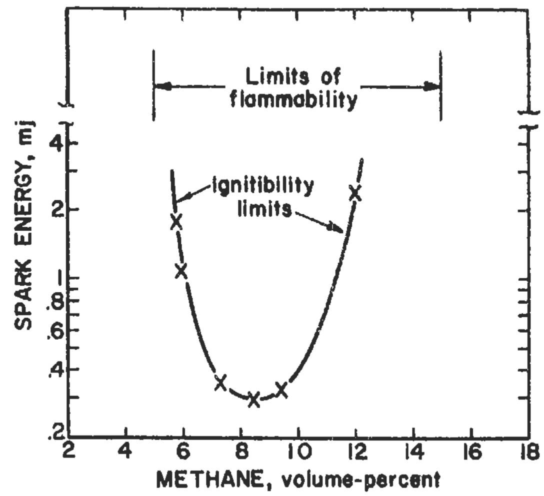


Figure 7-46. Ignitibility curve and limits of flammability for methane-air mixtures at atmospheric pressure and 26°C. By permission, U.S. Bureau of Mines, Bulletin 627 [43].

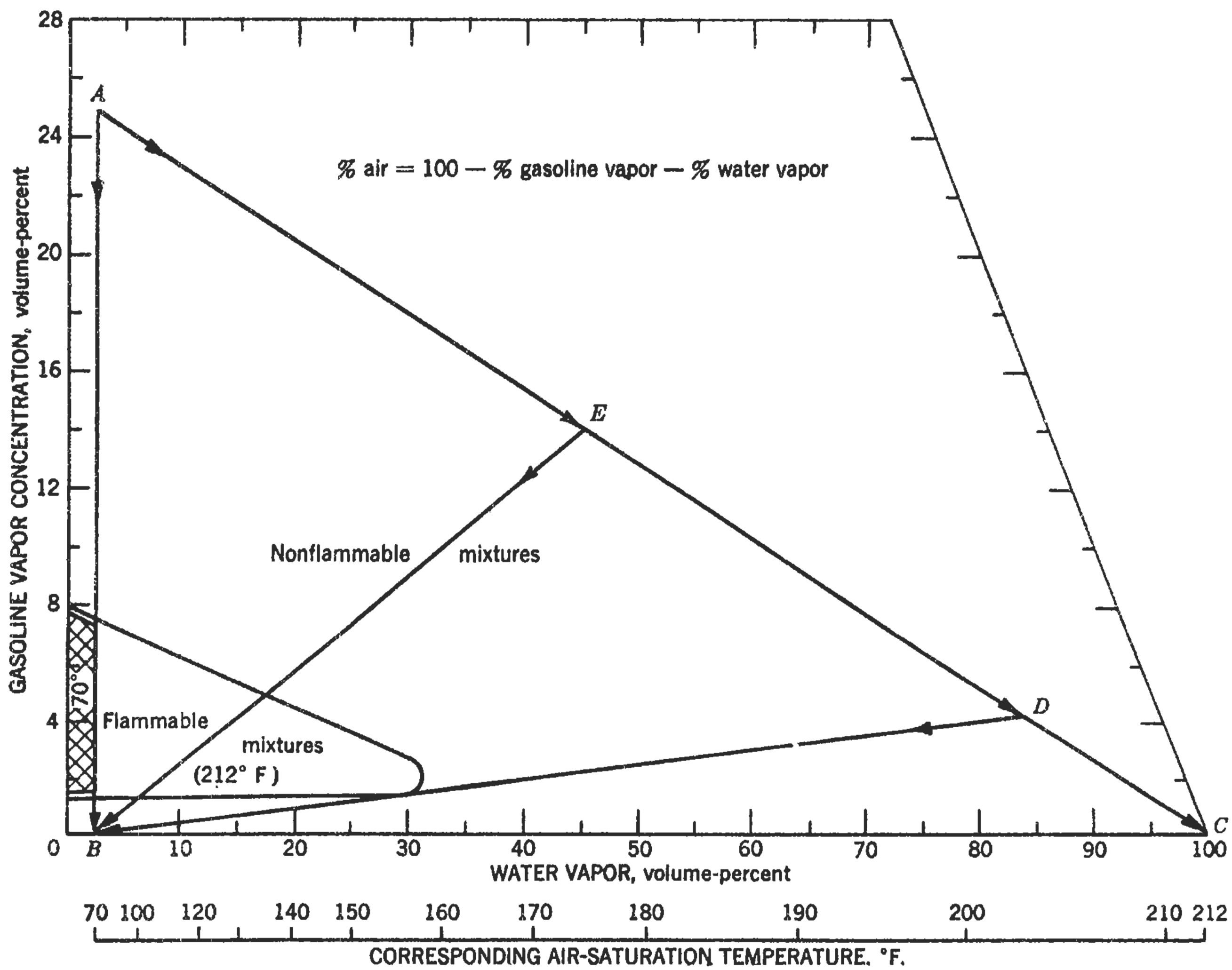


Figure 7-47. Flammability diagram for the system gasoline vapor-water vapor-air at 70°F and at 212°F and atmospheric pressure. By permission, U.S. Bureau of Mines, Bulletin 627 [43].

Table 7-20
Properties of Flammable Liquids, Gases, and Solids

Name	Synonym	Formula	Flash point, °F	
			Closed cup	Open cup
Ethyleneimine	Dimethylenimine	$NH(CH_2)_2$	12	...
Ethylene oxide	1,2-Epoxyethane	CH_2OCH_2	Gas	−4
Ethylethanolamine-1	Ethylaminoethanol	$C_2H_5NHCH_2CH_2OH$	...	160
Ethyl ether	See Diethyl ether			
Ethyl formate	Ethyl methanoate	$HCOOC_2H_5$	−4	10
Ethyl hexaldehyde	...	$C_4H_9CH(C_2H_5)CHO$	...	125
2-Ethylhexanediol-1,3	Ethylhexylene glycol	$C_3H_7CH(OH)CH(C_2H_5)CH_2OH$	...	260
2-Ethylhexanoic acid	Octoic acid	$CH_3(CH_2)_3CH(C_2H_5)CH_2OH$	...	260
Ethyl hexanol	2-Ethyl hexyl alcohol	$C_4H_9CH(C_2H_5)CH_2OH$	...	185
Ethylhexyl acetate	Octyl acetate	$CH_3COOCH_2CH(C_2H_5)C_4H_9$	180	190
Ethyl lactate	Ethyl 2-hydroxy propanoate	$CH_3CHOHCOOC_2H_5$	115	158
Ethyl mercaptan	Ethanethiol	C_2H_5SH	<80	...
Ethylmonobromoacetate	...	$BrCH_2COOC_2H_5$	118	...
Ethylmonochloroacetate	...	$CClH_2COOC_2H_5$	100	...
Ethyl morpholine	...	$CH_2CH_2OCH_2CH_2NCH_2CH_3$	...	90
Ethyl nitrate	Nitric ether	$C_2H_5ONO_2$	50	50
Ethyl nitrite	Nitrous ether	C_2H_5ONO	−31	...
Ethyl oxalate	Diethyl oxalate	$(COOC_2H_5)_2$	168	...
Ethyl phenyl ethanolamine	...	$C_6H_5NC_2H_5CH_2CH_2OH$	...	270
Ethyl phthallyl ethyl glycolate	...	$C_2H_5OCOC_6H_4COOCH_2COOC_2H_5$	365	385
Ethyl propionate	Propionic ether	$C_2H_5COOC_2H_5$	54	...
2-Ethyl-3-propylacrolein	2-Ethylhexenal	$C_3H_7CH:C(C_2H_5)CHO$	...	155
Ethyl *n*-propyl ether	1-Ethoxypropane	$C_2H_5OC_3H_7$	...	...
Ethyl silicate	Ethyl orthosilicate	$(C_2H_5)_4SiO_4$	...	125
Ethyl *p*-toluene sulfonamide	...	$C_7H_7SO_2NHC_2H_5$	260	380
Ethyl *p*-toluene sulfonate	...	$C_7H_7SO_3C_2H_5$	316	...
Fish oil	...	...	420	...
Fluorine	...	F_2	Gas	Gas
Formal	See Methylal			
Formaldehyde gas	Methanal	$HCOH$	Gas	Gas
Formaldehyde–37% in water	Formalin	...	130	200
Formic acid	Methanoic acid	$HCOOH$	156	...
Fuwl oil No. 1	Range oil; kerosene	...	114–185	...
Fuel oil No. 1-D	Diesel fuel, light	...	>100	...
Fuel oil No. 2	Domestic fuel oil	...	126–230	...
Fuel oil No. 2-D	Diesel fuel, medium	...	>100	...
Fuel oil No. 4	Light industrial fuel	...	154–240	...
Fuel oil No. 5	Medium industrial fuel	...	130–310	...
Fuel oil No. 6	Heavy industrial fuel; Bunker C.	...	150–430	...
Furan	Furfurane; oxole	$HC:CHCH:CHO$	−32	...
Furfural	Fural; 2-furaldehyde	C_4H_3OCHO	140	155
Furfuryl alcohol	Furfuryl carbinol	$C_4H_3OCH_2OH$	...	167
Furfurylamine	2-Furanmethylamine	$C_4H_3OCH_2NH_2$	...	99
Gas, blast furnace	...	...	Gas	Gas
Gas, coal gas	...	...	Gas	Gas
Gas, coke oven	...	...	Gas	Gas
Gas, manufactured, 540 Btu	...	...	Gas	Gas
Gas, manufactured, 815 Btu	...	...	Gas	Gas
Gas, natural, 1035 Btu	...	...	Gas	Gas
Gas, oil gas	...	...	Gas	Gas
Gas, producer	...	...	Gas	Gas
Gas, water	...	...	Gas	Gas
Gas, water-carbureted	...	...	Gas	Gas
Gas oil	(65 Diesel Index)	...	150–165	...
Gasoline, automotive—premium	...	...	−50±	...
Gasoline, automotive—regular	...	...	−50±	...
Gasoline, aviation—commercial	...	...	−50±	...
Gasoline, aviation—military	...	...	−50±	...
Glycerol	Glycerin	$HOCH_2CHOHCH_2OH$	320	350
Glyceryl triacetate	Triacetin	$(C_3H_5)(OOCCH_3)_3$	280	295
Glycol diacetate	See Ethylene glycol diacetate			
Glycol diformate	Ethylene glycol diformate	$HCOOCH_2CH_2OOCH$	...	200
Heat transfer oils:				
Heavy paraffinic	...	...	...	550
Light paraffinic	...	...	...	400
Light aromatic	...	...	...	280
Medium aromatic	...	...	...	360
Heptadecanol-*n*	Heptadecyl alcohol	$C_{17}H_{35}OH$	...	310
Heptane-*n*	Dipropylmethane	$CH_3(CH_2)_5CH_3$	25	30
Heptane-iso	...	$(CH_3)_2CHCH_2CH_2CH_2CH_3$	<0	...

Table 7-20 (continued)
Properties of Flammable Liquids, Gases, and Solids

Explosive limits in air, % by vol. Lower	Explosive limits in air, % by vol. Upper	Auto-ignition temperature, °F	Specific gravity (water = 1.0)	Vapor density (air = 1.0)	Melting point, °F	Boiling point or range, °F	Water solubility (miscibility)	Suitable extinguishing agents	Hazard
3.6	46	612	0.832	1.48	−97	132	∞	1	D, G, I
3	100	804	0.887	1.52	−168	51	∞	1, 3, 4	B, D, G, I
...	...	...	0.918	3.00	18	322	∞	1, 3	B, D, G
2.7	16.5	851	0.924	2.55	−112	130	s	2, 3	B, D, G
...	...	...	0.821	4.42	...	325	sl	2, 3	B, D, G
...	...	...	0.942	5.03	−40	472	sl	1, 2, 3	B
...	...	...	0.908	4.98	−180	440	sl	1, 2, 3	B
...	...	...	0.834	4.49	−105	359	sl	1, 2, 3	B, D, G
...	...	...	0.873	5.93	−37	390	sl	1, 2, 3	B, D, G
1.5 @ 212°F	...	752	1.03	4.07	...	309	∞	1, 3	B, D, G
2.8	18.2	570	0.839	2.11	−234	96	sl	3, 4	D, G
...	...	...	1.484	...	−40	318	i	2a, 3	D, G
...	...	...	1.156	...	−15	295	i	2a, 3	D, G
...	...	...	0.916	4.00	...	280	∞	1, 2, 3	B, D, G
3.8	...	...	1.105	3.14	−152	190	sl	2, 3	D, G
3.0	>50	194	0.900	2.59	...	63	sl	2, 3, 4	D, G
...	...	...	1.090	5.04	−41	367	d	1, 3	B, D, G
...	...	...	1.040	5.35	99	514	...	2, 3	B
...	...	...	1.180	9.60	...	608	...	1, 2, 3	B
1.8	11	890	0.891	3.52	−99	210	sl	2, 3	B, D, G
...	...	...	0.851	4.35	...	347	sl	1, 2, 3	B, D, G
1.9	24	...	0.747	...	<−110	147	s	2a, 3	B, D, G, I
...	...	...	0.936	7.22	−114	334	d	3	D, G
...	...	...	1.253	5.60	...	207	...	1, 3	B, D
...	...	...	1.17	6.91	93	439	i	1, 2, 3	B, D
...	...	...	...	...	...	...	...	1, 2, 3	C
...	...	...	...	1.31	−369	−305	d	4	A, E, F
7	73	806	...	1.07	−134	−6	vs	4	D, G, H, I
...	...	795	1.080	1.03	...	207	s	1, 3	D, G
...	...	1114	1.218	1.59	47	213	∞	1, 3	D, G
0.6	5.6	445–560	0.78–0.85	...	...	340–355	i	2, 3	B, D, G
1.3	6.0	350–625	<1.0	...	...	<590	i	2, 3	B, D, G
...	...	500–705	0.80–0.90	...	...	340–640	i	2, 3	B, D, G
1.3	6.0	490–545	0.81	...	...	380–650	i	2, 3	B, D, G
1	5	505	0.84–0.98	...	...	425–760	i	1, 2, 3	B, D, G
1	5	...	0.92–1.06	...	...	...	i	1, 2, 3	B, D, G
1	5	765	0.92–1.07	...	...	...	i	1, 2, 3	B, D, G
2.3	14.3	...	0.937	2.35	−122	88	i	3	B, D, F1, F2, G, H, I
2.1 @ 257°F	...	600	1.161	3.31	−34	322	s	1, 2, 3	B, D, F6
1.8	16.3	736	1.129	3.37	−18	340	∞	1, 2a, 3	B, D, H
...	...	915	1.050	3.35	−94	295	∞	2a, 3	B, D
35	74	...	...	...	...	...	...	4	D, H, I
5.3	32	1200	...	0.47	...	...	...	4	D, H, I
4.4	34	...	...	0.44	...	...	...	4	D, H, I
...	...	...	...	0.38	...	...	...	4	D, H, I
...	...	...	...	0.50	...	...	...	4	D, H, I
3.8–6.5	13–17	>1000	...	0.61	...	...	...	4	D, H, I
4.7	33	637	...	0.47	...	...	...	4	D, H, I
17–35	70–80	...	...	0.86	...	...	...	4	D, H, I
7.0	72	...	...	0.57	...	...	...	4	D, G, H, I
5.5	36	...	...	0.63	...	...	...	4	D, G, H, I
6.0	13.5	640–690	...	...	...	599–649	i	1, 2, 3	B, D, H
1.3–1.4	6.0–7.6	700	0.71–0.76	3.0–4.0	<−76	91–403	i	2, 3	B, D, G
1.3–1.4	6.0–7.6	700	0.70–0.75	3.0–4.0	<−76	91–401	i	2, 3	B, D, G
1	6.0–7.6	800–880	0.70–0.71	3.0–4.0	<−76	108–318	i	2, 3	B, D, G
1	6.0–7.6	800–880	0.70–0.71	3.0–4.0	<−76	107–319	i	2, 3	B, D, G
...	...	739	1.26	3.17	64	554	∞	1, 3	B, D
...	...	812	1.161	7.52	−108	496	s	1, 2a, 3	B, D
...	...	...	1.227	4.07	14	350	d	3	B, D
...	...	600	0.92	...	...	>550	i	1, 2, 3	B, D
...	...	550	0.89	...	...	>550	i	1, 2, 3	B, D
...	...	700+	0.99	...	...	>550	i	1, 2, 3	B, D
...	...	700+	0.97	...	...	>550	i	1, 2, 3	B, D
...	...	...	0.848	8.84	130	588	sl	1, 2, 3	B
1.2	6.7	433	0.688	3.45	−131	209	i	2, 3	B, D, G
1	6	...	0.725	3.45	−182	176–195	i	2, 3	B, D, G

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Table 7-21
Sample Listing of Properties of Flammable Liquids

PROPERTIES OF FLAMMABLE LIQUIDS						
Name	Flash Point °F		Explosive Limits in air % by Volume		Autoignition Temperature °F.	Vapor Density
	Closed Cup	Open Cup	Lower	Upper		Air = 1.0
Acetaldehyde	−36	—	4.0	55.0	365	1.52
Acetone	0	15	2.1	13.0	1000	2.00
Ammonia (Anhydrous)	Gas	Gas	15	28	1204	0.596
Amyl Acetate-n	76	80	1.1	7.5	714	4.49
Amyl Alcohol-n	91	120	1.2	—	572	3.04
Benzene (Benzol)	12	—	1.4	7.1	1044	2.77
Benzine	<0	—	1.4	5.9	550	2.50
Butyl Acetate-n	72	90	1.4	7.6	790	4.00
Butyl Alcohol-n	84	110	1.4	11.2	693	2.55
Camphor	150	200	—	—	871	5.24
Carbon Disulfide	−22	—	1.0	50	257	2.64
Carbon Tetrachloride	None	None	—	—	—	—
Cellosolve	104	120	2.6	15.7	460	3.10
Chloroform	None	None	—	—	—	4.13
Coal Tar Oil	80-160	—	—	—	—	—
Coal Tar Pitch	405	490	—	—	—	—
O-Cresol	178	—	1.3 at 300° F.	—	1038	3.72
Cyclohexanol	154	—	—	—	572	3.45
Denatured Alcohol — 95%	60	—	—	—	750	1.60
Ethyl Acetate	24	30	2.2	11	800	3.04
Ethyl Alcohol (Ethanol)	55	70	3.5	19	737	1.59
Ethylene Glycol	232	240	3.2	—	775	2.14
Ethyl n-propyl ether	—	—	1.9	24	—	—
Formaldehyde, 37% in water	130	200	—	—	795	1.03
Fuel oil No. 1	114-185	—	0.6	5.6	445-560	—
Fuel oil No. 1-D	100 min	—	1.3	6	350-625	—
Fuel oil No. 2	126-230	—	—	—	500-705	—
Fuel oil No. 2-D	100 min	—	1.3	6	490-545	—
Fuel oil No. 4	154-240	—	1	5	505	—
Fuel oil No. 5	130-310	—	1	5	—	—
Fuel oil No. 6	150-430	—	1	5	765	—
Gasoline Automotive premium	−50±	—	1.3-1.4	6.0-7.6	770	3.0-4.0
Gasoline Automotive regular	−50±	—	1.3-1.4	6.0-7.6	700	3.0-4.0
Gasoline Aviation, commercial	−50±	—	1	6.0-7.6	800-880	3.0-4.0
Gasoline Aviation military	−50±	—	1	6.0-7.6	800-880	3.0-4.0
Hexane-n	−7	—	1.2	7.5	453	2.91
Hexane-iso	<−20	—	1	7	—	3.00
Hydrogen sulfide	Gas	Gas	4.3	45.5	500	1.18
Jet fuel JP-1	110-125	—	0.6	5.6	442-560	—
Jet fuel JP-4	26-36	—	0.8	6.2	468	—
Kerosene	110-130	—	0.6	5.6	440-560	4.5
Lacquer	0-80	—	—	—	—	—
Maleic Anhydride	218	240	—	—	890	3.38
Methyl Acetate	15	20	3.1	16	935	2.56
Methyl Alcohol (Methanol)	54	60	5.5	36.5	878	1.11
Methyl Ethyl Ketone	30	—	1.8	10	960	2.48
Mineral spirits	100 min	110	0.77@212°F	—	475	3.9
Naphtha	100-110	—	0.8	5	440-500	—
Naphtha VM&P	20-45	—	0.9	6.0	450-500	3.75
Naphthalene	174	190	0.9	5.9	979	4.42
Petroleum crude	20-90	—	—	—	—	—
Petroleum ether	<0	—	1.4	5.9	550	2.50
Phenol	175	185	—	—	1319	3.24
Phthalic Anhydride	305	330	1.7	10.5	1083	5.10
Pine oil	172	175	—	—	—	—
Propane	<−100	Gas	2.2	9.6	871	1.56
Propyl Acetate-n	58	70	1.7	8.0	842	3.52
Propyl Alcohol-ISO	53	60	2.5	12	750	2.07
Quenching oil	365	405	—	—	—	—
Stoddard solvent	100-110	—	0.8	5	440-500	—
Styrene	90	—	1.1	6.1	914	3.60
Sulfur	405	440	—	—	450	—
Toluene	40	45	1.3	7.0	997	3.14
Trichloroethylene	Weakly Flammable		10 in O_2	65 in O_2	—	4.53
Turpentine	95	—	0.8	—	488	4.84
p-Xylene	77	—	1.1	7.0	984	3.66

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(text continued from page 486)

v_1, v_2, etc. = volume percent of each combustionable gas present in mixture, free from air and inert gas

$$v_1 + v_2 + v_3 = 100 \quad (7\text{-}55)$$

$L_1, L_2, L_3 \ldots$ = lower flammability limits, vol % for each flammable gas in mixture

Example 7-13: Calculation of LEL for Flammable Mixture

Assume mixture analysis (combustible with air):
Methane, 3.0%, LEL = 5.3%
Propane, 4.0%, LEL = 2.3%
Hexane, <u>1.0%</u>, LEL = 1.1%
Subtotal 8.0% combustible
Balance is air (92%)

For each component: v, combustible only on air-free basis.
Methane: $3/8$ (100) = 37.5%
Propane: $4/8$ (100) = 50.0%
Hexane: $1/8$ (100) = 12.5%
Percentage combustible only = 100.0%

$$\text{Mixture composite LEL} = \frac{100}{37.5/5.3 + 50/2.3 + 12.5/1.1}$$

$$= 2.48\% \text{ volume for mixture}$$

The UEL for a composite is determined in the same manner, using the respective component UEL values. For the overall mixture, the above can be used to calculate the composition. Also see Ref [52].

Pressure and Temperature Effects

The temperature and pressure of a liquid system are important in determining the effects created that result in a fire and explosion hazard. Because this relates to the flash point and flammability limits, see Tables 7-21, 7-22 and Figures 7-48, and 7-49A, and 7-49B [34].

- An increase in pressure raises the flash point, while a decrease lowers the flash point [39, 34].
- As temperature is increased on a liquid, its vapor pressure will increase and will therefore tend to vaporize at a greater rate [34].

In a closed container, equilibrium develops at any given temperature and pressure, while in an open condition (not enclosed) the liquid will continue to vaporize in air until the liquid is completely vaporized. In that situation, effects of temperature and pressure are valid only for enclosed conditions, such as tanks, vessels, piping and other processing equipment. See Figures 7-49A and 7-49B and Table 7-22 for LEL or UEL showing a variation of limits with temperatures and pressures.

Extreme care must be exercised in designing potentially flammable systems to use reliable flammability limits data and to recognize the effects of pressure/temperature on the data and its implications to the safety of the system in question. Unless otherwise indicated, most published data is at atmospheric pressure and ambient temperature and should be corrected for other conditions.

Figure 7-47 illustrates a gas-freeing system using gasoline-air-water-vapor (the water vapor could be steam). The mixture "A" represents a saturated gasoline-vapor-air-water-vapor mixture at 70°F. In a closed tank, a more volatile gasoline than the one diagrammed would give a saturated mixture with gasoline vapor and less air. A less volatile gasoline would give less gasoline vapor and more air. If a continuous supply of air saturated with water vapor is added to a tank containing mixture A, all compositions between A and B (air plus water vapor) will be formed until all the gasoline vapor has been flushed from the tank and only steam remains (at 212°F or higher). If the tank is cooled, the steam will condense and air will be drawn into the tank giving mixtures along C-B. At 70°F only air plus a small amount of water vapor will remain.

If hot water and water vapor at 175°F are used to flush mixture A from the tank, the mixture composition can only shift along AC to E. Mixtures between A and E are flushed from the tank, mixed with air to give a mixture between parts AE and B. After examining several other

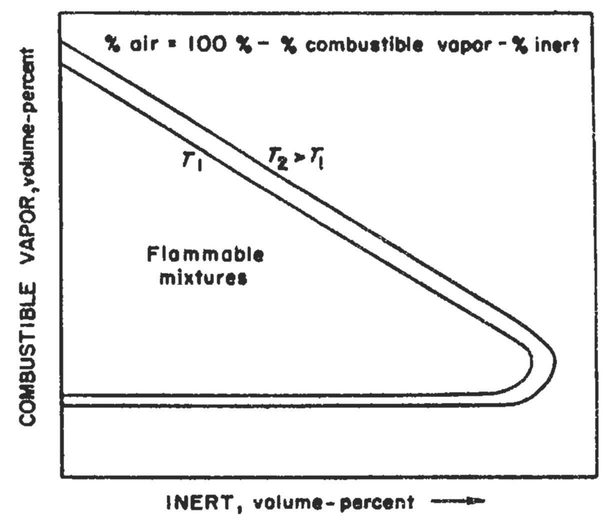
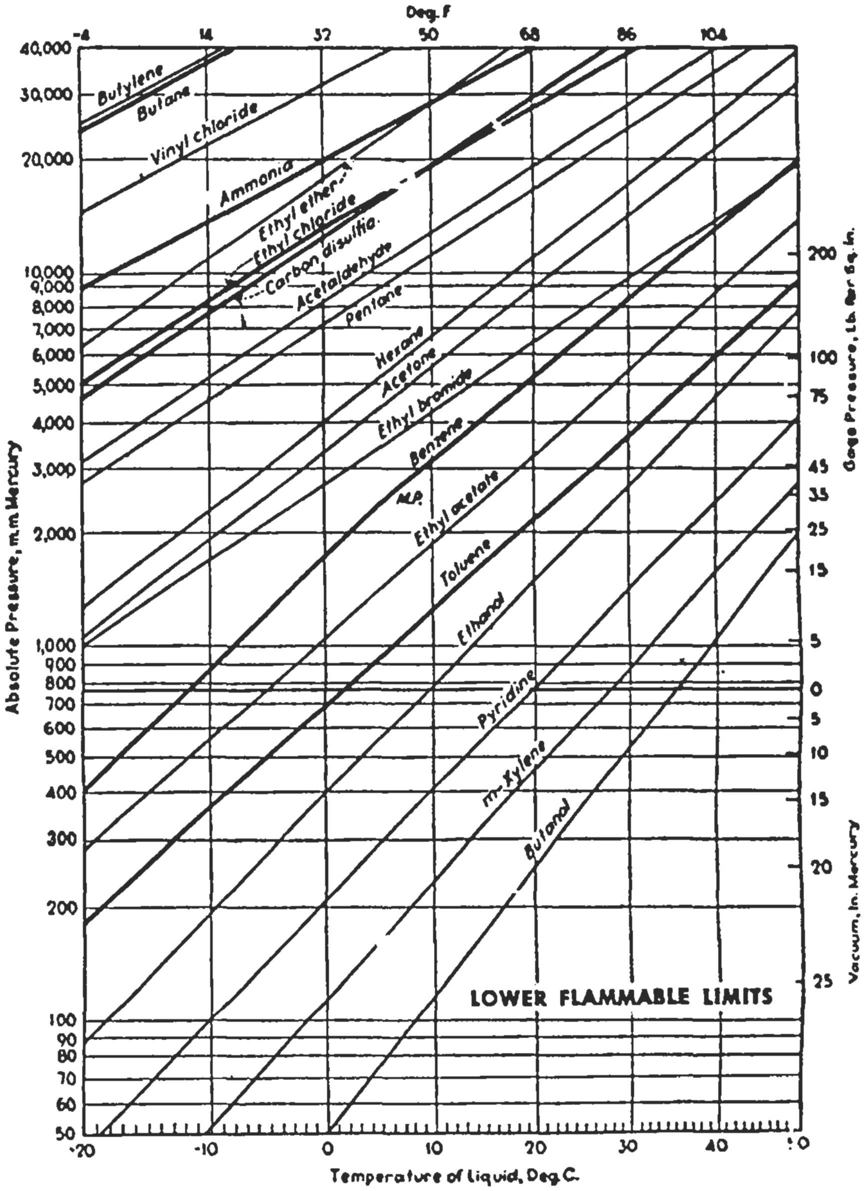


Figure 7-48. Effect of initial temperature on limits of flammability of a combustible vapor-inert-air system at atmospheric pressure. By permission, U.S. Bureau of Mines, Bulletin 627 [43].

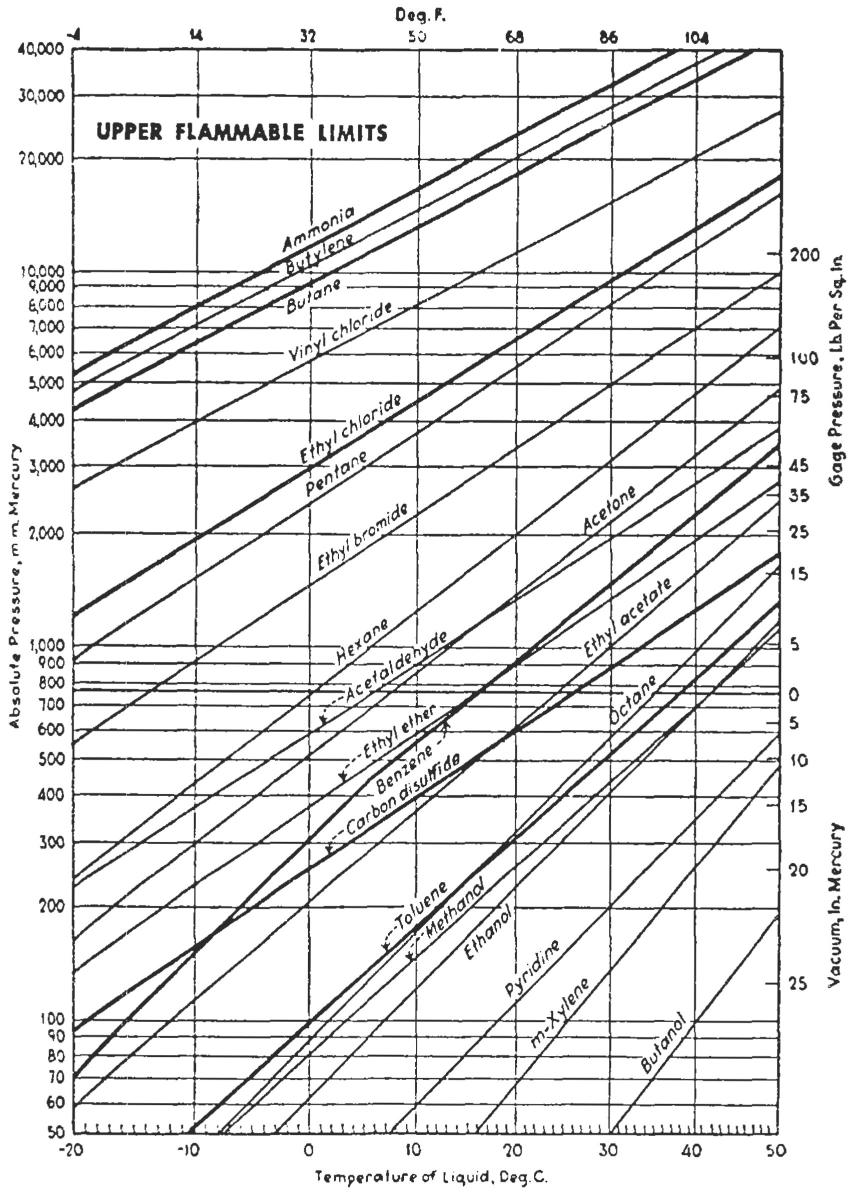


This chart is applicable only to flammable liquids or gases in equilibrium in a closed container. Mixtures of vapor and air will be too lean to burn at temperatures below and at pressures above the values shown by the line on the chart for any substance. Conditions represented by points to the left of and above the respective lines are accordingly nonflammable. Points where the diagonal lines cross the zero gauge pressure line (760 mm of mercury absolute pressure) indicate flash point temperatures at normal atmospheric pressure.

Figure 7-49A. Variation of lower flammable limits with temperature and pressure. Reprinted by permission from *Fire Protection Handbook*, 17th Ed. (1991), National Fire Protection Association, Quincy, MA 02269 [34].

variations with air and water vapor, the conclusion is that mixture A cannot be flushed from a tank without forming flammable mixtures unless steam or some other inert vapor or gas is used [43].

Figure 7-48 [43] shows the effects of temperature on limits of flammability at a constant initial pressure. As temperature is increased, the lower limit decreases and the upper limit increases. This is extremely important in evaluating the explosive potential for a mixture, recognizing that just reading numbers for UEL and LEL from a table is inadequate when temperatures are above normal atmospheric (or the temperature recorded for the date). Due to this situation, even a nonflammable mixture such as part A (Figure 7-47) can propagate a flame a short distance from a source because it can become flammable as temperature is raised, part B. The compositions along



This chart is applicable only to flammable liquids or gases in equilibrium in a closed container. Mixtures of vapor and air will be too "rich" to be flammable at temperatures above and pressures below the values shown by the lines on the chart for any substance. Conditions represented by points to the right of and below the respective lines are accordingly nonflammable.

Figure 7-49B. Variation of upper flammable limits with temperature and pressure. Reprinted by permission from *Fire Protection Handbook*, 17th Ed. (1991), National Fire Protection Association, Quincy, MA 02269 [34].

the "saturated vapor-air-mixture line and to the right of it make up the saturated and unsaturated combustible-oxidant system at a specified pressure" [43]. The engineer is urged to study Reference [43] because it deals with many more important topics of this subject than can be included here.

Figure 7-50 shows the effect of pressure on the limits of flammability of an ethane system.

Ignition of Flammable Mixtures

Ignition can take place for any flammable mixture within the concentration ranges for the respective LEL and UEL. The conditions for ignition may vary with the specific mixture, the type of oxidant (usually air or pure oxygen), the temperature, and pressure of the system. Ignition may result from electrical spark, static spark, contact with hot surfaces (autoignition) (see Figures 7-48,

Table 7-22
The Effect of Elevated Temperature on the Lower Flammable Limit of Combustible Solvents as Encountered in Industrial Ovens‡

Solvent	Flash Pt Closed Cup	Lower Flammable Limit Percent Vapor by Volume at Initial Temperature, °F						
		Room	212	392	437	482	572	662
Acetone	3	2.67	2.40	2.00*	—	—	—	—
Amyl Acetate, Iso	77	—	1.00	0.82	—	0.76*	—	—
Benzene	−4	1.32	1.10	0.93	—	—	0.80*	—
Butyl Alcohol, Normal	100	—	1.56	1.27	1.22*	—	—	—
Cresol, Meta-Para	202	—	1.06†	0.93	—	0.88*	—	—
Cyclohexane	−4	1.12	1.01	0.83*	—	—	—	—
Cyclohexanone	111	—	1.11	0.96	0.94	0.91*	—	—
Ethyl Alcohol	54	3.48	3.01	2.64	—	2.47	2.29*	—
Ethyl Lactate	131	—	1.55	1.29	—	1.22*	—	—
Gasoline	−45	1.07	0.94	0.77*	—	—	—	—
Hexane, Normal	−15	1.08	0.90	0.72*	—	—	—	—
High-Solvency Petroleum Naphtha	36	1.00	0.89	0.74	0.72	0.69*	—	—
Methyl Alcohol	52	6.70	5.80	4.81	—	4.62	4.44*	—
Methyl Ethyl Ketone	21	1.83	1.70	1.33*	—	—	—	—
Methyl Lactate	121	—	2.21	1.86	1.80	1.75*	—	—
Mineral Spirits, No. 10	104	—	0.77	0.63*	—	—	—	—
Toluene	48	1.17	0.99	0.82	—	—	0.72*	—
Turpentine	95	—	0.69	0.54*	—	—	—	—
V. M. & P. Naptha	28	0.92	0.76	0.67*	—	—	—	—

* Rapid and extensive thermal decomposition and oxidation reactions in vapor-air mixture at this temperature.
† Lower limit determined at 302°F.
‡ From NFPA *Quarterly*, April 1950; UL Bulletin of Research No. 43.

*Rapid and extensive thermal decomposition and oxidation reactions in vapor-air mixture at this temperature.
†Lower limit determined at 302°F
‡From NFPA Quarterly, April 1950; UL Bulletin of Research No. 43.
Reprinted by permission, *Fire Protection Handbook*, 17th Ed. (1991) p. 4-32. National Fire Protection Association [34].

7-51 and 7-52A and B) or other means. Established references include [44, 45, and 46].

Figures 7-51, 7-52A and B are convenient diagrams for studying the flammability of various compounds.

Where Autoignition Temperature (AIT) = the minimum temperature at which a material begins to self-heat at a high enough rate to result in combustion. Reported in the Data Guide (Figure 7-51) as the temperature in air at one atmosphere.

C_{st} = stoichiometric composition of combustionable vapor in air, expressed as a volume percent.
T_{st} = equilibrium temperature at which C_{st} exists over liquid in dry air at a temperature, °C or °F per chart.
T_L = equilibrium temperature at which the lower flammable limit composition exists over liquid in dry air at one atmosphere (theoretical flash point), °C or °F
T_u = equilibrium temperature at which the upper flammable limit composition exists over liquid in dry air at one atmosphere, °C or °F
MP = melting point (freezing point), °C or °F
BP = boiling point, °C or °F

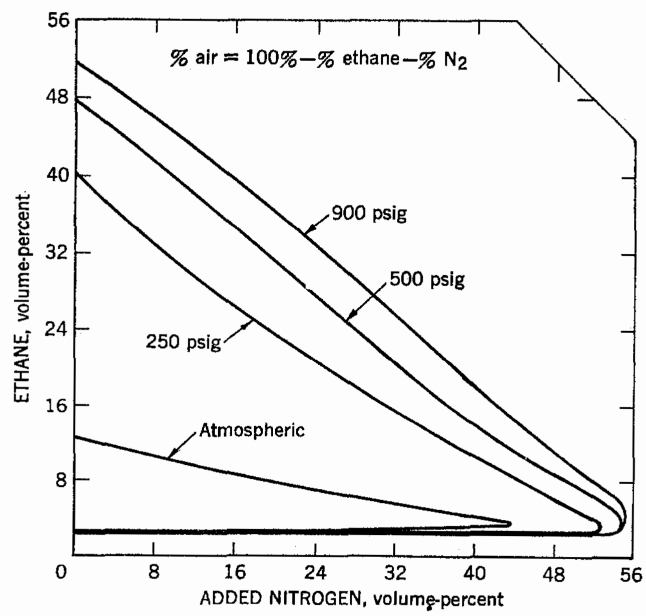


Figure 7-50. Effects of pressure on limits of flammability of ethane-nitrogen-air mixture at 26°C. By permission, U.S. Bureau of Mines, Bulletin 627 [43].

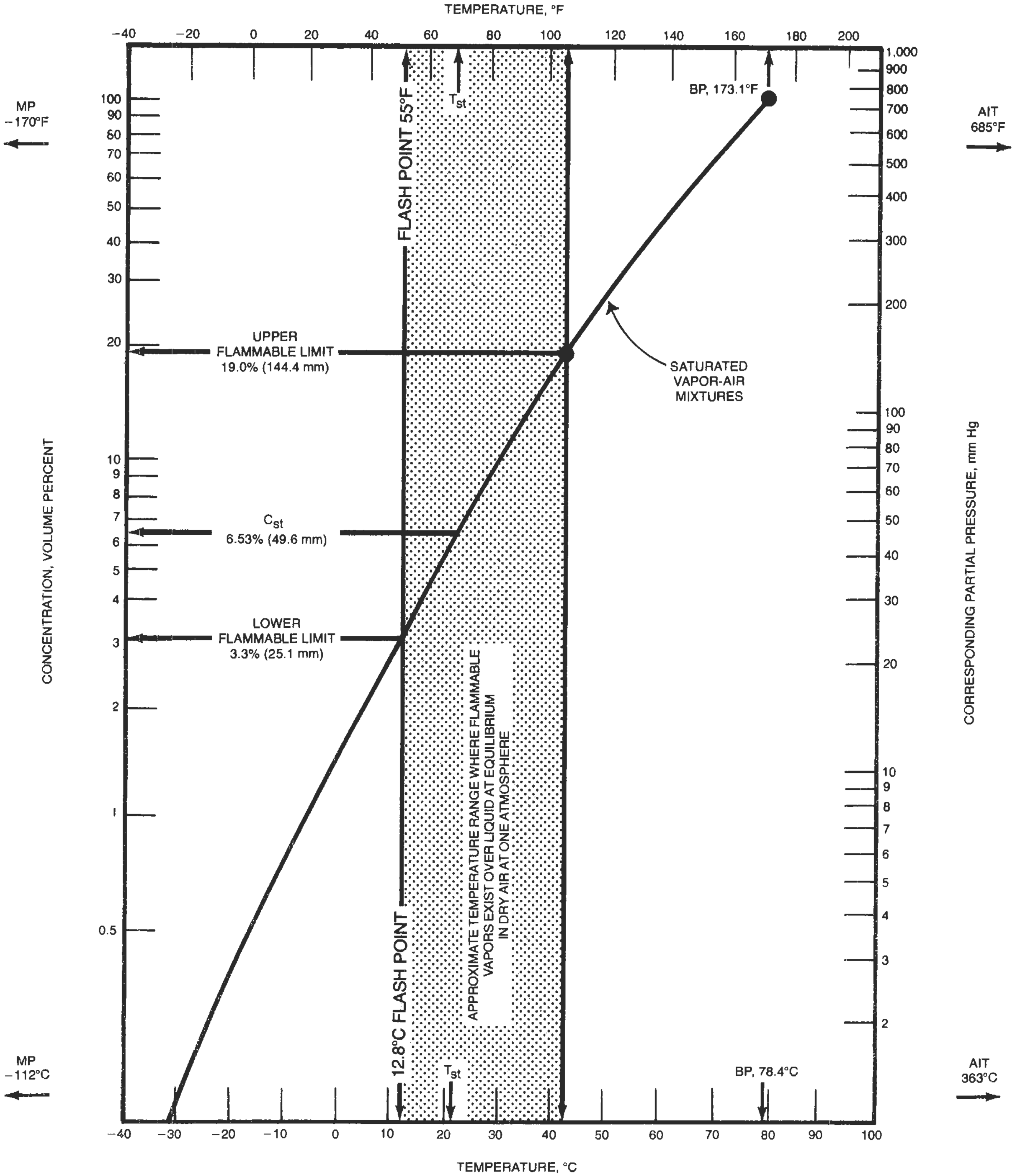


Figure 7-51. Example flammability guide for ethanol. By permission, Hercules, Inc.

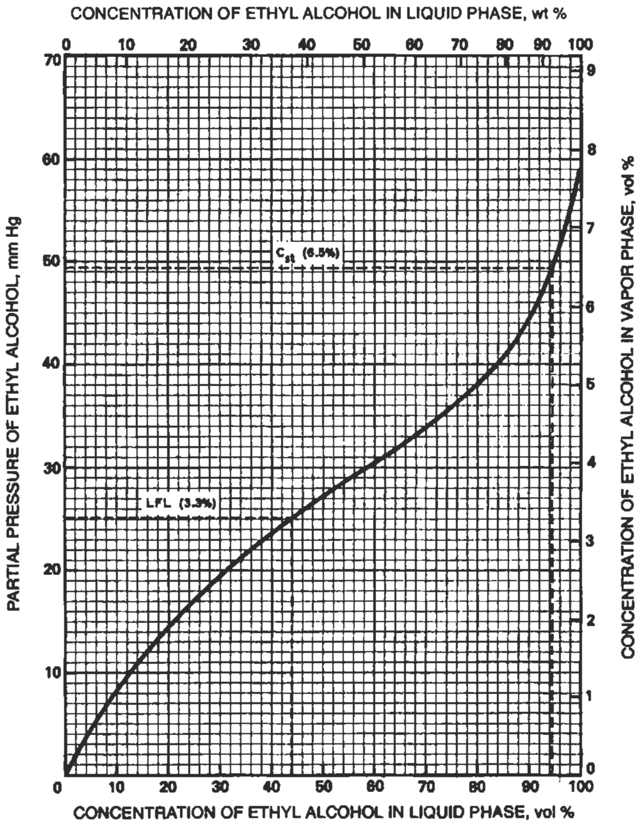


Figure 7-52A. Vapor-liquid data for solutions of ethyl alcohol relating to mixture flashpoints. By permission, Hercules, Inc.

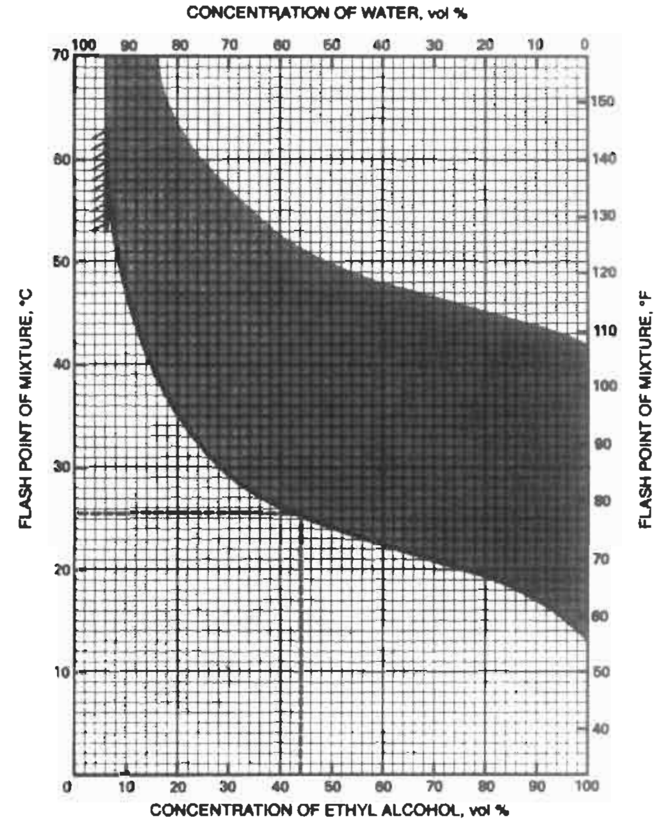


Figure 7-52B. Flashpoints of ethyl alcohol-water mixtures as a function of liquid phase concentration at one atmosphere total pressure. By permission, Hercules, Inc.

Aqueous Solutions of Flammable Liquids

Some organic compounds can be in solution with water and the mixture may still be a flammable mixture. The vapors above these mixtures such as ethanol, methanol, or acetone can form flammable mixtures with air. Bodurtha [39] and Albaugh and Pratt [47] discuss the use of Raoult's law (activity coefficients) in evaluating the effects. Figures 7-52A and B illustrate the vapor-liquid data for ethyl alcohol and the flash point of various concentrations, the shaded area of flammability limits, and the UEL. Note that some of the plots are calculated and bear experimental data verification.

Blast Pressures

Deflagrations and detonations produce pressures associated with the resulting shock/pressure waves. These pressures can be sufficiently large to damage and/or demolish enclosed vessels, equipment, and buildings. A deflagration can produce pressure rises in excess of 8:1 and rises of 40:1 when a reflected wave develops from a detonation, referenced to the initial pressure of the system. In a containment vessel, the peak pressure from a detonation beginning at atmospheric pressure may rise to about 19.7 times or 289.8 psia. Figure 7-53 illustrates detonation velocities and pressures. Barker, et. al [87] discuss blast loaded structures.

Table 7-23 presents U.S. Atomic Energy Commission data [43], which expresses the overpressure above normal required to do the damage indicated. Table 7-24 presents a collection of industrial overpressure damage situations [53]. The overpressuring $(P - P_o)$ is expressed [43]:

$$(P - P_o) = P_o\,[2k/(k - 1)]\,[v/v_s - 1] \tag{7-56}$$

k = ratio of specific heats
v_s = velocity of sound in the medium through which the shock wave travels, ft/sec
v = shock velocity (from test data or specific calculations), ft/sec
$P_i' = P_o$ = initial pressure of system, psia
$P_{max} = P$ = peak pressure of blast, psia

Other references on the subject of blast damage are [48] [49] [50]. Tables 7-25A and 7-25B provide several

Table 7-23
Conditions of Failure of Peak Overpressure for Selected Structural Components

Structural element	Failure	Approximate incident blast overpressure (psi)
Glass windows, large and small.	Usually shattering, occasional frame failure.	0.5–1.0
Corrugated asbestos siding.	Shattering.	1.0–2.0
Corrugated steel or aluminum paneling.	Connection failure, followed by buckling.	1.0–2.0
Wood siding panels, standard house construction.	Usually failure occurs at main connections, allowing a whole panel to be blown in.	1.0–2.0
Concrete or cinder-block wall panels, 8 or 12 inches thick (not reinforced).	Shattering of the wall.	2.0–3.0
Brick wall panel, 8 or 12 inches thick (not reinforced).	Shearing and flexure failures.	7.0–8.0

By permission, Zabetakis, M. G., U.S. Bureau of Mines, Bul. 627 [43].

miscellaneous conditions, including physiological effects of blast pressures on humans and structural damage to facilities.

Pressures of deflagration or detonation shock waves build upon the existing system pressure at the time of the initial blast. When a deflagration starts and then builds to a detonation, the resulting peak pressure can be quite high because the final pressure of the detonation builds on the peak pressure of the deflagration.

In an enclosure, a peak for initial pressure ratio for a deflagration generally can exceed 8:1 of the initial pressure. The pressures may build to a ratio of 40:1 (reflected pressure) times the initial pressure when a detonation develops. This is the reason detonations can be so disastrous. Their final pressure, when built on a deflagration peak pressure as a base or initial pressure, can be extremely high.

For example, see Figures 7-53 [43] and 7-54.

The overpressure levels, that is, final peak less initial starting pressure, shown in Table 7-23 [43] are low referenced to the magnitude of overpressures attainable from many industrial blasts.

The speed of a combustion reaction will be at a maximum at a certain fuel-air ratio that is generally close to the stoichiometric composition. It will be lower, however, for compositions closer to each of the explosive limits. The rate of pressure rise is a measure of the speed of flame propagation and accordingly the violence of the explosion. The rate of pressure rise is the slope of the tangent line (Figure 7-54) through the rising branch of the pressure, which is the time curve of an explosion [53]. The greatest rate of pressure rise will occur when an explosive mixture ignites in the center of a vessel. Ignition at any other location will result in a lower rate and somewhat reduced explosion pressure. The volume of the vessel influences the violence of the explosion (Figure 7-55 and Figure 7-57). For example, Bartknecht data [54] shows that for propane exploding in three separate but different sized vessels, the magnitude of the final maximum pressure is the same for each of about 7 bars, but the time to reach this maximum pressure is greater the larger the vessel (Figure 7-56).

The influence of vessel volume on the maximum rate of pressure rise for a specified gas is characterized by the Cubic Law, i.e., [54] the rate of pressure rise varies for each gas.

$$(dp/dt)_{max}\,(V)^{1/3} = \text{Constant} = K_G \qquad (7\text{-}57)$$

V = vessel volume, cubic meters
K_G = constant, (bar) (meters/second)

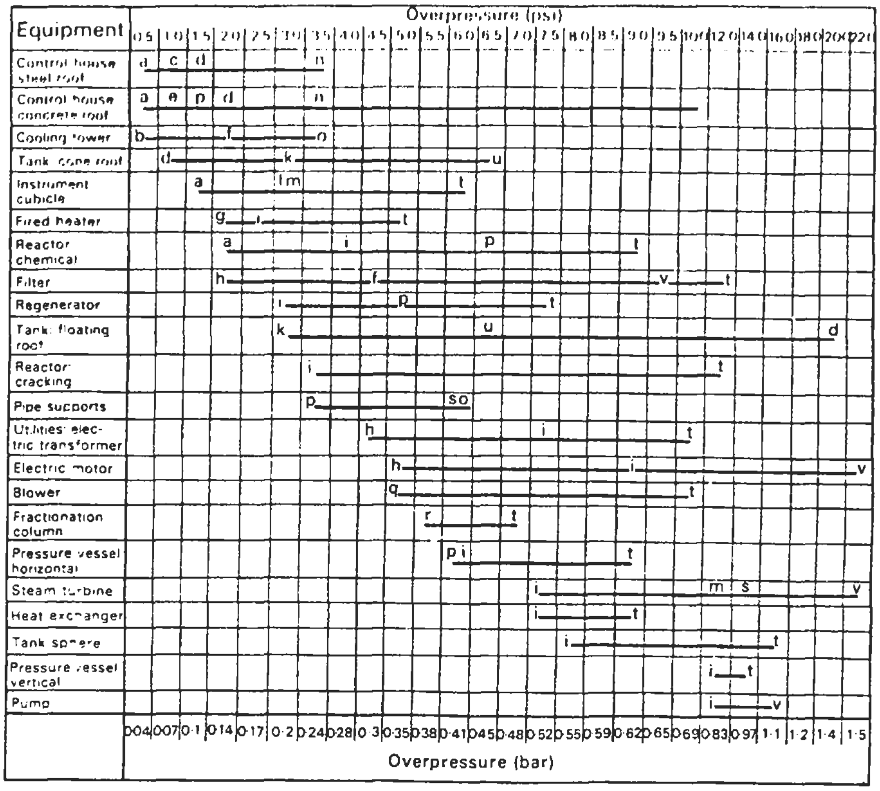
This is valid for the same degree of gas mixture turbulence and the same ignition source and is illustrated in Figure 7-58. Influence of the vessel shape is shown in Figure 7-56. The behavior of propane is considered representative of most flammable vapors including many solvents [54]. The maximum explosion pressure does not follow the cubic law and is almost independent of the volume of a vessel greater than 1 liter. For propane, town gas, and hydrogen, the volume relationship can be expressed:

$$\frac{1}{V^{1/3}} < 5.5, \text{ or}, \frac{\text{surface}}{\text{volume}} < \frac{1}{30 \text{ meters}} \cong \frac{dp}{dt} \qquad (7\text{-}58)$$

The violence of an explosion is influenced by the initial pressure or pressure of the system in which the explosion takes place. Figure 7-57 illustrates this point for propane and a constant ignition energy source. For low pressure below atmospheric, the explosion reactions are reduced until they will not propagate through the fuel-air mixture [54].

For some mixtures, unusual conditions seem to develop in the rate of pressure rise at peak explosion pressures due to possible changes in the violence of the reaction [54]. Similar results are reported for the level of energy required for ignition of a mixture related to the concentration range for ignition. Without examining the energy level versus concentration at various initial pressures, it

Table 7-24
Typical Damage Caused by Overpressure Effects of an Explosion



a Windows and gauges break
b Louvres fall at 0.3–0.5 psi
c Switchgear damaged by roof collapse
d Roof collapses
e Instruments damaged
f Inner parts damaged
g Brick cracks
h Debris-missile damage occurs
i Unit moves and pipes break
j Bracing fails
k Unit uplifts (half-filled)
l Power lines severed
m Controls damaged
n Block walls fail
o Frame collapses
p Frame deforms
q Case damaged
r Frame cracks
s Piping breaks
t Unit overturns or is destroyed
u Unit uplifts (0.9 filled)
v Unit moves on foundation

By permission, Wells, G. L., *Safety in Process Plant Design.* George Godwin Ltd., London: John Wiley & Sons, N.Y., 1980 [53].

might erroneously be concluded that some fuel-air mixture were "not explodable." For a pipeline of flammable gas/vapor with one end closed and the other open, an explosion originating at the closed end will produce a higher velocity and higher pressure explosion than if ignition originates at the open end [54].

The violence of an explosion increases when the shape of the vessel changes from spherical to a more elongated shape or has a ratio of length: diameter of 1:1 to 1:5, with length several times the diameter due to the flame front of the explosion moving swiftly in the axial direction and compressing at the end of the vessel thereby energizing the violence of the explosion (see Figures 7-56). Accordingly, the shape of the vessel is an important consideration when evaluating the explosion potential of a mixture. All remarks above are related to non-turbulent mixtures, because turbulence increases the violence of the explosion (Figure 7-58).

Figure 7-58 shows the differences in K_G value and maximum explosion pressures for turbulent and non-turbulent systems for the same fuel-air mixtures.

Mixtures of flammable gas/vapors plus oxygen when ignited can form more violent explosions with greater peak pressures than when ignited in air.

Table 7-25A
Selected Over-pressure Failure Situations

Damage Limits:	Avg Over-pressure (PSI)
Crater	280
Probable Total Destruction	10 & Above
Limit Serious Structural Damage	2.3
Limit Earth Wave Damage	1.2
Limit Minor Structural Damage	0.4
Missile Limit	0.3
Typical Glass Failure	0.15
Limit Glass Breakage	0.006

By permission, Stull [41] Dow Chemical Co. and American Institute of Chemical Engineers, *Monograph Series, No. 10,* V. 73 (1977).

Table 7-25B
Physiological Effects of Blast Over-pressures

Physiological Effects of Blast Pressures		Peak Over-pressure (PSI)
Knock Personnel Down		1
Eardrum Rupture	Threshold	5
	50%	15
Lung Damage	Threshold	30–40
	Severe	80 & up
Lethality	Threshold	100–120
	50%	130–180
	Near 100%	200–250

By permission, Stull [41] Dow Chemical Co. and American Institute of Chemical Engineers, *Monograph Series, No. 10,* V. 73 (1977).

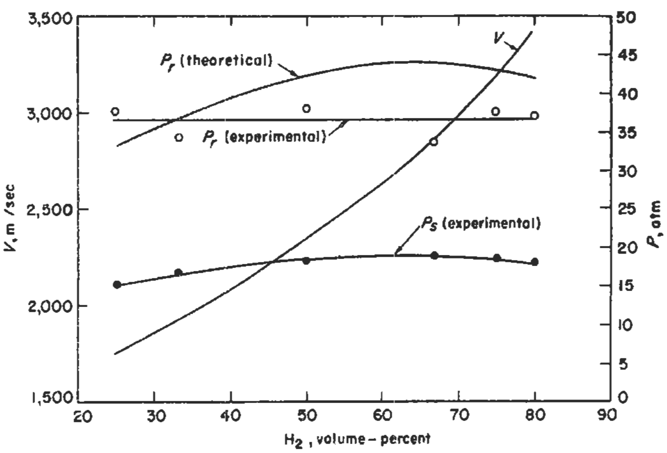


Figure 7-53. Detonation velocity, V, static pressure, P_s, and reflected pressure, P_r, developed by detonation wave propagating through hydrogen-oxygen mixtures in a cylindrical tube at atmospheric pressure at 18°C. By permission, U.S. Bureau of Mines, Bulletin 627 [43].

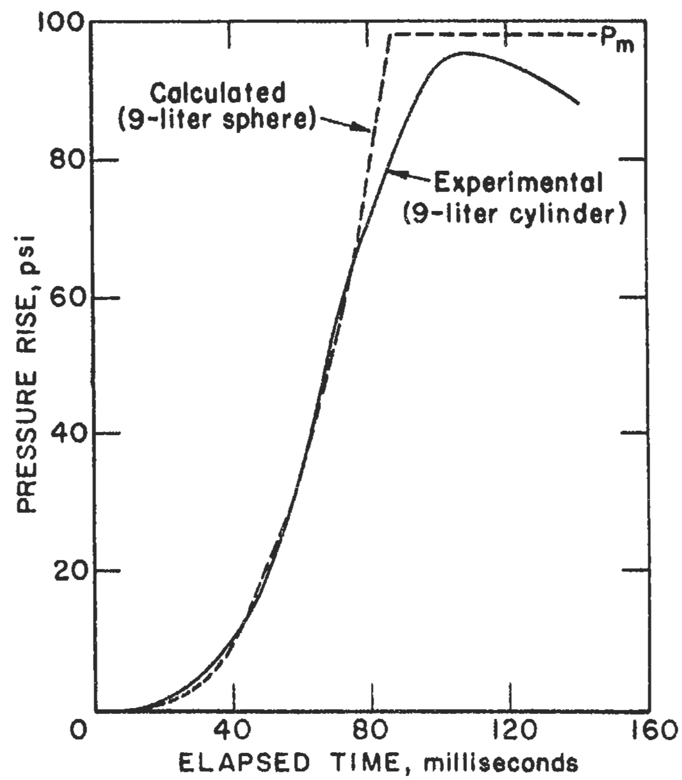


Figure 7-54. Pressure produced by ignition of a 9.6 volume percent methane-air mixture in a 9-liter cylinder (experimental). By permission, U.S. Bureau of Mines, Bulletin 627 [43].

Somewhat unusual and/or unexpected mixtures of gases/vapors can form explosive mixtures. Some, but not all inclusive, of these include [34]:

- Chlorine with hydrogen, ammonia, acetylene, turpentine, or powdered metals. Steel will burn in the presence of chlorine.
- Bromine causes fire in contact with combustible materials.
- Iodine is explosive with ammonia, turpentine, or lead triethyl.
- Fluorine reacts spontaneously with almost all elements, hydrogen, water vapor, and many organic compounds. Steel will melt and ignite in fluorine with a violent reaction.

See Ref. [34] for a more complete listing of corrosive chemicals, water and air-reactive chemicals, unstable chemicals, combustible chemicals, and oxidizing chemicals.

TNT (Tri-Nitro Toluene) Equivalence for Explosions

The explosion of a quantity of TNT has been established as the standard for defining or comparing the

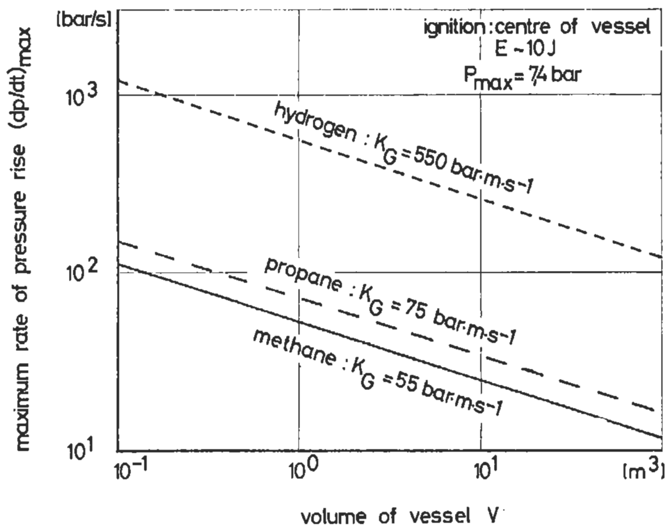


Figure 7-55. Influence of vessel volume on violence of explosion of flammable gases. Ignition at zero turbulence. By permission, Bartknect, W., *Explosions*, 2nd Ed. (1980), Springer-Verlag [54].

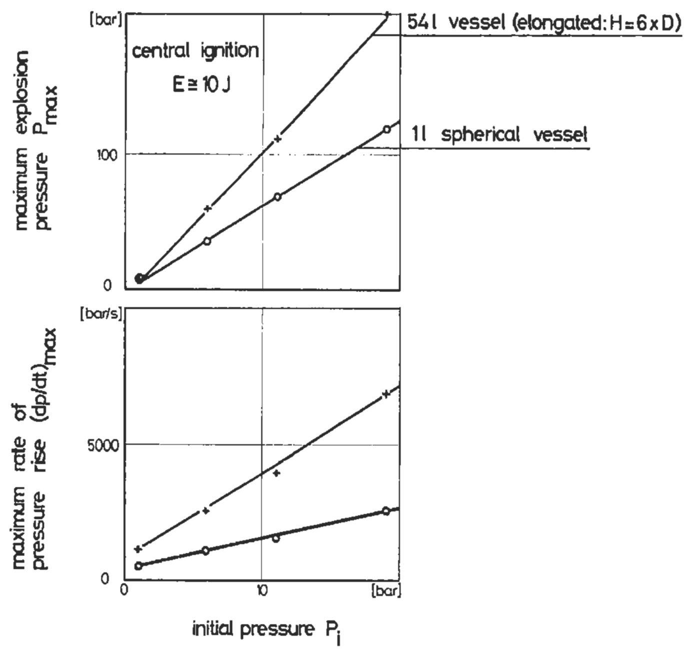


Figure 7-56. Influence on vessel shape on the explosion data of methane. By permission, Bartknect, W., *Explosions*, 2nd Ed. (1980), Springer-Verlag [54].

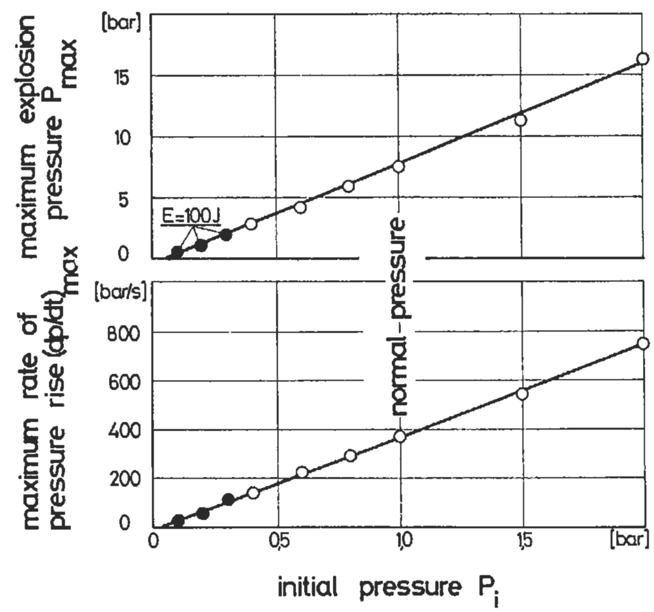


Figure 7-57. Influence of initial pressure on explosion data of propane, 7 liter vessel, E=10 J. By permission, Bartknect, W., *Explosions* 2nd, Ed. (1980), Springer-Verlag [54].

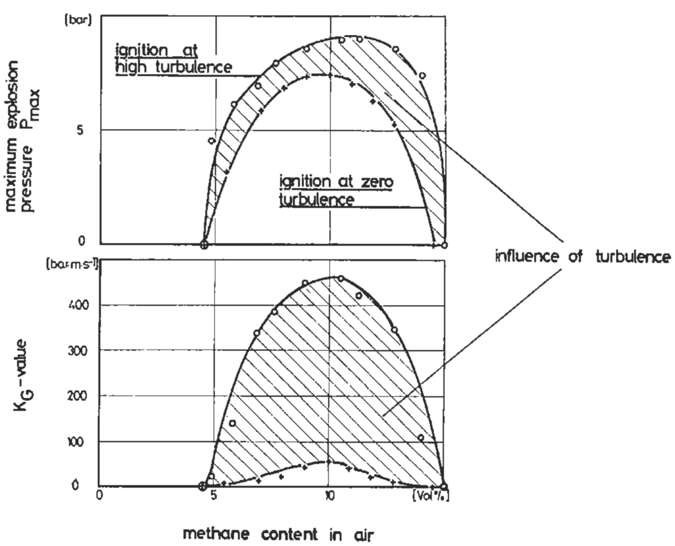


Figure 7-58. Influence of turbulence on the explosion data of methane at 10 joules. By permission, Bartknect, W., *Explosions*, 2nd Ed. (1980), Springer-Verlag [54].

blast effects of other explosive materials [49]. The blast wave generated by other chemical explosives generally differ somewhat from the waves of TNT explosions, including peak pressure and impulse, but are similar in other effects. The equivalent energy of TNT is 1120 calories/gram, (this value varies with some data). Other types of explosives are related by their charge weights and can be converted to their TNT equivalent, e_t:

$$W_{TNT} = (e_t)(W_c) \tag{7-59}$$

where W_{TNT} = equivalent charge weight of TNT, lb
e_t = equivalence factor, see Table 7-26.
W_c = charge weight of explosives of interest, lb

Table 7-26 [49] has been developed by ratio of relative heats of explosion. For close explosion, i.e., ($Z < 3.0$ ft./lb$^{1/3}$) and for shapes other than spherical, the TNT equivalent factor can be much greater than those from relative heats of explosion [49].

Pressure Piling

If two or more systems are connected together (such as a pipe length with an orifice plate, two or more vessels connected with pipe or duct, or a compartmented vessel) and an explosion develops in No. 1 area, which generally may be at equilibrium pressure with compartments No. 2 and 3 in equilibrium with No. 2, it can cause a pressure rise in front of the flame front in the unburnt gases in the interconnecting spaces (pipe, compartment). The increased pressure in compartment or area No. 1 becomes the starting pressure for an explosion in compartment No. 2 and, by the same analysis, this increased pressure in No. 2 becomes the starting pressure for an explosion in No. 3. This pressure buildup under these types of conditions is known as pressure piling [40]. From a pressure buildup standpoint,

1. when the initial pressure in compartment No. 1 is p_1, the final pressure will be (p_1) (x).
2. the final pressure in compartment No. 2 could be x^2p_1.
3. the final pressure in compartment No. 3 could be x^3p_1.

Where x = ratio of pressure increase, often with a value between 2 and 8 for a deflagration.

For example: If p_1 = 20 psig + 14.7 = 34.7 psia, assume x = 6.5.
Thus, final pressures in compartment No. 2 would be = $(6.5)^2$ (34.7) = 1466 psia.

Thus, it is easy to recognize that the pressure buildup in a process system can be dangerously large and requires attention to both pressure relieving and to the design pressures for vessels/equipment. This also helps explain why some process vessels fragment during an explosion and fragments impact on personnel, buildings, etc., to do damage. It also helps to explain *the shock wave effects.*

Table 7-26
TNT Equivalence Factors for Chemical Explosives

Explosive	e_t (TNT Equivalent)
Amatol 60/40 (60% ammonium nitrate, 40% TNT)	0.586
Baronal (50% barium nitrate, 35% TNT, 15% aluminum)	1.051
Comp B (60% RDX, 40% TNT)	1.148
C-4 (91% RDX, 9% plasticizer)	1.078
Explosive D (ammonium picrate)	0.740
H-6 (45% RDX, 30% TNT, 20% Al, 5% D-2 wax)	0.854
HBX-1 (40% RDX, 38% TNT, 17% Al, 5% D-2 wax)	0.851
HMX	1.256
Lead Azide	0.340
Lead Styphnate	0.423
Mercury Fulminate	0.395
Nitroglycerine (liquid)	1.481
Nitroguanidine	0.668
Octol, 70/30 (70% HMX, 30% TNT)	0.994
PETN	1.282
Pentolite, 50/50 (50% PETN, 50% TNT)	1.129
Picric Acid	0.926
RDX (Cyclonite)	1.185
Silver Azide	0.419
Tetryl	1.00
TNT	1.00
Torpex (42% RDX, 40% TNT, 18% Al)	1.667
Tritonal (80% TNT, 20% Al)	1.639

(Refs. 3-1 and 3-4 in original source)
By permission, U.S. Army Corps of Engineers, *Report HNDM-1110-1-2* (1977) [49].

Example 7-14: Estimating Blast Pressures and Destruction

A process petrochemical plant producing a synthesis gas high in hydrogen experiences an explosion that results in the destruction of a 1500 cubic foot storage vessel normally held at 50 psig. Unprotected glass windows (i.e., no wire mesh reinforcing, nor tempered) in the plant area 150 feet away from the tank are broken. What pressures were involved?

Using the equation for isentropic expansion of an ideal gas:

$$W_c = (P_1 V_1)/(k - 1)\ [1 - (P_2/P_1)^{(k-1)/k}],\ \text{ft. lb} \qquad (7\text{-}60)$$

V_1 = 1500 cu ft
k = 1.41 for hydrogen
P_1 = 50 + 14.7 = 64.7 psia
P_1 = 64.7 psia (144 in^2/ft^2) = 9316.8 lb/ft^2
P_2 = 14.7 psia final pressure

$$W_c = \left[\frac{(9316.8)(1500)}{1.41-1}\right]\left[1-\left(\frac{14.7}{64.7}\right)^{\frac{1.41-1}{1.41}}\right]$$

$$= 1.193 \times 10^7 \text{, ft lb}$$
$$= (1.193 \times 10^7)(0.3241 \text{ cal/ft-lb})$$
$$W_c = 0.38665 \times 10^7 \text{ calories}$$

Mass of TNT $= m_{TNT} = 0.38665 \times 10^7/1120$ cal/gmTNT
$= 3452$ gm TNT
lb TNT $= 3452/453$ gm/lb $= 7.610$ lb TNT

Using scaling: $Z = 150/(7.61)^{1/3} = 76.25$ ft/lb$^{1/3}$

(See following paragraph.)

Reading chart, Figure 7-59

Overpressure, $p^o = 0.37$ psi at a distance of 150 feet.

This overpressure would produce some glass breakage, ceiling damage, and minor structural damage. See Damage Tables 7-24 and 7-25A and B.

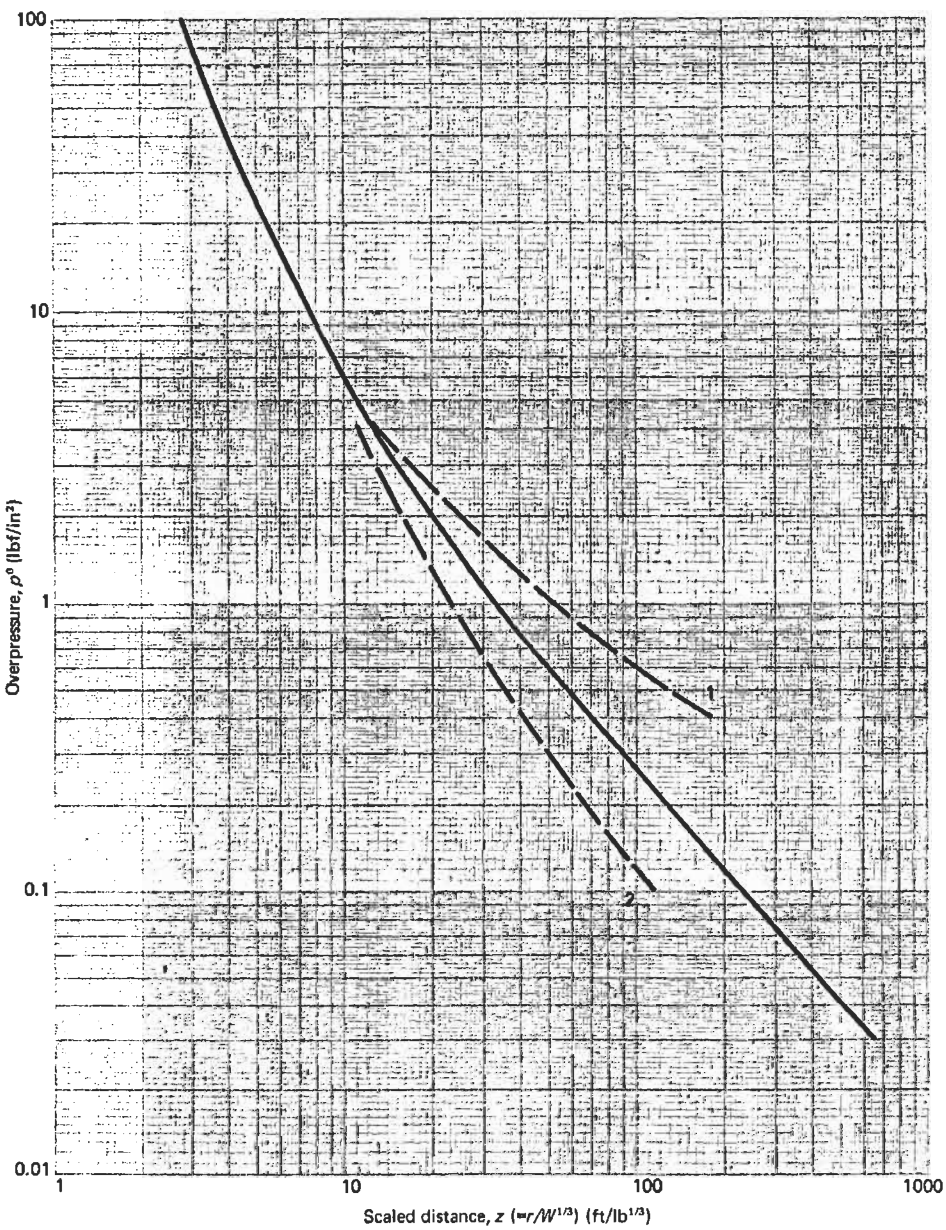


Figure 7-59. Peak overpressure vs. scaled distance for a blast wave from an explosion of TNT. By permission of the publishers, Butterworth-Heinemann, Ltd., Lees, F. P., *Loss Prevention in the Process Industries*, Vol. 1, p. 574 [40].

Blast Scaling

The Universal Hopkinson-Cranz and Sachs Laws of Blast Scaling have both been verified by experiment. These laws state that self-similar blast (shock) waves are produced at identical scaled distances when two explosive charges of similar geometry and the same explosive composition, but of different size, are detonated in the same atmosphere [49].

The scaled distance Z, which is a proportionately constant, is:

$$Z_{TNT} = \frac{R}{m_{TNT}^{1/3}}, \text{ for TNT only, ft/lb}^{1/3} \quad (7\text{-}61)$$

$$Z = R/W_c^{1/3}, \text{ ft/lb}^{1/3} = R_o/W_o^{1/3} \quad (7\text{-}62)$$

or $R_{exp} = R_o\lambda$

where R_{exp} = distance from the center of the explosion source to the point of interest, ft
m_{TNT} = mass of TNT, lb
Z_{TNT} = scaled distance to the point of interest, feet/(lb)$^{1/3}$
W_c = explosive charge weight, lb
λ = yield factor = $(W/W_o)^{1/3}$

Subscript, o, refers to reference value.

When accounting for effective charge weight at ground reflection the conversion is [49]:

$$W_e = 1.8W_c \quad (7\text{-}63)$$

W_e = effective charge weight in pounds of TNT for estimating surface burst effects with free air.

It is assumed that the energy released is proportional to the mass of a specific explosive [40].

At times it is necessary to have a feel for overpressure as it relates to shock front velocity [49]. (See Figure 7-60). Note especially that for a reasonable detonation velocity the peak overpressure could be in the range of 700 to 1000 psi and when referenced to Figure 7-60, the extent of industrial damage would be catastrophic. The use of scaled distance is illustrated in Ref. [41].

Example 7-15: Blast Scaling

Compare two different explosive charge weights of the same material. For an observed overpressure of 40 psi from a specific charge using the scaling equation above, the scaled distance is $Z = 5$ ft/lb$^{1/3}$. What is the distance for an overpressure of 40 psi with a charge of 500 lb?

$$Z = 5 = 5/(1 \text{ lb})^{1/3} = x/(500 \text{ lb})^{1/3}$$

$$x = (500)^{1/3}(5) = 7.93(5) = 39.6 \text{ ft}$$

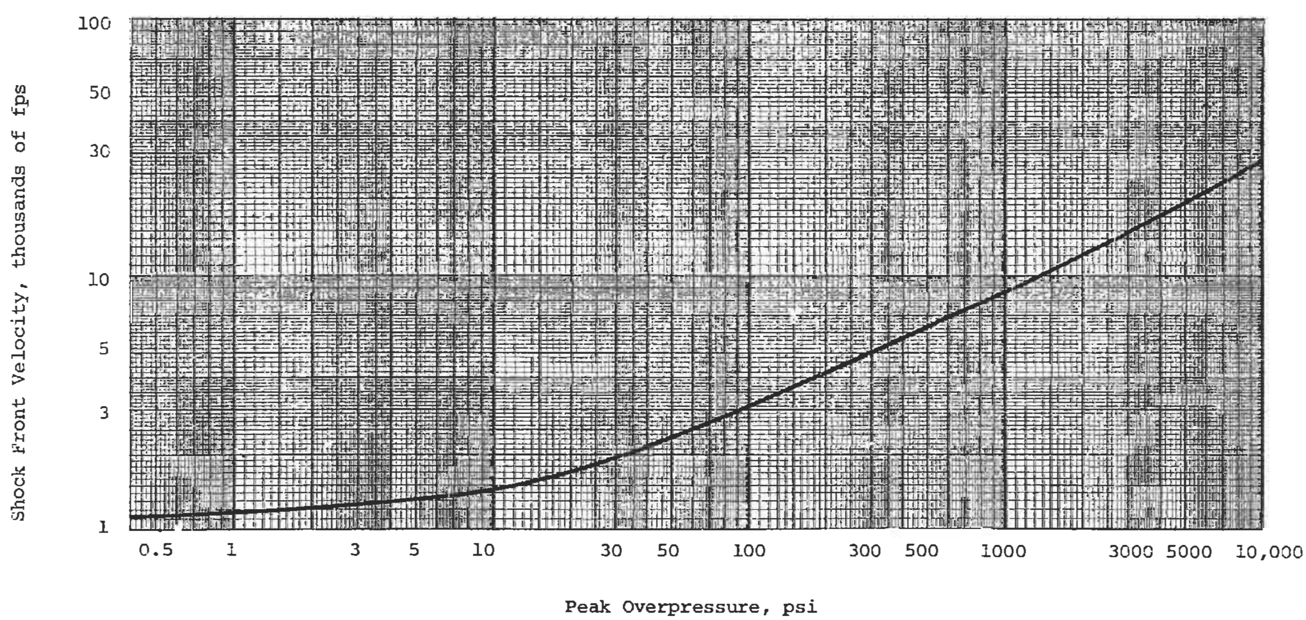


Figure 7-60. Shock front velocity as a function of peak overpressure at sea level. By permission, Report HNDM-1110-1-2, U.S. Army Corps of Engineers, 1977, Huntsville, AL [49].

The distance in which a 500 lb charge will develop a 40 psi overpressure is approximately 40 ft.

Example 7-16: Estimating Explosion Damage

An overpressure after an explosion is noted as 0.5 psi. The calculated scaled distance Z is 75 ft/(lb)$^{1/3}$. Thus for a one pound charge, windows are broken at a distance of 75 feet. How far will windows be broken for a 500 lb. charge?

$$Z = 75 = 75/(1)^{1/3} = x/(500)^{1/3}$$

$$x = 7.937\ (75) = 595 \text{ ft, windows broken for a 500 lb change}$$

In general, a reflected shock wave of 55 psi on a human for 400 milliseconds would be just about the tolerance limit [41] (see Table 7-25B). For a more detailed discussion of blast scaling and overpressure, see Ref [40].

Explosion Venting for Gases/Vapors (Not Dusts)

Unless there is sufficient explosion data for a specific chemical system, including air-mixing or run-away reactions, very few devices can be effective in relieving a confined vessel explosion other than a carefully designed rupture disk, with a good factor of overcapacity. Because detonation explosions initiate and travel so fast (see previous tables/charts), there is a limited chance to relieve the overpressure. Some fast microsecond electronic detectors based on rate of pressure rise or rate of temperature rise can and have been used to anticipate runaway reactions and thus create an "early" anticipated release or opening of the relieving device. When such explosive conditions are anticipated, special studies such as the DIERS technology [51, 67] can be justified to provide information for a safe response and control of an explosion. Other methods such as injection of inerting or reaction suppressants have proven to be beneficial (see Figures 7-61 and 7-62).

Bleves (Boiling Liquid Expanding Vapor Explosions).

This particular type of explosion is less known and understood, but nevertheless is an important type for damage consideration. This is a type of pressure release explosion and there are several descriptions.

Kirkwood [30] describes Bleves referenced to flammable liquids as occurring when a confined liquid is heated above its atmospheric boiling point by an external source of heat or fire and is suddenly released by the rupture of the container due to overpressurization by the expanding liquid. A portion of the superheated liquid immediately flashes to vapor and is ignited by the external heat source (see [81]).

NFPA [34] contains extensive descriptions of Bleves (also see [82]) and describes them in summary as paraphrased here with permission: liquefied gases stored in containers at temperatures above their boiling points at NTP will remain under pressure only as long as the container remains closed to the atmosphere. If the pressure is suddenly released to atmosphere due to failure from metal overstress by external fire or heat, corrosion penetration, or external impact (for examples), the heat stored in the liquid generates very rapid vaporization of a portion of the liquid proportional to the temperature difference between that of the liquid at the instant the container fails and the normal boiling point of the liquid. Often this can generate vapor from about one-third to one-half of the liquid in the container. The liquid vaporization is accompanied by a large liquid to vapor expansion, which provides the energy for propagation of vessel cracks, propulsion of pieces of the container, rapid mixing of the air and vapor resulting in a characteristic fireball upon ignition by the external fire or other source that caused the Bleve to develop in the first place, with atomization of the remaining cold liquid. Often the cold liquid from the vessel is broken into droplets that can burn as they fly out of the vessel. Often this cold liquid can escape ignition and may be propelled one-half mile or more from the initial site. In most Bleves, the failure originates in the vapor space above the liquid, and it is this space that is most subject to external overheating of the metal.

The transportation industry is subject to more impact failures of vessels that lead to Bleves. The Bleve occurs simultaneously with the impact in most recorded cases, but not all. Sometimes there is a delay due to lack of total penetration of the vessel with a hole or crack, and time is needed for the temperature in the container to rise. Failures of the vessel can lead to fireballs of more than several hundred feet in diameter.

The application of water externally to the vapor space of a vessel or application of insulation can often protect against Bleves.

A relief valve will not usually handle the vapor generated because its set pressure is usually higher than the boiling point pressure created by the hole or crack in the vessel; therefore, it will not relieve at the lower pressure.

Ref. [40] points out that the effects of a Bleve depends on whether the liquid in the vessel is flammable. The initial explosion may generate a blast wave and fragments from the vessel. For a flammable material, the conditions described in Ref. [34] above may result, and even a vapor cloud explosion may result.

Liquid Mist Explosions

When a flammable liquid is sprayed as fine droplets into the air, a flammable mixture can result, which may burn or explode. The mist or spray may be formed by condensation of saturated vapors or by mechanical means [40]. As the particle sizes of the liquid become greater than 0.01 mm diameter, the lower flammability limit of the material becomes lower; while above 0.01 mm, the LEL is about the same as the vapor. Mechanical engine crankcase explosions of oil mist in air are hazardous, and current practice is to apply explosion relief valves to the crankcase.

Compressed air system explosions in engines, pipelines, separators, etc., are characteristic of this same type of mist explosion.

Relief Sizing: Explosions of Gases and Vapors

The National Fire Protection Association has extensive codes that relate to fire and explosion prevention and protection for all major industries and/or occupations, for example [10, 26, 27, 33, 55]. NFPA Code No. 69 [55]

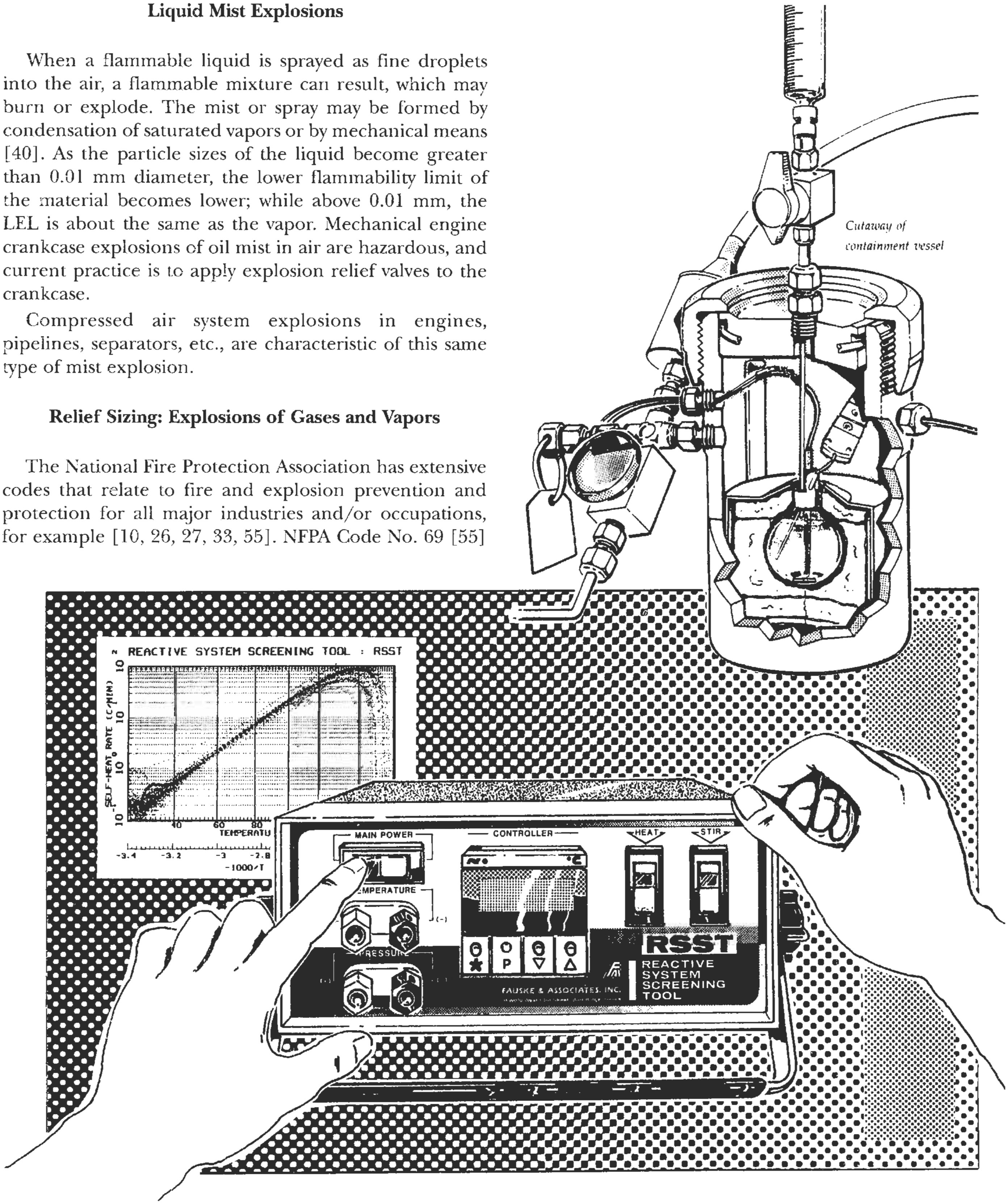


Figure 7-61. Reactive system screening tool (RSST) for evaluating runaway reaction potential. By permission, Fauske and Associates, Inc.

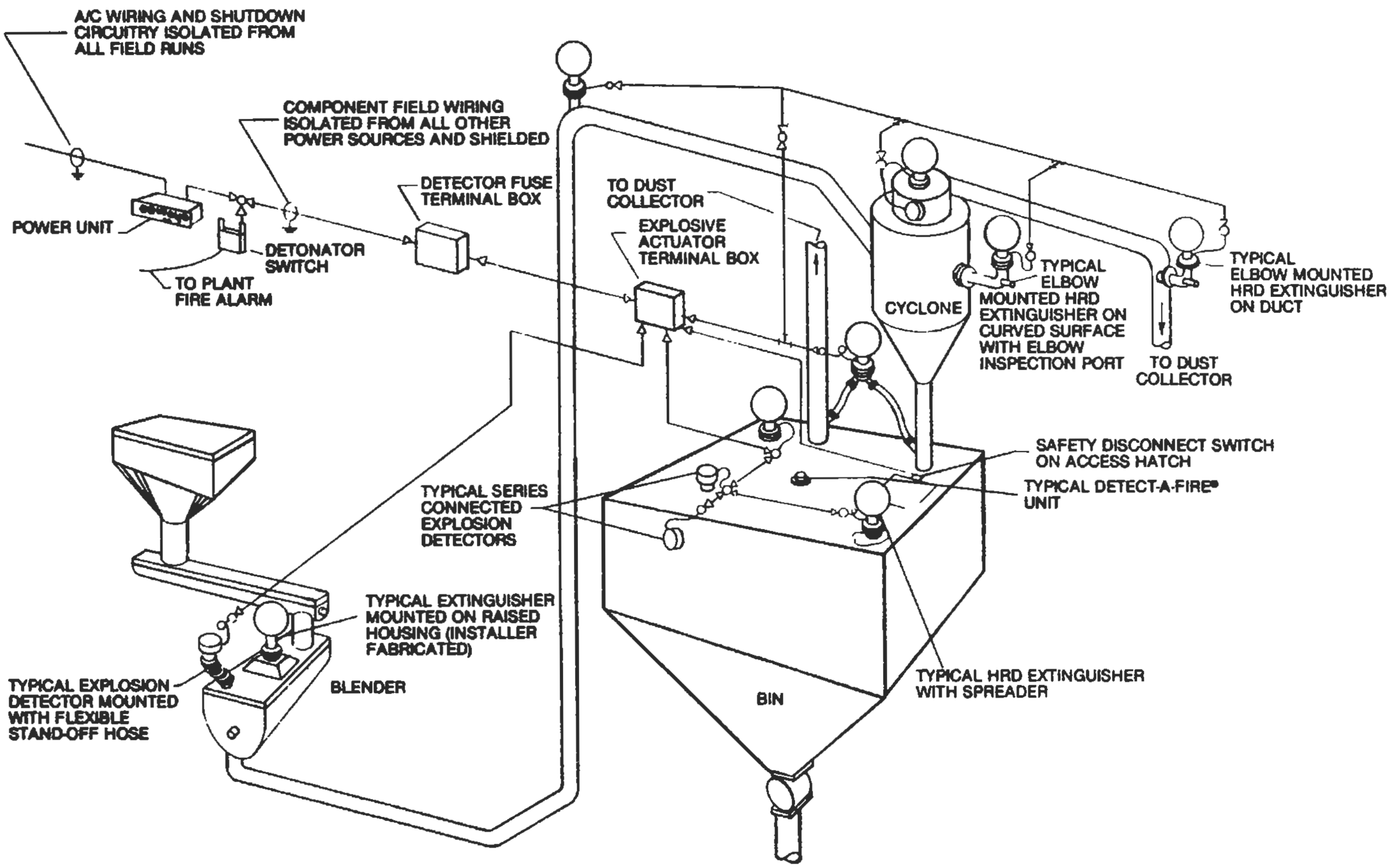


Figure 7-62. Typical Fenwal explosion suppression system. By permission, Fenwal Safety Systems, Inc.

relates to explosion prevention systems and the "design, construction, operation, maintenance and testing of systems for the prevention of deflagration explosions" and is a valuable document for designers. One significant requirement of this code relates to establishing the design pressure of a ASME code pressure vessel if the vessel is to possibly be subject to and contain a deflagration and its explosion pressure internally:

From NFPA-69 [55], extracted by permission. Design Pressure:

$$P_r = \frac{1.5\,[R\,(P_i + 14.7) - 14.7]}{F_u}, \text{ psig} \tag{7-64}$$

$$P_d = \frac{1.5\,[R\,(P_i + 14.7) - 14.7]}{F_y}, \text{ psig} \tag{7-65}$$

where P_r = the design pressure to prevent (vessel) rupture due to internal deflagration, psig

P_d = the design pressure to prevent deformation due to internal deflagration, psig

P_i = the maximum initial pressure at which the combustible atmosphere exists, psig

R = the ratio of the maximum deflagration pressure to the maximum initial pressure, as described in Code Par 5-3.3.1

F_u = the ratio of the ultimate stress of the vessel to the allowable stress of the vessel

F_y = the ratio of the yield stress of the vessel to the allowable stress of the vessel

Note: 1 psi = 6.89 kPa.

5-3.3.1 The dimensionless ratio R is the ratio of the maximum deflagration pressure, in absolute pressure units, to the maximum initial pressure, in consistent absolute pressure units. For gas/air mixtures, R shall be taken as 9.0; for dust/air mixtures, R shall be taken as 10.0.
Exception: A different value of R may be used if appropriate test data or calculations are available to confirm its suitability.

5-3.3.2 For operating temperatures below 25°C, the value of R shall be adjusted according to the following formula:

$$R' = R\,[298/(273 + T_i)] \tag{7-66}$$

where R is either 9.0 or 10.0 and T_i is the operating temperature in C.

5-3.3.3 For vessels fabricated of low carbon steel and low alloy stainless steel $F_u = 4.0$ and $F_y = 2.0$.

5-3.4 The presence of any pressure relief device on the system shall not cause the design pressure calculated by 5-3.3 to be reduced.

5-3.5* For systems handling gases and liquids, the maximum initial pressure, P_i, shall be the maximum pressure at which a combustible mixture can exist, but not higher than the setting of the pressure relief device plus its accumulation. For systems handling dusts, this maximum initial pressure shall be the maximum possible discharge pressure of the compressor or blower that is suspending or transporting the material or the setting of the pressure relief device on the vessel being protected plus its accumulation, whichever is greater. For gravity discharge of dusts, the maximum initial pressure shall be taken as 0.0 psig (0.0 kPa gage).

5-3.6 For systems operating under vacuum, the maximum initial pressure shall be taken as no less than atmospheric pressure (0.0 psig or 0.0 kPa gage).

5-3.7 The vessel design pressure shall be based on the wall thickness of the vessel, neglecting any allowance for corrosion or erosion.

5-3.8 The design must take into consideration the minimum operating temperature at which a deflagration may occur. This minimum temperature must be compared with the temperature characteristics of the material of construction of the vessel to ensure that brittle fracture will not result from a deflagration.
Note: * refers to Appendix A of the NFPA Code.

The NFAA-No. 68 [27] presents a design procedure for venting deflagrations within an enclosure in order to minimize the structural and mechanical damage. This deflagration may result from ignition of a combustible gas, vapor, mist or dust, but is not necessarily considered a detonation, because it acts so much more rapidly and with greater force than the deflagration. It does not apply to bulk autoignition of gases or unconfined deflagrations such as open-air or vapor cloud explosions. It also is not applicable to process situations where internal pressure develops from fire external to the vessel (see NFPA 30 [71] Flammable and Combustible Liquids Code) nor to runaway reactions.

Note that the venting design may not necessarily prevent a deflagration, but is intended to relieve the overpressure developed (see reference [72]).

Table 7-27
Fuel Characteristic Constant for Venting Equation

Fuel	$C(psi)^{1/2}$	$C(kPa)^{1/2}$
Anhydrous Ammonia	0.05	0.13
Methane	0.14	0.37
Gases with fundamental burning velocity less than 1.3 times that of propane	0.17	0.45
St-1 dusts	0.10	0.26
St-2 dusts	0.12	0.30
St-3 dusts	0.20	0.51

Supporting material on explosion protection methods against dust explosions is available for review at NFPA Headquarters, Batterymarch Park, Quincy, MA 02269.
Reprinted with permission, NFPA Code 68, *Venting of Deflagrations* (1988) National Fire Protection Association, Quincy, MA 02269 [27].
Note: This reprinted material is not the official position of the National Fire Protection Association on the referenced subject which is represented only by the standard in its entirety.

Vent or Relief Area Calculation [27] for Venting of Deflagrations in Low-Strength Enclosures

Low-strength enclosures not withstanding more than 1.5 psig (not applicable to end of elongated enclosure). Applicable more to rooms, buildings, and certain equipment enclosures (see Figure 7-8R).

$$A_v = CA_s/(P_{red})^{1/2}, \text{ sq ft} \tag{7-67}$$

where A_v = vent area, sq ft
A_s = internal strength-containing surface area of enclosure (not suspended ceilings, etc.) sq ft
C = venting equation constant, Table 7-27 NFPA-68 pp. 68–14
P_{red} = maximum internal overpressure that can be withstood by the weakest structural element, psi

For elongated enclosures, vent area should be applied evenly relative to the longest dimension. Length-to-diameter should not exceed 3 [27].

For other length-to-diameter ratio, refer to Ref. [27]. For cross sections other than circular or square, use the hydraulic diameter:

$$4A/P_{er}$$

where A = cross section area and P_{er} is the perimeter of the cross section.

For vent area limited or restricted to one end of an elongated enclosure (vertical tank, silo, etc.), the venting equation is limited to:

$$L_3 \leqq 12\ A/P_{er},\ \text{ft} \tag{7-68}$$

where L_3 = longest dimension of the enclosure, ft
A = cross section area, sq ft
P_{er} = perimeter of cross section, ft

For highly turbulent gas mixing in an enclosure and the vent area is restricted to one end of the elongated enclosure, ratio of length-to-diameter should not exceed 2, or

$$L_3 \leqq 8\ A/P_{er},\ \text{ft} \tag{7-69}$$

For other conditions, refer to the NFPA-68 Code. For the above relations to apply, the constant, C, should be referred to fuels having the characteristics of one of those in Table 7-27.

Ref. [27] presents a thorough discussion of limits to structural components strengths, and these should be observed. Ductile design practices should be used. The maximum allowable design stress should not exceed 25% of the ultimate strength. The strength of the enclosure should exceed the vent relief pressure by at least 0.35 psi.

The vent design must provide at least the area required to satisfy the volume of the enclosure (see NFPA-68) [27].

Example 7-17: Low Strength Enclosure Venting

Design to protect a large warehouse containing plastic materials that can emit ethylene and propylene vapors. The dimensions on the rectangular building with a flat roof are:

100 ft long × 50 ft wide × 20 ft tall.

The building design has been selected as good for 0.4 psi overpressure. See Table 7-23 for glass window shattering. Use known design figures when available.

Areas are:

Floor = 100 × 50	=	5,000 sq ft
Roof = same	=	5,000 sq ft
2 ends = 2 × 50 × 20	=	2,000 sq ft
2 sides = 2 × 100 × 20	=	4,000 sq ft
		16,000 sq ft

Vent area:

$$A_v = CA_s/(P_{red})^{1/2} \tag{7-67}$$

From Table 7-27:

$$C = 0.17\ (\text{psi})^{1/3}$$

$$P_{red} = 0.40 = 0.40\ \text{psi}$$

$$A_v = 0.17\ (16{,}000)/(0.40)^{1/2} = 4{,}300\ \text{sq ft}$$

This will require roof bursting panels, side wall bursting panel, or an end panel that could be hinged to blow out. The panel relieving pressure should be set for 0.40 psi − 0.35 psi = 0.05 psi to burst or relieve per code.

High Strength Enclosures for Deflagrations

This section and Chapters 6 and 7 of the code [27] apply to vessels and equipment capable of withstanding more than 1.5 psig internal pressure. These design procedures do not apply to a detonation that is not believed to be capable of being vented successfully [27].

The vent devices used to relieve the overpressure from the deflagration must be structurally sound, low in weight, and should not fragment to form missiles when the force hits the device.

The discharge from high pressure vessels must be vented out of the building to avoid fires and explosions and overpressure in the building, and the backpressure or pressure loss through the vent duct/pipe must be recognized as affecting the relieving pressure of the vessel. Never locate a vent duct discharge to atmosphere in an area where the discharge might be drawn into (a) fresh air intake to a building or, (b) fresh air intake to a compressor or gas fired engine.

Rupture disks when properly sized and located on the potentially overpressure vessel have been shown to provide the best protection for a deflagration but not a detonation [54].

Determination of Relief Areas for Deflagrations of Gases/Vapors/Mists in High Strength Enclosures

The nomographs of Figures 7-63 A, B, C, and D [27] were developed by Bartknecht [54] for the conditions of:

- no turbulence in vessel at time of ignition
- low ignition energy of 10 joules or less
- atmosphere pressure

To utilize the charts Figures 7-63A thru D, enter volume, read up to selected P_{red} value and across to vent pressure P_{stat}, and down to vent area required.

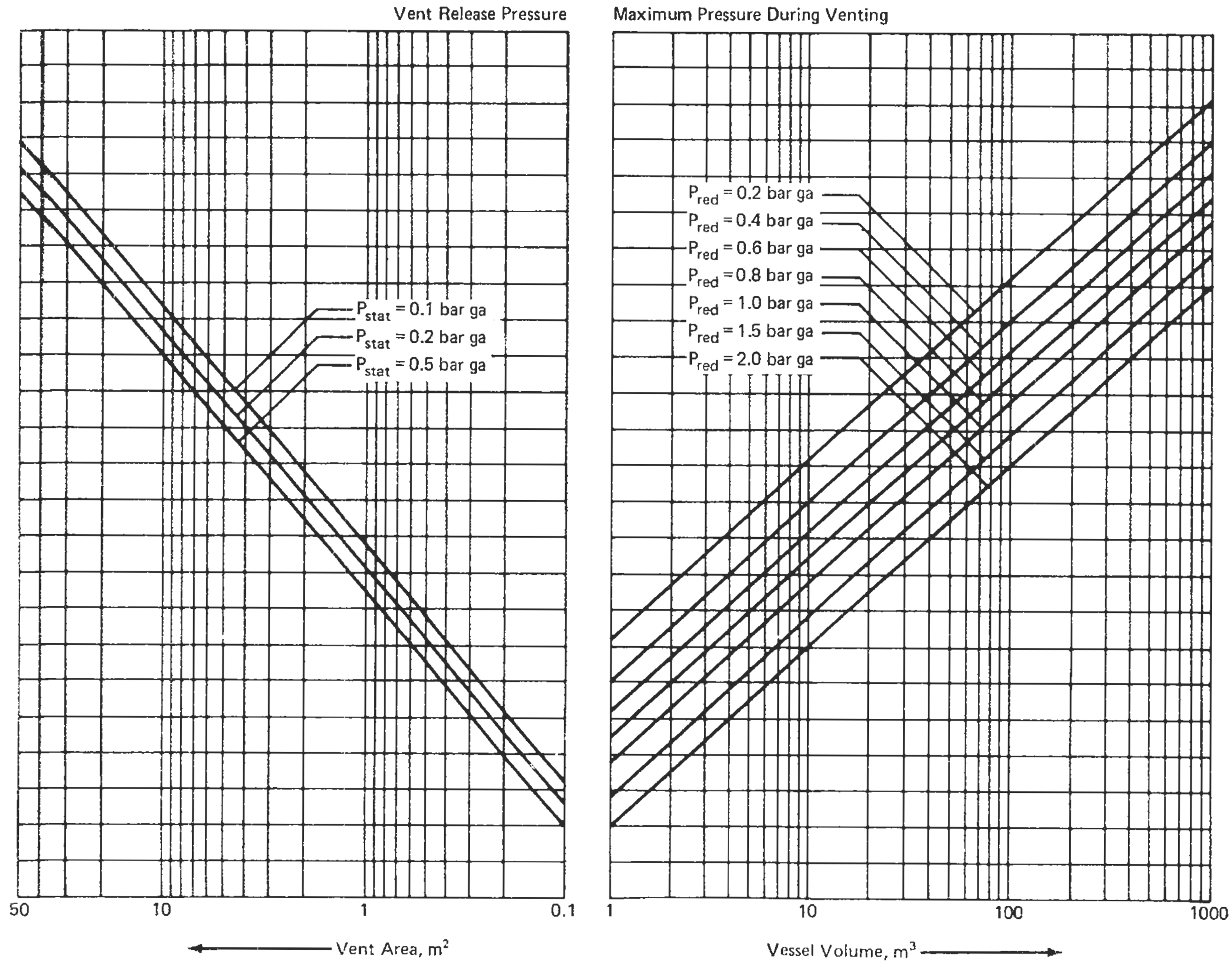


Figure 7-63A. Venting nomograph for methane. Reprinteed with permission, NFPA 68-1988, *Deflagration Venting,* (1988) National Fire Protection Association, Quincy, MA 02269. Note: This material is not the complete and official position of the National Fire Protection Association on the referenced subject, which is represented only by the standard in its entirety. Note from this author: this statement applies to all material referenced for use that originates with the National Fire Protection Association [27].

In order to calculate the same area results as the above noted charts for methane, propane, coke gas, and hydrogen, the following equation is presented by NFPA-68 [27]:

$$A_v = a\,(V)^b\,(e^{c(P_{stat})})\,(P_{red})^d \tag{7-70}$$

when
- A_v = required vent area, sq meters
- V = enclosure volume, cubic meters
- e = 2.718 (base natural log)
- a, b, c, d = coefficients, see Table to follow
- P_{red} = maximum pressure developed during venting, bar ga.
- P_{stat} = vent closure release pressure, bar ga.

***Coefficients/exponents [27]**

	a	b	c	d
Methane	0.105	0.770	1.230	−0.823
Propane	0.148	0.703	0.942	−0.671
Coke Gas	0.150	0.695	1.380	−0.707
Hydrogen	0.279	0.680	0.755	−0.393

* Do not use for extrapolation beyond the nomographs.

The nomographs (Figures 7-63A–D) apply for vessel/equipment with length/diameter ratio (L/D) of 5 or less, otherwise, a danger of detonation may exist. For equipment with L/D > 5, refer to NFPA-68, Chapter 8 [27], and NFPA-69 [55].

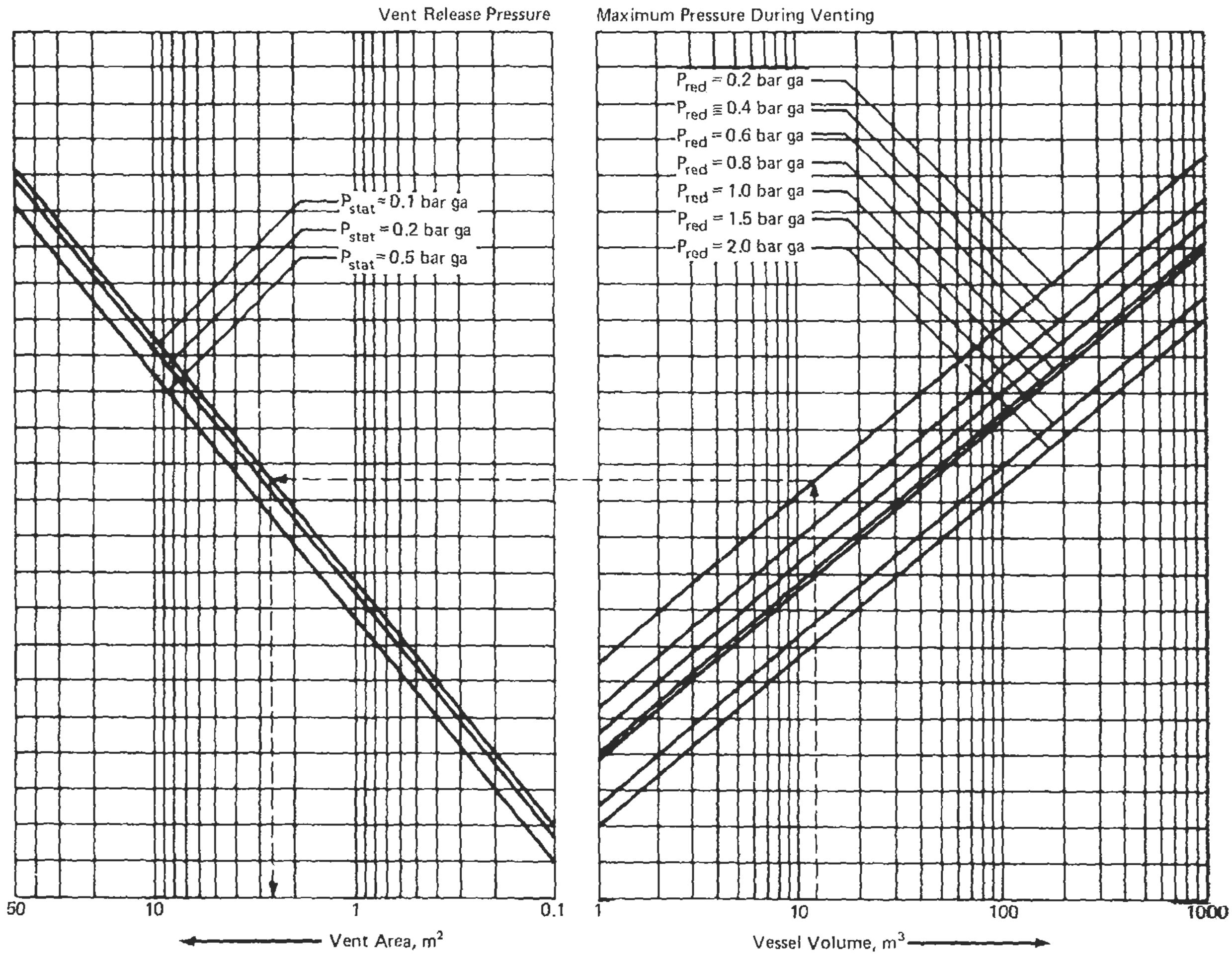


Figure 7-63B. *Venting nomograph for propane. Reprinted with permission, NFPA 68-1988, Deflagration Venting, (1988) National Fire Protection Association, Quincy, MA 02269. See note Figure 7-63A.*

For systems without test data or literature data

(1) When test data for a particular gas-air system is not available, the nomograph for hydrogen air system, Figure 7-63D, can be conservatively used. Ref. [27] reports the additional vent area determined in this manner will usually be small.

(2) Interpolation of existing nomographs to determine deflagration vent area of different gas-air mixtures: A_{ng} = area required for new gas or mixture.

Assume vessel volume = V′
Establish*, maximum allowable value for P_{red} = x
Establish**, P_{stat} = y
Maximum rate of pressure rise, K_G, for gas in question in a specific vessel = z bar/sec.

Note: K_G values are not constant and vary with test conditions. K_G provides means of comparing maximum rates of pressure rise for various gases, but should only be used for deflagration vent sizing if test data comes from test vessels of similar configuration and same type of ignitor and ignition energy [27]

*Must not exceed mechanical strength of vessel; use design or MAWP.
**Must be lower than x or not more than 10% above MAWP.

Refer to the propane and hydrogen nomographs (Figures 7-63B and 7-63D). The required vent area to protect the vessel specified from each of the charts will be a and b, square meters, respectively.

The maximum rates of pressure rise for propane and hydrogen are r and r′, respectively, bar/sec in the same vessel. For linear interpolation:

Required vent area for new gas =

$$[a + \left[\frac{(z - r)}{(r' - r)}\right](b - a)] = \text{vent area, sq meters} \qquad (7\text{-}71)$$

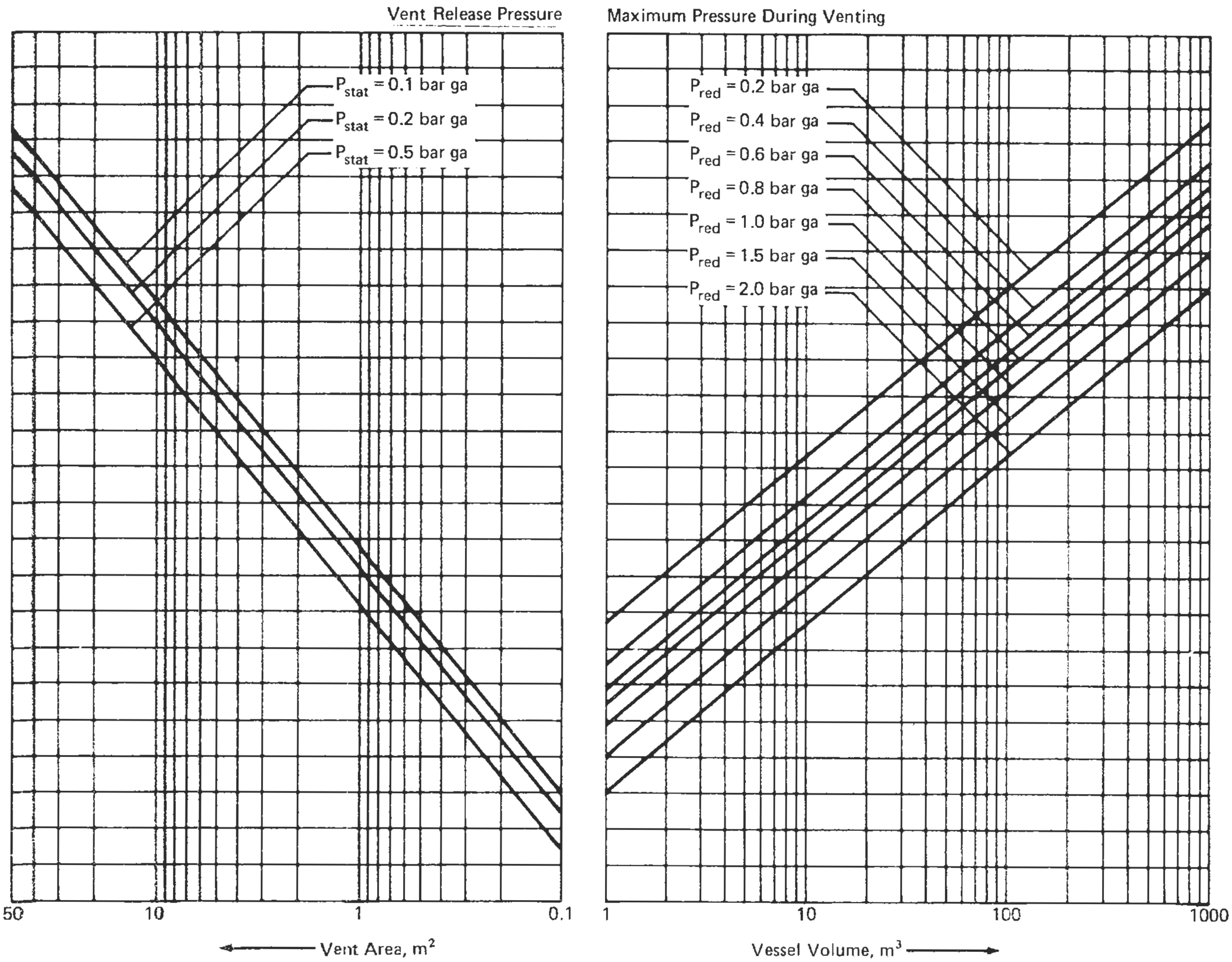


Figure 7-63C. Venting nomograph for coke gas. Reprinted with permission, NFPA 68-1988, *Deflagration Venting,* (1988) National Fire Protection Association, Quincy, MA 02269. See note Figure 7-63A.

Using the cubic equation presented earlier, Bartknecht [54] developed for vessels of different sizes for the same process system; in closed or vented vessels, valid for flammable gases and combustible dusts:

$$A_2 = \left[\left(\frac{V_1^{1/3}}{V_2^{1/3}}\right)\left(\frac{A_1}{V_1}\right)\right](V_2) = \left[\frac{V_1^{1/3}}{V_2^{1/3}}\right](f)(V_2)$$
$$(f_1)(V_2) \qquad (7\text{-}72)$$

where A_1 = initial vessel relief area, sq meter
A_2 = second vessel relief area, sq meter
V = volume of vessels 1 and 2, respectively, cu meters
f = specific relief area, sq meter/cu meter, from F = f V, and $f_1 V_1^{1/3} = f_2 V_2^{1/3}$ at same P_{stat} and P_{red}

When the relief device relieves, the explosion pressure falls off, but then it increases faster than in the beginning due to the opening of the relief when the flame front is distorted creating an acceleration of the combustion process [54]. Thus, there are two pressure peaks in the course of the relieving: (1) at the activation of the relief device due to pressure buildup in the vessel, and (2) at the end of the combustion reaction. The first pressure is always greater than the second. The second pressure rise is created by turbulence during the venting process [54].

K_g or K_{st} values, which are a measure of the characteristic for the course of an explosion (gas or dust) of vessels roughly cubic in shape, are termed deflagration indices.

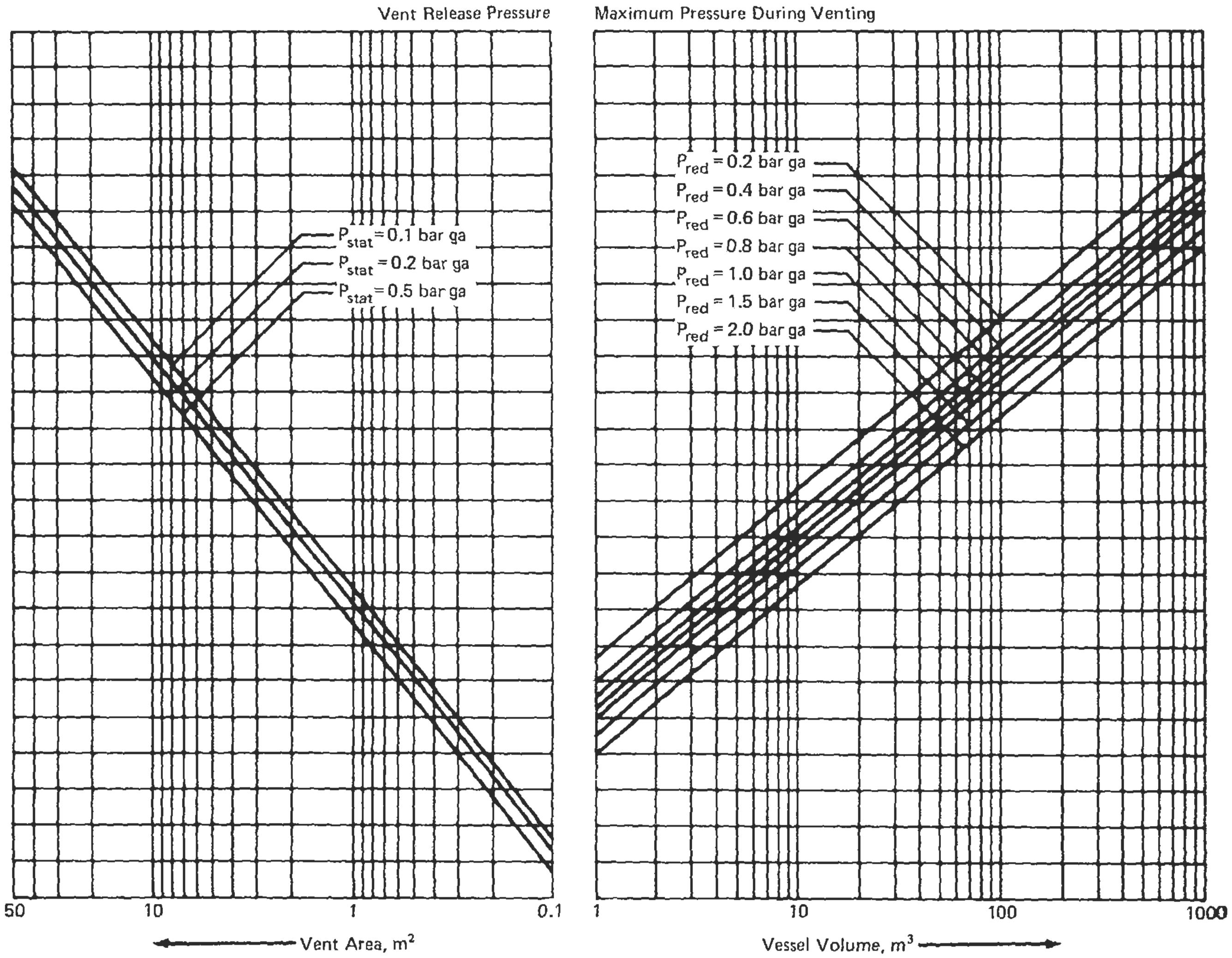


Figure 7-63D. *Venting nomograph for hydrogen.* Reprinted with permission, NFPA 68-1988, ***Deflagration Venting,*** (1988) National Fire Protection Association, Quincy, MA 02269. See note *Figure 7-63A.*

A few selected values for zero turbulence are [54]:

K_G, Bar meter/sec (average values)	
Methane	55
Propane	75
Coke Gas	140
Hydrogen	550

The nomograms, Figure 7-63A through D, are based on an operating pressure of 1 bar (absolute), but may be used without correction up to 1.2 bar absolute [54]. There is insufficient pressure data available to recommend using much above normal pressure. When the operating pressure is raised above normal, the reduced explosion pressure will show a proportional increase for a given constant relief venting area [54]. The nomographs are not applicable for detonations.

When systems involving solvent vapor are considered, use the nomogram for propane/air mixtures because most *common* solvent vapors have maximum explosion pressures of 7.1 to 7.6 bar, and the K_G falls between 40 and 75 bar meter/sec (see Ref. [54]).

The effects of turbulence must be taken into account when sizing a relief area. For example, the explosion violence of turbulent methane-air mixture is comparable to that of zero turbulence of hydrogen-air mixtures. From the investigations [54], the nomograms from Figure 7-63 can be applied for turbulent gas mixtures under the following conditions [54]:

- For large relief areas: $P_{red} \leqq 1$ bar, valid but influence of ignition energy must be considered.

- For small relief areas, $P_{red} > 1$ bar, independent of ignition energy. Relief area (vent area, A) from nomograms must be increased by a constant factor, $\Delta A = 0.08V^{2/3}$.

Recommend applying only for methane, propane, and solvent vapors due to insufficient data and for a static activation pressure of the relief device of $P_{stat} \leqq 0.1$ bar (see reference [54]).

Vessels with internal components may be susceptible to turbulence in the gas mixture, which can lead to detonations, which are not covered by this procedure. Hydrogen is particularly vulnerable to detonations; therefore, for such systems, a specialized expert should be consulted. The details of NFPA-68 should also be consulted as there are many factors that must be recognized. Also see reference [54].

The range of the nomographs can be extrapolated or extended for a specific vessel volume by cross-plotting as shown in Figure 7-64, limiting to a constant P_{red} for each chart at varying P_{stat}. Do not extrapolate below P_{stat} of 0.05 bar ga, nor below P_{red} of 0.1 bar ga. P_{red} should not be extrapolated above 2.0 bar ga; P_{stat} can be extrapolated but it must always be less than P_{red} by at least 0.05 bar [27].

Dust Explosions

It has been somewhat of a surprise to many engineers that fine dust particles are combustible and will explode

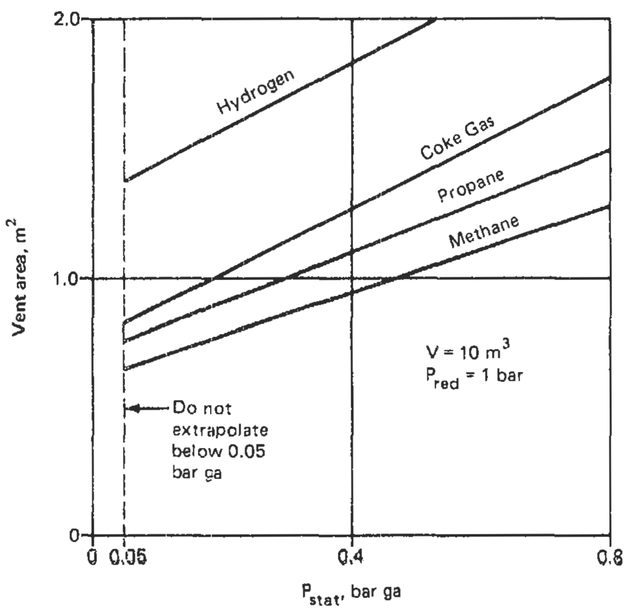


Figure 7-64. Extrapolation of nomographs for gases. Reprinted with permission, NFPA 68-1988, *Deflagration Venting,* (1988) National Fire Protection Association, Quincy, MA 02269. See note Figure 7-63A.

for many specific dust types, and the same laws of flammability apply as were previously presented for flammable gases/vapors, following the "cubic law" [27] for explosive violence [54]:

$$(dp/dt)_{max,V1}\,(V_1)^{1/3} = (dp/dt)_{max,V2}\,(V_2)^{1/3} \tag{7-73}$$
$$= K_G \text{ (gases), bar m/sec}$$
$$\text{or } K_{st} \text{ (dust), bar m/sec}$$

$$(dp/dt)_{P_{red},V1}\,(V_1)^{1/3} = (dp/dt)_{P_{red},V2}\,(V_2)^{1/3} \tag{7-74}$$
$$= (K_G)_{P_{red}} \text{ or } (K_{st})P_{red}\text{, bar m/sec}$$

when subscript G = for gas systems
st = for dust system
f = specific relief area
$(dp/dt)_{max}$ = maximum rate of pressure rise, bar/sec
$f_1\,(V_1)^{1/3} = f_2\,(V_2)^{1/3}$

Figures 7-65A through H are the venting area requirements for dust explosions. These are based on high energy ignition sources.

An equation from Ref. [27] represents the dust explosion nomographs, Figures 7-65A, B, and C. Because the equation was derived from the nomographs, it is no more accurate but may be more convenient:

$$A = (a)\,(V^{2/3})\,(K_{st})^b\,(P_{red})^c \tag{7-75}$$

where
$a = 0.000571\ e^{(2)\,(Pstat)}$
$b = 0.978\ e^{(-0.105)\,(Pstat)}$
$c = -0.687\ e^{(0.226)\,(Pstat)}$
A_v = vent area, sq meters
V = enclosure volume, cu meters
e = 2.718, natural logarithm
P_{red} = maximum pressure developed during venting, bar ga.
P_{stat} = vent closure release pressure, bar ga.
K_{st} = deflagration index for dust, bar m/sec., Table 7-28

Other equations are presented in Ref. [27] to represent the dust nomographs of Figures 7-65D through 65H.

Reference conversions:

1 - bar-meter/second	= 47.6 psi-ft/sec
1 - psi-ft/sec	= 0.021 bar-meter/sec
1 - atm	= 1.01 bars
1 - atm	= 14.7 psi

The dust hazard class, K_{st}, vessel volume and strength, and the relieving pressure of the vent closure are the key components of the relief determination using the nomographs [27]. Although stated by Ref. [27] to be non-exact

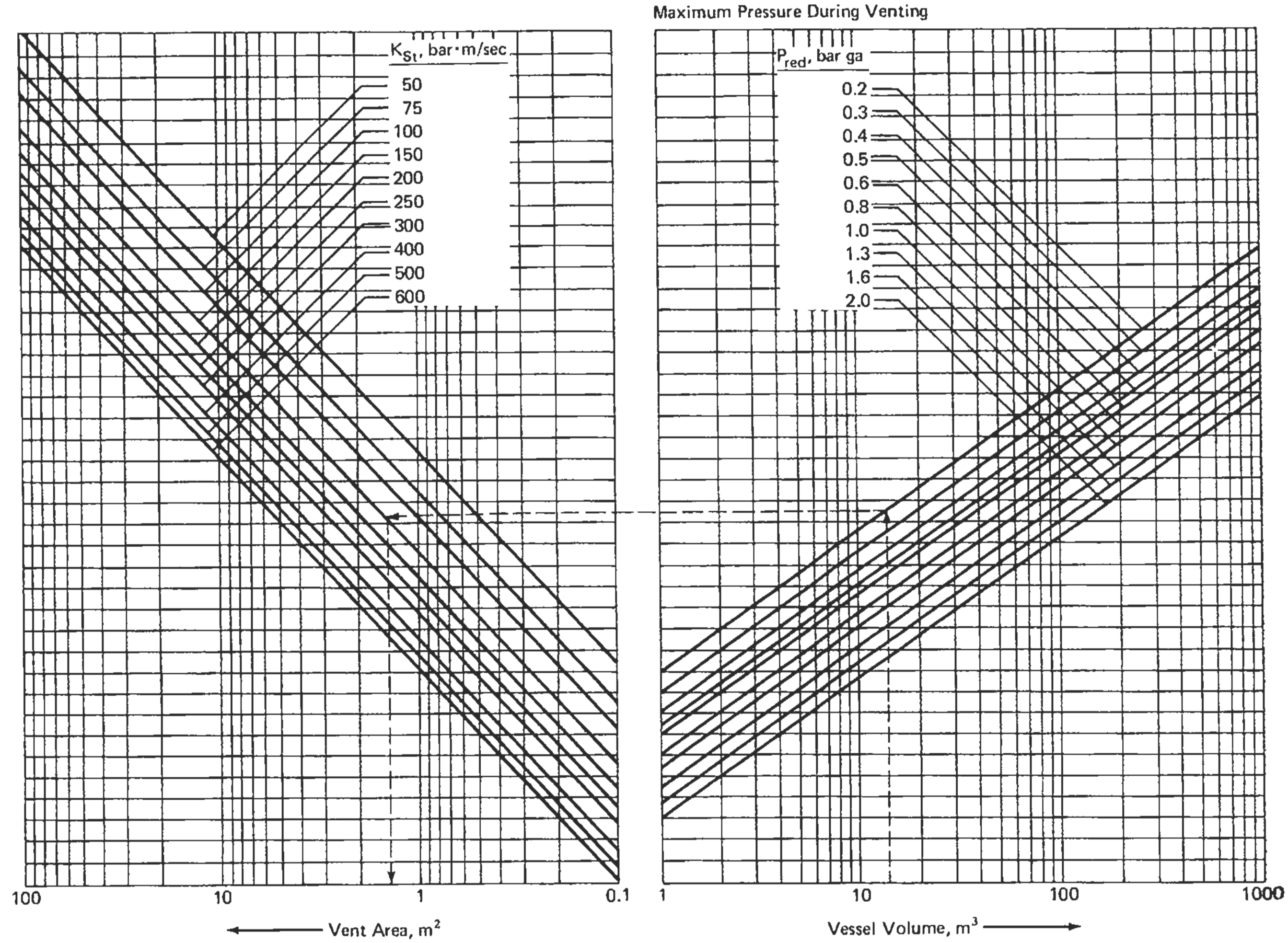


Figure 7-65A. Venting nomograph for dusts, P_{stat} = 0.1 bar ga. Reprinted with permission, NFPA 68-1988, *Deflagration Venting,* (1988) National Fire Protection Association, Quincy, MA 02269. See note Figure 7-63A.

but sufficiently accurate for industrial use, care must be used in selecting the conditions for design. Classification of a dust in a specific class is not assurance of the probability of a dust explosion [54].

The dust hazard classes for deflagrations are given in Table 7-28, with data in Tables 7-28 and 7-30A, B, C, D, and E.

Using a nomograph requires only the vessel volume in meters, selecting the dust class. St-1, St-2 or St-3 from Table 7-28. Using Tables 7-29 or 7-30 select the K_{st} value determined experimentally. The reduced pressure, P_{red}, (maximum pressure actually developed during a *vented* deflagration, termed reduced explosion pressure) must not exceed strength of vessel (see earlier discussion) and the P_{stat}, i.e., the vent device release pressure. Note that the static activation pressure, P_{st}, must be determined from experimental tests of the manufacture of relief panels such as rupture disks.

Example 7-18: Use of the Dust Nomographs

A storage silo for lignite particles (not lumps) is 10 ft diameter by 12 ft tall. What protection is needed to guard against destruction from a dust explosion (deflagration)?

From Table 7-30C, lignite is dust hazard class 1, with K_{st} = 151 bar-m/sec.

1. Vessel volume = π (10)^2 (12)/4 = 942.4 cu ft
   = 942.4/35.3 = 26.7 cu meters
   L/D = 12/10 = 1.2, length/diam. is ok.
2. P_{red}, bar ga = 2.0 = (2) (14.5) = 29 psig max. allowable vessel internal pressure. Note: 1 bar = 14.5 psi
3. Reading vent area (Figure 7-65D) = 0.70 sq meters
   = 0.70 × 10.8 sq ft/sq meter
   = 7.56 sq ft

select, P_{stat} = 0.1 bar ga, vent device release pressure
= (0.1) (14.5) = 1.45 psig.

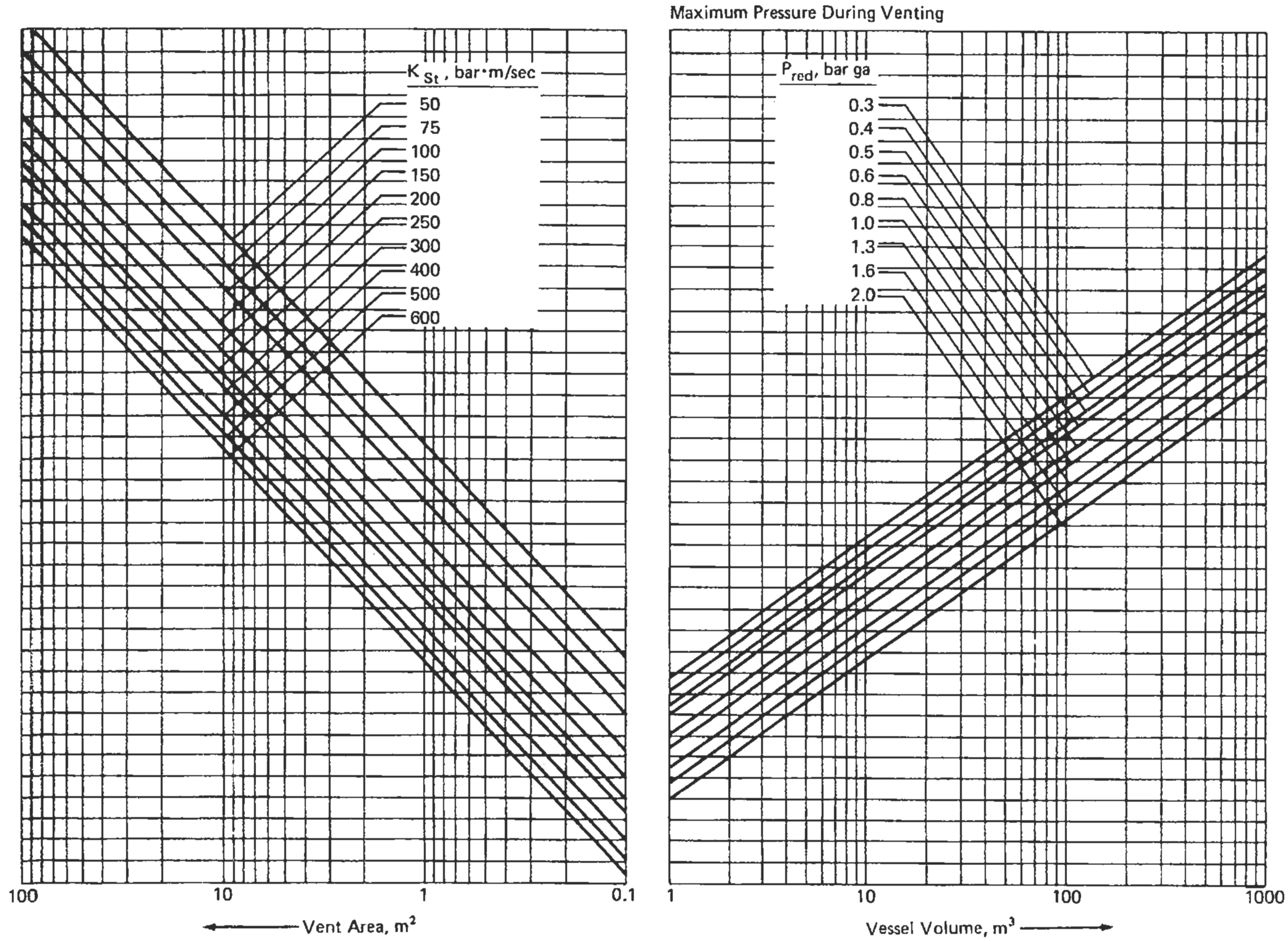


Figure 7-65B. Venting nomographs for dusts, P_{stat} = 0.2 bar ga. Reprinted with permission, NFPA 68-1988, *Deflagration Venting*, (1988) National Fire Protection Association, Quincy, MA 02269. See note Figure 7-63A.

Note: As P_{red} and P_{stat} change for a given class of dust, the vent area required will change. Examine charts.

The explosion characteristics of dusts of the specific type or identification varies with the size and shape of the dust particle [27]. These characteristics and a few other features [27] as well as moisture content are summarized in Table 7-31. These data are:

a) Lower flammability limit or minimum explosive concentration of the dust with air, averaging about 30 to 100 grams/cu meter.

b) Explosion characteristics vary with the degree of dispersion of the dust. Most dusts exhibit an optimum concentration for maximum explosion pressure and pressure rise.

c) Dust particles of < 420 microns may deflagrate. The smaller particles yielding maximum rate of pressure rise and maximum pressure, both increase as particle size decreases, and are more sensitive in the 200–420 micron range.

d) Minimum ignition energy that is very sensitive to particle size. The use of energy of ignition decreases with decreasing particle size. Decrease in particle size also increases the capacitance of a dust cloud; that is, the size of electrical charge on the cloud.

e) Mixtures of dusts, combustible gas/vapors and an oxidant such as air are termed "hybrid mixtures" and have unique characteristics. Some of these lead to an explosion under conditions not necessarily conforming to the gas or dust; therefore, when the possibilities of such mixtures develop, special care and investigation should be made.

f) Ignition energies for many combustible dusts have been established. Some of the values have been quite low, which allows various types of static charges to be eligible to ignite such systems.

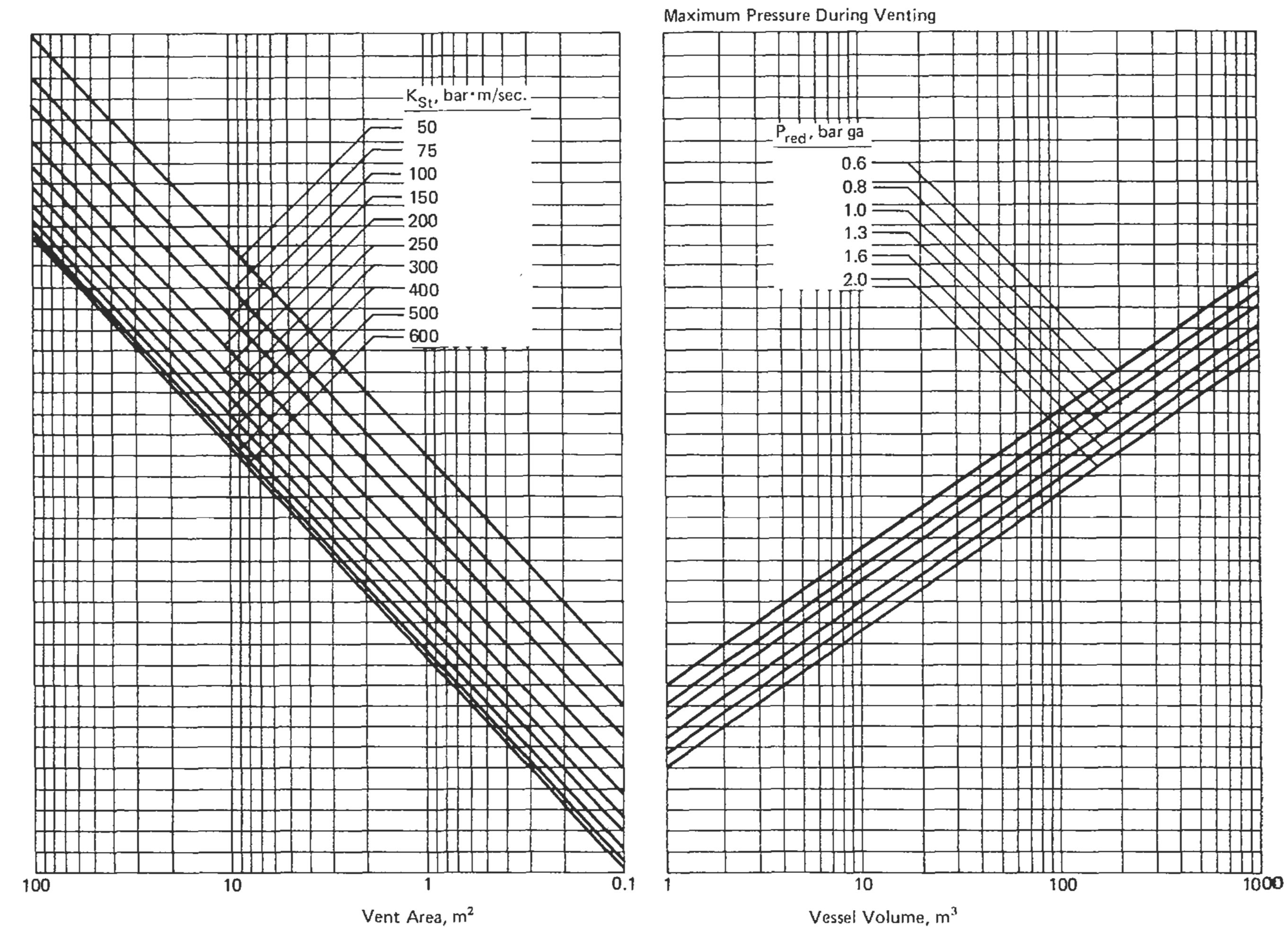


Figure 7-65C. Venting nomographs for dusts, P_{stat} = 0.5 bar ga. Reprinted with permission, NFPA 68-1988, *Deflagration Venting,* (1988) National Fire Protection Association, Quincy, MA 02269. See note Figure 7-63A.

g) Other than general atmospheric humidity, the moisture that is absorbed on a dust particles surface will usually raise its ignition temperature. Once an ignition has commenced, the humidity of the air has no effect.

Extrapolation/Interpolation of Dust Nomographs

Use the technique described earlier for gases.

Equations to Represent the Nomographs

NFPA-68 presents equations to represent the respective nomographs for dusts (Figures 7-65A through H) but they are stated to be no more accurate than the nomographs.

Venting of Bins, Silos, and Hoppers

For elongated vessels and pipelines, explosions create axial flow [54] resulting in definite differences compared to explosions in more cubical vessels.

Vessels are considered elongated when the height (H)/diameter (D) ratio > 5:1 [54].

In elongated confined vessels, with one end closed and the opposite end open or removable, when an explosion begins at or near the closed end, the rapid movement of the flame front caused by the high volume from combustion will cause displacement of the unburnt mixture ahead of it. Apparently this characteristic is independent of the nature of the combustible material [54], and the velocity can reach 80%–90% of the flame velocity, in part due to the high turbulence generated in the unburnt mixtures.

For these types of vessels or pipeline (see details, Ref. [54]), explosion venting should always include the entire cross-sectional area of the top or roof area of a silo (for

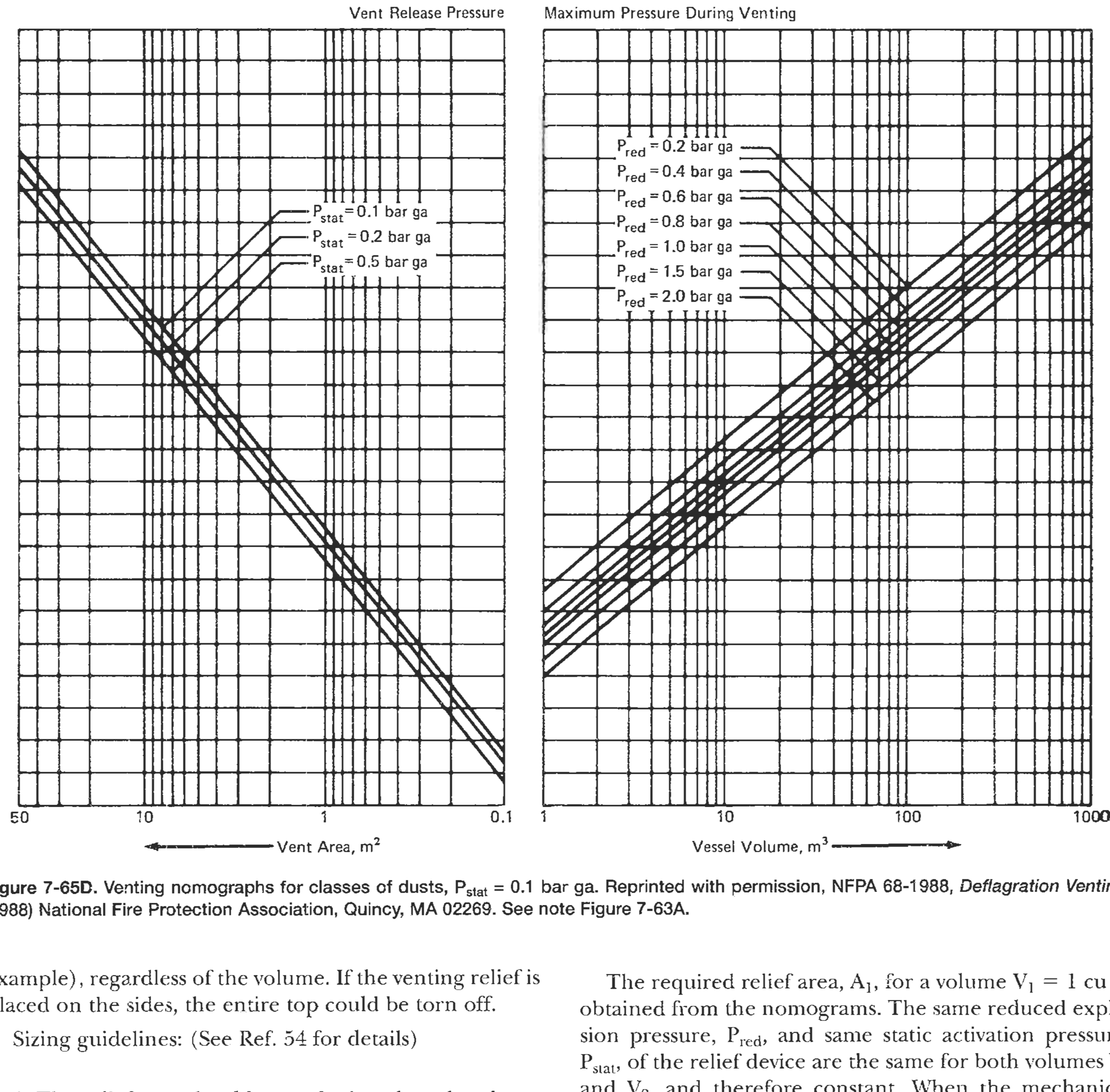


Figure 7-65D. Venting nomographs for classes of dusts, P_{stat} = 0.1 bar ga. Reprinted with permission, NFPA 68-1988, *Deflagration Venting*, (1988) National Fire Protection Association, Quincy, MA 02269. See note Figure 7-63A.

example), regardless of the volume. If the venting relief is placed on the sides, the entire top could be torn off.

Sizing guidelines: (See Ref. 54 for details)

1. The relief area should never be less than that determined from the nomograms. A silo's cross-sectional area establishes the maximum that can be protected.
2. The nomograms are applicable only for vessels with volumes up to 1,000 cu meters (35,315 cu ft).
3. Using the limiting relief area as the cross-sectional area of an elongated vessel, A_2, the greatest volume, V_2, that can be protected by this relief area can be calculated by the cubic law [54]:

$$V_2 = V_1 \sqrt{(A_2/A_1)^3} \tag{7-76}$$

The required relief area, A_1, for a volume V_1 = 1 cu m obtained from the nomograms. The same reduced explosion pressure, P_{red}, and same static activation pressure, P_{stat}, of the relief device are the same for both volumes V_1 and V_2, and therefore constant. When the mechanical strength of the vessel, P_{red}, is changed, the maximum volume and height will change with the hazard class, St-1, St-2, or St-3.

Dust Clouds

These can be readily ignited by flames, sparks, static electrical discharges (often the most likely), hot surfaces, and many other sources. Table 7-31 lists dust cloud ignition temperatures ranging from 572°F to 1112° F, and can be contrasted to flammable vapor-air ignition temperatures from 428°F to 1170°F. Generally, ignition tempera-

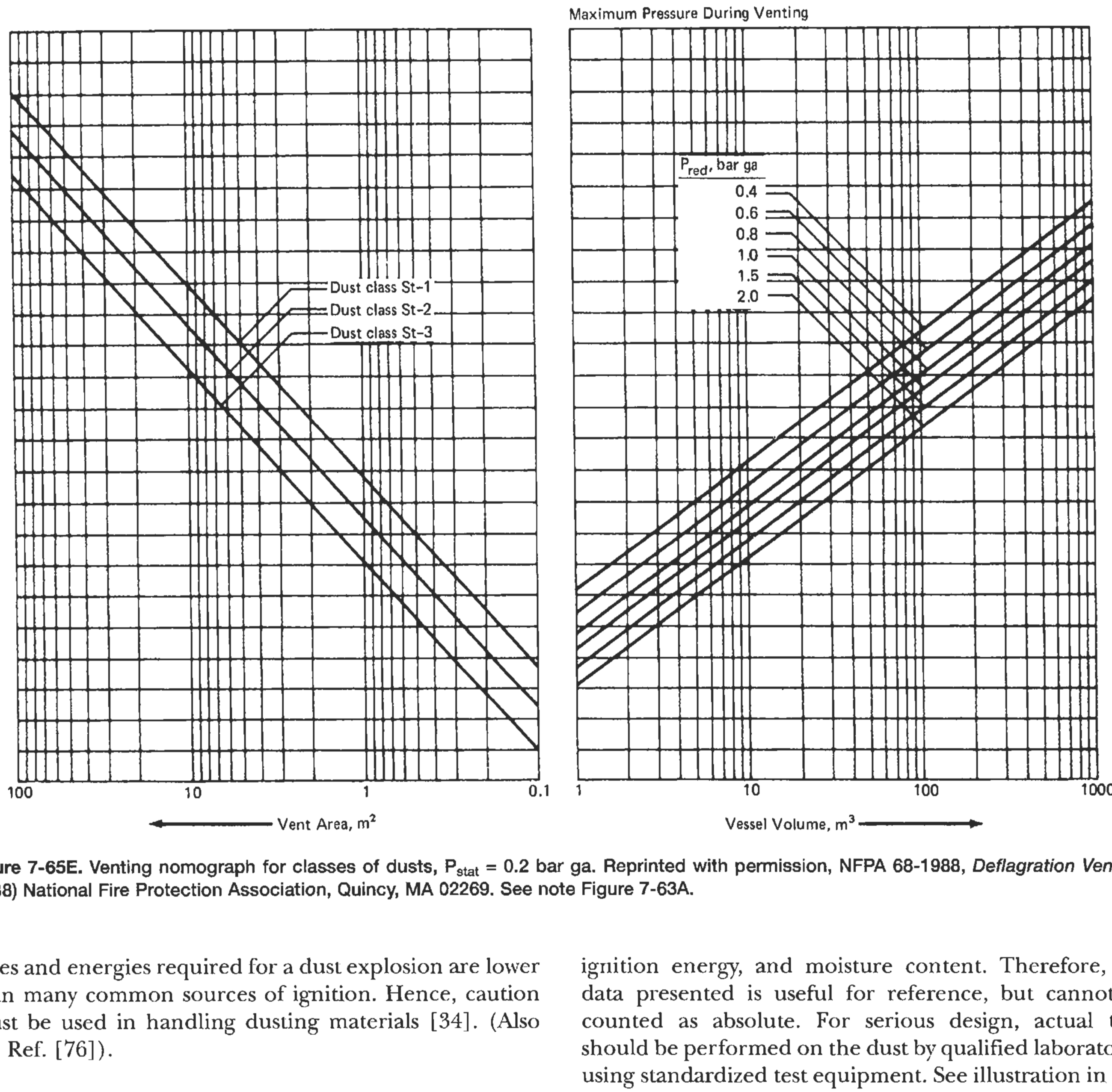


Figure 7-65E. Venting nomograph for classes of dusts, P_{stat} = 0.2 bar ga. Reprinted with permission, NFPA 68-1988, *Deflagration Venting,* (1988) National Fire Protection Association, Quincy, MA 02269. See note Figure 7-63A.

tures and energies required for a dust explosion are lower than many common sources of ignition. Hence, caution must be used in handling dusting materials [34]. (Also see Ref. [76]).

Dust Explosion Severity

Table 7-31 lists the explosibility index that is a relative measure of the potential damage from a dust explosion. A rating of 2 to 4 requires large vent areas. Above 4, for most cases, the explosion cannot be controlled by venting design and therefore requires the use of protection such as inert gas or explosive suppression systems, some of which are commercially available.

Unfortunately, rate of pressure rise and maximum explosion pressure listed in Table 7-31 are subject to uniqueness of the test conditions and are the function of particle size, dust concentration and uniformity, available ignition energy, and moisture content. Therefore, the data presented is useful for reference, but cannot be counted as absolute. For serious design, actual tests should be performed on the dust by qualified laboratories using standardized test equipment. See illustration in Ref. [34], pp. 4:94–95.

Explosion Suppression

Protection to guard against explosion has been classed as venting, suppression, and isolation. Ref. [54] discusses the subject rather thoroughly and Ref. 34 discusses some of the methods used to suppress explosions (see Figures 7-61, 7-62, 7-66 and 7-67). After an explosion has started special sensors are required, such as:

1. thermoelectrical
2. optical
3. pressure

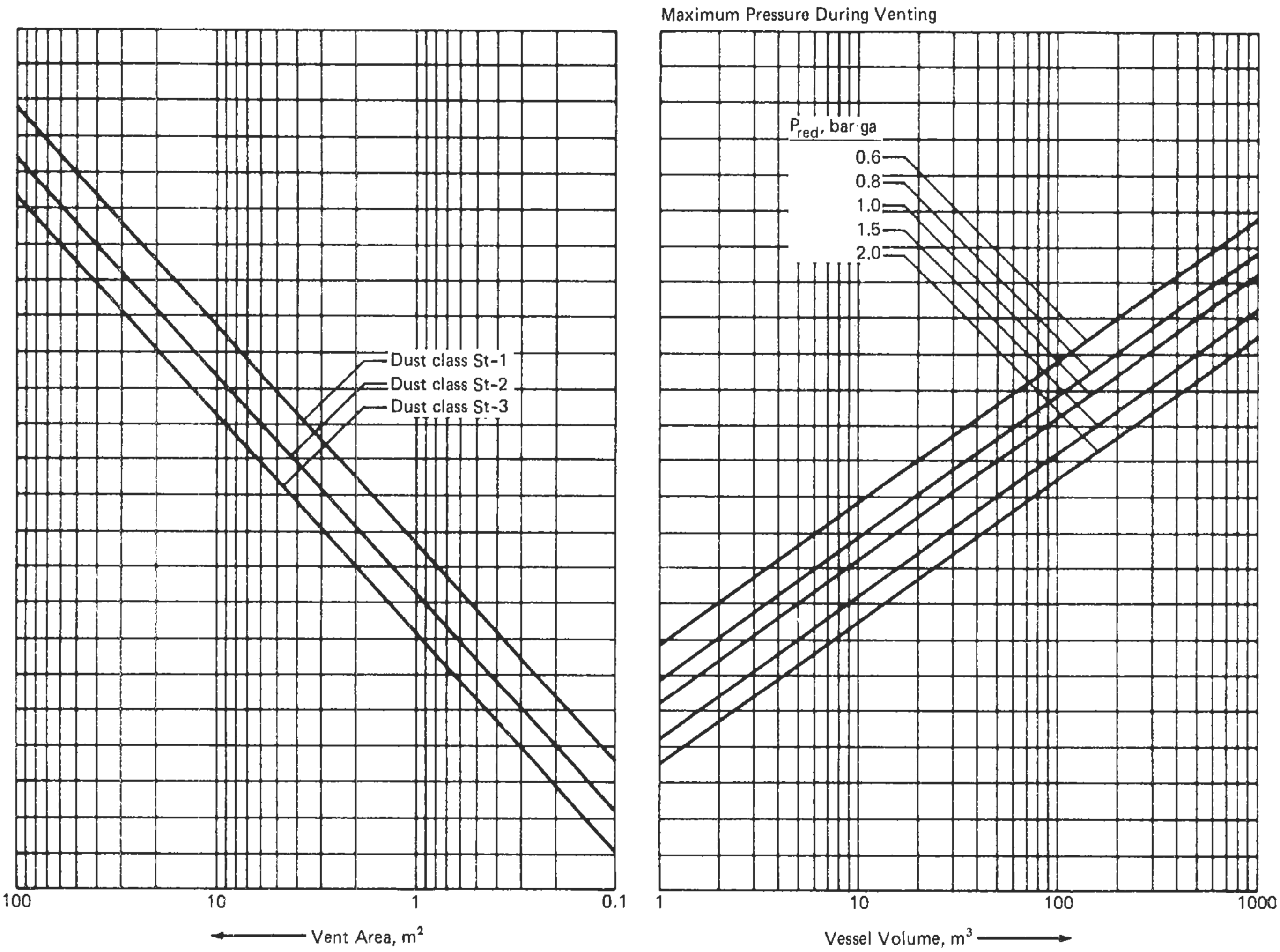


Figure 7-65F. Venting nomograph for classes of dusts, P_{stat} = 0.5 bar ga. Reprinted with permission, NFPA 68-1988, *Deflagration Venting,* (1988) National Fire Protection Association, Quincy, MA 02269. See note Figure 7-63A.

These require special electronics to cause responses of the systems to actually inject the suppression medium, such as [54]:

1. halons (halogenated hydrocarbons) See Figure 7-62
2. water, primarily for dust explosion
3. powders
4. ammonium phosphate for some organic peroxide

Special explosion characteristics must be understood before selecting the type of suppression medium.

For pipelines, bursting disks have been proven practical, especially when equipped with a sensor to pick up the explosion and a detonator to rupture the disks in advance of the pressure wave. The installation of a moveable explosion door carefully designed and weighted can also prove useful for pipelines.

The use of properly designed relief panels or "free floating" vessel covers are usually effective for dust explosions in silos tanks, filter housings, and the like. The design basis has been previously discussed.

A very careful study is required to develop a confidence for deflagration suppression in any flammable or dust system. The potential damage can be enormous.

The Fauske and Associates' *Reactive System Screening Tool* (RSST) was developed as a result of the DIERS studies and allows rapid evaluation of the potential for runaway reactions. It measures the rate of energy and gas release during the runaway and is valuable for screening various process systems before commercial designs are completed (see Figure 7-61).

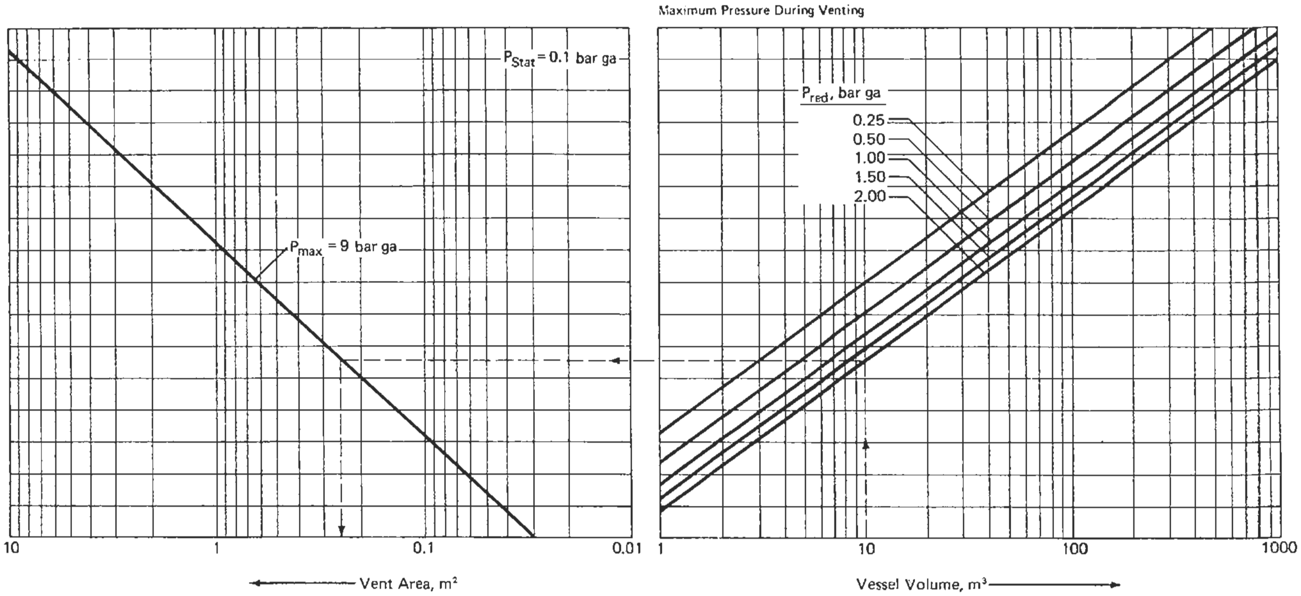


Figure 7-65G. Alternate venting nomograph for dusts of class St.-1 whose maximum deflagration pressure does not exceed 9 bar ga. Reprinted with permission, NFPA 68-1988, *Deflagration Venting,* (1988) National Fire Protection Association, Quincy, MA 02269. See note Figure 7-63A.

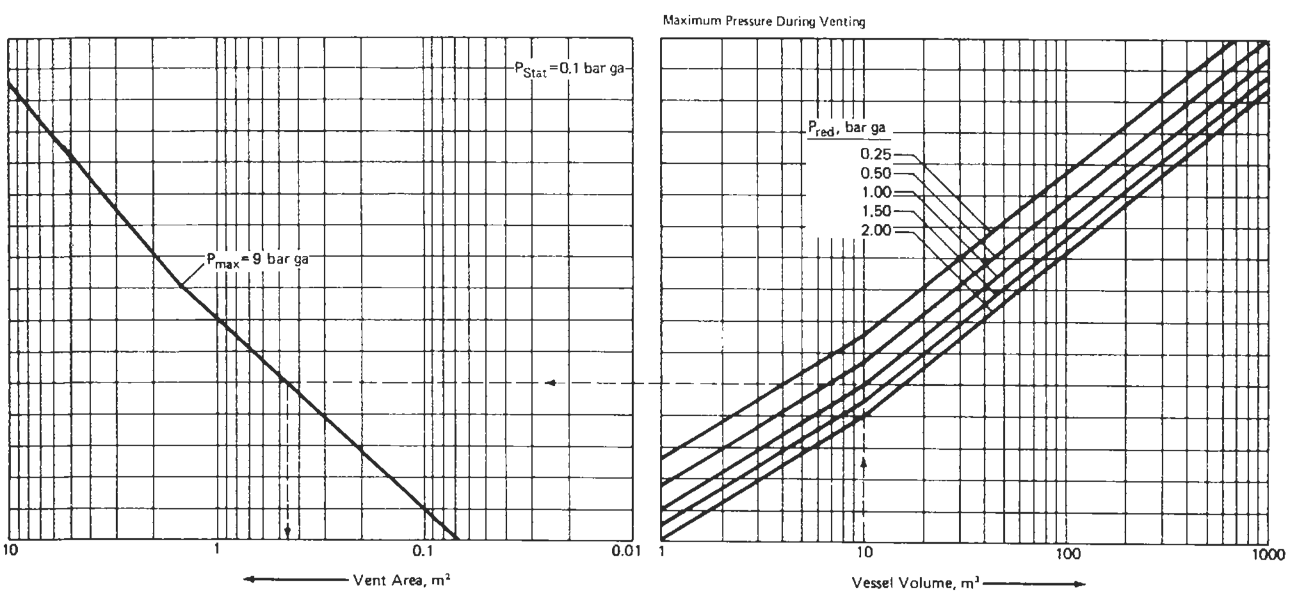


Figure 7-65H. Alternate venting nomograph for dusts of class St.-2 whose maximum deflagration pressure does not exceed 9 bar ga. Reprinted with permission, NFPA 68-1988, *Deflagration Venting,* (1988) National Fire Protection Association, Quincy, MA 02269. See note Figure 7-63A.

Unconfined Vapor Cloud Explosions

These explosions in air are usually the result of the release of flammable gas and/or mists by leaks, rupture of equipment, or rupture of safety relieving devices and release to the atmosphere, which become ignited by spark, static electricity, hot surfaces, and many other potential sources. They usually spread up into the air above the plant area and/or equipment. When the concentration is right, they ignite to form a deflagration and often a detonation that results in heavy damage. From some of these incidents, plant areas have been destroyed, glass broken up to or beyond five miles away, and people severely injured or killed.

Table 7-28
Hazard Class for Dusts

Hazard Class	K_{St} Bar meter/sec	*Max. Rate Pressure Rise $lb_f/in^2/sec$
St-1	$> 0 \leqq 200$	< 7300
St-2	201–300	7300–22,000
St-3	> 300	> 22,000
St-0	0, non-combustible	

Note: The nomographs are limited to an upper K_{st} value of 600. K_{st} values determined in an approximate spherical test vessel of at least 20 liter capacity. See Tables 7-29 and 7-30 for typical K_{st} values.
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*Added by this author.

Table 7-29
K_{St}-values of Technical Fine Dusts—High Ignition Energy

Type of dust	P_{max} (bar)	K_{St}-value ($bar \cdot m \cdot s^{-1}$)
PVC	6.7–8.5	27–98
Milk powder	8.1–9.7	58–130
Polyethylene	7.4–8.8	54–131
Sugar	8.2–9.4	59–165
Resin dust	7.8–8.9	108–174
Brown coal	8.1–10.0	93–176
Wood dusts	7.7–10.5	83–211
Cellulose	8.0–9.8	56–229
Pigments	6.5–10.7	28–344
Aluminum	5.4–12.9	16–750

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These clouds can travel with the prevailing wind currents and thereby explode considerable distances from the initial discharge of vapors.

Reference [40] presents a rather thorough review of the history and theoretical analysis of these types of explosions.

Atmosphere releases may form relatively still clouds, or they may plume and trail with the wind, or they may jet high into the atmosphere before forming a cloud, all depending to some extent on the unit. Wells [53] presents a thorough analysis of these phenomena.

Effects of Venting Ducts

Usually the relief of explosions cannot readily, safely, or conventionally be released right at the source, whether in a building or in a working plant area. Therefore, these reliefs are directed to some discharge point where the released material (dusts, gases, or mixtures), which may be very hot, can be safely released.

First and foremost these venting ducts should be as straight as possible, with few, if any elbows, and even these should be sweeping bends. There should be no valves of any type to keep flow resistance as low as possible, as this creates friction that creates backpressure on the relief device and raises burst conditions, which can be terribly dangerous. Figure 7-66 and 7-67 are used to assess the increased pressure due to ducts on relief discharge as affected by duct length. (This data is limited. See Ref. [56 and 53].)

Maximum Distance Between Vents

Figure 7-68 indicates the maximum allowable distance between vents on a vessel or pipe related to vent diameter when multiple vents are required. When distances are greater than indicated, a detonation should be anticipated in the design of the equipment strength. Figure 7-68 is applicable for systems with operating pressures up to 0.2 bar ga and for systems such as elongated vessels, pipes, and ducts with dusts or gases that are vented at one end, and have a velocity of less than 2 meters per second. Above 2 meters per second, alternate protection is recommended because a detonation is likely [27] [56]. See NFPA-68, Ref. [27] for details of application.

Runaway Reactions: DIERS

There is no standardized method for predicting or controlling runaway reaction that may lead to explosions (deflagrations or detonations), except possibly the Fauske approach (Figure 7-61).

Accordingly, to emphasize the safety problems affecting all industrial process plants and laboratories, the American Institute of Chemical Engineers established the industry-supported Design Institute for Emergency Relief Systems. The purposes of the Institute are [51]:

- Reduce the frequency, severity, and consequences of pressure producing accidents
- Promote the development of new techniques that will improve the design of emergency relief systems.
- Understand runaway reactions.
- Study the impact of two-phase flow on pressure relieving device systems.

The research and technical evaluations have provided industry with extremely valuable information and design procedures, including, but not limited to, two-phase flow phenomena and runaway reactions during safety/overpressure relief.

Table 7-30
Hazard Classes and K_{st} Values for Selected Types of Dusts

(A) Agricultural Products

Material	Median particle size, µm	Minimum explosive concentration g/m³	P_{max}, bar ga	$(dP/dt)_{max}$, bar/sec	K_{St} bar-m sec	Dust Hazard Class
Cellulose	33	60	9.7	229	229	2
Cellulose, pulp	42	30	9.9	62	62	1
Cork	42	30	9.6	202	202	2
Corn	28	60	9.4	75	75	1
Egg White	17	125	8.3	38	38	1
Milk, powdered	83	60	5.8	28	28	1
Milk, non-fat, dry	60	–	8.8	125	125	1
Soy Flour	20	200	9.2	110	110	1
Starch, corn	7	–	10.3	202	202	2
Starch, rice	18	60	9.2	101	101	1
Starch, wheat	22	30	9.9	115	115	1
Sugar	30	200	8.5	138	138	1
Sugar, milk	27	60	8.3	82	82	1
Sugar, beet	29	60	8.2	59	59	1
Tapioca	22	125	9.4	62	62	1
Whey	41	125	9.8	140	140	1
Wood Flour	29	–	10.5	205	205	2

(B) Chemical Dusts

Material	Median particle size, µm	Minimum explosive concentration g/m³	P_{max}, bar ga	$(dP/dt)_{max}$, bar/sec	K_{St} bar-m sec	Dust Hazard Class
Adipic Acid	<10	60	8.0	97	97	1
Anthraquinone	<10	–	10.6	364	364	3
Ascorbic Acid	39	60	9.0	111	111	1
Calcium Acetate	92	500	5.2	9	9	1
Calcium Acetate	85	250	6.5	21	21	1
Calcium Stearate	12	30	9.1	132	132	1
Carboxy-methyl-cellulose	24	125	9.2	136	136	1
Dextrin	41	60	8.8	106	106	1
Lactose	23	60	7.7	81	81	1
Lead Stearate	12	30	9.2	152	152	1
Methyl-cellulose	75	60	9.5	134	134	1
Paraform-aldehyde	23	60	9.9	178	178	1
Sodium Ascorbate	23	60	8.4	119	119	1
Sodium Stearate	22	30	8.8	123	123	1
Sulfur	20	30	6.8	151	151	1

(C) Carbonaceous Dusts

Material	Median particle size, µm	Minimum explosive concentration g/m³	P_{max}, bar ga	$(dP/dt)_{max}$, bar/sec	K_{St} bar-m sec	Dust Hazard Class
Charcoal activated	28	60	7.7	44	44	1
Charcoal, wood	14	60	9.0	10	10	1
Coal, bituminous	24	60	9.2	129	129	1
Coke, petroleum	15	125	7.6	47	47	1
Lampblack	<10	60	8.4	121	121	1
Lignite	32	60	10.0	151	151	1
Peat, 15% H_2O		58	60	10.9	157	1
Peat, 22% H_2O		46	125	8.4	69	1
Soot, pine	<10	–	7.9	26	26	1

(D) Metal Dusts

Material	Median particle size, µm	Minimum explosive concentration g/m³	P_{max}, bar ga	$(dP/dt)_{max}$, bar/sec	K_{St} bar-m sec	Dust Hazard Class
Aluminum	29	30	12.4	415	415	3
Bronze	18	750	4.1	31	31	1
Iron Carbonyl	<10	125	6.1	111	111	1
Magnesium	28	30	17.5	508	508	3
Zinc	10	250	6.7	125	125	1
Zinc	<10	125	7.3	176	176	1

To obtain updated listing of the published information on this research, contact the AIChE office in New York. The work is original and conducted by thoroughly qualified researchers/engineers. The work on runaway reactions is the first systematized examination of the subject and is really the only design approach available, but it requires careful study and is not just dropping numbers in equations.

Two-phase flow is an important aspect of venting relief as well as of runaway reactions, and is a complicated topic when related to liquid flashing in a vessel as it discharges on pressure relief. It cannot be adequately covered by conventional fluid two-phase flow.

Articles describing procedures related to the DIERS development of this entire subject have been published by some members of the DIERS group. These are referenced here, but to adequately describe the methodology requires more space than is suitable for this chapter. The detailed descriptions and illustrations in the noted articles can be most helpful to the potential user.

Table 7-30 continued (E) Plastics

Material	Median particle size, µm	Minimum explosive concentration g/m³	P_{max}, bar ga	$(dP/dt)_{max}$, bar/sec	K_{St} bar-m sec	Dust Hazard Class
(poly) Acrylamide	10	250	5.9	12	12	1
(poly) Acrylonitrile	25	–	8.5	121	121	1
(poly) Ethylene (Low Pressure Process)	<10	30	8.0	156	156	1
Epoxy Resin	26	30	7.9	129	129	1
Melamine Resin	18	125	10.2	110	110	1
Melamine, molded (Wood flour and Mineral-filled Phenol-Formaldehyde	15	60	7.5	41	41	1
Melamine, molded (Phenol-Cellulose)	12	60	10.0	127	127	1
(poly) Methyl Acrylate	21	30	9.4	269	269	2
(poly) Methyl Acrylate, Emulsion Polymer	18	30	10.1	202	202	2
Phenolic Resin	<10	15	9.3	129	129	1
(poly) Propylene	25	30	8.4	101	101	1
Terpene-Phenol Resin	10	15	8.7	143	143	1
Urea-Formaldehyde/Cellulose, Molded	13	60	10.2	136	136	1
(poly) Vinyl Acetate/Ethylene Copolymer	32	30	8.6	119	119	1
(poly) Vinyl Alcohol	26	60	8.9	128	128	1
(poly) Vinyl Butyral	65	30	8.9	147	147	1
(poly) Vinyl Chloride	107	200	7.6	46	46	1
(poly) Vinyl Chloride/Vinyl Acetylene Emulsion Copolymer	35	60	8.2	95	95	1
(poly)						

(E) Plastics continued

Material	Median particle size, µm	Minimum explosive concentration g/m³	P_{max}, bar ga	$(dP/dt)_{max}$, bar/sec	K_{St} bar-m sec	Dust Hazard Class
Vinyl Chloride/Ethylene/Vinyl Acetylene Suspension Copolymer	60	60	8.3	98	98	1

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DIERS Final Reports

Refer to the bibliography at the end of this chapter for a listing of the final reports of this program [67].

Flares/Flare Stacks

Flares are useful for the proper disposal of waste or emergency released gas/vapors and liquids. The effects on the environment and the thermal radiation from the flare must be recognized and designed for. Flares may be "ground" flares or they may be mounted on a tall stack to move the venting away from immediate plant areas. Figure 7-69 illustrates a plant flare stack system. The flow noted "from processes" could also include pressure relief valve discharges when properly designed for backpressure. This requires proper manifold design and, for safety, requires that the "worst case" volume condition be used, particularly assuming that all relief devices discharge at the same time and any other process vents are also flowing. The piping system sequence of entrance of the flare is important to backpressure determination for all respective relief devices.

The *knock-out drums or separator tanks/pots* can be designed using the techniques offered in the chapter on Mechanical Separation, and will not be repeated here. API-RP 521 [13] specifies 20–30 minutes holdup liquid capacity from relief devices plus a vapor space for dropout and a drain volume.

The unit should have backup instrumentation to ensure liquid level control to dispose of the waste recovered liquid.

The *seal tank/pot* is not a separator but a physical liquid seal (Figure 7-70) to prevent the possibilities of backflash from the flare from backing into the process manifolds. It is essential for every stack design.

The backpressure created by this drum is an additive to the pipe manifold pressure drops and the pressure loss through the separator. Therefore, it cannot be independently designed and not "integrated" into the backpressure system. The flow capacity of the relief valve(s) must

(text continued on page 526)

Table 7-31
Explosion Characteristics of Various Dusts
(Partial Listing)

(Compiled from the following reports of the U.S. Department of Interior, Bureau of Mines: RI 5753, The Explosibility of Agricultural Dusts; RI 6516, Explosibility of Metal Powders; RI 5971, Explosibility of Dusts Used in the Plastics Industry; RI 6597, Explosibility of Carbonaceous Dusts; RI 7132, Dust Explosibility of Chemicals, Drugs, Dyes and Pesticides; and RI 7208, Explosibility of Miscellaneous Dusts.)

Type of Dust	Explosibility Index	Ignition Sensitivity	Explosion Severity	Maximum Explosion Pressure psig	Max Rate of Pressure Rise psi/sec	Ignition Temperature Cloud °C	Ignition Temperature Layer °C	Min Cloud Ignition Energy joules	Min Explosion Conc oz/cu ft	Limiting Oxygen Percentage* (Spark Ignition)
Agricultural Dusts										
Alfalfa meal	0.1	0.1	1.2	66	1,100	530	—	0.32	0.105	—
Almond shell	0.3	0.9	0.3	101	1,400	450	210	0.08	0.065	—
Apricot pit	1.9	1.6	1.2	109	4,000	440	230	0.08	0.035	—
Cellulose	2.8	1.0	2.8	130	4,500	480	270	0.080	0.055	C13
Cellulose, alpha	>10	2.7	4.0	117	8,000	410	300	0.040	0.045	—
Cellulose, flock, fine cut	8.7	2.3	3.8	112	7,000	460	260	0.035	0.055	C13
Cereal grass	< 0.1	< 0.1	0.1	65	400	620	230	0.80	0.20	—
Cherry pit	4.4	2.0	2.2	113	4,400	430	220	0.08	0.03	—
Cinnamon	5.8	2.5	2.3	121	3,900	440	230	0.03	0.06	—
Citrus peel	0.6	0.7	0.9	51	1,200	500	330	0.10	0.06	—
Coca bean shell	13.7	3.6	3.8	77	3,300	470	370	0.03	0.04	—
Cocoa, natural 19% fat	0.6	0.5	1.1	68	1,200	510	240	0.10	0.075	—
Coconut shell	4.2	2.0	2.1	115	4,200	470	220	0.06	0.035	—
Coffee, raw bean	< 0.1	0.1	0.1	33	150	650	280	0.32	0.15	C17
Coffee, fully roasted	< 0.1	0.2	0.1	38	150	720	270	0.16	0.085	C17
Coffee, instant spray dried	< 0.1	0.1	0.1	68	500	410	350	†	0.28	—
Corn	6.9	2.3	3.0	113	6,000	400	250	0.04	0.055	—
Corncob grit	5.5	2.5	2.2	127	3,700	450	240	0.045	0.045	—
Corn dextrine, pure	12.1	3.1	3.9	124	5,500	410	390‡	0.04	0.04	—
Cornstarch commercial product	9.5	2.8	3.4	106	7,500	400	—	0.04	0.045	—
Cornstarch (thru No. 325 Sieve)	23.2	4.3	5.4	145	9,500	390	350	0.03	0.04	C11
Cork dust	>10	3.6	3.3	96	7,500	460	210	0.035	0.035	—
Cotton linter, raw	< 0.1	< 0.1	< 0.1	73	400	520	—	1.92	0.50	C21
Cottonseed meal	1.1	0.9	1.2	104	2,200	540	—	0.08	0.055	—
Cube root, South American	6.5	2.7	2.4	69	2,100	470	230	0.04	0.04	—
Egg white	< .1	< 0.1	0.2	58	500	610	—	0.64	0.14	—
Flax shive	0.2	0.7	0.3	108	1,500	430	230	0.08	0.08	—
Garlic, dehydrated	0.2	0.2	1.2	57	1,300	360	—	0.24	0.10	—
Grain dust, winter wheat, corn, oats	9.2	2.8	3.3	131	7,000	430	230	0.03	0.055	—
Grass seed, blue	< 0.1	0.1	0.1	51	400	490	180	0.26	0.29	—
Guar seed	2.4	1.7	1.4	70	1,200	500	—	0.06	0.04	—
Gum, arabic	1.1	0.7	1.6	84	1,500	500	260	0.10	0.06	—
Gum, karaya	0.3	0.2	1.5	83	1,100	520	240	0.18	0.10	—
Gum, Manila (copal)	18.0	6.2	2.9	63	2,800	360	390‡	0.03	0.03	—
Gum, tragacanth	8.1	2.6	3.1	88	2,400	490	260	0.045	0.04	—
Hemp hurd	20.5	3.8	5.4	121	10,000	440	220	0.035	0.04	—
Lycopodium	16.4	4.2	3.9	75	3,100	480	310	0.04	0.025	C13
Malt barley	5.5	2.6	2.1	95	4,400	400	250	0.035	0.055	—
Milk, skimmed	1.4	1.6	0.9	95	2,300	490	200	0.05	0.05	N15
Moss, Irish	< 0.1	< 0.1	< 0.1	35	400	480	230	†	§	—
Onion, dehydrated	< 0.1	< 0.1	< 0.1	35	500	410	—	†	0.13	—
Pea flour	4.0	1.8	2.2	68	1,900	560	260	0.04	0.05	—
Peach pit shell	7.1	3.1	2.3	115	4,700	440	210	0.05	0.03	—
Peanut hull	4.0	2.0	2.0	116	8,000	460	210	0.05	0.045	—
Peat, sphagnum, sun dried	2.0	2.0	1.0	104	2,200	460	240	0.05	0.045	—
Pecan nut shell	7.4	3.1	2.4	112	4,400	440	210	0.05	0.03	—
Pectin (from ground dried apple pulp)	10.3	2.2	4.7	132	8,000	410	200	0.035	0.075	—
Potato starch, dextrinated	20.9	5.1	4.1	120	8,000	440	—	0.025	0.045	—
Pyrethrum, ground flower leaves	0.4	0.6	0.6	95	1,500	460	210	0.08	0.10	—
Rauwolfia vomitoria root	9.2	2.2	4.2	106	7,500	420	230	0.045	0.055	—
Rice	0.3	0.5	0.5	47	700	510	450	0.10	0.085	—
Rice bran	1.4	1.1	1.3	61	1,300	490	—	0.08	0.045	—
Rice hull	2.7	1.6	1.7	109	4,000	450	220	0.05	0.055	—
Safflower meal	5.2	4.0	1.3	90	2,400	460	210	0.025	0.055	—
Soy flour	0.7	0.6	1.1	94	800	550	340	0.10	0.06	C15
Soy protein	4.0	1.2	3.3	98	6,500	540	—	0.06	0.05	C15
Sucrose, chemically pure	3.3	1.1	3.0	76	2,500	420	470‡	0.10	0.045	—
Sucrose	4.8	2.7	1.8	86	5,500	370	400‡	0.03	0.045	—
Sugar, powdered	9.6	4.0	2.4	109	5,000	370	400‡	0.03	0.045	—

* Numbers in this column indicate oxygen percentage while the letter prefix indicates the diluent gas. For example, the entry "C13" means dilution to an oxygen content of 13 percent with carbon dioxide as the diluent gas. The letter prefixes are: C = Carbon Dioxide; N = Nitrogen; A = Argon; and H = Helium.

† No ignition to 8.32 joules, the highest tried.

‡ Ignition denoted by flame, all others not so marked (‡) denoted by a glow.

§ No ignition to 2 oz per cu ft, the highest tried.

(continued)

Table 7-31
Explosion Characteristics of Various Dusts (Cont.)

Type of Dust	Explosibility Index	Ignition Sensitivity	Explosion Severity	Maximum Explosion Pressure psig	Max Rate of Pressure Rise psi/sec	Ignition Temperature Cloud °C	Ignition Temperature Layer °C	Min Cloud Ignition Energy joules	Min Explosion Conc oz/cu ft	Limiting Oxygen Percentage* (Spark Ignition)
1 Naphthyl-N-methylcarbamate ("Sevin") 15% (85% Inert)	>10	18.0	1.6	90	5,000	560	140	0.010	0.020	—
3, 4, 5, 6-tetrahydro-3, 5,-dimethyl-2H-1, 3, 5 thiadeazine 2 thione, ("Crag" No. 974) 5% (95% Inert)	>10	8.7	2.0	97	6,000	310	330	0.030	0.025	—
a, a' Trithiobis (N, N-dimethyl-thioformamide)	8.9	3.4	2.6	96	7,000	280	230	0.035	0.060	—
Thermoplastic Resins and Molding Compounds										
Group I. Acetal Resins										
Acetal, linear (Polyformaldehyde)	>10	6.5	1.9	113	4,100	440	—	0.020	0.035	C11
Group II. Acrylic Resins										
Methyl methacrylate polymer	6.3	7.0	0.9	84	2,000	480	—	0.020	0.030	C11
Methyl methacrylate-ethyl acrylate copolymer	>10	14.0	2.7	85	6,000	480	—	0.010	0.030	C11
Methyl methacrylate-ethyl acrylate-styrene copolymer	>10	9.2	1.7	90	4,400	440	—	0.020	0.025	—
Methyl methacrylate-styrene-butadiene-acrylonitrile copolymer	>10	8.4	1.4	87	4,700	480	—	0.020	0.025	C11
Methacrylic acid polymer, modified	0.6	1.0	0.6	97	1,800	450	290	0.100	0.045	—
Acrylamide polymer	2.5	4.1	0.6	85	2,500	410	240	0.030	0.040	—
Acrylonitrile polymer	>10	8.1	2.3	89	11,000	500	460	0.020	0.025	C13
Acrylonitrile-vinyl pyridine copolymer	>10	7.9	2.4	85	6,000	510	240	0.025	0.020	—
Acrylonitrile-vinyl chloride-vinylidene chloride copolymer (70–20–10)	>10	5.9	3.0	87	15,000	650	210	0.015	0.035	—
Group III. Cellulosic Resins										
Cellulose acetate	>10	8.0	1.6	85	3,600	420	—	0.015	0.040	C14
Cellulose triacetate	7.4	3.9	1.9	107	4,300	430	—	0.030	0.040	C12
Cellulose acetate butyrate	5.6	4.7	1.2	85	2,700	410	—	0.030	0.035	C14
Cellulose propionate, 0.3% free hydroxyl	7.5	2.9	2.6	107	4,700	460	—	0.060	0.025	—
Ethyl cellulose 5–10 micron dust	>10	21.8	3.4	120	6,500	370	350§	0.010	0.025	C12
Methyl cellulose	>10	9.3	3.1	133	6,000	360	340	0.020	0.030	C13
Carboxy methyl cellulose, low viscosity, 0.3 to 0.4% substitution, acid product	1.4	0.5	2.7	130	5,000	460	310	0.140	0.060	—
Hydroxyethyl cellulose-mono sodium phosphate sizing compound	1.7	2.1	0.8	110	4,000	390	340	0.035	0.070	—
Group IV. Chlorinated Polyether Resins										
Chlorinated polyether alcohol	0.2	0.6	0.3	88	1,900	460	—	0.160	0.045	—
Group V. Fluorocarbon Resins										
Tetrafluoroethylene polymer (micronized)	0.1‡	0.1‡	—	§	—	670	570†	§	‖	—
Monochlorotrifluoroethylene polymer	0.1‡	0.1‡	—	§	—	600	720†	§	‖	—
Group VI. Nylon (Polyamide) Resins										
Nylon (polyhexamethylene adipamide) polymer	>10	6.7	1.8	95	4,000	500	430	0.020	0.030	C13
Group VII. Polycarbonate Resins										
Polycarbonate	8.6	4.5	1.9	96	4,700	710	—	0.025	0.025	C15

* Numbers in this column indicate oxygen percentage while the letter prefix indicates the diluent gas. For example, the entry "C13" means dilution to an oxygen content of 13 percent with carbon dioxide as the diluent gas. The letter prefixes are: C = Carbon Dioxide; N = Nitrogen; A = Argon; and H = Helium.

† Ignition denoted by flame, all others not so marked (†) denoted by a glow.

‡ 0.1 designates materials presenting primarily a fire hazard as ignition of the dust cloud is not obtained by the spark of flame source but only by the intense heated surface source.

§ No ignition to 8.32 joules, the highest tried.

‖ No ignition to 2 oz per cu ft, the highest tried.

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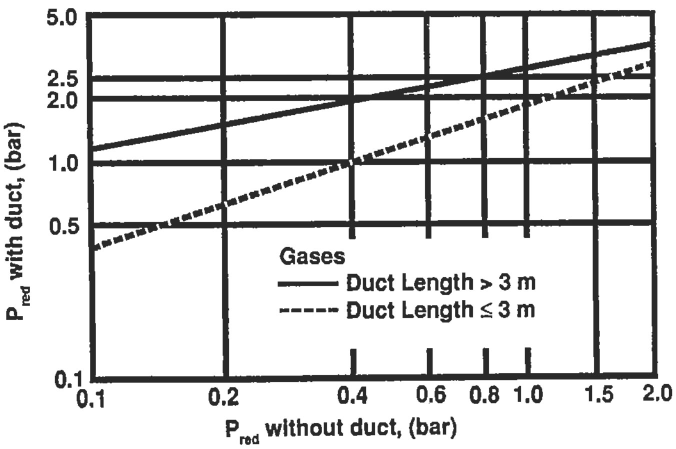


Figure 7-66. Increase in the reduced deflagration pressure for *gases* due to the effect of a vent duct. By permission, American Institute of Chemical Engineers, Meeting Mar. 6, 1988 by I. Swift (deceased) [56], with data from Ref. [54].

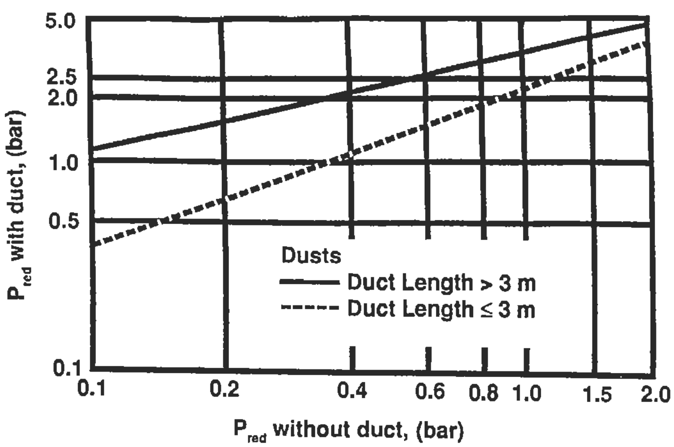


Figure 7-67. Increase in the reduced deflagration pressure for *dusts* due to the effect of a vent duct. By permission, American Institute of Chemical Engineers, Meeting Mar. 6, 1988, by I. Swift (deceased) [56], with data from Ref. [54].

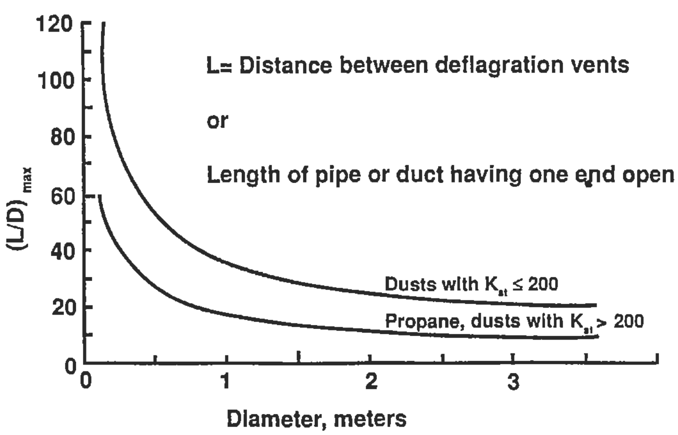


Figure 7-68. Maximum allowable length of a vessel vented at one end, or maximum distance between vents as a function of the vent diameter for the safe venting of deflagrations of dusts and gases. By permission, American Institute of Chemical Engineers, Meeting Mar. 6, 1988 by I. Swift (deceased) [56], with data from Ref. [54].

(text continued from page 523)

not be reduced due to backpressure on the valves' discharge side (outlet). The total backpressure of the system must be limited to 10% of the set pressure of each pressure relief valve that may be relieving concurrently [33]. When balanced relief valves are used, the manifold backpressure can be higher, less than 30% of the valve's set pressure, psia [33].

The key detail of a seal drum is the liquid seal:

$$h_l = 144P''/\rho, \text{ see Figure 7-70} \tag{7-77}$$

where h_l = seal, submerged, ft
P'' = maximum header exit pressure into seal, psig
ρ = density of seal liquid, lb/cu ft

Calculate the cross section of the drum volume for vapor above the liquid level, establishing the level referenced to h_l, plus clearance to drum bottom, h_l, normally

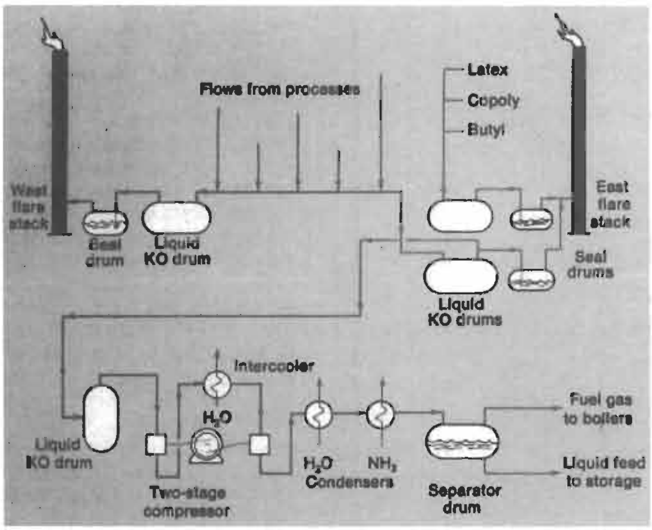


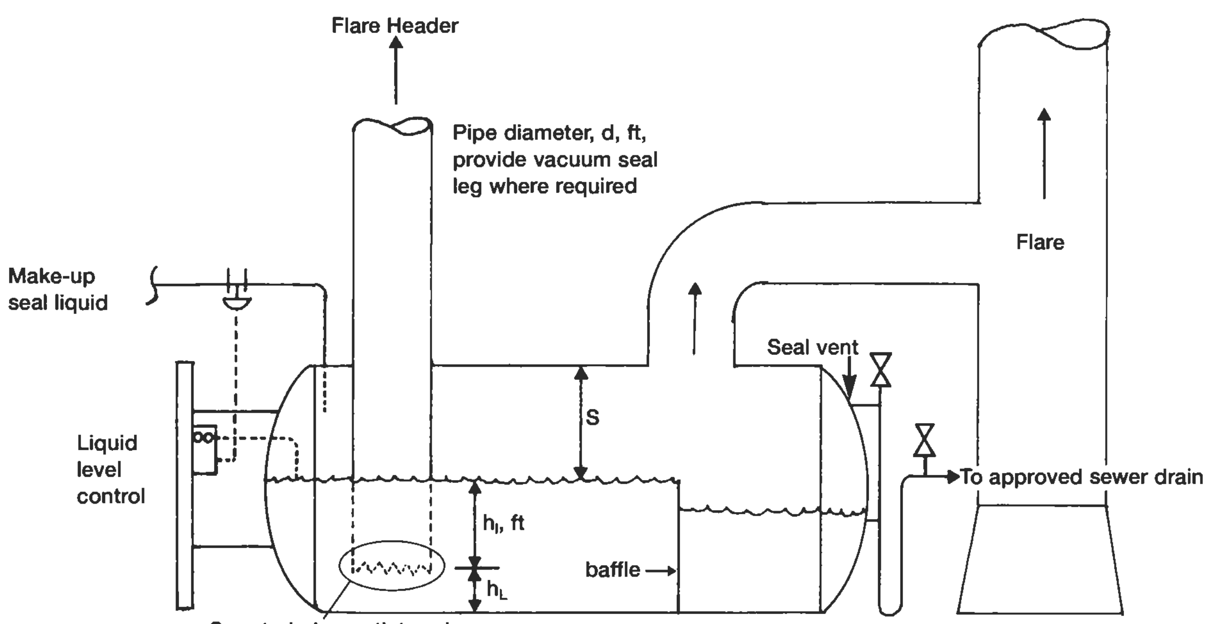
Figure 7-69. Illustration of one of many collection arrangements for process flow and/or relief valve discharge collections to relieve to one or more plant flare stacks. By permission, Livingston, D. D., *Oil & Gas Jour.*, Apr. 28, 1980.

12 inches to 18 inches. This would be a segment (horizontal vessel) of a circle (see Appendix Table 24). Ref. [33] recommends that the cross section area of the vapor space above the liquid be *at least* equivalent to that of a circle diameter [(2) (inlet pipe)]. Thus, the cross-section area of vapor space A_s with equivalent diameter S should be at least [(2) (a_p)] where a_p is the cross-sectional area of the inlet pipe [33]. To avoid bubble burst slugging, this author suggests that the cross-sectional area of S be calculated to have a vapor velocity of less than the entrainment velocity for a mist sized liquid particle, or that the vapor area be approximately one-third the cross-section area of the diameter of the horizontal drum. For a vertical drum, Ref. [33] recommends that the vapor disengaging height be three feet.

When vacuum can form in the system due to condensing/cooling hot vapor entering, the seal drum liquid volume and possibly the seal drum diameter/length must be adjusted to maintain a seal when/if the seal fluid is drawn up into the inlet piping. A vacuum seal leg should be provided on the inlet 1.2 times the expected equivalent vacuum height in order to maintain a seal.

The following design points should be considered:

- Provide liquid low level alarms to prevent loss of liquid by evaporation, entrainment, leaks, or failure of the makeup liquid system.
- Use a sealing liquid that has a relatively low vapor pressure, and is not readily combustible, and will not readily freeze. Quite often glycol or mixtures are



h_L = Seal pipe clearance to bottom, normally use 12 inch—18 inch
h_l = Submergence, seal, ft
S = Equivalent area diameter for vapor release

Figure 7-70. Suggested seal pot/drum for flare stack system. (See API RP-521, Fig. B-1, 3rd Ed., 1990.) Design adapted with permission by this author from API RP-521, 3rd Ed. (1990) American Petroleum Institute [33].

used. In freezing conditions the unit should receive personal inspection for condition of liquid.

- Provide overflow anti-siphon seal drains.
- Provide inlet vacuum seal legs.
- Some hydrocarbons may form gel clusters or layers with some sealant fluids; therefore, providing for cold weather heating and/or cleaning of the unit is necessary.
- Ref. [33] suggests minimum design pressure for such a seal vessel of 50 psig, ASME Code stamped (this author). Most flare seal drums operate at 0–5 psig pressure.
- Be extremely cautious and do not install light weight gauge glass liquid level columns. Rather use the heavier shatterproof style.
- Provide reliable seal liquid makeup, using liquid level gauging and monitoring with recording to ensure good records of performance. The liquid level must be maintained; otherwise, the hazards of a bleedthrough or backflow can become serious.

Flares

Flares are an attempt to deliberately burn the flammable safety relief and/or process vents from a plant. The height of the stack is important to the safety of the surroundings and personnel, and the diameter is important to provide sufficient flow velocity to allow the vapors/gases to leave the top of the stack at sufficient velocities to provide good mixing and dilution after ignition at the flare tip by pilot flames.

API [33] discusses factors influencing flare design, including the importance of proper stack velocity to allow jet mixing. Stack gases must not be diluted below the flammable limit. The exit velocity must not be too low to allow flammable gases to fall to the ground and become ignited. The atmospheric dispersion calculations are important for the safety of the plant. Computer models can be used to evaluate the plume position when the flare leaves the stack under various atmospheric wind conditions. This should be examined under alternate possibilities of summer through winter conditions. (Also see Ref. [78]).

The velocities of the discharge of relief devices through a stack usually exceed 500 feet/second. Because this stream exits as a jet into the air, it is sufficient to cause turbulent mixing [33].

For a flare stack to function properly and to handle the capacity that may be required, the flows under emergency conditions from each of the potential sources must be carefully evaluated. These include, but may not be limited to, pressure relief valves and rupture disks, process blowdown for startup, shutdown, upset conditions, and plant fires creating the need to empty or blowdown all or parts of a system.

Sizing

Diameter: sizing based on stack velocity [33c], solve for "d."

$$\text{Mach} = (1.702)(10^{-5})(W/P_t d^2)\sqrt{T/(kM)} \qquad (7\text{-}78)$$

where Mach = ratio of vapor velocity to sonic velocity in vapor, dimensionless. Mach = 0.5 for peak for short-term flow, and 0.2 for more normal and frequent conditions [33c].

W = vapor relief rate to stack, lbs/hr

P_t = pressure of the vapor just inside flare tips (at top), psia (For atmospheric release, P_t = 14.7 psia)

d = flare tip diameter, ft (end, or smallest diameter)

T = temperatures of vapors just inside flare tip, °R

k = ratio of specific heats, cp/cv for vapor being relieved

M = molecular weight of vapor

A peak velocity through the flare end (tip) of as much as 0.5 mach is generally considered a peak, short term. A more normal steady state velocity of 0.2 mach is for normal conditions and prevents flare/lift off [58]. Smokeless (with steam injection) flare should be sized for conditions of operating smokelessly, which means vapor flow plus steam flow [33c]. Pressure drops across the tip of the flare have been used satisfactorily up to 2 psi. It is important not to be too low and get flashback (without a molecular seal) or blowoff where the flame blows off the tip (see Ref. 57), Figure 7-71.

Another similar equation yielding close results [59]:

$$d_t^2 = (W/1370)\sqrt{T/M}, \text{ (generally for smokeless flares)} \qquad (7\text{-}79)$$

based on mach 0.2 limitation velocity, k = cp/cv = 0.2 and gas constant R = 1546 (Ft-lb force/(°R) (mole)

d_t = flare tip diameter, in.

W = gas vent rate, lb/hr

T = gas temperature in stack, °R

M = molecular weight of gas/vapor

For *non-smokeless* flares (no steam injection) about 30% higher capacity can be allowed [59]. Therefore, the diameter of a *non-smokeless* flare stack is approximately (0.85) (diameter of the smokeless flare stack).

The amount of steam injection required for smokeless

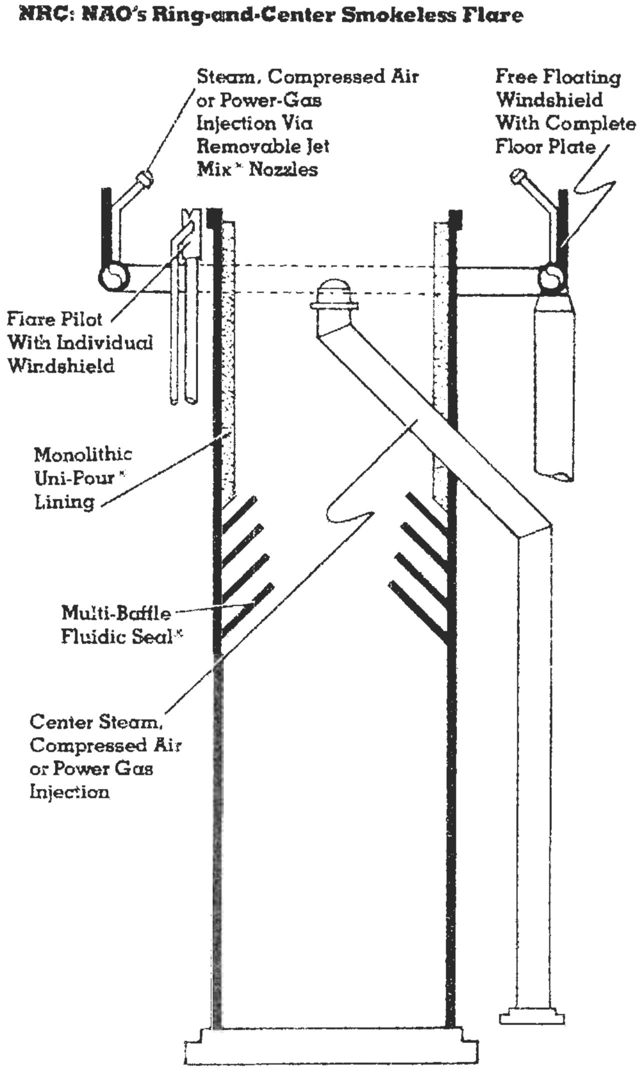


Figure 7-71. Flare stack arrangement for smokeless burning and backflash protection with Fluidic Seal® molecular seal. Steam can be injected into the flare to introduce air to the fuel by use of jets inside the stream and around the periphery. By permission, Straitz, J. F. III, "Make the Flare Protect the Environment," *Hydrocarbon Processing,* Oct. 1977, p. 131.

For specific details, consult a flare system design manufacturer.

$$W_{steam} = W_{hc}\ (0.68 - 10.8/M) \qquad (7\text{-}80)$$

where W_{steam} = steam injected, lbs/hr

W_{hc} = hydrocarbons to be flared, lbs/hr

M = molecular weight of hydrocarbons (average for mixture, hydrocarbons only)

This calculation is based on a steam-CO_2 weight ratio of approximately 0.7 [33A, Par 5.4.3.2.1].

These should be sized for conditions under which they will operate smokelessly.

Flame Length [33c]

$$\text{Heat liberated or released by flame, } Q_f = (W)\ (H_c) \qquad (7\text{-}81)$$

where Q_f = heat released by flame, BTU/hr

W_{hc} = W = gas/vapor flow rate, lb/hr

H_c = heat of combustion of gas/vapor, BTU/lb

Note: For many hydrocarbon-air mixtures, the value of H_c ranges 20,000 to 22,000 BTU/lb.

Referring to Figure 7-72 at the calculated heat release, H_c, read the flame length, and refer to dimensional diagram for flame plume from a stack, Figure 7-73.

Flame Distortion [33c] Caused by Wind Velocity

Referring to Figure 7-74, the flame distortion is determined as $\frac{\Delta x}{L}$ or $\frac{\Delta y}{L}$. Calculate U_j using the "d" determined for the selected Mach No. in earlier paragraph.

$$\frac{U_\infty}{U_j} = \frac{\text{wind velocity}}{\text{flare tip velocity}} \qquad (7\text{-}82)$$

$$U_j = (\text{flow})/(\pi d^2/4),\ \text{ft/sec} \qquad (7\text{-}83)$$

Flow, F_1 = (W/3600) (379.1/MW) [460+°F/520], cu ft/sec based on 60°F and 14.7 psia, and 359 cu ft/mol

U_∞ = lateral wind velocity, ft/sec

U_j = exit gas velocity from stack, ft/sec

°F = flowing temperature

Reference [60] presents an alternate calculation method for flame distortion.

Read, U_∞/U_j on Figure 7-74

and determine ratio: $\Delta y/L_f$ = a (vertical)

$\Delta x/L_f$ = b (horizontal)

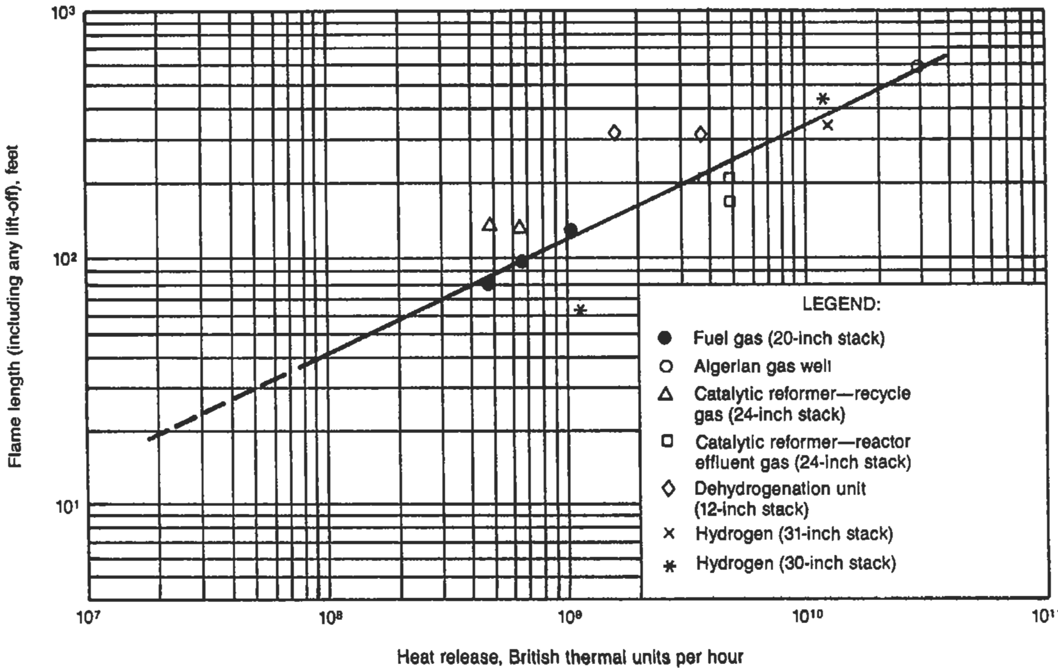
Then vertical: $\Delta y = L_f(a)$

Horizontal: $\Delta x = L_f(b)$

L_f = length of flame, ft

Flare Stack Height

The importance of the stack height (see Figure 7-73) is (a) to discharge the burning venting gases/vapors sufficiently high into the air so as to allow safe dispersion and



Note: Multiple points indicate separate observations or different assumptions of heat content.

Figure 7-72. Flame length versus heat release: industrial sizes and releases (customary units). Reprinted by permission, American Petroleum Institute, API RP-521, *Guide for Pressure Relieving and Depressuring Systems,* 3rd Ed., Nov. 1990 [33].

(b) to keep the flare flame of burning material sufficiently high to prevent the radiated heat from damaging equipment and facilities and from creating a life safety hazard to ground personnel. Figure 7-75 summarizes the accepted data for heat radiation related to human exposure time. Figure 7-76 summarizes the maximum radiation intensity related to a human escape time, allowing a 5-second reaction time to take action to escape, before the heat intensity injures the individual. Kent [60] suggests an escape velocity of 20 feet/second. The heat radiation is an important factor in locating/spacing of equipment with respect to one or more flares. The use of protective clothing and safety hard hats aids in extending the time of exposure when compared to bare skin.

The distance required between a flare stack venting and a point of exposure to thermal radiation is expressed [57] [33C]:

$$D_F = \sqrt{\tau F Q_r/(4\pi K)} \quad (7\text{-}84)$$

where D_F = minimum distance from the midpoint of a flame to the object, at ground level, ft (see Figure 7-73) (Note that this is not the flare stack height, but a part of calculation procedure)

F = fraction of heat radiated

This references to the total heat of combustion of a flame and selected values are [57, 33C]:

Hydrocarbon	F range	F range average
Methane	0.10 to 0.20*	0.15
Natural Gas	0.19 to 0.23	0.21
Propane	—	0.33**
Butane	0.21 to 0.30	0.28
Hydrogen	0.10 to 0.17	0.15

*0.20 used for methane with carbon weight ratio of 0.333.
**With weight ratio of 0.222.

When in doubt, to be safe use 0.4 [57] or 1.0 [33C].

τ = fraction heat intensity k transmitted through the atmosphere, usually assumed = 1.0 (see later equation for modifying) [33c].

Q_r = heat release (lower heating valve), BTU/hr

Kent [60] proposes total heat release:

$$Q_r = W\,\Sigma n h_c\,(379/M) \quad (7\text{-}85)$$

or (59) Q_n = 20,000 W

where M = molecular weight

h_c = net calorific heat value, BTU/SCF

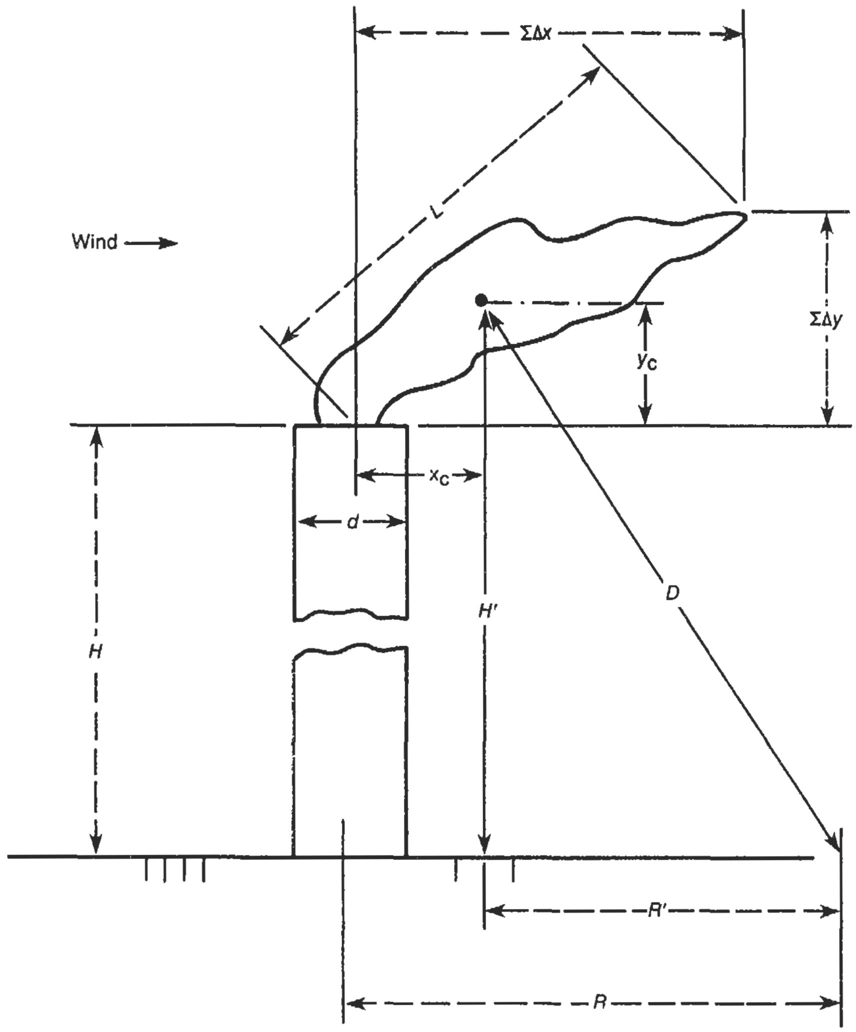


Figure 7-73. Dimensional references for sizing a flare stack. Reprinted by permission, American Petroleum Institute, API RP-521, *Guide for Pressure Relieving and Depressuring Systems,* 3rd Ed., Nov. 1990 [33].

n = mol fraction combustion compound(s)
f = fraction of radiated heat
$f = 0.20\ [h_c/900]^{1/2}$
h_c = 50 M + 100 for hydrocarbons BTU/SCF (LHV) @ 14.7 psia and 60°F
$h_c = \Sigma n h_c$ for gas mixtures, BTU/SCF
W = gas/vapors flow, lb/hr
K = allowable radiation, BTU/hr/sq ft (See Table 7-32) Select acceptable value for "conditions assumed"

Reasonable heat intensity, K, values are 1,500 BTU/hr/sq ft. When referred to Table 7-32

F = fraction of heat radiated

$\tau = 0.79\ (100/r)^{1/16}\ (100/D_F)^{1/16}$, from Ref. [33C].

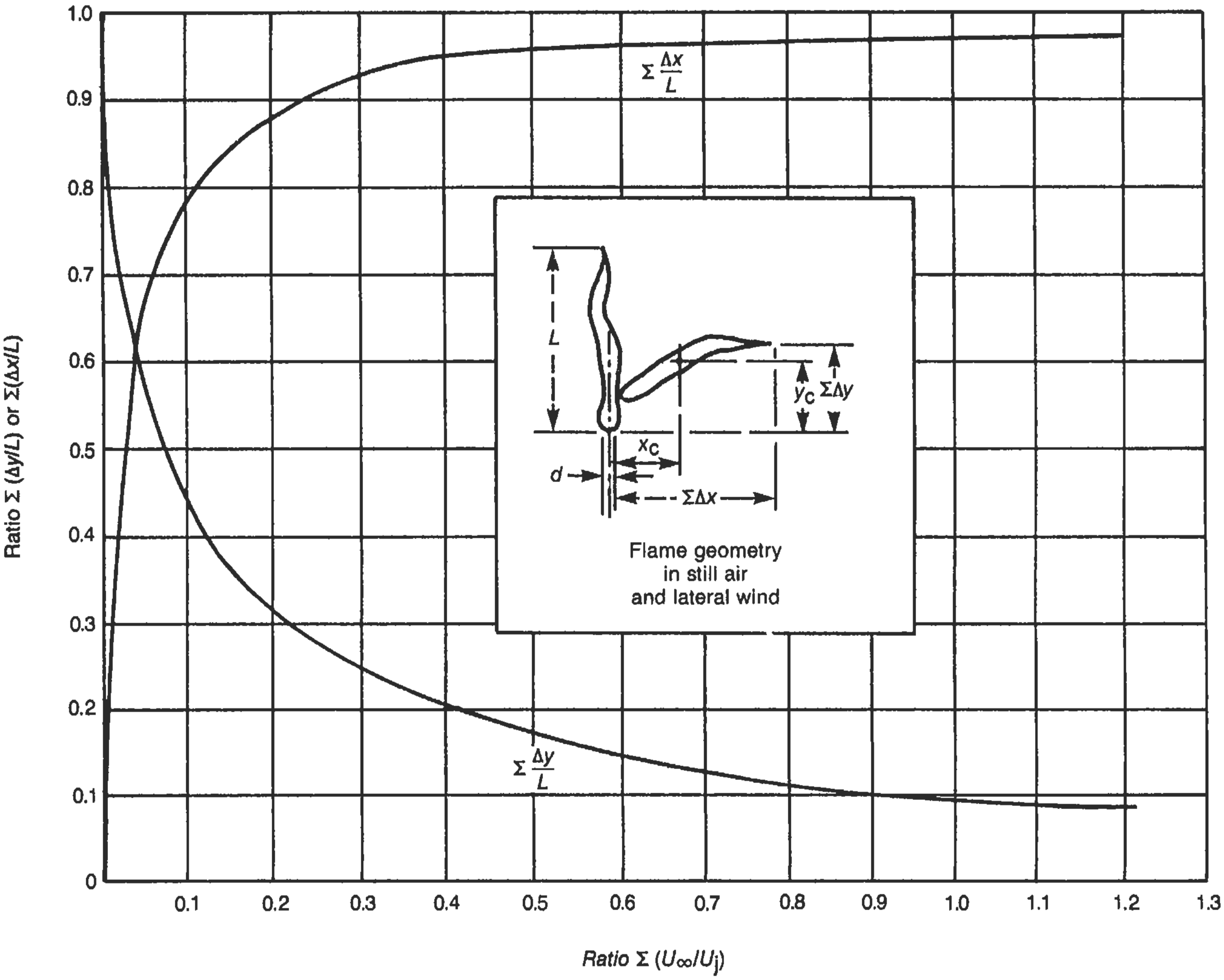
r_1 = relative humidity, %

When steam is injected at a rate of approximately 0.3 pounds of steam per pound of flare gas, the fraction of heat radiated is decreased by 20%. τ is based on hydrocarbon flame at 2240°F, 80 °F dry bulb air, relative humidity > 10%, distance from flame between 100 and 500 feet, and is acceptable to estimate under wide conditions.

A slightly altered form of the D_F equation above [61, 62] for spherical radiation:

$$I = (\text{flow})\ (\text{NHV})\ (\varepsilon)/(4\pi D_F^2),\ \text{BTU/hr/sq ft} \qquad (7\text{-}86)$$

where I = radiation intensity at point of object on ground level from midpoint of flame, Figures 7-73 and 7-74
Flow = gas flow rate, lb/hr (or SCFH)
NHV = net heating value of flare gas, BTU/lb or, (BTU/SCF)
ε = emissivity (see table, pg. 525)



U_∞ = lateral wind velocity.
U_j = exit gas velocity from stack.

Figure 7-74. Approximate flame distortion due to lateral wind on jet velocity from flare stack. Reprinted by permission, American Petroleum Institute, API RP-521, *Guide for Pressure Relieving and Depressuring Systems,* 3rd Ed., Nov. 1990 [33].

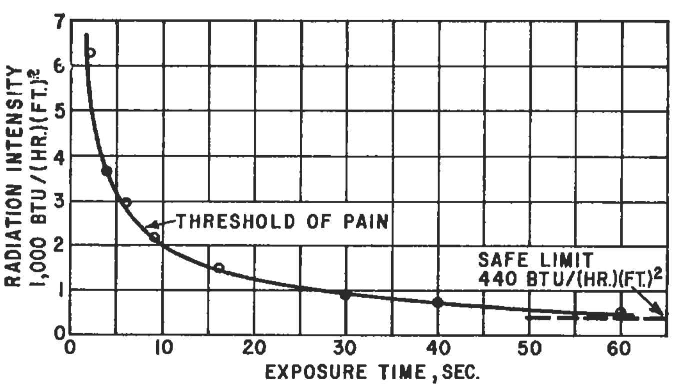


Figure 7-75. Heat radiation intensity vs. exposure time for bare skin at the threshold of pain. By permission, Kent, *Hydrocarbon Processing,* V. 43, No. 8 (1964), p. 121 [60].

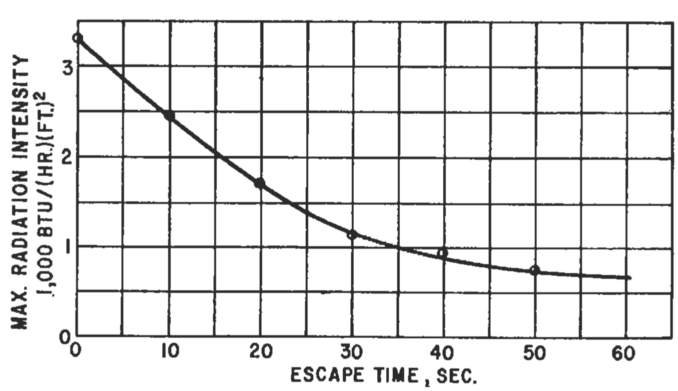


Figure 7-76. Maximum radiation intensity vs. escape time based on 5 seconds reaction time. By permission, Kent, *Hydrocarbon Processing,* V. 43, No. 8 (1964), p. 121 [60].

Table 7-32
Recommended Design Flare Radiation Levels Including Solar Radiation

Permissible Design Level (K)		
British Thermal Units per Hour per Square Foot	Kilowatts per Square Meter	Conditions
5000	15.77	Heat intensity on structures and in areas where operators are not likely to be performing duties and where shelter from radiant heat is available (for example, behind equipment)
3000	9.46	Value of K at design flare release at any location to which people have access (for example, at grade below the flare or a service platform of a nearby tower); exposure should be limited to a few seconds, sufficient for escape only
2000	6.31	Heat intensity in areas where emergency actions lasting up to 1 minute may be required by personnel without shielding but with appropriate clothing
1500	4.73	Heat intensity in areas where emergency actions lasting several minutes may be required by personnel without shielding but with appropriate clothing
500	1.58	Value of K at design flare release at any location where personnel are continuously exposed

Note:
On towers or other elevated structures where rapid escape is not possible, ladders must be provided on the side away from the flare, so the structure can provide some shielding when K is greater than 2000 British thermal units per hour per square foot (6.31 kilowatts per square meter).
Reprinted by permission, API RP-521, *Guide for Pressure Relieving and Depressuring Systems,* 3rd Ed., Nov. 1990, American Petroleum Institute [33].

Height of stack for still air [60]:

$$H = \left(L^2 + \frac{f Q_r}{\pi q_M} \right)^{0.5} - L \qquad (7\text{-}87)$$

The shortest stack exists when q_M = 3,300 BTU/hr sq ft (Figure 7-76). The limiting radial distance from the flame, allowing for speed of escape of 20 ft/sec is [60]:

$$y = 20\ t_e = [x^2 - H\ (H + L)]^{0.5} \qquad (7\text{-}88)$$

where H = height of flare stack, ft, Figure 7-73
L = height of flame (length of flame from top of stack to flame tip), ft
D = X = radial distance from flame core (center) to grade, ft
R = y = radial distance from base of stack, ft, to grade intersection with D
t_e = time interval for escape, sec
Note: For safety, personnel and equipment should be outside the "y" distance.
R = distance from flame center to point X on ground (see Figure 7-77)

This has been shown to be quite accurate for distances as close to the flame as one flame length [61].

Emissivity Values [62]	
Carbon Monoxide	0.075
Hydrogen	0.075
Hydrogen Sulfide	0.070
Ammonia	0.070
Methane	0.10
Propane	0.11
Butane	0.12
Ethylene	0.12
Propylene	0.13
Maximum	0.13

Length of Flame [62] See Figure 7-77

$$L_f = 10\ (D)\ (\Delta P_t/55)^{1/2} \qquad (7\text{-}89)$$

where L_f = length of flame, ft
D = flare tip diameter, in.
ΔP_t = pressure drop at the tip, in. of water

This gives flame length for conditions other than maximum flow.

The center of the flame is assumed to be located a distance of one-third the length of the flame from the tip, $L_f/3$ [62]. The flame angle is the vector addition of the wind velocity and the gas exit velocity.

$$V_{exit}, \text{ gas exit velocity} = 550\sqrt{\Delta P_t/55}\ , \text{ ft/sec} \qquad (7\text{-}90)$$

From Figure 7-77

$$X_c = (L_f/3)\ (\sin\theta)$$

$$Y_c = (L_f/3)\ (\cos\theta)$$

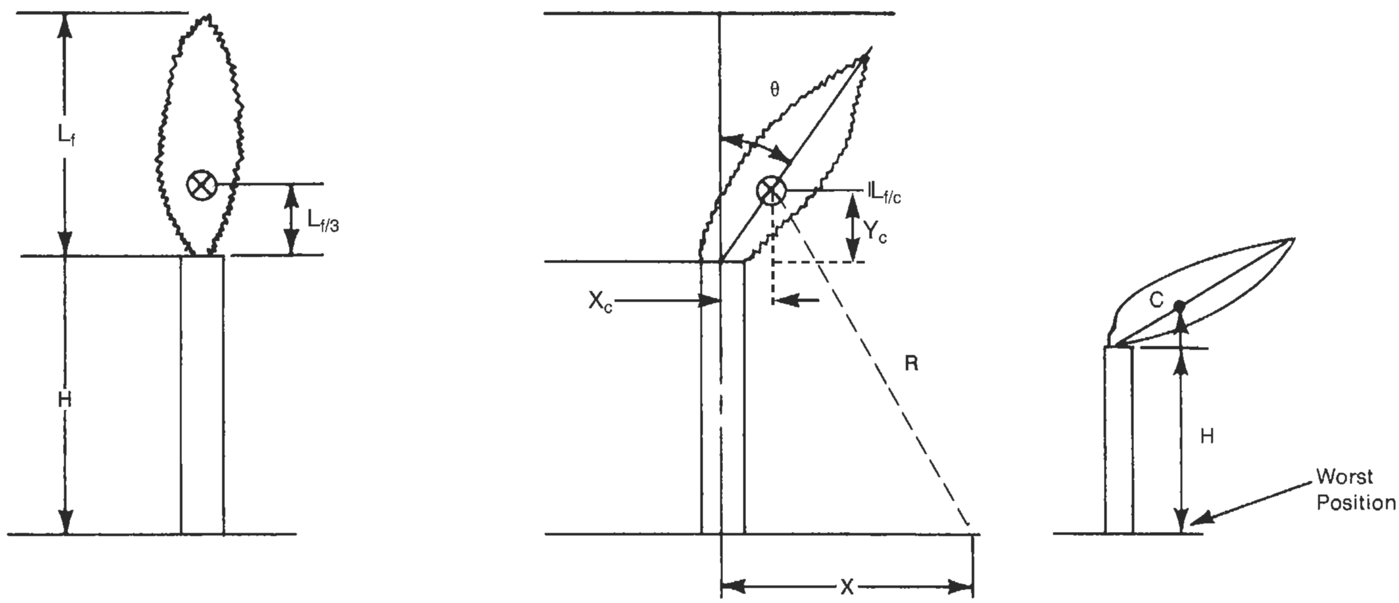


Figure 7-77. Diagrams for alternate flare stack designs of Straitz. By permission, Straitz, J. F. III and Altube, R. J., NAO, Inc. [62].

$$\text{Distance, } R = \sqrt{(X - X_c)^2 + (H + Y_c)^2} \qquad (7\text{-}91)$$

Worst condition of gas flow and wind velocities, *vertically* below flame center:

then $R = H + Y_c$
$H = R - (L_f/3)(\cos \theta)$
$\theta = \tan^{-1}(V_{wind}/V_{gas\ exit})$

This assumes that the flame length stays the same for any wind velocity that is not rigidly true. With a wind greater than 60 miles/hr, the flame tends to shorten. Straitz [62] suggests that practically this can be neglected.

Design values for radiation levels usually used [62]:

1. Equipment protection: 3,000 BTU/hr/sq ft
2. Personnel, short time exposure: 1,500 BTU/hr/sq ft
3. Personnel, continuous exposure: 440 BTU/hr/sq ft
4. Solar radiation adds to the exposure, so on sunny days, continuous personnel exposure: 200 − 300 BTU/hr/sq ft

Determine flare stack height above ground (grade):

Refer to Figure 7-73. Based on the mach velocity of the vapor/gases leaving the top tip of the flare stack (see Equation 7-76), determine the mach number, e.g., 0.2, then from Figure 7-73:

$$\text{where } H' = H + \tfrac{1}{2}(\Delta y) \qquad (7\text{-}92)$$

$$\text{and } R' = R - \tfrac{1}{2}(\Delta x) \qquad (7\text{-}93)$$

Δy and Δx from previous calculations under flame distortion

Refer to Table 7-32 and select the "condition" for radiation level, K, and ground distance, R, from stack.

Solve for R′ using the ground distance selected, R, from stack, and use the Δx previously calculated.

Then, determined height of stack, H, by

$$D^2 = (R')^2 + (H')^2 \qquad (7\text{-}94)$$

Substitute the previously calculated value of the distance from center of flame to grade, D and also R′.

First solve for H′, then

$$H \text{ (height of stack)} = H' - 1/2(\Delta y) \text{ (previously calculated)} \qquad (7\text{-}95)$$

Purging of Flare Stacks and Vessels/Piping

- Vacuum cycle
- Pressure cycle
- Continuous, flow through

There are several different approaches to purging:

Purging a system of flammable gas/vapor mixtures generally involves adding an inert gas such as nitrogen to the system. Sometimes the volumes of nitrogen are large, but it is still less expensive than most other nonflammable gas (even CO and CO2 have to be used cautiously) and certainly air cannot be used because it introduces oxygen that could aggravate the flammability problem of flammability limits. Also see [75].

Pressure Purging

The inert gas is added under pressure to the system to be purged. This is then vented or purged to the atmosphere, usually more than one cycle of pressurization followed by venting is necessary to drop the concentration of a specific flammable or toxic component to a pre-established level.

To determine the number of purge cycles and achieve a specified component concentration after "j" purge cycles of pressure (or vacuum) and relief [29]:

$$y_j = y_o\,(n_L/n_H)^j - y_o\,(P_L/P_H)^j \qquad (7\text{-}96)$$

Repeat the process as required to decrease the oxidant concentration to the desired level.

where P_H = initial high pressures, mmHg
P_L = initial low pressure or vacuum, mmHG
y_o = initial concentration of component (oxidant) under low pressure, mol fraction
n_H = number of mols at pressure condition
n_L = number of mols at atmospheric pressure or low pressure conditions
j = number of purge cycles (pressuring and relief)

Note: The above equation assumes pressure limits P_H and P_L are identical for each cycle and the total mols of nitrogen added for each cycle is constant [29].

Example 7-19. Purge Vessel by Pressurization following the method of Ref. [29].

A process vessel of 800 gallons capacity is to have the oxygen content reduced from 21% oxygen (air). The system before process startup is at ambient conditions of 14.7 psia and 80°F. Determine the number of purges to reduce the oxygen content to 1 ppm (10^{-6} lb mol) using purchased nitrogen and used at 70 psig and 80°F to protect the strength of the vessel. How much nitrogen would be required?

Using Equation 7-96:

y_o = initial mol fraction of oxygen. This is now the concentration of oxygen at end of the *first pressuring cycle* (not venting or purging).

At high pressure pressurization:

y_o = 21 lb mols oxygen/100 total mols in vessel (initial)

$$y_o = (0.21)\,(P_o/P_H), \text{ composition for the high pressure condition} \qquad (7\text{-}97)$$

P_o = beginning pressure in vessel, 14.7 psia
P_H = high pressure of the purge nitrogen
$y_o = (0.21)\,[14.7/(70 + 14.7)] = 0.03644$

The final oxygen concentration,

y_f is to be 1 ppm (10^{-6} lb mol/total mols)

$$y_f = y_o\,(P_L/P_H)^j \qquad (7\text{-}98)$$

$$10^{-6} = 0.21\,[14.7/(70 + 14.7)]^j$$

Solving by taking logarithms:

$$\ln\,[y_f/y_o] = j\,\ln\,[P_L/P_H]$$

$$\ln\,[10^{-6}/0.21] = j\,\ln\,[14.7/84.7]$$

$$j = 6.99 \text{ cycles}$$

Use seven minimum, perhaps use eight, for assurance that purging is complete. Note that the above relationships hold for vacuum purging. Keep in mind the relationships between high and low pressure of the system and use mmHg for pressure if more convenient. For sweep-through purging, see Ref. [29].

Total mols nitrogen required [29]:

$$\Delta n_{N_2} = j\,(P_H - P_L)\,[V/(R_g\,T)] \qquad (7\text{-}99)$$

$$= 7.0\,(84.7 - 14.7)\,[(800/7.48)/10.3\,(80 + 460)]$$
$$= 8.98 \text{ mols nitrogen}$$

lb nitrogen = 8.98 (28) = 125.72 lbs
V = 800 gal volume
R_g = 10.73 psi cu ft/lb mol °R
T = nitrogen temperature, 80°F + 460 = 540°R

Static Electricity

Static electrical charges cause major damage in chemical and refining plants, yet they are not often recognized in the planning and design details of many plant areas. One of the least commonly recognized situations is that dusts generated in plant operations can be ignited to explosive violence by static electrical charges built up on the small particles. Of course, there are other dangers of explosions/fires being ignited by static discharges involving flammable vapor and mists and liquid particles (larger than mists). (Also see Pratt [86].)

Although static discharges are small electrical phenomena, they are significantly different from a high voltage electrical discharge to ground from a power system or an arcing condition. These latter can certainly ignite dusts and vapors, but usually design efforts are made to properly insulate to prevent occurrences. Electrical codes aid in this design (see Chapter 14, Volume 3).

The use of intrinsically safe electrical and instrumentation equipment in appropriately designed environments can guard against many electrically related discharges. Reference should be made to authoritative books on this subject.

Static electricity is caused by the contact and separation of a good conductor material from a poor or nonconductor material or the separation of two nonconductor (or poor) materials. Static electricity is the accumulation of bound charges of the same sign and are prevented from reuniting quickly with charges of the opposite sign. This electrostatic phenomena is often characterized by the presence of high potential but small currents or charge quantities [63].

When two objects/particles separate after being in contact (equal charges), one particle loses electrons and becomes positively charged while the other gains electrons and becomes negatively charged.

Low humidity allows the resistance of insulating surfaces to increase to a very high level, and this allows electrostatic charge separation and accumulation to occur [64]. Static electricity is usually present in some degree in many industrial situations, but ignitions caused by static discharge are preventable. The charging process arises at an interface between dissimilar materials, that is, between hydrocarbons and metal or hydrocarbon and water [64]. The charge separation process occurs at the molecular level, but does not occur while the materials are in contact. When the charges are separated by moving the materials apart, the voltage potential rises. In a pipeline there is a "streaming current" established by charges off the inner pipe wall being carried in the fluid by the flow.

In storage or process tanks, a charge generation can occur if a liquid enters above the liquid surface by the spraying or splashing of the liquid and a charged mist may form [64] and the bulk liquid will become charged.

Figures 7-78A and 7-78B illustrate that when a charged fuel enters a metal tank, it attracts a charge of equal magnitude but opposite sign to the inside surface of the tank shell. At the same time a charge of the same magnitude and sign as the charge in the fuel is repelled to the outside surface of the tank. Note that Bustin [64] states the choice of signs on the fuel is arbitrary.

If the tank is grounded, the repelled charge is neutralized; hence, the tank stays at zero voltage (see Figure 7-78B). The charge from this neutralization current is equal in magnitude and sign to the charge carried into the tank by the liquid. The ammeter is exactly equal to the "streaming" current entering the tank [64].

Inside the tank a voltage difference exists between the negative charge on the shell and the positive charge in the liquid. For a grounded tank, the voltage is zero at the shell. Note that grounding a closed metal tank has no effect on the voltage difference between the two parts in the tank. Grounding a metal tank does not alter the risk of an electrostatic spark being generated within, but it does eliminate the possibility of an external spark discharge from the tank to ground.

Static electricity is classed as (a) spark discharges and (b) corona discharges. The spark is a quick, instantaneous release of charge across an air gap from one "electrode source" to another. The corona is a discharge that branches in a diffuse manner, spreading over a large area of a poor conductor or ending in space [64]. The current is weaker (less) from a corona than a spark [64]. For a flammable mixture to ignite, the electrical discharge must release sufficient minimum energy to allow ignition to take place, and this minimum energy varies between flammable hydrocarbons and between dusts.

To avoid electrostatic discharging or even charging, the following list of conditions suggested by Haase [63] should be considered:

- electrostatic grounding of all conducting surfaces
- increasing the conductivity of the materials

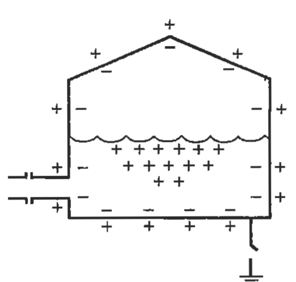


Figure 7-78A. Electrical charge induction in tank shell. By permission, Bustin & Dukek, *Electrostatic Hazards in the Petroleum Industry,* Research Studies Press, Somerset, England [64].

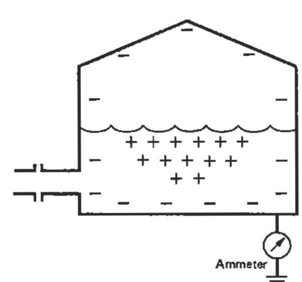


Figure 7-78B. Electrical induced positive charge, grounded tank. By permission, Bustin & Dukek, *Electrostatic Hazards in the Petroleum Industry,* Research Studies Press, Somerset, England [64].

- increasing the surface conductivity through the raising of the relative humidity or through surface treatment
- increasing the conductivity of the air
- relative low working speeds
- proper choice of contact materials
- proper control of the contact temperatures of the surfaces
- or a combination of the above

Note that charges can be transported by persons or containers from a nonhazardous area into an unsuspecting unsafe (or hazardous) area and ignition could then take place [63].

It is essential that the process hazardous atmosphere and the process system and handling of combustible hydrocarbons/chemicals be recognized in the physical designs by conforming to the appropriate class of atmosphere/environment codes specified by the National Electrical Code [71, 83, and 84].

Nomenclature

a = area, sq in.
a_p = cross-sectional area of the inlet pipe, sq ft
A = area, sq meters, sq ft, or sq in.; consistent with equation units
or A = nozzle throat area, or orifice flow area, effective discharge area (calculations required), or from manufacturer's standard orifice areas, sq in.
A_1 = initial vessel relief area, sq in. or sq meters
A_2 = second vessel relief area, sq in. or sq meters
A_3 = exposed surface area of vessel, sq ft
A_s = internal surface area of enclosure, sq ft or sq meters
A_v = vent area, sq meters or sq ft
A_w = total wetted surface area, sq ft
AIT = auto-ignition temperature
B = cubical expansion coefficient per °F of liquid at expected temperature (see tabulation in text)
BP = boiling point, °C or °F
B.P. = burst pressure, either psig or psi abs
Bar = 14.5 psi = 0.987 atmosphere = 100 k Pa atmosphere; 14.7 psia = 1.01 bars
$C_1 = c = C$ = gas/vapor flow constant depending on ratio of specific heats c_p/c_v (see Figure 7-25 sonic)
c_p/c_v = ratio of specific heats
C_{st} = stoichiometric composition of combustible vapor in air expressed as a volume percent
C_o = sonic flow discharge orifice constant, varying with Reynolds number
$C(psi)^{1/2}$ = venting equation constant, fuel characteristic constant for explosion venting equation, Table 7-27 (constant characterizes the fuel and clears the dimensional units (gases and vapors))
C_h = specific heat of trapped fluid, Btu/lb/°F
C_2 = subsonic flow constant for gas or vapor, function of $k = c_p/c_v$, Table 7-11
c = orifice coefficients for liquids
d = diameter, inches (usually of pipe)
d' = flare tip diameter, ft
$D = d_t$ = flare tip diameter, in.
D_F = minimum distance from midpoint of flame to the object, ft
dp/dt = rate of pressure rise, bar/sec or psi/sec
E = joint efficiency in cylindrical or spherical shells or ligaments between openings (see ASME Code Par.UW-12 or UG-53)
e = natural logarithm base, $e = 2.718$
e_t = TNT equivalent (explosion) (see Table 7-26)
F = environment factor for Table 7-8
F'_{gs} = relief valve factor for non-insulated vessels in gas service exposed to open fires
$F = F_h$ = fraction of heat radiated
F' = operating environment factor for safety relief of gas only vessels (see pg. 446)
F = Flow gas/vapor, cubic feet per minute at 14.7 psia and 60°F
F_u = The ratio of the ultimate stress of the vessel to the allowable stress of the vessel
F_y = ratio of the yield stress of the vessel to the allowable stress of the vessel
F_1 or F_2 = relief area for vessels 1 or 2 resp., sq ft
F_2 = coefficient of subcritical flow (see Figure 7-29)
°F = temperature, °Fahrenheit
f = specific relief area, sq meter/cubic meter, or area/unit volume
f_q = steam quality, dryness fraction
G = specific gravity of gas (air = 1), or specific gravity of liquid (water = 1) at actual discharge temperature
GPM = gallons per minute flow
g = acceleration of gravity, 32.0 ft/sec/sec
H_c = heat of combustion of gas/vapor, Btu/lb
H = total heat transfer rate, Btu/hr
h_1 = seal, submerged, ft (Figure 7-70)
h_L = seal pipe clearance, (Figure 7-70)
h = head of liquid, ft
h_c = net calorific heat value, Btu/SCF
j = number of purge cycles (pressurizing and relief)
K = permissible design level for flare radiation (including solar radiation), Btu/hr/sq ft, Table 7-32
K_p = liquid capacity correction factor for overpressures lower than 25% from Figure 7-22. Non-code equations only.
K_b = vapor or gas flow correction factor for constant back pressures above critical pressure from curve on Figure 7-26
K_v = vapor or gas flow factor for variable back pressures from Figure 7-27A or 27B. Applies to Balanced Seal valves only.
K_w = liquid correction factor for variable back pressures from Figure 7-28. Applies to balanced seal valves only. Conventional valves require no correction.
K_u = liquid viscosity correction factor from chart Figure 7-24
K_{sh} = steam superheat correction factor from Table 7-7
K_n = Napier steam correction factor for set pressures between 1500 and 2900 psig from Table 6
$K = K_d$ = coefficient of discharge:*

0.975 for air, steam, vapors and gases
0.724 for ASME Code liquids**
0.64 for non-ASME Code liquids
0.62 for bursting/rupture disk
*Where the pressure relief valve is used in series with a rupture disk, a combination capacity factor of 0.8 must be applied to the denominator of the above valve equations. Consult the valve manufacturer (also see specific section this chapter of text) for higher factors based on National Board flow test results conducted with various rupture disk designs/arrangements (see Table 7-12).
**For saturated water see ASME Code, Appendix 11-2.

K_D = discharge coefficient orifice or nozzle
K_G = deflagration index, maximum rate of pressure rise for gases, bar-meter/second = bar-m/sec
K_{st} = deflagration index, maximum rate of pressure rise for dusts, bar-meters/sec = bar-m/sec
K_w = variable or constant back pressure sizing factor, balanced valves, liquids only (Figure 7-28)
k = ratio of specific heats, c_p/c_v
L = liquid flow, gallons per minute
L_f = length of flame, ft
L_v = L = latent heat of vaporization, Btu/lb
L_x = distance between adjacent vents, meters or feet
LEL = lower explosive, or lower flammable limit, percent of mixture of flammable gases only in air
L_1, L_2, etc = lower flammability limits, vol % for each flammable gas in mixture
L_3 = longest dimension of the enclosure, ft
L/D = length-to-diameter ratio, dimensionless
M = molecular weight
m = meter or percent moisture, or 100 minus steam quality
MAWP = maximum allowable working pressure of a pressure vessel, psi gauge (or psi absolute if so specifically noted)
MP = melting point (freezing point), °C or °F
MR = universal gas constant = 1544 ft lb/lb sec-sec. Units depend on consistency with other symbols in equation, or manufacturing range for metal bursting/rupture disks.
mj = spark energy, milli-joules
m_{TNT} = mass of TNT, lb
n = moles of specified components
n_H = total number moles at pressure or atmospheric condition
n_L = total number mols at atmospheric pressure or low pressure or vacuum condition
P = relieving pressure, psia = valve set pressure + permissible overpressure, psig, + 14.7, or any pressure, bar (gauge), or a consistent set of pressure units. Minimum overpressure is 3 psi
$P_1 = P'$ = pressure, psia
p″ = maximum header exit pressure into seal, psig
P_b = stamped bursting pressure, plus overpressure allowance (ASME 10% or 3 psi, whichever is greater) plus atmospheric pressure (14.7), psia
$P_c = P_{crit}$ = critical pressure of a gas system, psi abs
P_d = design pressure of vessel or system to prevent deformation due to internal deflagration, psig
P_d = ASME Code design pressure (or maximum allowable working pressure), psi
P_{do} = pressure on outlet side of rupture disk, psia
P_e = exit or back pressure, psia, stamped burst pressure
P_{er} = perimeter of a cross section, ft or meters
P_H = Initial high pressure, mmHg
P_i = maximum initial pressure at which the combustible atmosphere exists, psig
P'_i = initial pressure of system, psia
P_L = initial low pressure or vacuum, mmHg
P_{max} = maximum explosion pressure, bar, or other consistent pressure units
P_{uv} = maximum pressure developed in an unvented vessel, bar (gauge) or psig
P_{op} = Normal expected or maximum expected operating pressure, psia
P_i; P_o = relieving pressure, psia, or sometimes upstream pressure, psi abs, or initial pressure of system
P_r = design pressure to prevent rupture due to internal deflagration, psig
P_{red} = reduced pressure termed maximum internal overpressure that can be withstood by the weakest structural element, psig, or bar ga, or kPa (gas/vapor), or maximum pressure actually developed during a vented deflagration
P_{red} = maximum pressure developed during venting, bar ga (dusts)
P_{stat} = vent closure release pressure, bar ga (dusts)
P_η = normal operating gas pressure, psia
ΔP_t = pressure drop at flare tip, inches water
P_t = pressure of the vapor just inside flare tips (at top), psia
P_1 = upstream relieving pressure, or set pressure at inlet to safety relief device, psig (or psia, if consistent)
P_2 = back pressure or downstream at outlet of safety relief device, psig, or psia, depending on usage
p = rupture pressure for disk, psig or psia
p° = overpressure (explosion), lb force/sq in.
p′ = pressure, psi abs
ΔP = pressure differential across safety relief valve, inlet pressure minus back pressure or down stream pressure, psi. Also = set pressure + overpressure, psig-back pressure, psig. At 10% overpressure delta P equals 1.1 $P_1 - P_2$. Below 30 psig set delta P equals $P_1 + 3 - P_2$.
Δp = differential pressure across liquid relief rupture disk, usually equals p, psig
ΔP(dusts) = pressure differential, bar or psi
Q = total heat absorption from external fire (input) to the wetted surface of the vessel, Btu/hr
Q′ = liquid flow, cu ft/sec
Q_A = required flow, cu ft/min at actual flowing temperature and pressure, ACFM
Q_f = heat released by flame, Btu/hr
Q_r = heat release, lower heating valve, Btu/hr
Q_s = required flow, cu ft/min at standard conditions of 14.7 psia and 60°F, SCFM
q = Average unit heat absorption, Btu/hr/sq ft of wetted surface
R = ratio of the maximum deflagration pressure to the maximum initial pressure, as described in NFPA Code-69, Par 5-3.3.1
also R = individual gas constant = MR/M = 1544/M

R' = adjusted value of R, for NFPA Code-69
R_{exp} = distance from center of explosion source to the point of interest, ft
R_e = Reynolds number (or sometimes, R)
R_g = Universal gas constant = 1544 = MR
R_{gc} = individual gas constant = MR/M
R_i = inside radius of vessel, no corrosion allowance added, in.
°R = temperature, absolute, degrees Rankin
$r = r_c$ = ratio of back pressure to upstream pressure, P_2/P_1, or critical pressure ratio, P_c/P_1
r_1 = relative humidity, percent
S = maximum allowable stress in vessel wall, from ASME Code, psi., UCS-23.1–23.5; UHA-23, UHT-23
S' = SpGr = specific gravity of liquid, referenced to water at the same temperature
$S_G = S_g$ = SpGr of gas relative to air, equals ratio of mol wt of gas to that of air, or liquid fluid specific gravity relative to water, with water = 1.0 at 60°F
SpGr = specific gravity of fluid, relative to water = 1.0
St = dust hazard class
St St = stainless steel
SSU = viscosity Saybolt universal seconds
°S = degrees of superheat, °F
T = absolute inlet or gas temperature, degrees Rankin °R = °F + 460, or temperature of relief vapor [26], °R
T_η = normal operating gas temperature, °R
T_i = operating temperature, °C (NFPA Code-69)
T_o = temperature of service, °R
T_L = equilibrium temperature at which the lower flammable limit composition exists over liquid in dry air at one atmosphere (theoretical flash point), °C or °F
T_{st} = equilibrium temperature at which C_{st} exists over liquid in dry air at one atmosphere, °C or °F
T_u = equilibrium temperature at which the upper flammable limit composition exists over liquid in dry air at one atmosphere, °C or °F
T_w = vessel wall temperature, °R
T_1 = gas temperature, °R, at the upstream pressure, determined from $T_1 = (P_1/P_\eta)\ (T_\eta)$
t = minimum required thickness of shell of vessel, no corrosion, inches
U_∞ = viscosity at flowing temperature, Saybolt universal seconds
U = lateral wind velocity, ft/sec
U_j = flare tip velocity, ft/sec
UEL = upper explosive or flammable limit, percent of mixture of flammable gases only in air
V = velocity, ft/sec, or dust vessel volume, cu ft
or, V = vessel volume, cubic meters or cubic feet, or required gas capacity in SCFM
or, V = vapor flow required through valve (sub-critical), Std cu ft/min at 14.7 psia and 60°F
$\overline{V}$ = specific volume of fluid, cu ft/lb
V_a = required air capacity, SCFM
V_c = cubic feet of free air per hour from Table 7-17, which is 14.7 psia and 60°F, or from Equation 7-49, for wetted area $A_w > 2800$ sq ft
V' = venting requirement, cubic feet free air per hour at 14.7 psia and 60°F
V_L = flow rate at flowing temperature, U.S. gallons per minute, or required liquid capacity in U.S. gallons per minute
v = shock velocity, ft/sec or ft/min (depends on units selected)
v_1 = specific volume of gas or vapor at upstream or relief pressure and temperature conditions, cu ft/lb
v_s = sonic velocity of gas, ft/sec
v_1, v_2 = volume percent of each combustible mixture, free from air or inert gas
W = required vapor capacity in pounds per hour, or any flow rate in pounds per hour, vapor relief rate to flare stack, lbs/hr
W_c = charge weight of explosive, lb
W_e = effective charge weight, pounds of TNT for estimating surface burst effects in free air
W_s = required steam capacity flow or rate in pounds per hour, or other flow rate, lb/hr
W_{hc} = hydrocarbon to be flared, lbs/hr
W_{TNT} = equivalent charge weight of TNT, lb
W_L = liquid flow rate, gal per min (gpm)
W_{steam} = steam injected into flare, lbs/hr
w = charge weight of explosives of interest, lb
y_f = final oxidant concentration, mol fraction
y_j = specified component concentration after "j" purges
y_o = initial concentration of component (oxidant) under low pressure, mol fraction
Z = compressibility factor, deviation of actual gas from perfect gas law. Usually Z = 1.0 at low pressure below 300 psig.
Z, or Z_{TNT} = scaled distance for explosive blasts, ft/(lb)$^{1/3}$
z = actual distance for explosion damage, feet

Subscripts

1 = condition 1
2 = condition 2

Greek Symbols

β = beta ratio orifice diameter to pipe diameter (or nozzle inlet diameter)
ε = (epsilon) emissivity value
λ = (lambda) yield factor, $(W/W_o)^{1/3}$, with subscript "o" referring to reference value
μ = (mu) absolute viscosity at flowing temperature, centipoise (cp)
π = (pi), 3.1418
ρ = (rho) fluid density, lb/cu ft
τ = (tau) fraction heat intensity transmitted

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Appendix

A-1.
Alphabetical Conversion Factors

A

Acre	x 10	= Square chain (Gunters)
Acre	x 160	= Rods
Acre	x 1 x 10^5	= Square links (Gunters)
Acre	x 0.4047	= Hectare or square hectometer
Acres	x 43,560	= Square feet
Acres	x 4,047	= Square meters
Acres	x 1.562 x 10^{-3}	= Square miles
Acres	x 4,840	= Square yards
Acre-feet	x 43,560	= Cubic feet
Acre-feet	x 3.259 x 10^5	= Gallons
Amperes/square centimeters	x 6.452	= Amperes/square inch
Amperes/square centimeters	x 10^4	= Amperes/square meter
Amperes/square inch	x 0.1550	= Amperes/square centimeters
Amperes/square inches	x 1,550	= Amperes/square meter
Amperes/square meter	x 10^{-4}	= Amperes/square centimeter
Amperes/square meter	x 6.452 x 10^{-4}	= Amperes/square inch
Ares	x 0.02471	= Acre (USA)
Ares	x 119.60	= Square yards
Ares	x 100	= Square meters
Atmospheres	x 14.7	= Pounds/square inch
Atmospheres	x 1.058	= Tons/square foot
Atmospheres	x 29.92	= Inches of mercury at 0°C
Atmospheres	x 76	= Centimeters of mercury
Atmospheres	x 33.90	= Feet of water at 4°C
Atmospheres	x 1.0333	= Kilograms/square centimeter
Atmospheres	x 10,332	= Kilograms/square meter
Atmospheres	x 1,013.2	= Millibar
Atmospheres	x 760	= Millimeters of mercury
Atmospheres	x 10.332	= Meters of water at 4°C

(Continued on next page)

A-1.
(Continued). Alphabetical Conversion Factors

B

Barrels (USA, dry)	x 7,056	= Cubic inches
Barrels (USA, dry)	x 105	= Quarts (dry)
Barrels (USA, liquid)	x 31.5	= Gallons
Barrels (oil)	x 42	= Gallons (oil)
Barrels (oil)	x 0.159	= Cubic meters
Barrels (oil)	x 159	= Liters
Barrels/day	x 6.6245 x 10^{-3}	= Cubic meters/hour
Barrels (oil)	x 5.6154	= Cubic feet
Barrels/day	x 29.167 x 10^{-3}	= Gallons per minute
Bars	x 0.9869	= Atmospheres
Bars	x 10^{6}	= Dynes/square centimeter
Bars	x 1.020 x 10^{4}	= Kilograms/square meter
Bars	x 2,089	= Pounds/square foot
Bars	x 14.50	= Pounds/square inch
Board feet x 144 sq.in.	x 1 in.	= Cubic inches
Board feet	x 0.0833	= Cubic feet
Btu	x 1.055 x 10^{10}	= Ergs
Btu	x 778.3	= Foot-pounds
Btu	x 252	= Gram-calories
Btu	x 3.931 x 10^{-4}	= Horsepower-hours
Btu	x 1,054.8	= Joules
Btu	x 0.252	= Kilogram-calories
Btu	x 107.5	= Kilogram-meters
Btu	x 2.928 x 10^{-4}	= Kilowatt-hours
Btu/hour	x 0.2162	= Foot-pounds/second
Btu/hour	x 0.070	= Gram-calorie/second
Btu/hour	x 3.929 x 10^{-4}	= Horsepower-hours (British)
Btu/hour	x 0.2931	= Watts
Btu/hour foot °F	x 1.4882	= Kilocalorie/meter hour °C
Btu/hour foot^2	x 2.7125	= Kilocalorie/meter^2 hour
Btu/pound	x 0.5556	= Kilocalorie/kilogram
Btu/pound °F	x 1	= Kilocalorie/kilogram °C
Btu inches/hour foot^2 °F	x 0.12402	= Kilocalorie/meter hour °C
Btu/minute	x 12.96	= Foot-pounds/second
Btu/minute	x 0.02356	= Horsepower
Btu/minute	x 0.01757	= Kilowatts
Btu/minute	x 17.57	= Watts
Btu/square foot/minute	x 0.1221	= Watts/square inch
Bucket (British-dry)	x 1.818 x 10^{4}	= Cubic centimeters
Bushels	x 1.2445	= Cubic feet
Bushels	x 2,150.4	= Cubic inches
Bushels	x 0.03524	= Cubic meters
Bushels	x 35.24	= Liters
Bushels	x 4	= Pecks
Bushels	x 64	= Pints (dry)
Bushels	x 32	= Quarts (dry)

A-1.
(Continued). Alphabetical Conversion Factors

C

Calories, gram (mean)	x 3.9685×10^{-3}	= Btu (mean)
Calories/gram	x 1.8	= Btu/pound
Calories/gram-mol-°C	x 1.0	= Btu/pound-mol-°F
Candle/square centimeter	x 3.142	= Lamberts
Candle/square inch	x 0.487	= Lamberts
Centares	x 1.0	= Square meters
Centigrade	x 9/5 + 32	= Fahrenheit
Centiliter	x 0.3382	= Fluid ounce (USA)
Centiliter	x 0.6103	= Cubic inch
Centiliter	x 2.705	= Drams
Centimeters	x 3.281×10^{-2}	= Feet
Centimeters	x 0.3937	= Inches
Centimeters	x 1.094×10^{-2}	= Yards
Centimeters	x 393.7	= mils
Centimeters of mercury	x 0.1934	= Pounds/square inch
Centimeters of mercury	x 0.01316	= Atmospheres
Centimeters of mercury	x 0.4461	= Feet of water
Centimeters of mercury	x 136	= Kilograms/square meter
Centimeters of mercury	x 27.85	= Pounds/square foot
Centimeters of mercury	x 0.1934	= Pounds/square inch
Centimeters/second	x 1.1969	= Feet/minute
Centimeters/second	x 0.03281	= Feet/second
Centimeters/second	x 0.036	= Kilometers/hour
Centimeters/second	x 0.1943	= Knots
Centimeters/second	x 0.6	= Meters/minute
Centimeters/second	x 0.02237	= Miles/hour
Chain	x 792	= Inches
Chain	x 20.12	= Meters
Circumference	x 6.283	= Radians
Cords	x 8	= Cord feet
Cord feet	x 16	= Cubic feet
Coulomb	x 2.998×10^{9}	= Statcoulombs
Coulombs	x 1.036×10^{-5}	= Faradays
Coulombs/square centimeter	x 64.52	= Coulombs/square inch
Coulombs/square centimeter	x 10^{4}	= Coulombs/square meter
Coulombs/square inch	x 0.155	= Coulombs/square centimeter
Coulombs/square inch	x 1,550	= Coulombs/square meter
Cubic centimeters	x 3.531×10^{-5}	= Cubic feet
Cubic centimeters	x 0.06102	= Cubic inches
Cubic centimeters	x 10^{-6}	= Cubic meters
Cubic centimeters	x 1.308×10^{-6}	= Cubic yards
Cubic centimeters	x 2.624×10^{-4}	= Gallons (USA liquid)
Cubic centimeters	x 0.001	= Liters
Cubic centimeters	x 2.113×10^{-3}	= Pints (USA liquid)
Cubic centimeters	x 1.057×10^{-3}	= Quarts (USA liquid)
Cubic feet	x 0.8036	= Bushels (dry)

A-1.
(Continued). Alphabetical Conversion Factors

Cubic feet	x 28,320	= Cubic centimeters
Cubic feet	x 1,728	= Cubic inches
Cubic feet	x 0.02832	= Cubic meters
Cubic feet	x 0.03704	= Cubic yards
Cubic feet	x 7.48052	= Gallons (USA liquid)
Cubic feet	x 6.232	= Gallons (imperial)
Cubic feet	x 6.428	= Gallons (USA dry)
Cubic feet	x 62.425	= Pounds (water)
Cubic feet	x 28.32	= Liters
Cubic feet	x 59.84	= Pints (USA liquid)
Cubic feet	x 29.92	= Quarts (USA liquid)
Cubic feet	x 0.1781	= Barrels (oil, USA)
Cubic feet/minute	x 472	= Cubic centimeters/second
Cubic feet/minute	x 0.1247	= Gallons/second
Cubic feet/minute	x 0.4720	= Liters/second
Cubic feet/minute	x 62.43	= Pounds of water/minute
Cubic feet/minute	x 1.6989	= Cubic meters/hour
Cubic feet/minute	x 4.719×10^{-4}	= Cubic meters/second
Cubic feet/second	x 0.646317	= Million gallons/day
Cubic feet/second	x 448.831	= Gallons/minute
Cubic feet/second	x 101.94	= Cubic meters/hour
Cubic inches	x 16.39	= Cubic centimeters
Cubic inches	x 5.787×10^{-4}	= Cubic feet
Cubic inches	x 1.639×10^{-5}	= Cubic meters
Cubic inches	x 2.143×10^{-5}	= Cubic yards
Cubic inches	x 4.329×10^{-3}	= Gallons (U.S.)
Cubic inches	x 0.01639	= Liters
Cubic inches	x 1.061×10^{5}	= Mil-feet
Cubic inches	x 0.03463	= Pints (USA liquid)
Cubic inches	x 0.01732	= Quarts (USA liquid)
Cubic meters	x 6.290	= Barrels (USA oil)
Cubic meters	x 28.38	= Bushes (dry)
Cubic meters	x 10^{6}	= Cubic centimeters
Cubic meters	x 35.314	= Cubic feet
Cubic meters	x 61,023	= Cubic inches
Cubic meters	x 1.308	= Cubic yards
Cubic meters	x 264.17	= Gallons (USA liquid)
Cubic meters	x 220	= Gallons (imperial)
Cubic meters	x 1,000	= Liters
Cubic meters	x 2,113	= Pints (USA liquid)
Cubic meters	x 1,057	= Quarts (USA liquid)
Cubic meters/hour	x 9.810×10^{-3}	= Cubic feet/second
Cubic meters/hour	x 0.5886	= Cubic feet/minute
Cubic meters/hour	x 4.4033	= Gallons/minute (USA)
Cubic meters/hour	x 150.95	= Barrels/day
Cubic meters/hour	x 3.6651	= Imperial gallons/minute
Cubic meters/hour	x 35.31	= Cubic feet/hour

A-1.
(Continued). Alphabetical Conversion Factors

Cubic meters/hour	x 277.8	= Cubic centimeters/second
Cubic yards	x 7.646×10^5	= Cubic centimeters
Cubic yards	x 27	= Cubic feet
Cubic yards	x 46,656	= Cubic inches
Cubic yards	x 0.7646	= Cubic meters
Cubic yards	x 202	= Gallons (USA liquid)
Cubic yards	x 764.6	= Liters
Cubic yards	x 1,615.9	= Pints (USA liquid)
Cubic yards	x 807.9	= Quarts (USA liquid)
Cubic yards/minute	x 0.45	= Cubic feet/second
Cubic yards/minute	x 3.367	= Gallons/second
Cubic yards/minute	x 12.74	= Liters/second
Cubit, Bible	x 21.8	= Inch
Cup	x 0.5	= Pint
Cup	x 16	= Tablespoon

D

Dalton	x 1.650×10^{-24}	= Gram
Days	x 1,440	= Minutes
Days	x 86,400	= Seconds
Decigrams	x 0.1	= Grams
Deciliters	x 0.1	= Liters
Decimeters	x 0.1	= Meters
Degrees (Angle)	x 60	= Minutes
Degrees (Angle)	x 3600	= Seconds
Degrees (Angle)	x 0.01745	= Radians
Degrees (Angle)	x 0.01111	= Quadrants
Degrees/second	x 0.1667	= Revolutions/minute
Degrees/second	x 2.778×10^{-3}	= Revolutions/second
Dekagrams	x 10	= Grams
Dekaliters	x 10	= Liters
Dekameters	x 10	= Meters
Drams (apothecaries or troy)	x 0.1371429	= Ounces (avoirdupois)
Drams (apothecaries or troy)	x 0.125	= Ounces (troy)
Drams	x 27.34375	= Grains
Drams	x 1.771845	= Grams
Drams	x 0.0625	= Ounces
Dyne/centimeter	x 0.01	= Erg/square millimeters
Dyne/square centimeters	x 9.869×10^{-7}	= Atmospheres
Dyne/square centimeters	x 2.953×10^{-5}	= Inch of mercury at 0°C
Dyne/square centimeters	x 4.015×10^{-4}	= Inch of water at 4°C
Dynes/square centimeters	x 10^{-6}	= Bars
Dynes	x 1.020×10^{-3}	= Grams
Dynes	x 10^{-7}	= Joules/centimeters
Dynes	x 10^{-5}	= Joules/meters (newtons)
Dynes	x 1.020×10^{-6}	= Kilograms

A-1.
(Continued). Alphabetical Conversion Factors

Dynes	x 7.233 x 10^{-5}	= Poundals
Dynes	x 2.248 x 10^{-6}	= Pounds
E		
Ell	x 114.30	= Centimeters
Ell	x 45	= Inches
Em, pica	x 0.167	= Inch
Em, pica	x 0.4233	= Centimeter
Erg/second	x 1	= Dyne-centimeter/second
Erg/second	x 1.0 x 10^{-7}	= Watt
Erg	x 9.480 x 10^{-11}	= Btu
Erg	x 1	= Dyne-centimeter
Erg	x 7.367 x 10^{-8}	= Foot-pounds
Erg	x 1.0 x 10^{-7}	= Joules
Expansion coefficient, °F	x 1.8	= Expansion coefficient, °C
F		
Fahrenheit minus 32°F	x .5556	= Centigrade
Famm	x 5.8455	= Foot, USA
Famm	x 1.7814	= Meter
Faradays	x 26.8	= Ampere-hour
Fathom, British	x 6.08	= Feet
Fathom, British	x 1.8532	= Meter
Fathom, USA	x 6	= Feet
Fathom, USA	x 1.8288	= Meter
Fathom, USA	x 2	= Yard
Feet	x 30.48	= Centimeters
Feet	x 1.645 x 10^{-4}	= Miles (nautical)
Feet, USA	x 0.3048	= Meters
Feet, USA	x 0.3333	= Yards
Feet, USA	x 0.18939 x 10^{-3}	= Miles, USA statute
Feet, USA	x 12	= Inches
Feet, USA	x 1.2 x 10^{4}	= Mils
Feet, USA	x 0.0606	= Rod
Feet of water	x 0.0295	= Atmospheres
Feet of water	x 0.8826	= Inches of mercury
Feet of water	x 0.03048	= Kilograms/square centimeters
Feet of water	x 304.8	= Kilograms/square meter
Feet of water	x 62.43	= Pounds/square foot
Feet of water	x 0.4335	= Pounds/square inch
Feet/hour	x 0.01666	= Feet/minute
Feet/hour	x 0.2777 x 10^{-3}	= Feet/second
Feet/hour	x 0.1894 x 10^{-3}	= Miles/hour
Feet/minute	x 0.5080	= Centimeter/second
Feet/minute	x 0.01666	= Feet/second

A-1.
(Continued). Alphabetical Conversion Factors

Feet/minute	x 0.18288	= Kilometer/hour
Feet/minute	x 0.009868	= Knot
Feet/minute	x 0.3048	= Meter/minute
Feet/minute	x 0.00508	= Meter/second
Feet/minute	x 0.01136	= Mile/hour
Feet/minute	x 0.1894×10^{-3}	= Mile/minute
Feet/second	x 30.48	= Centimeters/second
Feet/second	x 1.097	= Kilometers/hour
Feet/second	x 0.5921	= Knots
Feet/second	x 18.29	= Meters/minute
Feet/second	x 0.681818	= Miles/hour
Feet/second	x 0.0113636	= Miles/minute
Feet/second	x 3600	= Feet/hour
Feet/second	x 60	= Feet/minute
Feet/second/second	x 30.48	= Centimeters/second/second
Feet/second/second	x 1.097	= Kilometers/hour/second
Feet/second/second	x 0.3048	= Meters/second/second
Feet/second/second	x 0.6818	= Miles/hour/second
Feet/100 feet	x 1	= Percent grade
Firkin	x 9	= Gallon, liquid, USA
Firkin	x 34.06798	= Liter
Foot-candle	x 10.764	= Lumen/square meter
Foot-candle	x 1	= Lumen/square foot
Foot-candle	x 10.764	= Lux
Foot-candle	x 1.076	= Milliphot
Foot-candle	x 0.001076	= Phot
Foot-candle	x distance in feet^2	= Candlepower
Foot-Lambert	x 0.3425×10^{-3}	= Candle/square centimeters
Foot-Lambert	x 0.3183	= Candle/square foot
Foot-Lambert	x 0.00221	= Candle/square inch
Foot-Lambert	x 0.001076	= Lambert
Foot-Lambert	x square foot Area	= Lumen
Foot-Lambert	x 1.076	= Millilambert
Foot-Lambert	x 0.342×10^{-3}	= Stilb
Foot-pound	x 1.2853×10^{-3}	= Btu
Foot-pound	x 1.356×10^{7}	= Ergs
Foot-pound	x 0.32389	= Gram-calorie
Foot-pound	x 5.0505×10^{-7}	= Horsepower-hours, USA
Foot-pound	x 5.12×10^{-7}	= Horsepower-hours, metric
Foot-pound	x 12	= Inch-pound
Foot-pound	x 1.35582	= Joule absolute
Foot-pound	x 1.3554	= Joule international
Foot-pound	x 3.238×10^{-4}	= Kilogram-calories
Foot-pound	x 0.1383	= Kilogram-meters
Foot-pound	x 3.766×10^{-7}	= Kilowatt-hours
Foot-pound	x 0.001356	= Kilowatt-second
Foot-pound	x 0.01338	= Liter-atmosphere

A-1.
(Continued). Alphabetical Conversion Factors

Foot-pound	x 0.3766 x 10^{-3}	= Watt-hour
Foot-pound	x 1.356	= Watt-second
Foot-pound/minute	x 0.077118	= Btu/hour
Foot-pound/minute	x 1.286 x 10^{-3}	= Btu/minute
Foot-pound/minute	x 2.259 x 10^{5}	= Erg/second
Foot-pound/minute	x 0.01666	= Foot-pound/second
Foot-pound/minute	x 3.066 x 10^{-5}	= Horsepower, metric
Foot-pound/minute	x 3.0303 x 10^{-5}	= Horsepower, USA
Foot-pound/minute	x 2.2597 x 10^{-5}	= Kilowatt
Foot-pound/minute	x 0.022597	= Watt
Foot-pound/second	x 0.0771	= Btu/minute
Foot-pound/second	x 4.6263	= Btu/hour
Foot-pound/second	x 1.843 x 10^{-3}	= Horsepower, metric
Foot-pound/second	x 1.818 x 10^{-3}	= Horsepower, USA
Foot-pound/second	x 1.356	= Joule
Foot-pound/second	x 1.3558 x 10^{-3}	= Kilowatts
Foot-pound/second	x 1.3558	= Watt
Fot	x 0.974	= Foot, USA
Fot	x 100	= Lines
Fot	x 0.2969	= Meter
Fot	x 10	= Turn
Foute	x 1	= Foot, USA
Furlong	x 6.6	= Chain, engineer
Furlong	x 10	= Chain, Gunter
Furlong	x 660	= Feet
Furlong	x 201.168	= Meters
Furlong	x 0.125	= Mile, statue, USA
Furlong	x 220	= Yards
Furlong	x 40	= Rods
Fuss	x 0.9842	= Foot, USA
Fuss	x 0.300	= Meter

G

Gallon, British, Imperial Liquid	x 0.125	= Bushel, dry, British
Gallon, British, Imperial Liquid	x 4546	= Cubic centimeter
Gallon, British, Imperial Liquid	x 0.16046	= Cubic foot
Gallon, British, Imperial Liquid	x 0.0045	= Cubic meter
Gallon, British, Imperial Liquid	x 1.032	= Gallon, dry, USA
Gallon, British, Imperial Liquid	x 1.20095	= Gallon, liquid, USA
Gallon, British, Imperial Liquid	x 4.54596	= Kilogram
Gallon, British, Imperial Liquid	x 10	= Pound, water, 62°F
Gallon, dry, USA	x 0.125	= Bushel, USA
Gallon, dry, USA	x 4404.92	= Cubic centimeter
Gallon, dry, USA	x 0.155555	= Cubic foot
Gallon, dry, USA	x 268.803	= Cubic inch
Gallon, dry, USA	x 1.16365	= Gallon, liquid, USA

A-1.
(Continued). Alphabetical Conversion Factors

Gallon, dry, USA	x 4.4049	= Liter
Gallon, dry, USA	x 0.05	= Peck
Gallon, dry, USA	x 8	= Pint
Gallon, dry, USA	x 4.6546	= Quart, liquid, USA
Gallon, liquid, USA	x 0.0238	= Barrel, oil
Gallon, liquid, USA	x 3785.434	= Cubic centimeter
Gallon, liquid, USA	x 3.785434	= Cubic decimeter
Gallon, liquid, USA	x 0.13368	= Cubic foot
Gallon, liquid, USA	x 231	= Cubic inch, water, 62°F
Gallon, liquid, USA	x 3.7854×10^{-3}	= Cubic meter
Gallon, liquid, USA	x 4.951×10^{-3}	= Cubic yard
Gallon, liquid, USA	x 0.859365	= Gallon, dry, USA
Gallon, liquid, USA	x 0.832673	= Gallon, liquid, British
Gallon, liquid, USA	x 3.7853	= Liter
Gallon, liquid, USA	x 8	= Pint, liquid, USA
Gallon, liquid, USA	x 4	= Quart, liquid, USA
Gallon, liquid, USA	x 8.3453	= Pounds, water
Gallons/hour, USA	x 0.1337	= Cubic feet/hour
Gallons/hour, USA	x 2.228×10^{-3}	= Cubic feet/minute
Gallons/hour, USA	x 0.01666	= Gallons/minute
Gallons/hour, USA	x 2.777×10^{-4}	= Gallons/second
Gallons/minute, USA	x 34.2857	= Barrels/day, oil
Gallons/minute, USA	x 1.42857	= Barrels/hour, oil
Gallons/minute, USA	x 0.023809	= Barrels/minute, oil
Gallons/minute, USA	x 192.49999	= Cubic feet/day
Gallons/minute, USA	x 8.021	= Cubic feet/hour
Gallons/minute, USA	x 0.13368	= Cubic feet/minute
Gallons/minute, USA	x 2.228×10^{-3}	= Cubic feet/second
Gallons/minute, USA	x 0.2271	= Cubic meters/hour
Gallons/minute, USA	x 1440	= Gallons/day
Gallons/minute, USA	x 60	= Gallons/hour
Gallons/minute, USA	x 0.01666	= Gallons/second
Gallons/minute, USA	x 5.35565	= Tons, long, water, 62°F/day
Gallons/minute, USA	x 5.99839	= Tons, short, water, 62°F/day
Gallons/minute, USA	x 0.06308	= Liters/second
Gallons/second, USA	x 481	= Cubic feet/hour
Gallons/second, USA	x 8.02	= Cubic feet/minute
Gallons/second, USA	x 0.1337	= Cubic feet/second
Gallons/second, USA	x 60	= Gallons/minute, USA
Gills, British	x 142.07	= Cubic centimeter
Gills, British	x 0.1183	= Liters
Gills, British	x 0.25	= Pints, liquid
Grade	x 0.0025	= Circle
Grade	x 9,000	= Degree
Grade	x 54	= Minute
Grade	x 0.01571	= Radian
Grain	x 0.01666	= Dram, apothecary

A-1.
(Continued). Alphabetical Conversion Factors

Grain	x 0.03657	= Dram, avoirdupois
Grain (troy)	x 1	= Grain (avdp)
Grain (troy)	x 0.0648	= Grams
Grain (troy)	x 2.0833×10^{-3}	= Ounces (troy)
Grain (troy)	x 2.286×10^{-3}	= Ounces (avdp)
Grains/U.S. gallon	x 17.118	= Parts/million
Grains/U.S. gallon	x 142.86	= Pounds/million gallon
Grains/Imperial gallon	x 14.286	= Parts/million
Gram	x 5	= Carat
Gram	x 3.858	= Carat, metric
Gram	x 100	= Centigram
Gram	x 0.2572	= Dram, apothecary
Gram	x 0.56438	= Dram, avoirdupois
Gram	x 980.665	= Dyne
Gram	x 15.4324	= Grain
Gram	x 9.807×10^{-5}	= Joules/centimeters
Gram	x 9.807×10^{-3}	= Joules/meter (newtons)
Gram	x 0.001	= Kilograms
Gram	x 1000	= Milligrams
Gram	x 0.03527	= Ounces, avoirdupois
Gram	x 0.03215	= Ounces, troy
Gram	x 0.07093	= Poundals
Gram	x 2.205×10^{-3}	= Pounds
Grams	x 1/MW	= Gram-mols
Grams/centimeter	x 5.6×10^{-3}	= Pounds/inch
Grams/cubic centimeter	x 62.43	= Pounds/cubic foot
Grams/cubic centimeter	x 0.03613	= Pounds/cubic foot
Grams/liter	x 58.417	= Grains/gallon, USA
Grams/liter	x 8.345	= Pounds/1,000 gallons
Grams/liter	x 0.062427	= Pounds/cubic foot
Grams/liter	x 1,000	= Parts/million
Grams/square centimeter	x 2.0481	= Pounds/square foot
Gram-calories	x 3.968×10^{-3}	= Btu
Gram-calories	x 4.1868×10^{7}	= Ergs
Gram-calories	x 3.088	= Foot-pounds
Gram-calories	x 1.55856×10^{-6}	= Horsepower/hours
Gram-calories	x 1.163×10^{-6}	= Kilowatt-hours
Gram-calories	x 1.163×10^{-3}	= Watt-hours
Gram/calories/second	x 14.286	= Btu/hour
Gram-centimeters	x 9.29658×10^{-8}	= Btu
Gram-centimeters	x 980.7	= Ergs
Gram-centimeters	x 9.807×10^{-5}	= Joules
Gram-mol	x 22.414	= Liters at 0°C and 1 atm
Gross	x 12	= Dozen
Gross, great	x 144	= Dozen
Gross, great	x 12	= Gross

A-1.
(Continued). Alphabetical Conversion Factors

H

Hand	x 10.16	= Centimeter
Hand	x 4	= Inch
Hand	x 48	= Foot
Hand	x 1,016	= Meter
Head, feet elevation, water	x 0.433	= Pounds/square inch
Hectare	x 2.471	= Acre
Hectare	x 100	= Are
Hectare	x 1.07639 x 10^5	= Square feet
Hectare	x 0.01	= Square kilometer
Hectare	x 10,000	= Square meters
Hectare	x 3.861 x 10^{-3}	= Square miles
Hectare	x 11,960	= Square yard
Hectogram	x 100	= Gram
Hectoliter	x 3.532	= Cubic feet
Hectoliter	x 0.1	= Cubic meter
Hectoliter	x 0.1308	= Cubic yard
Hectoliter	x 26.42	= Gallon, USA
Hectoliter	x 100	= Liter
Hectometer	x 328.089	= Feet
Hectometer	x 100	= Meter
Hectometer	x 0.06214	= Mile, statute, USA
Hectometer	x 109.36	= Yard
Hectowatts	x 100	= Watts
Henries	x 1,000	= Millihenries
Hogsheads, British	x 10.114	= Cubic feet
Hogsheads, USA	x 8.42184	= Cubic feet
Hogsheads, USA	x 63	= Gallons, USA
Hogsheads, USA	x 238.476	= Liter
Hogsheads, USA	x 504	= Pint
Hogsheads, USA	x 252	= Quart
Horsepower, USA	x 42.44	= Btu/minute
Horsepower, USA	x 33,000	= Foot-pounds/minute
Horsepower, USA	x 550	= Foot-pounds/second
Horsepower, USA	x 0.7457	= Kilowatts
Horsepower, USA	x 1.014	= Horsepower (metric)
Horsepower, boiler	x 33,479	= Btu/hour
Horsepower, boiler	x 34.5	= Pounds water/hour
Horsepower, boiler	x 9.803	= Kilowatts
Horsepower, electric	x 0.7072	= Btu/second
Horsepower, electric	x 746	= Joule/second
Horsepower, electric	x 0.746	= Kilowatts
Horsepower, electric	x 746	= Watts
Horsepower, (metric)	x 0.98632	= Horsepower (USA)
Horsepower, (metric)	x 542.5	= Foot-pounds/second

A-1.
(Continued). Alphabetical Conversion Factors

Horsepower, hours, USA	x 2,547	= Btu
Horsepower, hours, USA	x 2.6845×10^{13}	= Ergs
Horsepower, hours, USA	x 1.98×10^{6}	= Foot-pounds
Horsepower, hours, USA	x 641,190	= Gram-calories
Horsepower, hours, USA	x 1.01387	= Horsepower-hour, metric
Horsepower, hours, USA	x $2,376 \times 10^{4}$	= Inch-pounds
Horsepower, hours, USA	x 26.8453×10^{5}	= Joule
Horsepower, hours, USA	x 0.7457	= Kilowatt-hour
Horsepower-hours, metric	x 2509.83	= Btu
Horsepower-hours, metric	x 1.9529×10^{6}	= Foot-pounds
Horsepower-hours, metric	x 0.98632	= Horsepower-hour, USA
Horsepower-hours, metric	x 632,467	= Gram-calories
Horsepower-hours, metric	x 26.4761	= Joule
Horsepower-hours, metric	x 0.73545	= Kilowatt-hour
Hours	x 0.0417	= Day
Hours	x 60	= Minute
Hours	x 0.00137	= Month
Hours	x 0.1142×10^{-3}	= Year
Hours	x 5.952×10^{-3}	= Week
Hundredweight (long)	x 112	= Pounds
Hundredweight (long)	x 0.05	= Tons, long
Hundredweight (short)	x 1.8	= Cubic foot
Hundredweight (short)	x 45.36	= Kilograms
Hundredweight (short)	x 100	= Pounds
Hundredweight (short)	x 0.05	= Ton, short
Hundredweight (short)	x 0.04536	= Tons, metric
Hundredweight (short)	x 0.044643	= Tons, long

I

Inch	x 254×10^{6}	= Angstrom
Inch	x 2.54	= Centimeter
Inch	x 0.833×10^{-3}	= Chain, engineer
Inch	x 1.2626×10^{-3}	= Chain, Gunter
Inch	x 0.254	= Decemeter
Inch	x 0.08333	= Foot, USA
Inch	x 0.0254	= Meter
Inch	x 1.578×10^{-5}	= Mile, statute, USA
Inch	x 25.4	= Millimeters
Inch	x 1,000	= Mils
Inch	x 5.05×10^{-3}	= Rods
Inch	x 0.02778	= Yards
Inch, mercury	x 0.03342	= Atmospheres
Inch, mercury	x 1.133	= Feet of water
Inch, mercury	x 13.61	= Inch height, water
Inch, mercury	x 70.73	= Pound/square foot
Inch, mercury	x 0.49116	= Pound/square inch

A-1.
(Continued). Alphabetical Conversion Factors

Inch, mercury	x 0.03453	= Kilograms/square centimeter
Inches of mercury	x 345.3	= Kilograms/square meter
Inch-pound	x 1.07×10^{-4}	= Btu
Inch-pound	x 0.0833	= Foot-pound
Inch, water, 4°C	x 2.458×10^{-3}	= Atmospheres
Inch, water, 4°C	x 0.07355	= Inches of mercury
Inch, water, 4°C	x 2.54×10^{-3}	= Kilograms/square centimeter
Inch, water, 4°C	x 0.5781	= Ounces/square inch
Inch, water, 4°C	x 5.204	= Pounds/square foot
Inch, water, 4°C	x 0.03613	= Pounds/square inch

J

Joule	x 9.48×10^{-4}	= Btu
Joule	x 10^{7}	= Erg
Joule	x 0.7376	= Foot-pound
Joule	x 2.389×10^{-4}	= Kilogram-calorie
Joule	x 1.0197×10^{4}	= Gram-centimeter
Joule	x 2.778×10^{-4}	= Watt-hours

K

Kilogram	x 1,000	= Grams
Kilogram	x 70.93	= Poundals
Kilogram	x 2.205	= Pounds (avdp)
Kilogram	x 9.842×10^{-4}	= Tons, long
Kilogram	x 1.102×10^{-3}	= Tons, short
Kilogram	x 0.001	= Tons, metric
Kilocalories	x 3.968	= Btu
Kilocalories	x 3,088	= Foot-lbs.
Kilocalories	x 1.56×10^{-3}	= Hp.-hrs.
Kilocalories	x 1.163×10^{-3}	= Kilowatt-hrs.
Kilocalories/square meter/hr/°C	x 0.205	= Btu/square foot/hr/°F
Kilogram-meters	x 9.294×10^{-3}	= Btu
Kilograms/cubic meter	x 0.001	= Grams/cubic centimeter
Kilograms/cubic meter	x 0.06243	= Pounds/cubic foot
Kilograms/cubic meter	x 3.613×10^{-5}	= Pounds/cubic inch
Kilograms/cubic meter	x 3.405×10^{-10}	= Pounds/mil-foot
Kilograms/cubic meter	x 8.428×10^{-3}	= Ton, short/cubic yard
Kilograms/meter	x 10	= Gram/centimeter
Kilograms/meter	x 391.983	= Gram/inch
Kilograms/meter	x 0.672	= Pounds/foot
Kilograms/meter	x 0.056	= Pounds/inch
Kilograms/square centimeter	x 0.9678	= Atmospheres
Kilograms/square centimeter	x 32.81	= Feet of water
Kilograms/square centimeter	x 28.96	= Inch of mercury
Kilograms/square centimeter	x 2,048	= Pounds/square foot

A-1.
(Continued). Alphabetical Conversion Factors

Kilograms/square centimeter	x 14.223	= Pounds/square inch
Kilograms/square meter	x 9.678 x 10^{-5}	= Atmospheres
Kilograms/square meter	x 3.281 x 10^{-3}	= Feet of water
Kilograms/square meter	x 2.896 x 10^{-3}	= Inches of mercury
Kilograms/square meter	x 0.03937	= Inches of water
Kilograms/square meter	x 0.2048	= Pounds/square foot
Kilograms/square meter	x 1.422 x 10^{-3}	= Pounds/square inch
Kilometers	x 3281	= Feet
Kilometers	x 3.937 x 10^{4}	= Inches
Kilometers	x 1,000	= Meters
Kilometers	x 0.6214	= Miles (statute)
Kilometers	x 0.5295	= Miles (nautical)
Kilometers	x 1,094	= Yards
Kilometers/hour	x 27.78	= Centimeters/second
Kilometers/hour	x 54.68	= Feet/minute
Kilometers/hour	x 0.9113	= Feet/second
Kilometers/hour	x 0.5396	= Knots
Kilometers/hour	x 16.67	= Meters/minute
Kilometers/hour	x 0.6214	= Miles/hour
Kilowatts	x 56.92	= Btu/minute
Kilowatts	x 1.35972	= Horsepower, metric
Kilowatts	x 1.341	= Horsepower, USA
Kilowatts	x 1,000	= Watts
Kilowatts	x 4.426 x 10^{4}	= Foot-lbs./minute
Kilowatts	x 737.6	= Foot-lbs./second
Kilowatt-hrs.	x 2.655 x 10^{6}	= Foot-lbs.
Kilowatt-hours	x 1000	= Watt hours
Kilowatt-hours	x 3,413	= Btu
Kilowatt-hours	x 3.6 x 10^{13}	= Ergs
Kilowatt-hours	x 1.36	= Horsepower-hour, metric
Kilowatt-hours	x 1.341	= Horsepower-hour, USA
Kilowatt-hours	x 3.6 x 10^{6}	= Joules
Kilowatt-hours	x 860.5	= Kilogram-calories
Kilowatt-hours	x 3.671 x 10^{5}	= Kilogram-meters
Kilowatt-hours	x 3.53	= Pounds of water*
Kilowatt-hours	x 22.75	= Pounds of water†
Kip	x 1	= Kilopound
Kip	x 1,000	= Pound
Knott, USA	x 51.48	= Centimeter/second
Knott, USA	x 6080.2	= Feet/hour
Knott, USA	x 1.8532	= Kilometer/hour
Knott, USA	x 30.887	= Meter/minute
Knott, USA	x 1.15155	= Mile/hour (statute)
Knott, USA	x 2027	= Yards/hour

*Evaporated from and at 212°F
†Raised from 62°F to 212°F

A-1.
(Continued). Alphabetical Conversion Factors

L

League, land	x 24	= Furlong
League, land	x 4.828	= Kilometer
League, land	x 3	= Mile
League, marine	x 5.56	= Kilometer
League, marine	x 3	= Mile, nautical
League, marine	x 3.45	= Mile, statute
Light year	x 5.9×10^{12}	= Miles
Light year	x 9.46091×10^{12}	= Kilometers
Links, engineers'	x 12	= Inches
Links, surveyors'	x 7.92	= Inches
Links, surveyors'	x 0.66	= Feet
Links, surveyors'	x 0.22	= Yard
Liters	x 0.02838	= Bushels, USA, dry
Liters	x 100	= Centiliters
Liters	x 1,000	= Cubic centimeters
Liters	x 0.035316	= Cubic feet
Liters	x 61.02	= Cubic inches
Liters	x 6.291×10^{-3}	= Barrels, oil, USA
Liters	x 61.027	= Cubic inches
Liters	x 0.001	= Cubic meter
Liters	x 1.308×10^{-3}	= Cubic yard
Liters	x 0.2642	= Gallon, USA, liquid
Liters	x 0.2199	= Gallons (Imperial)
Liters	x 1.7598	= Pint, USA, dry
Liters	x 2.1134	= Pint, USA, liquid
Liters	x 2.202	= Pounds of water
Liters/minute	x 5.886×10^{-4}	= Cubic foot/second
Liters/minute	x 4.403×10^{-3}	= Gallons/second
Liters/second	x 2.1186	= Cubic feet/minute
Lumen	x 0.07958	= Candlepower
Lumen	x 1.47×10^{-3}	= Watt
Lumens/square foot	x 1	= Foot-candles
Lux	x 0.0929	= Foot-candles

M

Maas	x 1.5	= Liter
Meter	x 10^{10}	= Angstrom units
Meter	x 100	= Centimeter
Meter	x 3.2808	= Feet, USA
Meter	x 0.01	= Hectometer
Meter	x 39.37	= Inches
Meter	x 0.001	= Kilometer
Meter	x 5.396×10^{-4}	= Miles, nautical
Meter	x 6.214×10^{-4}	= Miles, statute

A-1.
(Continued). Alphabetical Conversion Factors

Meter	x 1000	= Millimeters
Meter	x 1.094	= Yards
Meter	x 1.179	= Vara
Meters/minute	x 1.667	= Centimeters/second
Meters/minute	x 3.281	= Feet/minute
Meters/minute	0.05468	= Feet/second
Meters/minute	x 0.06	= Kilometers/hour
Meters/minute	x 0.03238	= Knots
Meters/minute	x 0.03728	= Miles/hour
Meters/second	x 196.8	= Feet/minute
Meters/second	x 3.281	= Feet/second
Meters/second	x 3.6	= Kilometers/hour
Meters/second	x 0.06	= Kilometers/minute
Meters/second	x 2.237	= Miles/hour
Meters/second	x 0.03728	= Miles/minute
Microns	x 39.37×10^{-6}	= Inches
Microns	x 1×10^{-6}	= Meters
Micron	x 0.0001	= Centimeter
Micron	x 1000	= Millimicron
Mile, USA, nautical	x 6,080.2	= Feet, USA
Mile, USA, nautical	x 6,080	= Feet, British
Mile, USA, nautical	x 72,962.5	= Inches
Mile, USA, nautical	x 1.853	= Kilometer
Mile, USA, nautical	x 0.333	= League
Mile, USA, nautical	x 1,853.248	= Meter
Mile, USA, nautical	x 1.15155	= Mile, USA, statute
Mile, USA, nautical	x 2,026.73	= Yard
Miles, USA, statute	x 5,280	= Feet, USA
Miles, USA, statute	x 8	= Furlongs
Miles, USA, statute	x 63,360	= Inches
Miles, USA, statute	x 1.60935	= Kilometer
Miles, USA, statute	x 8,000	= Link
Miles, USA, statute	x 1,609.35	= Meters
Miles, USA, statute	x 0.8684	= Mile, USA, nautical
Miles, USA, statute	x 1,900.8	= Vara
Miles, USA, statute	x 1,706	= Yard
Miles/hour	x 44.7	= Centimeters/second
Miles/hour	x 88	= Feet/minute
Miles/hour	x 1.467	= Feet/second
Miles/hour	x 1.609	= Kilometers/hour
Miles/hour	x 0.02682	= Kilometers/minute
Miles/hour	x 0.8684	= Knots
Miles/hour	x 26.82	= Meters/minute
Miles/hour	x 0.4470	= Meters/second
Miles/hour	x 0.01667	= Miles/minute
Miles/minute	x 5,280	= Feet/minute
Miles/minute	x 316,800	= Feet/hour

A-1.
(Continued). Alphabetical Conversion Factors

Miles/minute	x 88	= Feet/second
Miles/minute	x 60	= Miles/hour
Miles/minute	x 1.609	= Kilometers/minute
Miles/minute	x 0.8684	= Knots/minute
Millimeter	x 0.1	= Centimeter
Millimeter	x 3.281×10^{-3}	= Feet
Millimeter	x 0.03937×10^{-2}	= Inches
Millimeter	x 10^{-6}	= Kilometers
Millimeter	x 0.001	= Meters
Millimeter	x 6.214×10^{-7}	= Miles
Millimeter	x 39.37	= Mils
Millimeter	x 1.094×10^{-3}	= Yards
Million gallons/day	x 1.54723	= Cubic feet/second
Mils	x 2.540×10^{-3}	= Centimeters
Mils	x 8.333×10^{-5}	= Feet
Mils	x 0.001	= Inches
Mils	x 2.540×10^{-8}	= Kilometers
Mils	x 2.778×10^{-5}	= Yards
N		
Nail	x 2.5	= Inch
Nepers	x 8.686	= Decibels
Newton	x 1×10^{5}	= Dynes
O		
Ounces (avdp)	x 16	= Drams
Ounces (avdp)	x 437.5	= Grains
Ounces (avdp)	x 28.349527	= Grams
Ounces (avdp)	x 0.0625	= Pounds
Ounces (avdp)	x 0.9115	= Ounces, troy
Ounces	x 2.79×10^{-5}	= Tons, long
Ounces	x 2.835×10^{-5}	= Tons, metric
Ounces, fluid	x 1.805	= Cubic inches
Ounces, fluid	x 0.02957	= Liters
Ounces, troy	x 480	= Grains
Ounces, troy	x 31.103481	= Grams
Ounces, troy	x 1.09714	= Ounces, avoirdupois
Ounces, troy	x 0.08333	= Pounds, troy
Ounces/square inch	x 0.0625	= Pounds/square inch
P		
Parsec	x 19×10^{12}	= Miles, USA, statute
Parsec	x 3.084×10^{13}	= Kilometers
Parts/million	x 0.05833	= Grains/gallon, USA

A-1.
(Continued). Alphabetical Conversion Factors

Parts/million	x 0.07016	= Grains/gallon, British
Parts/million	x 8.345	= Pounds/million gallons, USA
Peck, British	x 554.6	= Cubic inches
Peck, British	x 2	= Gallons, British
Peck, British	x 9.0919	= Liters
Peck, USA	x 0.25	= Bushels
Peck, USA	x 537.605	= Cubic inches
Peck, USA	x 8.809582	= Liters
Peck, USA	x 8	= Quarts, dry
Peck, USA	x 9.3092	= Quarts, liquid
Pennyweights, troy	x 24	= Grains
Pennyweights, troy	x 0.05	= Ounces, troy
Pennyweights, troy	x 1.55517	= Grams
Pennyweights, troy	x 4.1667×10^{-3}	= Pounds, troy
Percent	x 10^4	= Parts/million
Pfund, Germany	x 500	= Gram
Pint, USA, dry	x 0.015625	= Bushel
Pint, USA, dry	x 550.6136	= Cubic centimeter
Pint, USA, dry	x 0.01945	= Cubic feet
Pint, USA, dry	x 33.6	= Cubic inches
Pint, USA, dry	x 2	= Cup
Pint, USA, dry	x 0.125	= Gallon, USA, dry
Pint, USA, dry	x 0.145545	= Gallon, USA, liquid
Pint, USA, dry	x 0.5506	= Liter
Pint, USA, dry	x 0.0625	= Peck
Pint, USA, dry	x 0.5	= Quart, USA, dry
Pint, USA, dry	x 0.58182	= Quart, USA, liquid
Pint, USA, liquid	x 437.2	= Cubic centimeters
Pint, USA, liquid	x 0.01671	= Cubic feet
Pint, USA, liquid	x 28.875	= Cubic inch
Pint, USA, liquid	x 2	= Cup
Pint, USA, liquid	x 0.1074	= Gallon, USA, dry
Pint, USA, liquid	x 0.125	= Gallon, USA, liquid
Pint, USA, liquid	x 4	= Gill
Pint, USA, liquid	x 0.4732	= Liters
Pint, USA, liquid	x 16	= Ounces
Pint, USA, liquid	x 0.5	= Quarts, USA, liquid
Pint, USA, liquid	x 0.42968	= Quarts, USA, dry
Pint, USA, liquid	x 128	= Dram, fluid
Poise	x 100	= Centipoise
Pole	x 16.5	= Feet
Pole	x 5.0292	= Meter
Pole	x 1	= Rod
Pole	x 5.5	= Yard
Ponce	x 2.71	= Centimeter
Pood	x 1,000	= Cubic inch
Pood	x 40	= Funt

A-1.
(Continued). Alphabetical Conversion Factors

Pood	x 4.32	= Gallon, USA
Pood	x 16.3805	= Kilogram
Poundals	x 13,826	= Dynes
Poundals	x 14.098	= Grams
Poundals	x 1.383×10^{-3}	= Joules/centimeter
Poundals	x 0.1383	= Joules/meter
Poundals	x 0.0141	= Kilograms
Poundals	x 0.1383	= Newton
Poundals	x 0.03108	= Pound-force
Pound-mol	x 359.05	= Cubic feet at 32°F and 1 atm
Pounds	x 1/Mol. Wt.	= Pound-mols
Pounds	x 2267.9616	= Carats
Pounds (avdp)	x 256	= Drams
Pounds (avdp)	x 7,000	= Grains
Pounds (avdp)	x 453.5924	= Grams (metric)
Pounds	x 0.04448	= Joules/centimeters
Pounds (avdp)	x 0.4536	= Kilograms
Pounds (avdp)	x 16	= Ounces
Pounds (avdp)	x 14.5833	= Ounces, troy
Pounds	x 32.174	= Poundals
Pounds (avdp)	x 1.21528	= Pounds, troy
Pounds	x 4.464×10^{-4}	= Tons, long
Pounds	x 4.536×10^{-4}	= Tons, metric
Pounds	x 5×10^{-4}	= Tons, short
Pounds, troy	x 5,760	= Grains
Pounds, troy	x 373.24177	= Grams
Pounds, troy	x 13.1657	= Ounces, avoirdupois
Pounds, troy	x 12	= Ounces, troy
Pounds, troy	x 240	= Pennyweights, troy
Pounds, troy	x 0.822857	= Pounds, avoirdupois
Pounds, troy	x 3.6735×10^{-4}	= Tons, long
Pounds, troy	x 3.7324×10^{-4}	= Tons, metric
Pounds, troy	x 4.1143×10^{-4}	= Tons, short
Pounds of water	x 0.01602	= Cubic feet
Pounds of water	x 27.68	= Cubic inches
Pounds of water	x 0.1198	= Gallons
Pounds of water/minute	x 2.67×10^{-4}	= Cubic feet/second
Pounds/cubic foot	x 0.01602	= Grams/cubic centimeters
Pounds/cubic foot	x 16.02	= Kilograms/cubic meter
Pounds/cubic foot	x 5.787×10^{-4}	= Pounds/cubic inch
Pounds/cubic foot	x 27	= Pounds/cubic yard
Pounds/cubic inch	x 27.68	= Grams/cubic centimeter
Pounds/cubic inch	x 2.768×10^{4}	= Kilograms/cubic meter
Pounds/cubic inch	x 1,728	= Pounds/cubic foot
Pounds/cubic inch	x 46,656	= Pounds/cubic yard
Pounds/hour	x 10.714×10^{-3}	= Tons/day, long
Pounds/hour	x 12×10^{-3}	= Tons/day, short

A-1.
(Continued). Alphabetical Conversion Factors

Pounds/hour	x 10.886×10^{-3}	= Tons/day, metric
Pounds/hour	x 0.45359	= Kilograms/hour
Pounds/square foot	x 4.725×10^{-4}	= Atmospheres
Pounds/square foot	x 0.01602	= Feet of water
Pounds/square foot	x 0.01414	= Inches of mercury
Pounds/square foot	x 4.8824	= Kilograms/square meter
Pounds/square foot	x 0.1111	= Ounce/square inch
Pounds/square foot	x 0.107638	= Pound/square centimeter
Pounds/square foot	x 6.944×10^{-3}	= Pound/square inch
Pounds/square foot	x 10.76387	= Pound/square meter
Pounds/square inch	x 0.068046	= Atmospheres
Pounds/square inch	x 2.307	= Feet of water
Pounds/square inch	x 27.7	= Inch of water
Pounds/square inch	x 2.036	= Inch of mercury
Pounds/square inch	x 0.0703	= Kilogram/square centimeter
Pounds/square inch	x 703.1	= Kilogram/square meter
Pounds/square inch	x 51.714	= Millimeters of mercury
Pounds/square inch	x 2,304	= Ounce/square foot
Pounds/square inch	x 144	= Pound/square foot

Q

Quadrant	x 0.25	= Circumference
Quadrant	x 90	= Degrees
Quadrant	x 5,400	= Minutes
Quadrant	x 1.571	= Radians
Quarts, USA, dry	x 0.03125	= Bushel
Quarts, USA, dry	x 1,101.2	= Cubic centimeter
Quarts, USA, dry	x 0.03889	= Cubic foot
Quarts, USA, dry	x 67.20	= Cubic inches
Quarts, USA, dry	x 1.1012	= Liter
Quarts, USA, dry	x 1.16365	= Quart, USA, liquid
Quarts, USA, liquid	x 946.331	= Cubic centimeter
Quarts, USA, liquid	x 0.03342	= Cubic foot
Quarts, USA, liquid	x 57.75	= Cubic inches
Quarts, USA, liquid	x 9.464×10^{-4}	= Cubic meters
Quarts, USA, liquid	x 1.238×10^{-3}	= Cubic yard
Quarts, USA, liquid	x 4	= Cup
Quarts, USA, liquid	x 256	= Dram fluid
Quarts, USA, liquid	x 0.25	= Gallons
Quarts, USA, liquid	x 0.946331	= Liter
Quarts, USA, liquid	x 5.9523×10^{-3}	= Oil, barrel
Quarts, USA, liquid	x 32	= Ounces
Quarts, USA, liquid	x 2	= Pint
Quarts, USA, liquid	x 0.8594	= Quart, USA, dry

A-1.
(Continued). Alphabetical Conversion Factors

R

Radians	x 57.3	= Degrees
Radians	x 3,438	= Minutes
Radians	x 0.6366	= Quadrants
Radians	x 2.063×10^5	= Seconds
Revolutions/minute	x 6.0	= Degrees/second
Revolutions/minute	x 0.1047	= Radians/second
Revolutions/minute	x 0.01667	= Revolutions/second
Rod	x 0.165	= Chain, engineer
Rod	x 0.25	= Chain, Gunters
Rod	x 16.5	= Foot
Rod	x 0.025	= Furlong
Rod	x 198	= Inch
Rod	x 25	= Link
Rod	x 5.029	= Meter
Rod	5.5	= Yard

S

Seconds, angle	x 2.778×10^{-4}	= Degrees
Seconds, angle	x 16.67×10^{-3}	= Minutes
Seconds, time	x 2.777×10^{-4}	= Hour
Seconds, time	x 0.166	= Minutes
Slugs	x 14.59	= Kilograms
Slugs	x 32.17	= Pounds
Slugs/cubic foot	x 0.5154	= Gm/cubic centimeter
Snow, cubic foot	x 7.2	= Pounds, 32°F
Snow, inch deep	x 0.1	= Inch, water
Square centimeter	x 1.076×10^{-3}	= Square foot
Square centimeter	x 0.155	= Square inch
Square centimeter	x 0.0001	= Square meter
Square centimeter	x 3.861×10^{-11}	= Square miles
Square centimeter	x 100	= Square millimeters
Square centimeter	x 1.196×10^{-4}	= Square yards
Square feet, USA	x 2.296×10^{-5}	= Acre
Square feet, USA	x 9.29×10^{-4}	= Are
Square feet, USA	x 929.034	= Square centimeters
Square feet, USA	x 144	= Square inches
Square feet, USA	x 0.0929	= Square meter
Square feet, USA	x 3.587×10^{-8}	= Square miles
Square feet, USA	x 9.29×10^{4}	= Square millimeters
Square feet, USA	x 0.1111	= Square yards
Square inches	x 6.452	= Square centimeters
Square inches	x 6.944×10^{-3}	= Square feet

A-1.
(Continued). Alphabetical Conversion Factors

Square inches	x 645.2	= Square millimeters
Square inches	x 7.716×10^{-4}	= Square yard
Square kilometer	x 247.1	= Acre
Square kilometer	x 100	= Hectare
Square kilometer	x 10.76×10^{6}	= Square feet
Square kilometer	x 1.55×10^{9}	= Square inches
Square kilometer	x 10^{6}	= Square meters
Square kilometer	x 0.3861	= Square mile, USA
Square kilometer	x 1.196×10^{6}	= Square yards
Square meters	x 2.471×10^{-4}	= Acre
Square meters	x 0.01	= Are
Square meters	x 0.0001	= Hectare
Square meters	x 10,000	= Square centimeters
Square meters	x 10.764	= Square feet
Square meters	x 1,550	= Square inches
Square meters	x 3.861×10^{-7}	= Square miles
Square meters	x 1.196	= Square yards
Square miles	x 640	= Acre
Square miles	x 259	= Hectare
Square miles	x 27.88×10^{6}	= Square feet
Square miles	x 2.59	= Square kilometers
Square miles	x 2.59×10^{6}	= Square meters
Square miles	x 3.098×10^{6}	= Square yards
Square millimeters	x 0.01	= Square centimeters
Square millimeters	x 1.076×10^{-5}	= Square feet
Square millimeters	x 1.55×10^{-3}	= Square inches
Square rods	x 0.00625	= Acre
Square rods	x 272.25	= Square feet
Square rods	x 25.293	= Square meter
Square rods	x 30.25	= Square yard
Square vara	x 7.716	= Square feet
Square yard	x 2.066×10^{-4}	= Acres
Square yard	x 8361	= Square centimeter
Square yard	x 9	= Square feet
Square yard	x 1,296	= Square inches
Square yard	x 0.8361	= Square meters
Square yard	x 3.228×10^{-7}	= Square miles
Square yard	x 8.361×10^{5}	= Square millimeters
Square yard	x 0.03306	= Square rods
Stone	x 14	= Pound
Stone	x 6.35	= Kilogram

T

Tablespoon	x 0.0625	= Cup
Tablespoon	x 3	= Teaspoon

A-1.
(Continued). Alphabetical Conversion Factors

Teaspoon	x 0.0208	= Cup
Teaspoon	x 0.333	= Tablespoon
Temperature, °C + 17.78	x 1.8	= °F
Temperature, °F − 32	x 0.5556	= °C
Ton, long	x 1,016	= Kilogram
Ton, long	x 2,240	= Pounds
Ton, long	x 1.016	= Metric tons
Ton, long	x 1.12	= Short tons
Ton, metric	x 7.454	= Barrel, oil, 36 API
Ton, metric	x 1,000	= Kilograms
Ton, metric	x 2,205	= Pounds
Ton, metric	x 0.9842	= Ton, long
Ton, metric	x 1.1023	= Ton, short
Ton, shipping, USA	x 40	= Cubic feet
Ton, shipping, USA	x 2.8317	= Cubic meter
Ton, shipping, USA	x 1.050	= Ton, shipping, British
Ton, short	x 40	= Cubic feet
Ton, short	x 268.8	= Gallons, USA, liquid
Ton, short	x 4	= Hogshead
Ton, short	x 907.18486	= Kilograms
Ton, short	x 1,000	= Liter
Ton, short	x 32,000	= Ounces
Ton, short	x 2,000	= Pounds
Ton, short	x 0.89286	= Tons, long
Ton, short	x 0.907	= Tons, metric
Tons, short/square foot	x 9,765	= Kilograms/square meter
Tons, short/square foot	x 2,000	= Pounds/square inch
Tons, short/day	x 83.333	= Pounds/hour
Tons, short/day	x 0.16643	= Gallons/minute
Tons, short/day	x 0.9072	= Tons, metric/day
Tons, short/day	x 0.8929	= Tons, long/day
Tons, short/day	x 37.8	= Kilograms/hour
Tons, metric/day	x 91.859	= Pounds/hour
Tons, metric/day	x 41.667	= Kilograms/hour
Tons, metric/day	x 0.9843	= Tons, long/day
Tons, metric/day	x 1.1023	= Tons, short/day
Tons, long/day	x 1.12	= Tons, short/day
Tons, long/day	x 1.016	= Tons, metric/day
Tons, long/day	x 93.333	= Pounds/hour
Tons, long/day	x 42.335	= Kilograms/hour

V

Vara	x 2.7777	= Feet
Vara	x 33.3333	= Inch
Vara	x 0.9259	= Yard

A-1.
(Concluded). Alphabetical Conversion Factors

Volt/inch	x 0.3937	= Volt/centimeter
W		
Water, 62°F, Gallon	x 8.3311	= Pound
Water height in feet	x 0.4335	= Pound/square inch
Water height in feet	x 0.03048	= Kilograms/square centimeters
Water height in inches	x 0.03613	= Pound/square inch
Water height in inches	x 0.00254	= Kilograms/square centimeter
Water height in meters	x 1.42067	= Pound/square inch
Water height in meters	x 0.100	= Kilograms/square centimeters
Watts	x 3.4128	= Btu/hour
Watts	x 0.05688	= Btu/minute
Watts	x 107	= Ergs/second
Watts	x 44.27	= Foot-pounds/minute
Watts	x 0.7378	= Foot-pounds/second
Watts	x 1.341×10^{-3}	= Horsepower, USA
Watts	x 1.36×10^{-3}	= Horsepower, metric
Watts	x 0.001	= Kilowatt
Watts	x 1	= Joules/second
Watts (abs.)	x 0.056884	= Btu (mean)/minute
Watt-hours	x 3.4128	= Btu
Watt-hours	x 3.60×10^{10}	= Ergs
Watt-hours	x 2,656	= Foot-pounds
Watt-hours	x 858.85	= Gram-calories
Watt-hours	x 1.341×10^{-3}	= Horsepower-hours, USA
Watt-hours	x 1.3596×10^{-3}	= Horsepower-hours, metric
Watt-hours	x 0.8605	= Kilogram-calories
Watt-hours	x 367.2	= Kilogram-meters
Watt-hours	x 0.001	= Kilowatt-hours
Y		
Yard, USA	x 91.4402	= Centimeter
Yard, USA	x 3	= Feet
Yard, USA	x 36	= Inch
Yard, USA	x 9.144×10^{-4}	= Kilometer
Yard, USA	x 0.9144	= Meter
Yard, USA	x 4.934×10^{-4}	= Mile, nautical, USA
Yard, USA	x 5.682×10^{-4}	= Mile, statute, USA
Yard, USA	x 914.402	= Millimeters
Yard, USA	x 0.1818	= Rod
Year	x 8,765	= Hours
Year	x 525,948	= Minutes

A-2.
Physical Property Conversion Factors

Acceleration of gravity = 32.172 ft./sec./sec.
= 980.6 cm./sec./sec.

Electrical conductance;
1 mho = 1 ohm^{-1}
= 10^{-6} megamho
= 10^{6} micromho

Heat Value of Fuel

Lower heating value
= Higher heating value − 10.3 ($9H_2 + H_2O$), Btu/lb.
where: H_2 = weight % hydrogen in fuel
H_2O = weight % water vapor in fuel
GPM = (pounds/hour)/(500 × Sp.Gr.)

$$\text{Velocity, feet/sec.} = \frac{0.321\ (\text{GPM})}{(\text{Flow Area, sq.in.})}$$

Head, feet = 2.31 (Pressure or head, psi)/Sp.Gr.

$$\text{Brake horsepower, BHP} = \frac{(\text{GPM})\ (\text{Sp.Gr.})\ (\text{Head, feet})}{3960\ (\text{Efficiency, fraction})}$$

Weight/Volume (avoirdupois unless otherwise stated)

Density of sea water = 1.025 grams/cc.

1 gram-molecular volume of a gas at 760 mm Hg and 0° C. = 22.4 liters

1 U. S. gallon = (8.34 × Sp.Gr. of fluid), pounds

Weight of one cu.ft. liquid = (62.32 pounds × Sp.Gr. of fluid), pounds/cu.ft.

1 pound avoirdupois = 1.2153 pound apothecaries'

1 grain avoirdupois = 1 grain troy = 1 grain apothecaries' weight

Air Analysis*

	By Weight %	By Volume %
Nitrogen	75.47	78.2
Oxygen	23.19	21.0

* Neglects trace gases such as argon, xenon, helium, krypton and assumes dry basis.

Gas Constants, (R), Universal

R = 0.0821 (atm) (liter)/(g-mol) (°K)
= 1.987 (g-cal.)/(g-mol) (°K)
= 1.987 Btu/(lb.-mol) (°R)
= 1.987 (Chu)/(lb.-mol) (°K)
= 8.314 joules/(g-mol) (°K)
= 1,546 (ft.) (lb.force)/(lb.-mol) (°R)
= 10.73 (lb.-force/sq. in abs.) (cu.ft.)/(lb.-mol) (°R)
= 18,510 (lb.-force/sq.in.) (cu.in.)/(lb.-mol) (°R)
= 0.7302 (Atm) (cu.ft.)/(lb.-mol) (°R)

R_1 = R/mol.wt. gas
where: R_1 = individual gas constant

Avogadro Constant, N_a = 6.02252 × 10^{23} molecules/mol

Density, Vapor or Gases (Ideal), ρ

$$\rho = \left(\frac{\text{mol. wt., vapor}}{359}\right)\left(\frac{14.7 + p}{14.7}\right)\left(\frac{460 + 32}{460 + {}^\circ F}\right), \text{ lbs./cu.ft.}$$

where: p = gage pressure at actual condition, psig
°F = fahrenheit temperature at actual condition

$$\rho = \frac{144\ P}{R_1\ T}, \text{ pounds/cu.ft.}$$

where: P = absolute pressure, pounds/sq. in. abs.
T = absolute temperature, °Rankine, °R
o = standard conditions (0°C & 760 mm Hg)

$V = V_o\ (P_o/P'')(T/T_o)$

$P''V$ = 1543 nT; P'' = PSF abs.; V = cu. ft.

T = °R; n = Lb. moles

$$\text{cu.ft.} = \frac{\text{lb}}{\text{MW}}\ (359)\left(\frac{273 + {}^\circ C}{273}\right)\left(\frac{14.7}{p + 14.7}\right) \text{ at p, °C}$$

Specific Volume, Gas or Vapor

$\overline{V} = 1/\rho$, cu.ft./pound

Velocity of sound in dry air @ 0° C. and 1 atm. = 1,089 ft./sec.

Density of dry air @ 0° C. and 1 atm.
= 0.001293 gm/cu.cm.
= 0.0808 lb./cu.ft.

Viscosity (Dynamic)

1 Poise = 1 gram/cm.-sec. = 1 dyne-sec./sq:cm.
= 0.1 kg/meter-sec.

1 Poise × 100 = Centipoise (μ)

Poise × 2.09 × 10^{-3} = slugs/ft.-sec.
= pounds (force)-sec./sq.ft.

Poise × 0.10 = pascal-sec.

(Continued on next page)

A-2.
(Continued). Physical Property Conversion Factors

Poise $\times$ 0.0672 = pounds (mass)/(ft.-sec.) = poundal-sec./sq.ft.
Poise $\times$ 0.10 = Newton-sec./sq. meter
Centipoise $\times$ 0.01 = gm./cm.-sec.
Centpoise $\times$ 6.72 $\times 10^{-4}$ = pound/ft.-sec .
Centipoise $\times$ 2.4 = pound/ft.-hr.
Millipoise $\times$ 1000 = poise
Micropoise $\times$ 1,000,000 = poise
Slugs/ft.-sec. $\times$ 47,900 = centipoise
1 centistoke = 1.076 $\times 10^{-5}$ ft.^{2}/sec.
1 centipoise (cp) = 0.01 gm./cm. sec.
Slugs/ft.-sec. $\times$ 32.2 = pounds (mass)/ft.-sec.
Pounds/ft.-sec. $\times$ 3600 = lb./ft.-hr.
Pounds (mass)/ft.-sec. $\times$ 1487 = centipoise
Pounds (mass)/ft.-sec. $\times$ 0.0311 = slugs/ft.-sec. = pounds (force)-sec./sq.ft.
Viscosity of air @ 68° F. = 180.8 $\times 10^{-6}$ poise
Viscosity of water @ 66° F = 0.010087 poise

Viscosity (Kinematic)

Kinematic viscosity, centistokes $\times$ 1.076 $\times 10^{-5}$ = ft.^{2}/sec.

$$\text{Kinematic viscosity, centistokes } (\nu) = \frac{\text{Dynamic viscosity, centipoise}}{\text{Fluid density, gm./cu.cm.}}$$

$$= \frac{\text{Centipoise}}{\text{Sp.Gr. of liquid relative to water at 39.2° F. (4° C.)}}$$

Centistokes $\times$ 0.01 = stokes, sq.cm./sec.
Centistokes $\times$ 1.076 $\times 10^{-5}$ = sq.ft./sec.
Centistokes $\times$ 0.01 = Stokes, sq.cm./sec.

Thermal Conductivity (through a homogeneous material)

$$\frac{\text{Btu (ft.)}}{\text{(sq.ft.) (°F.) (hr.)}} \times 4.134 \times 10^{-3} = \frac{\text{(g.-cal.) (cm.)}}{\text{(sq.cm.) (°C.) (sec.)}}$$

$$\times 1.200 \times 10 = \frac{\text{(Btu) (in.)}}{\text{(sq.ft.) (°F.) (hr.)}}$$

$$\times 3.518 \times 10^{-3} = \frac{\text{(kilowatt hrs.) (in.)}}{\text{(sq.ft.) (°F.) (hr.)}}$$

$$\frac{\text{(g.-cal.) (cm.)}}{\text{(sq.cm.) (°C.) (hr.)}} \times 8.063 \times 10^{-1} = \frac{\text{Btu (in.)}}{\text{(sq.ft.) (°F.) (hr.)}}$$

$$\times 6.719 \times 10^{-2} = \frac{\text{Btu (ft.)}}{\text{(sq.ft.) (°F.) (hr.)}}$$

$$\frac{\text{(g.-cal.) (cm.)}}{\text{(sq.cm.) (°C.) (sec.)}} \times 2.903 \times 10^{3} = \frac{\text{Btu (in.)}}{\text{(sq.ft.) (°F.) (hr.)}}$$

$$\times 8.063 \times 10^{-1} = \frac{\text{Btu (in.)}}{\text{(sq.ft.) (°F.) (sec.)}}$$

$$\times 8.506 \times 10^{2} = \frac{\text{(joules) (in.)}}{\text{(sq.ft.) (°F.) (sec.)}}$$

Specific Gravity (Liquid)

$$s = \frac{\rho \text{ of liquid @ 60° F.*}}{\rho \text{ of water @ 60° F.*}}$$

* or at other specified temperature

Oil

$$\text{s at 60° F./60° F.} = \frac{141.5}{131.5 + \text{degrees API}}$$

Liquids Lighter Than Water

$$\text{s @ 60° F./60° F.} = \frac{140}{130 + \text{degrees Baume'}}$$

Liquids Heavier Than Water

$$\text{s @ 60° F./60° F.} = \frac{145}{145 - \text{degrees Baume'}}$$

Specific Gravity (Gases)

$$S_g = \frac{\text{R of air}}{\text{R of gas}} = \frac{53.3}{\text{R of gas}}, \text{ where R = gas constant}$$

$$S_g = \frac{\text{mol. wt. (gas)}}{\text{mol. wt. (air)}} = \frac{\text{mol. wt. (gas)}}{29}$$

Density, Liquid ρ

Density liquid, ρ = (62.3 lb./cu. ft. water) (Sp. Gr. liquid), pounds /cu. ft.

Metric

1 gram = 10 decigrams
= 100 centigrams
= 1,000 milligrams
= 1,000,000 microgram
= 0.001 kilogram
= 10^{-6} megagram
1 liter = 10 deciliters = 1.0567 liquid quarts
10 liters = 1 dekaliter = 2.6417 liquid gallons
10 dekaliters = 1 hectoliter = 2.8375 U. S. bushels

(Continued on next page)

A-2.
(Concluded). Physical Property Conversion Factors

1 meter = 10 decimeters = 39.37 inches
= 100 centimeters
= 1,000 millimeters
= 1,000,000 miscrons = 1,000,000 micrometers
= 1/1,000 kilometer
= 10^{10} Angstrom units

10 millimeters = 1 centimeter = 0.3937 inches
10 centimeters = 1 decimeter = 3.937 inches
25.4 millimeters = 1 inch

Specific Heat

$$\frac{\text{(gram-cal.)}}{\text{(gram)(°C.)}} \times 1.8 = \frac{\text{Btu}}{\text{(pound)(°C.)}}$$

$$\times 1.0 = \frac{\text{Btu}}{\text{(pound)(°F.)}}$$

$$\times 4.186 = \frac{\text{joules}}{\text{(gram)(°C.)}}$$

$$\times 1055 = \frac{\text{joules}}{\text{(pound)(°F.)}}$$

$$\times 1.163 \times 10^{-3} = \frac{\text{kilowatt-hours}}{\text{(kilogram)(°C.)}}$$

$$\times 2.930 \times 10^{-4} = \frac{\text{kilowatt-hours}}{\text{(pound)(°F.)}}$$

Specific heat of water at 1 atm. = 0.238 cal./gm-°C.
Btu/lb. – ° F. × 0.2390 = Btu/lb. – ° R

Heat Transfer Coefficient

PCU/(hr.)(sq. ft.)(°C.) × 1.0
= Btu/(hr.)(sq.ft.)(°F)

Kg-cal./(hr.)(sq. m.)(°C.) × 0.2048
= Btu/(hr.)(sq.ft.)(°F)

G-cal./(sec.)(sq. cm.)(°C.) × 7,380
= Btu/(hr.)(sq.ft.)(°F)

Watts/(sq. in.)(°F.) × 490 = Btu/(hr.)(sq. ft.)(°F.)

Energy Units

Pound-Centigrade-Unit(PCU) × 1.8 = Btu
× 0.45359 = calorie
× 1400.4 = ft.-lb.
× 0.0005276 = kilowatt-hr.
× 1899.36 = joules

Calories × 3.9683 = Btu
× 3091.36 = ft.-lb.
× 0.001559 = horsepower-hr.
× 0.001163 = kilowatt-hr.
× 4187.37 = joules

Pressure

1 mm Hg = 1,333 dynes/sq. cm.
750 mm Hg = 10 dynes/sq. cm. = 1 megabar @ °C. and g = 980.6

A-3.
Synchronous Speeds

$$\text{Synchronous Speed} = \frac{\text{Frequency x 120}}{\text{No. of Poles}}$$

	FREQUENCY				FREQUENCY	
Poles	60 cycle	50 cycle	25 cycle	Poles	60 cycle	50 cycle
2	3600	3000	1500	42	171.4	142.9
4	1800	1500	750	44	163.6	136.4
6	1200	1000	500	46	156.5	130.4
8	900	750	375	48	150	125
10	720	600	300	50	144	120
12	600	500	250	52	138.5	115.4
14	514.3	428.6	214.3	54	133.3	111.1
16	450	375	187.5	56	128.6	107.1
18	400	333.3	166.7	58	124.1	103.5
20	360	300	150	60	120	100
22	327.2	272.7	136.4	62	116.1	96.8
24	300	250	125	64	112.5	93.7
26	276.9	230.8	115.4	66	109.1	90.9
28	257.1	214.3	107.1	68	105.9	88.2
30	240	200	100	70	102.9	85.7
32	225	187.5	93.7	72	100	83.3
34	211.8	176.5	88.2	74	97.3	81.1
36	200	166.7	83.3	76	94.7	78.9
38	189.5	157.9	78.9	78	92.3	76.9
40	180	150	75	80	90	75

Courtesy Ingersoll-Rand Co.

A-4.
Conversion Factors

Units of Length	Multiply units in left column by proper factor below							
	in.	ft.	yd.	mile	mm.	cm.	m.	km.
1 inch	1	0.0833	0.0278	—	25.40	2.540	0.0254	—
1 foot	12	1	0.3333	—	304.8	30.48	0.3048	—
1 yard	36	3	1	—	914.4	91.44	0.9144	—
1 mile	—	5280	1760	1	—	—	1609.3	1.609
1 millimeter	0.0394	0.0033	—	—	1	0.100	0.001	—
1 centimeter	0.3937	0.0328	0.0109	—	10	1	0.01	—
1 meter	39.37	3.281	1.094	—	1000	100	1	0.001
1 kilometer	—	3281	1094	0.6214	—	—	1000	1

(1 micron = 0.001 millimeter)

Courtesy Ingersoll-Rand Co.

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A-4.
(Continued). Conversion Factors

Units of Weight	Multiply units in left column by proper factor below						
	grain	oz.	lb.	ton	gram	kg.	metric ton
1 grain	1	—	—	—	0.0648	—	—
1 ounce	437.5	1	0.0625	—	28.35	0.0283	—
1 pound	7000	16	1	0.0005	453.6	0.4536	—
1 ton	—	32,000	2000	1	—	907.2	0.9072
1 gram	15.43	0.0353	—	—	1	0.001	—
1 kilogram	—	35.27	2.205	—	1000	1	0.001
1 metric ton	—	35,274	2205	1.1023	—	1000	1

Units of Density	Multiply units in left column by proper factor below				
	lb/cu. in.	lb/cu. ft.	lb/gal.	g/cu. cm.	g/liter
1 pound/cu. in.	1	1728	231.0	27.68	27,680
1 pound/cu. ft.	—	1	0.1337	0.0160	16.019
1 pound/gal.	0.00433	7.481	1	0.1198	119.83
1 gram/cu. cm.	0.0361	62.43	8.345	1	1000.0
1 gram/liter	—	0.0624	0.00835	0.001	1

Units of Area	Multiply units in left column by proper factor below						
	sq. in.	sq. ft.	acre	sq. mile	sq. cm.	sq. m.	hectare
1 sq. inch	1	0.0069	—	—	6.452	—	—
1 sq. foot	144	1	—	—	929.0	0.0929	—
1 acre	—	43,560	1	0.0016	—	4047	0.4047
1 sq. mile	—	—	640	1	—	—	259.0
1 sq. centimeter	0.1550	—	—	—	1	0.0001	—
1 sq. meter	1550	10.76	—	—	10,000	1	—
1 hectare	—	—	2.471	—	—	10,000	1

Units of Volume	Multiply units in left column by proper factor below							
	cu. in.	cu. ft.	cu. yd.	cu. cm.	cu. meter	liter	U.S. gal.	Imp. gal.
1 cu. inch	1	—	—	16.387	—	0.0164	—	—
1 cu. foot	1728	1	0.0370	28,317	0.0283	28.32	7.481	6.229
1 cu. yard	46,656	27	1	—	0.7646	764.5	202.0	168.2
1 cu. centimeter	0.0610	—	—	1	—	0.0010	—	—
1 cu. meter	61,023	35.31	1.308	1,000,000	1	999.97	264.2	220.0
1 liter	61.025	0.0353	—	1000.028	0.0010	1	0.2642	0.2200
1 U.S. gallon	231	0.1337	—	3785.4	—	3.785	1	0.8327
1 Imperial gallon	277.4	0.1605	—	4546.1	—	4.546	1.201	1

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A-4.
(Concluded). Conversion Factors

Units of Pressure	Multiply units in left column by proper factor below						
	lb/sq. in.	lb/sq. ft.	int. atm.	kg/cm^2	mm Hg at 32°F	in. Hg at 32°F	ft. water at 39.2°F
1 pound/sq. in.	1	144	—	0.0703	51.713	2.0359	2.307
1 pound/sq. ft.	0.00694	1	—	—	0.3591	0.01414	0.01602
1 intern. atmosphere	14.696	2116.2	1	1.0333	760	29.921	33.90
1 kilogram/sq. cm.	14.223	2048.1	0.9678	1	735.56	28.958	32.81
1 millimeter-mercury— 1 torr (torricelli)—	0.0193	2.785	—	—	1	0.0394	0.0446
1 inch mercury	0.4912	70.73	0.0334	0.0345	25.400	1	1.133
1 foot water	0.4335	62.42	—	0.0305	22.418	0.8826	1

Units of Energy	Multiply units in left column by proper factor below					
	ft.-lb.	Btu	g. cal.	Joule	kw-hr.	hp-hr.
1 foot-pound	1	0.001285	0.3240	1.3556	—	—
1 Btu	778.2	1	252.16	1054.9	—	—
1 gram calorie	3.0860	0.003966	1	4.1833	—	—
1 int. Joule	0.7377	0.000948	0.2390	1	—	—
1 int. kilowatt-hour	2,655,656	3412.8	860,563	—	1	1.3412
1 horsepower-hour	1,980,000	2544.5	641,617	—	0.7456	1

Units of Specific Energy	Multiply units in left column by proper factor below				
	absolute Joule/g	int. Joule/g	cal/g	int. cal/g	Btu/lb.
1 absolute Joule/gram	1	0.99984	0.23901	0.23885	0.42993
1 int. Joule/gram	1.000165	1	0.23904	0.23892	0.43000
1 calorie/gram	4.1840	4.1833	1	0.99935	1.7988
1 int. calorie/gram	4.1867	4.1860	1.00065	1	1.8000
1 Btu/lb.	2.3260	2.3256	0.55592	0.55556	1

Units of Power (rates of energy use)	Multiply units in left column by proper factor below								
	hp	watt	kw	Btu/min.	Btu/hr.	ft-lb/sec.	ft-lb/min.	g. cal/sec.	metric hp
1 horsepower	1	745.7	0.7475	42.41	2544.5	550	33.000	178.2	1.014
1 watt	—	1	0.001	0.0569	3.413	0.7376	44.25	0.2390	0.00136
1 kilowatt	1.3410	1000	1	56.88	3412.8	737.6	44,254	239.0	1.360
1 Btu per minute	—	—	—	1	60	12.97	778.2	4.203	0.0239
1 metric hp	0.9863	735.5	0.7355	41.83	2509.6	542.5	32.550	175.7	1

Units of Refrigeration	Multiply units in left column by factor below					
	Btu(IT)/min.	Btu(IT)/hr.	kg cal/hr.	ton (U.S.) comm	ton (BRIT.) comm	frigorie/hr.
1 ton (U.S.) comm	200	12,000	3025.9	1	0.8965	3025.9
1 ton (Brit) comm	223.08	13,385	3375.2	1.1154	1	3375.2
1 frigorie/hr.	0.06609	3.9657	1	0.0003305	0.0002963	1

Note:—Btu is International Steam Table Btu(IT). 1 frigorie = 1 kg cal (**Not** IT).

A-5.
Temperature Conversion

NOTE: The center column of numbers in boldface refers to the temperature in degrees, either Centigrade or Fahrenheit, which it is desired to convert into the other scale. If converting from Fahrenheit to Centigrade degrees, the equivalent temperature will be found in the left column, while if converting from degrees Centigrode to degrees Fahrenhiet, the answer will be found in the column on the right.

Centigrade		Fahrenheit
−273.17	**−459.7**	
−268	**−450**	
−262	**−440**	
−257	**−430**	
−251	**−420**	
−246	**−410**	
−240	**−400**	
−234	**−390**	
−229	**−380**	
−223	**−370**	
−218	**−360**	
−212	**−350**	
−207	**−340**	
−201	**−330**	
−196	**−320**	
−190	**−310**	
−184	**−300**	
−179	**−290**	
−173	**−280**	
−169	**−273**	−459.4
−168	**−270**	−454
−162	**−260**	−436
−157	**−250**	−418
−151	**−240**	−400
−146	**−230**	−382
−140	**−220**	−364
−134	**−210**	−345
−129	**−200**	−328
−123	**−190**	−310
−118	**−180**	−292
−112	**−170**	−274
−107	**−160**	−256
−101	**−150**	−238
−96	**−140**	−220
−90	**−130**	−202
−84	**−120**	−184
−79	**−110**	−166
−73.3	**−100**	−148.0
−67.8	**−90**	−130.0
−62.2	**−80**	−112.0
−59.4	**−75**	−103.0
−56.7	**−70**	−94.0
−53.9	**−65**	−85.0
−51.1	**−60**	−76.0
−48.3	**−55**	−67.0
−45.6	**−50**	−58.0
−42.8	**−45**	−49.0
−40.0	**−40**	−40.0
−37.2	**−35**	−31.0
−34.4	**−30**	−22.0
−31.7	**−25**	−13.0
−28.9	**−20**	−4.0
−26.1	**−15**	5.0
−23.3	**−10**	.4.0

Centigrade		Fahrenheit
−20.6	**−5**	23.0
−17.8	**0**	32.0
−17.2	**1**	33.8
−16.7	**2**	35.6
−16.1	**3**	37.4
−15.6	**4**	39.2
−15.0	**5**	41.0
−14.4	**6**	42.8
−13.9	**7**	44.6
−13.3	**8**	46.4
−12.8	**9**	48.2
−12.2	**10**	50.0
−11.7	**11**	51.8
−11.1	**12**	53.6
−10.6	**13**	55.4
−10.0	**14**	57.2
−9.4	**15**	59.0
−8.9	**16**	60.8
−8.3	**17**	62.6
−7.8	**18**	64.4
−7.2	**19**	66.2
−6.7	**20**	68.0
−6.1	**21**	69.8
−5.6	**22**	71.6
−5.0	**23**	73.4
−4.4	**24**	75.2
−3.9	**25**	77.0
−3.3	**26**	78.8
−2.8	**27**	80.6
−2.2	**28**	82.4
−1.7	**29**	84.2
−1.1	**30**	86.0
−0.6	**31**	87.8
0.0	**32**	89.6
0.6	**33**	91.4
1.1	**34**	93.2
1.7	**35**	95.0
2.2	**36**	96.8
2.8	**37**	98.6
3.3	**38**	100.4
3.9	**39**	102.2
4.4	**40**	104.0
5.0	**41**	105.8
5.6	**42**	107.6
6.1	**43**	109.4
6.7	**44**	111.2
7.2	**45**	113.0
7.8	**46**	114.8
8.3	**47**	116.6
8.9	**48**	118.4
9.4	**49**	120.2
10.0	**50**	122.0
10.6	**51**	123.8

Centigrade		Fahrenheit
11.1	**52**	125.6
11.7	**53**	127.4
12.2	**54**	129.2
12.8	**55**	131.0
13.3	**56**	132.8
13.9	**57**	134.6
14.4	**58**	136.4
15.0	**59**	138.2
15.6	**60**	140.0
16.1	**61**	141.8
16.7	**62**	143.6
17.2	**63**	145.4
17.8	**64**	147.2
18.3	**65**	149.0
18.9	**66**	150.8
19.4	**67**	152.6
20.0	**68**	154.4
20.6	**69**	156.2
21.1	**70**	158.0
21.7	**71**	159.8
22.2	**72**	161.6
22.8	**73**	163.4
23.3	**74**	165.2
23.9	**75**	167.0
24.4	**76**	168.8
25.0	**77**	170.6
25.6	**78**	172.4
26.1	**79**	174.2
26.7	**80**	176.0
27.2	**81**	177.8
27.8	**82**	179.6
28.3	**83**	181.4
28.9	**84**	183.2
29.4	**85**	185.0
30.0	**86**	186.8
30.6	**87**	188.6
31.1	**88**	190.4
31.7	**89**	192.2
32.2	**90**	194.0
32.8	**91**	195.8
33.3	**92**	197.6
33.9	**93**	199.4
34.4	**94**	201.2
35.0	**95**	203.0
35.6	**96**	204.8
36.1	**97**	206.6
36.7	**98**	208.4
37.2	**99**	210.2
37.8	**100**	212.0
40.6	**105**	221
43.3	**110**	230
46.1	**115**	239
48.9	**120**	248
51.7	**125**	257

Centigrade		Fahrenheit
54.4	**130**	266
57.2	**135**	275
60.0	**140**	284
62.8	**145**	293
65.6	**150**	302
68.3	**155**	311
71.1	**160**	320
73.9	**165**	329
76.7	**170**	338
79.4	**175**	347
82.2	**180**	356
85.0	**185**	365
87.8	**190**	374
90.6	**195**	383
93.3	**200**	392
96.1	**205**	401
98.9	**210**	410
100.0	**212**	414
102	**215**	419
104	**220**	428
107	**225**	437
110	**230**	446
113	**235**	455
116	**240**	464
118	**245**	473
121	**250**	482
124	**255**	491
127	**260**	500
129	**265**	509
132	**270**	518
135	**275**	527
138	**280**	536
141	**285**	545
143	**290**	554
146	**295**	563
149	**300**	572
154	**310**	590
160	**320**	608
166	**330**	626
171	**340**	644
177	**350**	662
182	**360**	680
188	**370**	698
193	**380**	716
199	**390**	734
204	**400**	752
210	**410**	770
216	**420**	788
221	**430**	806
227	**440**	824
232	**450**	842
238	**460**	860
243	**470**	878
249	**480**	896
254	**490**	914
260	**500**	932

The formulas at the right may also be used for converting Centigrade or Fahrenheit degrees into the other scales.

Degrees Cent., $°C = \frac{5}{9}(°F + 40) - 40$

$= \frac{5}{9}(°F - 32)$

Degrees Fahr., $°F = \frac{9}{5}(°C + 40) - 40$

$= \frac{9}{5}\ C + 32$

Degrees Kelvin, $°K = °C + 273.2$

Degrees Rankine, $°R = °F + 459.7$

Courtesy Ingersoll-Rand Co.

A-6.
Altitude and Atmospheric Pressures

Altitude above Sea Level			Temperature**		Barometer*		Atmospheric Pressure	
Feet*	Miles	Meters*	°F	°C	Inches Hg Abs.	mm Hg Abs.	PSIA	Kg/sq cm Abs.
-5000	——	-1526	77	25	35.58	903.7	17.48	1.229
-4500	——	-1373	75	24	35.00	889.0	17.19	1.209
-4000	——	-1220	73	23	34.42	874.3	16.90	1.188
-3500	——	-1068	71	22	33.84	859.5	16.62	1.169
-3000	——	-915	70	21	33.27	845.1	16.34	1.149
-2500	——	-763	68	20	32.70	830.6	16.06	1.129
-2000	——	-610	66	19	32.14	816.4	15.78	1.109
-1500	——	-458	64	18	31.58	802.1	15.51	1.091
-1000	——	-305	63	17	31.02	787.9	15.23	1.071
-500	——	-153	61	16	30.47	773.9	14.96	1.052
0	——	0	59	15	29.92	760.0	14.696	1.0333
500	——	153	57	14	29.38	746.3	14.43	1.015
1000	——	305	55	13	28.86	733.0	14.16	.956
1500	——	458	54	12	28.33	719.6	13.91	.978
2000	——	610	52	11	27.82	706.6	13.66	.960
2500	——	763	50	10	27.32	693.9	13.41	.943
3000	——	915	48	9	26.82	681.2	13.17	.926
3500	——	1068	47	8	26.33	668.8	12.93	.909
4000	——	1220	45	7	25.84	656.3	12.69	.892
4500	——	1373	43	6	25.37	644.4	12.46	.876
5000	0.95	1526	41	5	24.90	632.5	12.23	.860
6000	1.1	1831	38	3	23.99	609.3	11.78	.828
7000	1.3	2136	34	1	23.10	586.7	11.34	.797
8000	1.5	2441	31	-1	22.23	564.6	10.91	.767
9000	1.7	2746	27	-3	21.39	543.3	10.50	.738
10,000	1.9	3050	23	-5	20.58	522.7	10.10	.710
15,000	2.8	4577	6	-14	16.89	429.0	8.29	.583
20,000	3.8	6102	-12	-24	13.76	349.5	6.76	.475
25,000	4.7	7628	-30	-34	11.12	282.4	5.46	.384
30,000	5.7	9153	-48	-44	8.903	226.1	4.37	.307
35,000	6.6	10,679	-66		7.060	179.3	3.47	.244
40,000	7.6	12,204	-70	-57	5.558	141.2	2.73	.192
45,000	8.5	13,730	-70	-57	4.375	111.1	2.15	.151
50,000	9.5	15,255	-70	-57	3.444	87.5	1.69	.119
55,000	10.4	16,781	-70	-57	2.712	68.9	1.33	.0935
60,000	11.4	18,306	-70	-57	2.135	54.2	1.05	.0738
70,000	13.3	21,357	-67	-55	1.325	33.7	.651	.0458
80,000	15.2	24,408	-62	-52	†8.273^{-1}	21.0	.406	.0285
90,000	17.1	27,459	-57	-59	5.200^{-1}	13.2	.255	.0179
100,000	18.9	30,510	-51	-46	3.290^{-1}	8.36	.162	.0114
120,000	22.8	36,612	-26	-48	1.358^{-1}	3.45	——	——
140,000	26.6	42,714	4	-16	5.947^{-2}	1.51	——	——
160,000	30.4	48,816	28	-2	2.746^{-2}	†6.97^{-1}	——	——
180,000	34.2	54,918	19	-7	1.284^{-2}	3.26^{-1}	——	——
200,000	37.9	61,020	-3	-19	5.846^{-3}	1.48^{-1}	——	——
220,000	41.7	67,122	-44	-42	2.523^{-3}	6.41^{-2}	——	——
240,000	45.5	73,224	-86	-66	9.955^{-4}	2.53^{-2}	——	——
260,000	49.3	79,326	-129	-90	3.513^{-4}	8.92^{-3}	——	——
280,000	53.1	85,428	-135	-93	1.143^{-4}	3.67^{-3}	——	——
300,000	56.9	91,530	-127	-88	3.737^{-5}	9.49^{-4}	——	——
400,000	75.9	122,040	——	——	6.3^{-7}	1.60^{-5}	——	——
500,000	94.8	152,550	——	——	1.4^{-7}	3.56^{-6}	——	——
600,000	114	183,060	——	——	5.9^{-8}	1.50^{-6}	——	——
800,000	152	244,080	——	——	1.6^{-8}	4.06^{-7}	——	——
1,000,000	189	305,100	——	——	5.1^{-9}	1.30^{-7}	——	——
1,200,000	228	366,120	——	——	2.0^{-9}	5.08^{-8}	——	——
1,400,000	266	427,140	——	——	8.2^{-10}	2.08^{-8}	——	——
1,600,000	304	488,160	——	——	3.8^{-10}	9.65^{-9}	——	——
1,800,000	342	549,180	——	——	1.8^{-10}	4.57^{-9}	——	——
2,000,000	379	610,200	——	——	9.2^{-11}	2.34^{-9}	——	——

Data from NASA Standard Atmosphere (1962).

*Temperature and barometer are approximate for negative altitudes.

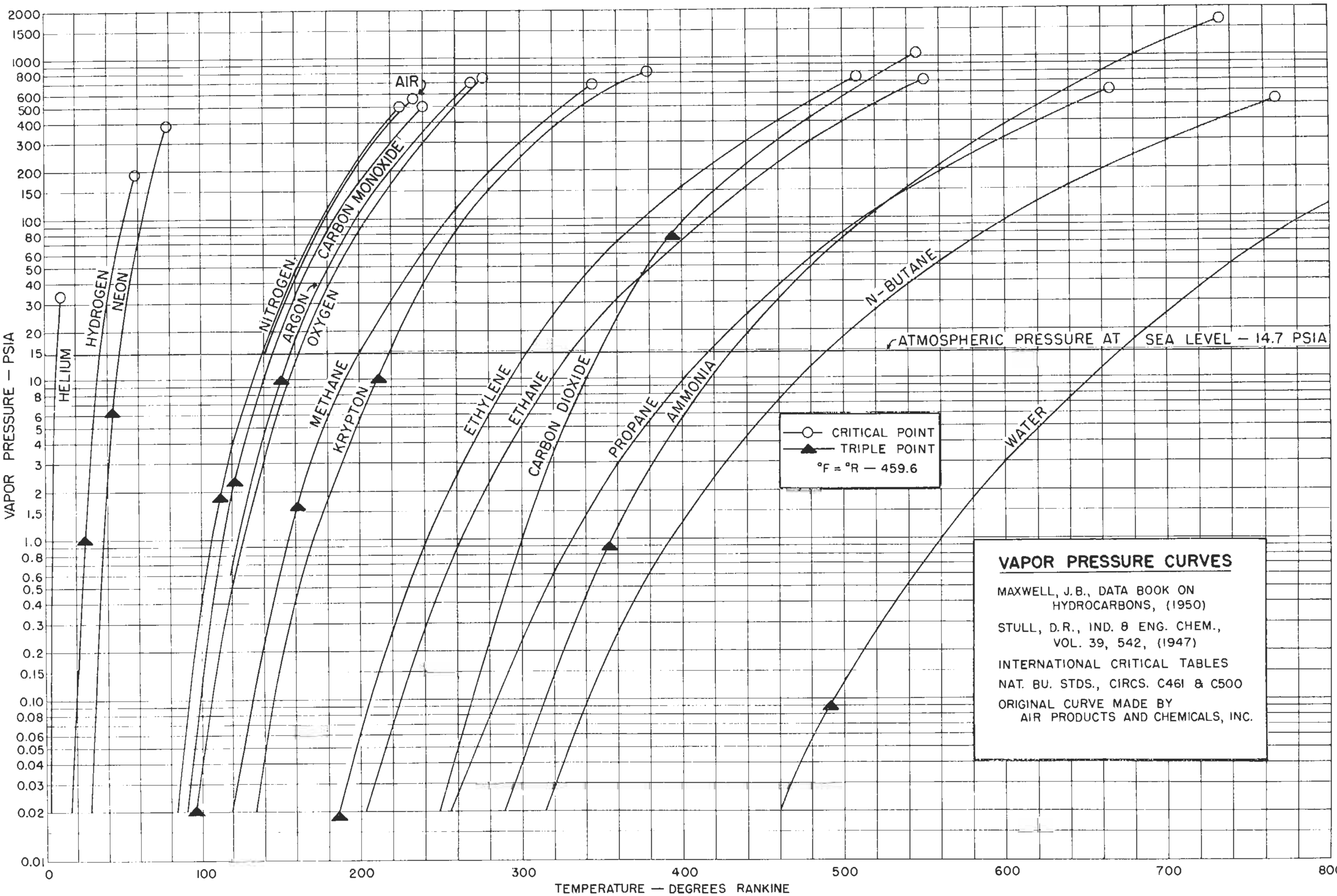
**Temperatures are average existing at 40° latitude and are rounded to even numbers.

†Negative exponent shows number of spaces the decimal point must be moved to the left.

Courtesy Ingersoll-Rand Co.

A-7.

Vapor Pressure Curves. (Courtesy Ingersoll-Rand Co.)



A-8.

Pressure Conversion Chart

MULTIPLY GIVEN NUMBER OF

BY FACTOR TO OBTAIN ⟶

GIVEN	lb/in²	in H_2O (at +39.2°F)	cm H_2O (at +4°C)	in Hg (at +32°F)	mm Hg (Torr) (at 0°C)	dyne/cm² (1μ bar)	newton/m² (PASCAL)	kgm/cm²	bar	atm. (A_n)	lb/ft²	ft H_2O (at +39.2°F)
lb/in²	1.0000	2.7680×10^{1}	7.0308×10^{1}	2.0360	5.1715×10^{1}	6.8948×10^{4}	6.8948×10^{3}	7.0306×10^{-2}	6.8947×10^{-2}	6.8045×10^{-2}	1.4400×10^{2}	2.3067
in H_2O (at +39.2°F)	3.6127×10^{-2}	1.0000	2.5400	7.3554×10^{-2}	1.8683	2.4908×10^{3}	2.4908×10^{2}	2.5399×10^{-3}	2.4908×10^{-3}	2.4582×10^{-3}	5.2022	8.3333×10^{-2}
cm H_2O (at +4°C)	1.4223×10^{-2}	0.3937	1.0000	2.8958×10^{-2}	0.7355	9.8064×10^{2}	9.8064×10^{1}	9.9997×10^{-4}	9.8064×10^{-4}	9.6781×10^{-4}	2.0481	3.2808×10^{-2}
in Hg (at +32°F)	4.9116×10^{-1}	1.3596×10^{1}	3.4532×10^{1}	1.0000	2.5400×10^{1}	3.3864×10^{4}	3.3864×10^{3}	3.4532×10^{-2}	3.3864×10^{-2}	3.3421×10^{-2}	7.0727×10^{1}	1.1330
mm Hg (Torr) (at 0°C)	1.9337×10^{-2}	5.3525×10^{-1}	1.3595	3.9370×10^{-2}	1.0000	1.3332×10^{3}	1.3332×10^{2}	1.3595×10^{-3}	1.3332×10^{-3}	1.3158×10^{-3}	2.7845	4.4605×10^{-2}
dyne/cm² (1μ bar)	1.4504×10^{-5}	4.0147×10^{-4}	1.0197×10^{-3}	2.9530×10^{-5}	7.5006×10^{-4}	1.0000	1.0000×10^{-1}	1.0197×10^{-6}	1.0000×10^{-6}	9.8692×10^{-7}	2.0886×10^{-3}	3.3456×10^{-5}
newton/m² (PASCAL)	1.4504×10^{-4}	4.0147×10^{-3}	1.0197×10^{-2}	2.9530×10^{-4}	7.5006×10^{-3}	1.0000×10^{1}	1.0000	1.0197×10^{-5}	1.0000×10^{-5}	9.8692×10^{-6}	2.0885×10^{-2}	3.3456×10^{-4}
kgm/cm²	1.4224×10^{1}	3.9371×10^{2}	1.00003×10^{3}	2.8959×10^{1}	7.3556×10^{2}	9.8060×10^{5}	9.8060×10^{4}	1.0000	9.8060×10^{-1}	9.678×10^{-1}	2.0482×10^{3}	3.2809×10^{1}
bar	1.4504×10^{1}	4.0147×10^{2}	1.0197×10^{3}	2.9530×10^{1}	7.5006×10^{2}	1.0000×10^{6}	1.0000×10^{5}	1.0197	1.0000	9.8692×10^{-1}	2.0885×10^{3}	3.3456×10^{1}
atm. (A_n)	1.4696×10^{1}	4.0679×10^{2}	1.0333×10^{3}	2.9921×10^{1}	7.6000×10^{2}	1.0133×10^{6}	1.0133×10^{5}	1.0332	1.0133	1.0000	2.1162×10^{3}	3.3900×10^{1}
lb/ft²	6.9445×10^{-3}	1.9223×10^{-1}	4.882×10^{-1}	1.4139×10^{-2}	3.591×10^{-1}	4.7880×10^{2}	4.7880×10^{1}	4.8824×10^{-4}	4.7880×10^{-4}	4.7254×10^{-4}	1.0000	1.6019×10^{-2}
ft H_2O (at +39.2°F)	4.3352×10^{-1}	1.2000×10^{1}	3.0480×10^{1}	8.826×10^{-1}	2.2419×10^{1}	2.9890×10^{4}	2.9890×10^{3}	3.0479×10^{-2}	2.9890×10^{-2}	2.9499×10^{-2}	6.2427×10^{1}	1.0000

A-9.
Vacuum Conversion

Torr	Absolute Pressure		Inches Hg	Psia	Vacuum*
	Microns Hg	Mm Hg	(Abs.)		Inches Hg
		762	30.00	14.74	—
		750	29.53	14.50	0.47
		700	27.56	13.54	2.44
		650	25.59	12.57	4.41
		600	23.62	11.60	6.38
		550	21.65	10.64	8.35
		500	19.68	9.67	10.32
		450	17.72	8.70	12.28
		400	15.75	7.74	14.25
		350	13.78	6.77	16.22
		300	11.81	5.80	18.19
		250	9.84	4.84	20.16
		200	7.84	3.87	22.13
		150	5.91	2.900	24.09
		100	3.94	1.934	26.06
		50	1.97	.967	28.03
		40	1.57	.774	28.43
		30	1.181	.580	28.82
		20	0.787	.3868	
		10	0.394	.1934	
		5	.197	.0967	
		4	.158	.0774	
		3	.1181	.0580	
		2	.0787	.0387	
1.0	1000	1	.0392	.0193	
0.5	500	0.50	.0197		Low Vacuum
1×10^{-1}	100	0.10	.0039		
5×10^{-2}	50	0.050			
1×10^{-2}	10	0.010			
5×10^{-3}	5	0.005			
1×10^{-3}	1	0.001			
1×10^{-4} to 1×10^{-6}					High Vacuum
1×10^{-6} to 1×10^{-9}					Very High Vac.
1×10^{-9} and beyond					Ultra High Vac.

*Refers to 30″ Barometer

Conversion Factors:
1 millimeter = 1000 microns 1 inch Hg = 25.4 mm Hg
1 Torr = 1 mm Hg Abs. 1 atmosphere = 14.7 pounds per sq. in. = 760 mm Hg = 29.92 in. Hg

Courtesy Pfaudler Co., Div. of Sybron Corp.

A-10.
Decimal and Millimeter Equivalents of Fractions

Inches: Fractions	Inches: Decimals	Millimeters	Inches: Fractions	Inches: Decimals	Millimeters
1/64	.015625	.397	33/64	.515625	13.097
1/32	.03125	.794	17/32	.53125	13.494
3/64	.046875	1.191	35/64	.546875	13.891
1/16	.0625	1.588	9/16	.5625	14.288
5/64	.078125	1.984	37/64	.578125	14.684
3/32	.09375	2.381	19/32	.59375	15.081
7/64	.109375	2.778	39/64	.609375	15.478
1/8	.125	3.175	5/8	.625	15.875
9/64	.140625	3.572	41/64	.640625	16.272
5/32	.15625	3.969	21/32	.65625	16.669
11/64	.171875	4.366	43/64	.671875	17.066
3/16	.1875	4.763	11/16	.6875	17.463
13/64	.203125	5.159	45/64	.703125	17.859
7/32	.21875	5.556	23/32	.71875	18.256
15/64	.234375	5.953	47/64	.734375	18.653
1/4	.250	6.350	3/4	.750	19.050
17/64	.265625	6.747	49/64	.765625	19.447
9/32	.28125	7.144	25/32	.78125	19.844
19/64	.296875	7.541	51/64	.796875	20.241
5/16	.3125	7.938	13/16	.8125	20.638
21/64	.328125	8.334	53/64	.828125	21.034
11/32	.34375	8.731	27/32	.83475	21.431
23/64	.359375	9.128	55/64	.859375	21.828
3/8	.375	9.525	7/8	.875	22.225
25/64	.390625	9.922	57/64	.890625	22.622
13/32	.40625	10.319	29/32	.90625	23.019
27/64	.421875	10.716	59/64	.921875	23.416
7/16	.4375	11.113	15/16	.9375	23.813
29/64	.453125	11.509	61/64	.953125	24.209
15/32	.46875	11.906	31/32	.96875	24.606
31/64	.484375	12.303	63/64	.984375	25.003
1/2	.500	12.700	1	1.000	25.400

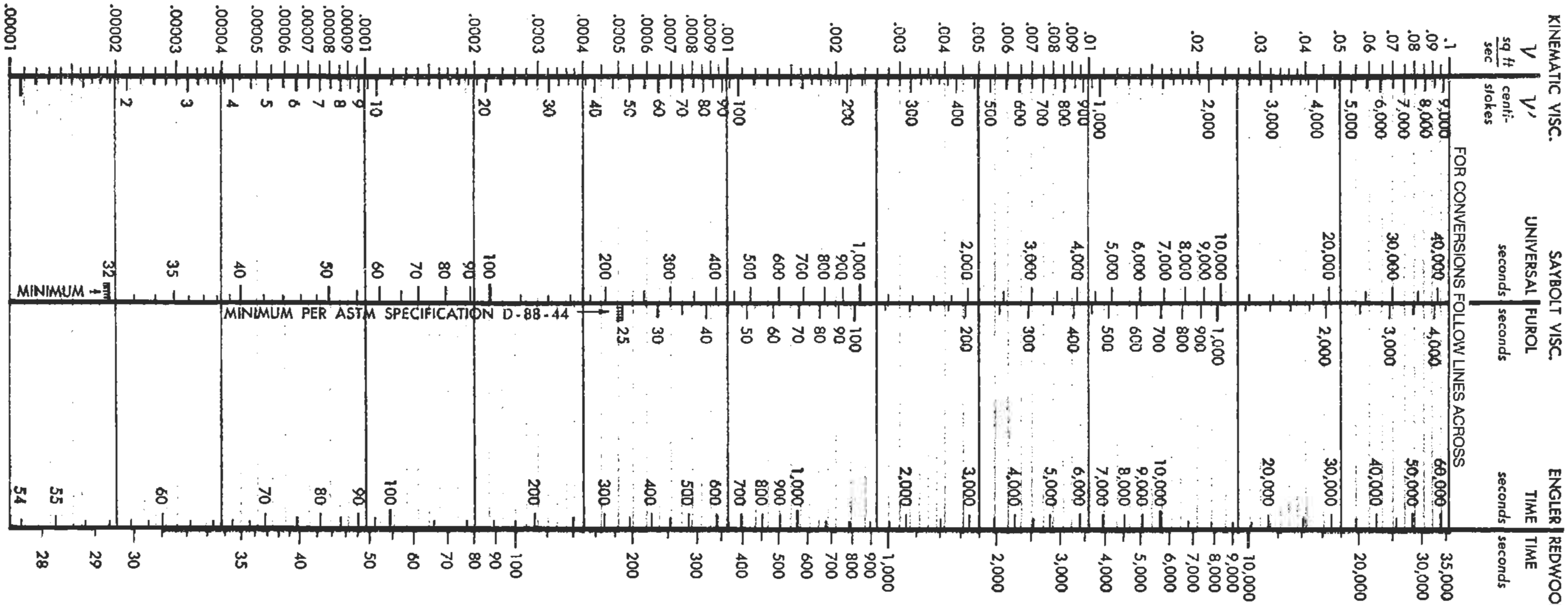
A-11.
Particle Size Measurement

Meshes/Lineal Inch US and ASTM Std. Sieve No.	Actual Opening: Inches	Actual Opening: Microns	Meshes/Lineal Inch US and ASTM Std. Sieve No.	Actual Opening: Inches	Actual Opening: Microns
10	.0787	2000	170	.0035	88
12	.0661 1/6	1680	200	.0029	74
14	.0555	1410		.0026	65
16	.0469 3/64	1190	230	.0024	62
18	.0394	1000	270	.0021	53
20	.0331 1/32	840		.0020	50
25	.0280	710	325	.0017	44
30	.0232	590		.0016	40
35	.0197 1/64	500	400	.00142	36
40	.0165	420		.00118	30
45	.0138	350	550	.00099	25
50	.0117	297	625	.00079	20
60	.0098	250		.00059	15
70	.0083	210	1,250	.000394	10
80	.0070	177	1,750	.000315	8
100	.0059	149	2,500	.000197	5
120	.0049	125	5,000	.000099	2.5
140	.0041	105	12,000	.0000394	1

* 1 micron (μ) = 1 micrometer (μm), new National Bureau of Standards terminology
1 micron = one-millionth of a meter
Inches $\times$ 25,400 = microns or micrometers
Reference ASTM E 11-70

A-12.

Viscosity Conversions. (By permission, Tube Turns Div., Chemetron Corp., Bull. TT 725.)



To convert other units into kinematic viscosity in English units ν (sq ft per sec) or in Metric units ν' (centistokes), use the chart or the formulas to the right:

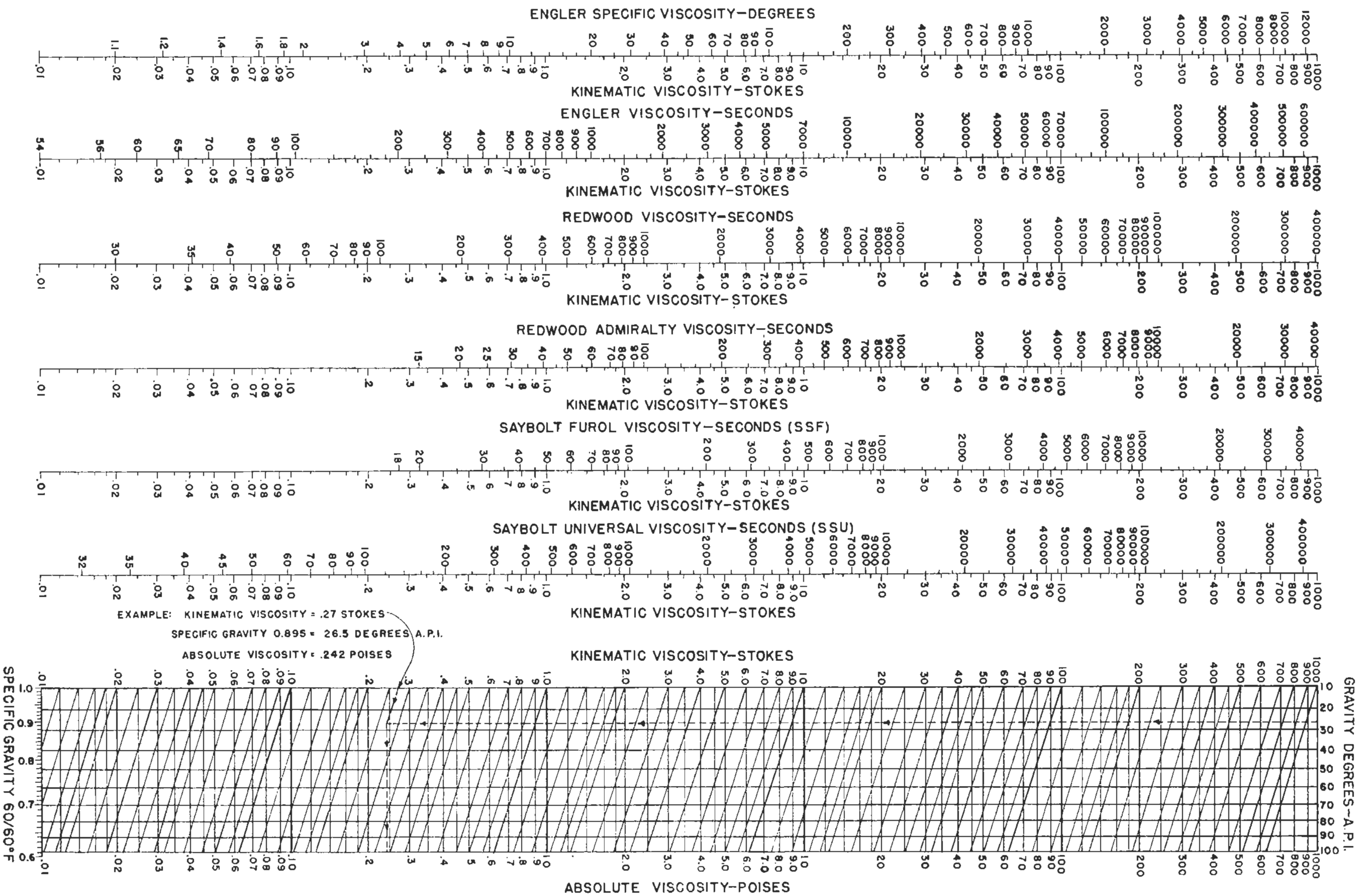
To convert:	into centistokes (ν')	into sq ft per sec (ν)
from Metric units (centistokes)	....	$\nu = 0.000\,010\,76\ \nu'$
from English units (sq ft per sec)	$\nu' = 92\,900\ \nu$	....
from Saybolt Universal (seconds)	see Table I in ASTM Spec. D-446-39 (plotted for basic temperature 100 F)	converted from ASTM Spec. D-446-39
from Saybolt Furol (seconds)	see Table I in ASTM Spec. D-666-44 (plotted for std temp of 122 F)	converted from ASTM Spec. D-666-44
from Engler (seconds)	$\nu' = 0.147\ \text{Engler} - \dfrac{374}{\text{Engler}}$	$\nu = 0.000\,001\,58\ \text{Engler} - \dfrac{0.00403}{\text{Engler}}$
from Redwood standard (seconds)	$\nu' = 0.260\ \text{Redwood} - \dfrac{171.5}{\text{Redwood}}$	$\nu = 0.000\,002\,80\ \text{Redwood} - \dfrac{0.00185}{\text{Redwood}}$
from absolute viscosity	$\nu' = \dfrac{\text{centipoises}}{\text{density}}$	$\nu = 32.2\ \mu\ \dfrac{(\text{lb sec per sq ft})}{\rho\ (\text{lb per cu ft})}$

To convert degrees API and Baumé into Specific Gravity, use the formulas to right:

Liquids lighter than water (API Formula)	Liquids heavier than water (U.S. Bureau of Stds.)
$\text{Specific gravity 60/60F} = \dfrac{141.5}{131.5 + \text{Degrees API}}$	$\text{Specific gravity} = \dfrac{145}{145 - \text{Degrees Baumé}}$

A-13.

Viscosity Conversions. (Courtesy Kinney Vacuum Div., The New York Air Brake Co.)



1 CENTISTOKE = (STOKE/100) 1 STOKE = 100 CENTISTOKES

1 CENTIPOISE = (POISE/100) 1 POISE = 100 CENTIPOISES

$$\text{KINEMATIC VISCOSITY (STOKES)} = \frac{\text{ABSOLUTE VISCOSITY (POISES)}}{\text{DENSITY AT GIVEN TEMPERATURE}}$$

A-14.
Commercial Wrought Steel Pipe Data
(Based on ANSI B36.10 wall thicknesses)

Schedule	Nominal Pipe Size	Outside Diameter	Thickness	Inside Diameter		Inside Diameter Functions (In Inches)				Transverse Internal Area	
				d	D	d^2	d^3	d^4	d^5	a	A
	Inches	Inches	Inches	Inches	Feet					Sq. In.	Sq. Ft.
Schedule 10	14	14	0.250	13.5	1.125	182.25	2460.4	33215.	448400.	143.14	0.994
	16	16	0.250	15.5	1.291	240.25	3723.9	57720.	894660.	188.69	1.310
	18	18	0.250	17.5	1.4583	306.25	5359.4	93789.	1641309.	240.53	1.670
	20	20	0.250	19.5	1.625	380.25	7414.9	144590.	2819500.	298.65	2.074
	24	24	0.250	23.5	1.958	552.25	12977.	304980.	7167030.	433.74	3.012
	30	30	0.312	29.376	2.448	862.95	25350.	744288.	21864218.	677.76	4.707
Schedule 20	8	8.625	0.250	8.125	0.6771	66.02	536.38	4359.3	35409.	51.85	0.3601
	10	10.75	0.250	10.25	0.8542	105.06	1076.9	11038.	113141.	82.52	0.5731
	12	12.75	0.250	12.25	1.021	150.06	1838.3	22518.	275855.	117.86	0.8185
	14	14	0.312	13.376	1.111	178.92	2393.2	32012.	428185.	140.52	0.9758
	16	16	0.312	15.376	1.281	236.42	3635.2	55894.	859442.	185.69	1.290
	18	18	0.312	17.376	1.448	301.92	5246.3	91156.	1583978.	237.13	1.647
	20	20	0.375	19.250	1.604	370.56	7133.3	137317.	2643352.	291.04	2.021
	24	24	0.375	23.25	1.937	540.56	12568.	292205.	6793832.	424.56	2.948
	30	30	0.500	29.00	2.417	841.0	24389.	707281.	20511149.	660.52	4.587
Schedule 30	8	8.625	0.277	8.071	0.6726	65.14	525.75	4243.2	34248.	51.16	0.3553
	10	10.75	0.307	10.136	0.8447	102.74	1041.4	10555.	106987.	80.69	0.5603
	12	12.75	0.330	12.09	1.0075	146.17	1767.2	21366.	258304.	114.80	0.7972
	14	14	0.375	13.25	1.1042	175.56	2326.2	30821.	408394.	137.88	0.9575
	16	16	0.375	15.25	1.2708	232.56	3546.6	54084.	824801.	182.65	1.268
	18	18	0.438	17.124	1.4270	293.23	5021.3	85984.	1472397.	230.30	1.599
	20	20	0.500	19.00	1.5833	361.00	6859.0	130321.	2476099.	283.53	1.969
	24	24	0.562	22.876	1.9063	523.31	11971.	273853.	6264703.	411.00	2.854
	30	30	0.625	28.75	2.3958	826.56	23764.	683201.	19642160.	649.18	4.508
Schedule 40	⅛	0.405	0.068	0.269	0.0224	0.0724	0.0195	0.005242	0.00141	0.057	0.00040
	¼	0.540	0.088	0.364	0.0303	0.1325	0.0482	0.01756	0.00639	0.104	0.00072
	⅜	0.675	0.091	0.493	0.0411	0.2430	0.1198	0.05905	0.02912	0.191	0.00133
	½	0.840	0.109	0.622	0.0518	0.3869	0.2406	0.1497	0.09310	0.304	0.00211
	¾	1.050	0.113	0.824	0.0687	0.679	0.5595	0.4610	0.3799	0.533	0.00371
	1	1.315	0.133	1.049	0.0874	1.100	1.154	1.210	1.270	0.864	0.00600
	1¼	1.660	0.140	1.380	0.1150	1.904	2.628	3.625	5.005	1.495	0.01040
	1½	1.900	0.145	1.610	0.1342	2.592	4.173	6.718	10.82	2.036	0.01414
	2	2.375	0.154	2.067	0.1722	4.272	8.831	18.250	37.72	3.355	0.02330
	2½	2.875	0.203	2.469	0.2057	6.096	15.051	37.161	91.75	4.788	0.03322
	3	3.500	0.216	3.068	0.2557	9.413	28.878	88.605	271.8	7.393	0.05130
	3½	4.000	0.226	3.548	0.2957	12.59	44.663	158.51	562.2	9.886	0.06870
	4	4.500	0.237	4.026	0.3355	16.21	65.256	262.76	1058.	12.730	0.08840
	5	5.563	0.258	5.047	0.4206	25.47	128.56	648.72	3275.	20.006	0.1390
	6	6.625	0.280	6.065	0.5054	36.78	223.10	1352.8	8206.	28.891	0.2006
	8	8.625	0.322	7.981	0.6651	63.70	508.36	4057.7	32380.	50.027	0.3474
	10	10.75	0.365	10.02	0.8350	100.4	1006.0	10080.	101000.	78.855	0.5475
	12	12.75	0.406	11.938	0.9965	142.5	1701.3	20306.	242470.	111.93	0.7773
	14	14.0	0.438	13.124	1.0937	172.24	2260.5	29666.	389340.	135.28	0.9394
	16	16.0	0.500	15.000	1.250	225.0	3375.0	50625.	759375.	176.72	1.2272
	18	18.0	0.562	16.876	1.4063	284.8	4806.3	81111.	1368820.	223.68	1.5533
	20	20.0	0.593	18.814	1.5678	354.0	6659.5	125320.	2357244.	278.00	1.9305
	24	24.0	0.687	22.626	1.8855	511.9	11583.	262040.	5929784.	402.07	2.7921
Schedule 60	8	8.625	0.406	7.813	0.6511	61.04	476.93	3725.9	29113.	47.94	0.3329
	10	10.75	0.500	9.750	0.8125	95.06	926.86	9036.4	88110.	74.66	0.5185
	12	12.75	0.562	11.626	0.9688	135.16	1571.4	18268.	212399.	106.16	0.7372
	14	14.0	0.593	12.814	1.0678	164.20	2104.0	26962.	345480.	128.96	0.8956
	16	16.0	0.656	14.688	1.2240	215.74	3168.8	46544.	683618.	169.44	1.1766
	18	18.0	0.750	16.500	1.3750	272.25	4492.1	74120.	1222982.	213.83	1.4849
	20	20.0	0.812	18.376	1.5313	337.68	6205.2	114028.	2095342.	265.21	1.8417
	24	24.0	0.968	22.064	1.8387	486.82	10741.	236994.	5229036.	382.35	2.6552
Schedule 80	⅛	0.405	0.095	0.215	0.0179	0.0462	0.00994	0.002134	0.000459	0.036	0.00025
	¼	0.540	0.119	0.302	0.0252	0.0912	0.0275	0.008317	0.002513	0.072	0.00050
	⅜	0.675	0.126	0.423	0.0353	0.1789	0.0757	0.03200	0.01354	0.141	0.00098
	½	0.840	0.147	0.546	0.0455	0.2981	0.1628	0.08886	0.04852	0.234	0.00163
	¾	1.050	0.154	0.742	0.0618	0.5506	0.4085	0.3032	0.2249	0.433	0.00300
	1	1.315	0.179	0.957	0.0797	0.9158	0.8765	0.8387	0.8027	0.719	0.00499
	1¼	1.660	0.191	1.278	0.1065	1.633	2.087	2.6667	3.409	1.283	0.00891

Courtesy Crane Co., Technical Manual 410, Flow of Fluids.

(Continued on next page)

A-14.
(Continued). Commercial Wrought Steel Pipe Data
(Based on ANSI B36.10 wall thicknesses)

Schedule	Nominal Pipe Size	Outside Diameter	Thickness	Inside Diameter d	Inside Diameter D	Inside Diameter Functions (In Inches) d^2	d^3	d^4	d^5	Transverse Internal Area a	A
	Inches	Inches	Inches	Inches	Feet					Sq. In.	Sq. Ft.
Schedule 80—cont.	1½	1.900	0.200	1.500	0.1250	2.250	3.375	5.062	7.594	1.767	0.01225
	2	2.375	0.218	1.939	0.1616	3.760	7.290	14.136	27.41	2.953	0.02050
	2½	2.875	0.276	2.323	0.1936	5.396	12.536	29.117	67.64	4.238	0.02942
	3	3.5	0.300	2.900	0.2417	8.410	24.389	70.728	205.1	6.605	0.04587
	3½	4.0	0.318	3.364	0.2803	11.32	38.069	128.14	430.8	8.888	0.06170
	4	4.5	0.337	3.826	0.3188	14.64	56.006	214.33	819.8	11.497	0.07986
	5	5.563	0.375	4.813	0.4011	23.16	111.49	536.38	2583.	18.194	0.1263
	6	6.625	0.432	5.761	0.4801	33.19	191.20	1101.6	6346.	26.067	0.1810
	8	8.625	0.500	7.625	0.6354	58.14	443.32	3380.3	25775.	45.663	0.3171
	10	10.75	0.593	9.564	0.7970	91.47	874.82	8366.8	80020.	71.84	0.4989
	12	12.75	0.687	11.376	0.9480	129.41	1472.2	16747.	190523.	101.64	0.7058
	14	14.0	0.750	12.500	1.0417	156.25	1953.1	24414.	305176.	122.72	0.8522
	16	16.0	0.843	14.314	1.1928	204.89	2932.8	41980.	600904.	160.92	1.1175
	18	18.0	0.937	16.126	1.3438	260.05	4193.5	67626.	1090518.	204.24	1.4183
	20	20.0	1.031	17.938	1.4948	321.77	5771.9	103536.	1857248.	252.72	1.7550
	24	24.0	1.218	21.564	1.7970	465.01	10027.	216234.	4662798.	365.22	2.5362
Schedule 100	8	8.625	0.593	7.439	0.6199	55.34	411.66	3062.	22781.	43.46	0.3018
	10	10.75	0.718	9.314	0.7762	86.75	807.99	7526.	69357.	68.13	0.4732
	12	12.75	0.843	11.064	0.9220	122.41	1354.4	14985.	165791.	96.14	0.6677
	14	14.0	0.937	12.126	1.0105	147.04	1783.0	21621.	262173.	115.49	0.8020
	16	16.0	1.031	13.938	1.1615	194.27	2707.7	37740.	526020.	152.58	1.0596
	18	18.0	1.156	15.688	1.3057	246.11	3861.0	60572.	950250.	193.30	1.3423
	20	20.0	1.281	17.438	1.4532	304.08	5302.6	92467.	1612438.	238.83	1.6585
	24	24.0	1.531	20.938	1.7448	438.40	9179.2	192195.	4024179.	344.32	2.3911
Schedule 120	4	4.50	0.438	3.624	0.302	13.133	47.595	172.49	625.1	10.315	0.07163
	5	5.563	0.500	4.563	0.3802	20.82	95.006	433.5	1978.	16.35	0.1136
	6	6.625	0.562	5.501	0.4584	30.26	166.47	915.7	5037.	23.77	0.1650
	8	8.625	0.718	7.189	0.5991	51.68	371.54	2671.	19202.	40.59	0.2819
	10	10.75	0.843	9.064	0.7553	82.16	744.66	6750.	61179.	64.53	0.4481
	12	12.75	1.000	10.750	0.8959	115.56	1242.3	13355.	143563.	90.76	0.6303
	14	14.0	1.093	11.814	0.9845	139.57	1648.9	19480.	230137.	109.62	0.7612
	16	16.0	1.218	13.564	1.1303	183.98	2495.5	33849.	459133.	144.50	1.0035
	18	18.0	1.375	15.250	1.2708	232.56	3546.6	54086.	824804.	182.66	1.2684
	20	20.0	1.500	17.000	1.4166	289.00	4913.0	83521.	1419857.	226.98	1.5762
	24	24.0	1.812	20.376	1.6980	415.18	8459.7	172375.	3512313.	326.08	2.2645
Schedule 140	8	8.625	0.812	7.001	0.5834	49.01	343.15	2402.	16819.	38.50	0.2673
	10	10.75	1.000	8.750	0.7292	76.56	669.92	5862.	51291.	60.13	0.4176
	12	12.75	1.125	10.500	0.8750	110.25	1157.6	12155.	127628.	86.59	0.6013
	14	14.0	1.250	11.500	0.9583	132.25	1520.9	17490.	201136.	103.87	0.7213
	16	16.0	1.438	13.124	1.0937	172.24	2260.5	29666.	389340.	135.28	0.9394
	18	18.0	1.562	14.876	1.2396	221.30	3292.0	48972.	728502.	173.80	1.2070
	20	20.0	1.750	16.5	1.3750	272.25	4492.1	74120.	1222981.	213.82	1.4849
	24	24.0	2.062	19.876	1.6563	395.06	7852.1	156069.	3102022.	310.28	2.1547
Schedule 160	½	0.840	0.187	0.466	0.0388	0.2172	0.1012	0.04716	0.02197	0.1706	0.00118
	¾	1.050	0.218	0.614	0.0512	0.3770	0.2315	0.1421	0.08726	0.2961	0.00206
	1	1.315	0.250	0.815	0.0679	0.6642	0.5413	0.4412	0.3596	0.5217	0.00362
	1¼	1.660	0.250	1.160	0.0966	1.346	1.561	1.811	2.100	1.057	0.00734
	1½	1.900	0.281	1.338	0.1115	1.790	2.395	3.205	4.288	1.406	0.00976
	2	2.375	0.343	1.689	0.1407	2.853	4.818	8.138	13.74	2.241	0.01556
	2½	2.875	0.375	2.125	0.1771	4.516	9.596	20.39	43.33	3.546	0.02463
	3	3.50	0.438	2.624	0.2187	6.885	18.067	47.41	124.4	5.408	0.03755
	4	4.50	0.531	3.438	0.2865	11.82	40.637	139.7	480.3	9.283	0.06447
	5	5.563	0.625	4.313	0.3594	18.60	80.230	346.0	1492.	14.61	0.1015
	6	6.625	0.718	5.189	0.4324	26.93	139.72	725.0	3762.	21.15	0.1469
	8	8.625	0.906	6.813	0.5677	46.42	316.24	2155.	14679.	36.46	0.2532
	10	10.75	1.125	8.500	0.7083	72.25	614.12	5220.	44371.	56.75	0.3941
	12	12.75	1.312	10.126	0.8438	102.54	1038.3	10514.	106461.	80.53	0.5592
	14	14.0	1.406	11.188	0.9323	125.17	1400.4	15668.	175292.	98.31	0.6827
	16	16.0	1.593	12.814	1.0678	164.20	2104.0	26961.	345482.	128.96	0.8956
	18	18.0	1.781	14.438	1.2032	208.45	3009.7	43454.	627387.	163.72	1.1369
	20	20.0	1.968	16.064	1.3387	258.05	4145.3	66590.	1069715.	202.67	1.4074
	24	24.0	2.343	19.314	1.6095	373.03	7204.7	139152.	2687582.	292.98	2.0346

A-14.
(Concluded). Commercial Wrought Steel Pipe Data
(Based on ANSI B36.10 wall thicknesses)

Nominal Pipe Size	Outside Diameter	Thickness	Inside Diameter		Inside Diameter Functions (In Inches)				Transverse Internal Area	
			d	D	d^2	d^3	d^4	d^5	a	A
Inches	Inches	Inches	Inches	Feet					Sq. In.	Sq. Ft.
					Standard Wall Pipe					
1/8	0.405	0.068	0.269	0.0224	0.0724	0.0195	0.00524	0.00141	0.057	0.00040
1/4	0.540	0.088	0.364	0.0303	0.1325	0.0482	0.01756	0.00639	0.104	0.00072
3/8	0.675	0.091	0.493	0.0411	0.2430	0.1198	0.05905	0.02912	0.191	0.00133
1/2	0.840	0.109	0.622	0.0518	0.3869	0.2406	0.1497	0.0931	0.304	0.00211
3/4	1.050	0.113	0.824	0.0687	0.679	0.5595	0.4610	0.3799	0.533	0.00371
1	1.315	0.133	1.049	0.0874	1.100	1.154	1.210	1.270	0.864	0.00600
1 1/4	1.660	0.140	1.380	0.1150	1.904	2.628	3.625	5.005	1.495	0.01040
1 1/2	1.900	0.145	1.610	0.1342	2.592	4.173	6.718	10.82	2.036	0.01414
2	2.375	0.154	2.067	0.1722	4.272	8.831	18.250	37.72	3.355	0.02330
2 1/2	2.875	0.203	2.469	0.2057	6.096	15.051	37.161	91.75	4.788	0.03322
3	3.500	0.216	3.068	0.2557	9.413	28.878	88.605	271.8	7.393	0.05130
3 1/2	4.000	0.226	3.548	0.2957	12.59	44.663	158.51	562.2	9.886	0.06870
4	4.500	0.237	4.026	0.3355	16.21	65.256	262.76	1058.	12.730	0.08840
5	5.563	0.258	5.047	0.4206	25.47	128.56	648.72	3275.	20.006	0.1390
6	6.625	0.280	6.065	0.5054	36.78	223.10	1352.8	8206.	28.891	0.2006
8	8.625	0.277	8.071	0.6725	65.14	525.75	4243.0	34248.	51.161	0.3553
	8.625S	0.322	7.981	0.6651	63.70	508.36	4057.7	32380.	50.027	0.3474
	10.75	0.279	10.192	0.8493	103.88	1058.7	10789.	109876.	81.585	0.5666
10	10.75	0.307	10.136	0.8446	102.74	1041.4	10555.	106987.	80.691	0.5604
	10.75S	0.365	10.020	0.8350	100.4	1006.0	10080.	101000.	78.855	0.5475
12	12.75	0.330	12.090	1.0075	146.17	1767.2	21366.	258300.	114.80	0.7972
	12.75S	0.375	12.000	1.000	144.0	1728.0	20736.	248800.	113.10	0.7854
					Extra Strong Pipe					
1/8	0.405	0.095	0.215	0.0179	0.0462	0.00994	0.002134	0.000459	0.036	0.00025
1/4	0.540	0.119	0.302	0.0252	0.0912	0.0275	0.008317	0.002513	0.072	0.00050
3/8	0.675	0.126	0.423	0.0353	0.1789	0.0757	0.03201	0.01354	0.141	0.00098
1/2	0.840	0.147	0.546	0.0455	0.2981	0.1628	0.08886	0.04852	0.234	0.00163
3/4	1.050	0.154	0.742	0.0618	0.5506	0.4085	0.3032	0.2249	0.433	0.00300
1	1.315	0.179	0.957	0.0797	0.9158	0.8765	0.8387	0.8027	0.719	0.00499
1 1/4	1.660	0.191	1.278	0.1065	1.633	2.087	2.6667	3.409	1.283	0.00891
1 1/2	1.900	0.200	1.500	0.1250	2.250	3.375	5.062	7.594	1.767	0.01225
2	2.375	0.218	1.939	0.1616	3.760	7.290	14.136	27.41	2.953	0.02050
2 1/2	2.875	0.276	2.323	0.1936	5.396	12.536	29.117	67.64	4.238	0.02942
3	3.500	0.300	2.900	0.2417	8.410	24.389	70.728	205.1	6.605	0.04587
3 1/2	4.000	0.318	3.364	0.2803	11.32	38.069	128.14	430.8	8.888	0.06170
4	4.500	0.337	3.826	0.3188	14.64	56.006	214.33	819.8	11.497	0.07986
5	5.563	0.375	4.813	0.4011	23.16	111.49	536.6	2583.	18.194	0.1263
6	6.625	0.432	5.761	0.4801	33.19	191.20	1101.6	6346.	26.067	0.1810
8	8.625	0.500	7.625	0.6354	58.14	443.32	3380.3	25775.	45.663	0.3171
10	10.75	0.500	9.750	0.8125	95.06	926.86	9036.4	88110.	74.662	0.5185
12	12.75	0.500	11.750	0.9792	138.1	1622.2	19072.	223970.	108.434	0.7528
					Double Extra Strong Pipe					
1/2	0.840	0.294	0.252	0.0210	0.0635	0.0160	0.004032	0.00102	0.050	0.00035
3/4	1.050	0.308	0.434	0.0362	0.1884	0.0817	0.03549	0.01540	0.148	0.00103
1	1.315	0.358	0.599	0.0499	0.3588	0.2149	0.1287	0.07711	0.282	0.00196
1 1/4	1.660	0.382	0.896	0.0747	0.8028	0.7193	0.6445	0.5775	0.630	0.00438
1 1/2	1.900	0.400	1.100	0.0917	1.210	1.331	1.4641	1.611	0.950	0.00660
2	2.375	0.436	1.503	0.1252	2.259	3.395	5.1031	7.670	1.774	0.01232
2 1/2	2.875	0.552	1.771	0.1476	3.136	5.554	9.8345	17.42	2.464	0.01710
3	3.500	0.600	2.300	0.1917	5.290	12.167	27.984	64.36	4.155	0.02885
3 1/2	4.000	0.636	2.728	0.2273	7.442	20.302	55.383	151.1	5.845	0.04059
4	4.500	0.674	3.152	0.2627	9.935	31.315	98.704	311.1	7.803	0.05419
5	5.563	0.750	4.063	0.3386	16.51	67.072	272.58	1107.	12.966	0.09006
6	6.625	0.864	4.897	0.4081	23.98	117.43	575.04	2816.	18.835	0.1308
8	8.625	0.875	6.875	0.5729	47.27	324.95	2234.4	15360.	37.122	0.2578

A-15.
Stainless Steel Pipe Data
(Based on ANSI B36.19 wall thicknesses)

Nominal Pipe Size	Outside Diameter	Thickness	Inside Diameter		Inside Diameter Functions (In Inches)				Transverse Internal Area	
			d	D	d^2	d^3	d^4	d^5	a	A
Inches	Inches	Inches	Inches	Feet					Sq. In.	Sq. Ft.
Schedule 5 S										
½	0.840	0.065	0.710	0.0592	0.504	0.358	0.254	0.1804	0.396	0.00275
¾	1.050	0.065	0.920	0.0767	0.846	0.779	0.716	0.659	0.664	0.00461
1	1.315	0.065	1.185	0.0988	1.404	1.664	1.972	2.337	1.103	0.00766
1¼	1.660	0.065	1.530	0.1275	2.341	3.582	5.480	8.384	1.839	0.01277
1½	1.900	0.065	1.770	0.1475	3.133	5.545	9.815	17.37	2.461	0.01709
2	2.375	0.065	2.245	0.1871	5.040	11.31	25.40	57.03	3.958	0.02749
2½	2.875	0.083	2.709	0.2258	7.339	19.88	53.86	145.9	5.764	0.04003
3	3.500	0.083	3.334	0.2778	11.12	37.06	123.6	411.9	8.733	0.06065
3½	4.000	0.083	3.834	0.3195	14.70	56.36	216.1	828.4	11.545	0.08017
4	4.500	0.083	4.334	0.3612	18.78	81.41	352.8	1529.	14.750	0.1024
5	5.563	0.109	5.345	0.4454	28.57	152.7	816.2	4363.	22.439	0.1558
6	6.625	0.109	6.407	0.5339	41.05	263.0	1685.	10796.	32.241	0.2239
8	8.625	0.109	8.407	0.7006	70.68	594.2	4995.	41996.	55.512	0.3855
10	10.750	0.134	10.482	0.8375	109.9	1152.	12072.	126538.	86.315	0.5994
12	12.750	0.156	12.438	1.0365	154.7	1924.	23933.	297682.	121.50	0.8438
Schedule 10 S										
⅛	0.405	0.049	0.307	0.0256	0.0942	0.0289	0.00888	0.00273	0.074	0.00051
¼	0.540	0.065	0.410	0.0342	0.1681	0.0689	0.02826	0.01159	0.132	0.00092
⅜	0.675	0.065	0.545	0.0454	0.2970	0.1619	0.08822	0.04808	0.233	0.00162
½	0.840	0.083	0.674	0.0562	0.4543	0.3062	0.2064	0.1391	0.357	0.00248
¾	1.050	0.083	0.884	0.0737	0.7815	0.6908	0.6107	0.5398	0.614	0.00426
1	1.315	0.109	1.097	0.0914	1.203	1.320	1.448	1.589	0.945	0.00656
1¼	1.660	0.109	1.442	0.1202	2.079	2.998	4.324	6.235	1.633	0.01134
1½	1.900	0.109	1.682	0.1402	2.829	4.759	8.004	13.46	2.222	0.01543
2	2.375	0.109	2.157	0.1798	4.653	10.04	21.65	46.69	3.654	0.02538
2½	2.875	0.120	2.635	0.2196	6.943	18.30	48.21	127.0	5.453	0.03787
3	3.500	0.120	3.260	0.2717	10.63	34.65	112.9	368.2	8.347	0.05796
3½	4.000	0.120	3.760	0.3133	14.14	53.16	199.9	751.5	11.11	0.07712
4	4.500	0.120	4.260	0.3550	18.15	77.31	329.3	1403.	14.26	0.09899
5	5.563	0.134	5.295	0.4413	28.04	148.5	786.1	4162.	22.02	0.1529
6	6.625	0.134	6.357	0.5298	40.41	256.9	1633.	10382.	31.74	0.2204
8	8.625	0.148	8.329	0.6941	69.37	577.8	4813.	40083.	54.48	0.3784
10	10.750	0.165	10.420	0.8683	108.6	1131.	11789.	122840.	85.29	0.5923
12	12.750	0.180	12.390	1.0325	153.5	1902.	23566.	291982.	120.6	0.8372
Schedule 40 S										
⅛ to 12	Values are the same, size for size, as those shown on the facing page for Standard Wall Pipe (heaviest weight on 8, 10, and 12-inch sizes).									
Schedule 80 S										
⅛ to 12	Values are the same, size for size, as those shown on the facing page for Extra Strong Pipe.									

Courtesy Crane Co., Technical Manual 410, Flow of Fluids.

A-16.
Properties of Pipe

Tabulated below are the most generally required data used in piping design. This table is believed to be the most comprehensive published up to this time. Many thicknesses traditionally included in such tables have been omitted because of their having become obsolete through disuse and lack of coverage by any Standard.

Sizes and thicknesses listed herein are covered by the following Standards:—

1) American National Standard Institute B36.10
2) American National Standard Institute B36.19
3) American Petroleum Institute Standard API 5L
4) American Petroleum Institute Standard API 5LX
5) New United States Legal Standard for Steel Plate Gauges.

Sizes and thicknesses to which no Standard designation applies are largely the more commonly used dimensions to which Taylor Forge Electric Fusion Welded Pipe is produced for a wide variety of applications including river crossings, penstocks, power plant and other piping.

All data is computed from the *nominal* dimensions listed and the effect of tolerances is not taken into account. Values are computed by application of the following formulas:

Radius of Gyration: $R = \frac{\sqrt{D^2 + d^2}}{4}$

Moment of Inertia: $I = R^2 A$

Section Modulus: $Z = \frac{I}{0.5\ D}$

ANSI American National Standards Institute

Nominal Pipe Size	Nominal Outside Diam. D	Designation	Wall Thickness	Inside Diam. d	Weight per Foot	Wt. of Water per Ft. of Pipe	Sq. Ft. Outside Surface per Ft.	Sq. Ft. Inside Surface per Ft.	Transverse Area in.^{2} a	Area of Metal in.^{2} A	Moment of Inertia in.^{4} I	Section Modulus in.^{3} Z	Radius of Gyration in. R
⅛	.405	10S	.049	.307	.186	.0320	.106	.0804	.0740	.0548	.00090	.00440	.1270
		Std.	.068	.269	.244	.0246	.106	.0705	.0568	.0720	.00106	.00530	.1215
		X-Stg.	.095	.215	.314	.0157	.106	.0563	.0364	.0925	.00122	.00600	.1146
¼	.540	10S	.065	.410	.330	.0570	.141	.1073	.1320	.0970	.00280	.01030	.1695
		Std.	.088	.364	.424	.0451	.141	.0955	.1041	.1250	.00331	.01230	.1628
		X-Stg.	.119	.302	.535	.0310	.141	.0794	.0716	.1574	.00378	.01395	.1547
⅜	.675	10S	.065	.545	.423	.1010	.177	.1427	.2333	.1245	.00590	.01740	.2160
		Std.	.091	.493	.567	.0827	.177	.1295	.1910	.1670	.00730	.02160	.2090
		X-Stg.	.126	.423	.738	.0609	.177	.1106	.1405	.2173	.00862	.02554	.1991
½	.840	10S	.083	.670	.671	.1550	.220	.1764	.3568	.1974	.01430	.03410	.2692
		Std.	.109	.622	.850	.1316	.220	.1637	.3040	.2503	.01710	.04070	.2613
		X-Stg.	.147	.546	1.087	.1013	.220	.1433	.2340	.3200	.02010	.04780	.2505
		160	.138	.464	1.310	.0740	.220	.1220	.1706	.3836	.02213	.05269	.2402
		XX-Stg.	.294	.252	1.714	.0216	.220	.0660	.0499	.5043	.02424	.05772	.2192
¾	1.050	10S	.083	.884	.857	.2660	.275	.2314	.6138	.2522	.02970	.05660	.3430
		Std.	.113	.824	1.130	.2301	.275	.2168	.5330	.3326	.03704	.07055	.3337
		X-Stg.	.154	.742	1.473	.1875	.275	.1948	.4330	.4335	.04479	.08531	.3214
		160	.219	.612	1.940	.1280	.275	.1607	.2961	.5698	.05270	.10038	.3041
		XX-Stg.	.308	.434	2.440	.0633	.275	.1137	.1479	.7180	.05792	.11030	.2840
1	1.315	10S	.109	1.097	1.404	.4090	.344	.2872	.9448	.4129	.07560	.1150	.4282
		Std.	.133	1.049	1.678	.3740	.344	.2740	.8640	.4939	.08734	.1328	.4205
		X-Stg.	.179	.957	2.171	.3112	.344	.2520	.7190	.6388	.10560	.1606	.4066
		160	.250	.815	2.850	.2261	.344	.2134	.5217	.8364	.12516	.1903	.3868
		XX-Stg.	.358	.599	3.659	.1221	.344	.1570	.2818	1.0760	.14050	.2136	.3613
1¼	1.660	10S	.109	1.442	1.806	.7080	.434	.3775	1.633	.5314	.1606	.1934	.5499
		Std.	.140	1.380	2.272	.6471	.434	.3620	1.495	.6685	.1947	.2346	.5397
			.191	1.278	2.996	.5553	.434	.3356	1.283	.8815	.2418	.2913	.5237
			.250	1.160	3.764	.4575	.434	.3029	1.057	1.1070	.2833	.3421	.5063
			.382	.896	5.214	.2732	.434	.2331	.6305	1.5340	.3411	.4110	.4716

Courtesy Taylor Forge Division, Energy Products Group, Gulf and Western Mfg. Co., by permission.

A-16.
(Continued). Properties of Pipe

Nominal Pipe Size	Nominal Outside Diam. D	Designation	Wall Thickness	Inside Diam. d	Weight per Foot	Wt. of Water per Ft. of Pipe	Sq. Ft. Outside Surface per Ft.	Sq. Ft. Inside Surface per Ft.	Transverse Area in.^{2} a	Area of Metal in.^{2} A	Moment of Inertia in.^{4} I	Section Modulus in.^{3} Z	Radius of Gyration in. R
1½	1.900	10S	.109	1.682	2.085	.9630	.497	.4403	2.221	.613	.2469	.2599	.6344
		Std.	.145	1.610	2.717	.8820	.497	.4213	2.036	.800	.3099	.3262	.6226
		X-Stg.	.200	1.500	3.631	.7648	.497	.3927	1.767	1.068	.3912	.4118	.6052
		160	.281	1.337	4.862	.6082	.497	.3519	1.405	1.430	.4826	.5080	.5809
		XX-Stg.	.400	1.100	6.408	.4117	.497	.2903	.950	1.885	.5678	.5977	.5489
2	2.375	10S	.109	2.157	2.638	1.583	.622	.5647	3.654	.775	.5003	.4213	.8034
		Std.	.154	2.067	3.652	1.452	.622	.5401	3.355	1.075	.6657	.5606	.7871
		X-Stg.	.218	1.939	5.022	1.279	.622	.5074	2.953	1.477	.8679	.7309	.7665
		- -	.250	1.875	5.673	1.196	.622	.4920	2.761	1.669	.9555	.8046	.7565
		160	.344	1.687	7.450	.970	.622	.4422	2.240	2.190	1.162	.9790	.7286
		XX-Stg.	.436	1.503	9.029	.769	.622	.3929	1.774	2.656	1.311	1.1040	.7027
2½	2.875	10S	.120	2.635	3.53	2.360	.753	.6900	5.453	1.038	.9878	.6872	.9755
		Std.	.203	2.469	5.79	2.072	.753	.6462	4.788	1.704	1.530	1.064	.9474
		X-Stg.	.276	2.323	7.66	1.834	.753	.6095	4.238	2.254	1.924	1.339	.9241
		160	.375	2.125	10.01	1.535	.753	.5564	3.547	2.945	2.353	1.638	.8938
		XX-Stg.	.552	1.771	13.69	1.067	.753	.4627	2.464	4.028	2.871	1.997	.8442
3	3.500	10S	.120	3.260	4.33	3.62	.916	.853	8.346	1.272	1.821	1.041	1.196
		API	.125	3.250	4.52	3.60	.916	.851	8.300	1.329	1.900	1.086	1.195
		API	.156	3.188	5.58	3.46	.916	.835	7.982	1.639	2.298	1.313	1.184
		API	.188	3.125	6.65	3.34	.916	.819	7.700	1.958	2.700	1.545	1.175
		Std.	.216	3.068	7.57	3.20	.916	.802	7.393	2.228	3.017	1.724	1.164
		API	.250	3.000	8.68	3.06	.916	.785	7.184	2.553	3.388	1.936	1.152
		API	.281	2.938	9.65	2.94	.916	.769	6.780	2.842	3.819	2.182	1.142
		X-Stg.	.300	2.900	10.25	2.86	.916	.761	6.605	3.016	3.892	2.225	1.136
		160	.438	2.624	14.32	2.34	.916	.687	5.407	4.214	5.044	2.882	1.094
		XX-Stg.	.600	2.300	18.58	1.80	.916	.601	4.155	5.466	5.993	3.424	1.047
3½	4.000	10S	.120	3.760	4.97	4.81	1.047	.984	11.10	1.46	2.754	1.377	1.372
		API	.125	3.750	5.18	4.79	1.047	.982	11.04	1.52	2.859	1.430	1.371
		API	.156	3.688	6.41	4.63	1.047	.966	10.68	1.88	3.485	1.743	1.360
		API	.188	3.624	7.71	4.48	1.047	.950	10.32	2.27	4.130	2.065	1.350
		Std.	.226	3.548	9.11	4.28	1.047	.929	9.89	2.68	4.788	2.394	1.337
		API	.250	3.500	10.02	4.17	1.047	.916	9.62	2.94	5.201	2.601	1.329
		API	.281	3.438	11.17	4.02	1.047	.900	9.28	3.29	5.715	2.858	1.319
		X-Stg.	.318	3.364	12.51	3.85	1.047	.880	8.89	3.68	6.280	3.140	1.307
		XX-Stg.	.636	2.728	22.85	2.53	1.047	.716	5.84	6.72	9.848	4.924	1.210
4	4.500	10S	.120	4.260	5.61	6.18	1.178	1.115	14.25	1.65	3.97	1.761	1.550
		API	.125	4.250	5.84	6.15	1.178	1.113	14.19	1.72	4.12	1.829	1.548
		API	.156	4.188	7.24	5.97	1.178	1.096	13.77	2.13	5.03	2.235	1.537
		API	.188	4.124	8.56	5.80	1.178	1.082	13.39	2.52	5.86	2.600	1.525
		API	.219	4.062	10.02	5.62	1.178	1.063	12.96	2.94	6.77	3.867	1.516
		Std.	.237	4.026	10.79	5.51	1.178	1.055	12.73	3.17	7.23	3.214	1.510
		API	.250	4.000	11.35	5.45	1.178	1.049	12.57	3.34	7.56	3.360	1.505
		API	.281	3.938	12.67	5.27	1.178	1.031	12.17	3.73	8.33	3.703	1.495
		API	.312	3.876	14.00	5.12	1.178	1.013	11.80	4.11	9.05	4.020	1.482
		X-Stg.	.337	3.826	14.98	4.98	1.178	1.002	11.50	4.41	9.61	4.271	1.477
		120	.438	3.624	19.00	4.47	1.178	.949	10.32	5.59	11.65	5.177	1.444
		- -	.500	3.500	21.36	4.16	1.178	.916	9.62	6.28	12.77	5.676	1.425
		160	.531	3.438	22.60	4.02	1.178	.900	9.28	6.62	13.27	5.900	1.416
		XX-Stg.	.674	3.152	27.54	3.38	1.178	.826	7.80	8.10	15.28	6.793	1.374

A-16.

(Continued). Properties of Pipe

Nominal Pipe Size	Nominal Outside Diam. D	Designation	Wall Thickness	Inside Diam. d	Weight per Foot	Wt. of Water per Ft. of Pipe	Sq. Ft. Outside Surface per Ft.	Sq. Ft. Inside Surface per Ft.	Transverse Area in.^{2} a	Area of Metal in.^{2} A	Moment of Inertia in.^{4} I	Section Modulus in.^{3} Z	Radius of Gyration in. R
5	5.563	10S	.134	5.295	7.77	9.54	1.456	1.386	22.02	2.29	8.42	3.028	1.920
		API	.156	5.251	9.02	9.39	1.456	1.375	21.66	2.65	9.70	3.487	1.913
		API	.188	5.187	10.80	9.16	1.456	1.358	21.13	3.17	11.49	4.129	1.902
		API	.219	5.125	12.51	8.94	1.456	1.342	20.63	3.68	13.14	4.726	1.891
		Std.	.258	5.047	14.62	8.66	1.456	1.321	20.01	4.30	15.16	5.451	1.878
		API	.281	5.001	15.86	8.52	1.456	1.309	19.64	4.66	16.31	5.862	1.870
		API	.312	4.939	17.51	8.31	1.456	1.293	19.16	5.15	17.81	6.402	1.860
		API	.344	4.875	19.19	8.09	1.456	1.276	18.67	5.64	19.28	6.932	1.849
		X-Stg.	.375	4.813	20.78	7.87	1.456	1.260	18.19	6.11	20.67	7.431	1.839
		120	.500	4.563	27.10	7.08	1.456	1.195	16.35	7.95	25.74	9.253	1.799
		160	.625	4.313	32.96	6.32	1.456	1.129	14.61	9.70	30.03	10.800	1.760
		XX-Stg.	.750	4.063	38.55	5.62	1.456	1.064	12.97	11.34	33.63	12.090	1.722
6	6.625	12 Ga.	.104	6.417	7.25	14.02	1.734	1.680	32.34	2.13	11.33	3.42	2.31
		10S	.134	6.357	9.29	13.70	1.734	1.660	31.75	2.73	14.38	4.34	2.29
		8 Ga.	.164	6.297	11.33	13.50	1.734	1.649	31.14	3.33	17.38	5.25	2.28
		API	.188	6.249	12.93	13.31	1.734	1.639	30.70	3.80	19.71	5.95	2.28
		6 Ga.	.194	6.237	13.34	13.25	1.734	1.633	30.55	3.92	20.29	6.12	2.27
		API	.219	6.187	15.02	13.05	1.734	1.620	30.10	4.41	22.66	6.84	2.27
		API	.250	6.125	17.02	12.80	1.734	1.606	29.50	5.01	25.55	7.71	2.26
		API	.277	6.071	18.86	12.55	1.734	1.591	28.95	5.54	28.00	8.46	2.25
		Std.	.280	6.065	18.97	12.51	1.734	1.587	28.90	5.58	28.14	8.50	2.24
		API	.312	6.001	21.05	12.26	1.734	1.571	28.28	6.19	30.91	9.33	2.23
		API	.344	5.937	23.09	12.00	1.734	1.554	27.68	6.79	33.51	10.14	2.22
		API	.375	5.875	25.10	11.75	1.734	1.540	27.10	7.37	36.20	10.90	2.21
		X-Stg.	.432	5.761	28.57	11.29	1.734	1.510	26.07	8.40	40.49	12.22	2.19
		- -	.500	5.625	32.79	10.85	1.734	1.475	24.85	9.63	45.60	13.78	2.16
		120	.562	5.501	36.40	10.30	1.734	1.470	23.77	10.74	49.91	15.07	2.15
		160	.719	5.187	45.30	9.16	1.734	1.359	21.15	13.36	58.99	17.81	2.10
		XX-Stg.	.864	4.897	53.16	8.14	1.734	1.280	18.83	15.64	66.33	20.02	2.06
8	8.625	12 Ga.	.104	8.417	9.47	24.1	2.26	2.204	55.6	2.78	25.3	5.86	3.01
		10 Ga.	.134	8.357	12.16	23.8	2.26	2.188	54.8	3.57	32.2	7.46	3.00
		10S	.148	8.329	13.40	23.6	2.26	2.180	54.5	3.94	35.4	8.22	3.00
		8 Ga.	.164	8.297	14.83	23.4	2.26	2.172	54.1	4.36	39.1	9.06	2.99
		API	.188	8.249	16.90	23.2	2.26	2.161	53.5	5.00	44.5	10.30	2.98
		6 Ga.	.194	8.237	17.48	23.1	2.26	2.156	53.3	5.14	45.7	10.60	2.98
		API	.203	8.219	18.30	23.1	2.26	2.152	53.1	5.38	47.7	11.05	2.98
		API	.219	8.187	19.64	22.9	2.26	2.148	52.7	5.80	51.3	11.90	2.97
		3 Ga.	.239	8.147	21.42	22.6	2.26	2.133	52.1	6.30	55.4	12.84	2.96
		20	.250	8.125	22.40	22.5	2.26	2.127	51.8	6.58	57.7	13.39	2.96
		30	.277	8.071	24.70	22.2	2.26	2.115	51.2	7.26	63.3	14.69	2.95
		API	.312	8.001	27.72	21.8	2.26	2.095	50.3	8.15	70.6	16.37	2.94
		Std.	.322	7.981	28.55	21.6	2.26	2.090	50.0	8.40	72.5	16.81	2.94
		API	.344	7.937	30.40	21.4	2.26	2.078	49.5	8.94	76.8	17.81	2.93
		API	.375	7.875	33.10	21.1	2.26	2.062	48.7	9.74	83.1	19.27	2.92
		60	.406	7.813	35.70	20.8	2.26	2.045	47.9	10.48	88.8	20.58	2.91
		API	.438	7.749	38.33	20.4	2.26	2.029	47.2	11.27	94.7	21.97	2.90
		X-Stg.	.500	7.625	43.39	19.8	2.26	2.006	45.6	12.76	105.7	24.51	2.88
		100	.594	7.437	50.90	18.8	2.26	1.947	43.5	14.96	121.4	28.14	2.85
		- -	.625	7.375	53.40	18.5	2.26	1.931	42.7	15.71	126.5	29.33	2.84
		120	.719	7.187	60.70	17.6	2.26	1.882	40.6	17.84	140.6	32.61	2.81
		140	.812	7.001	67.80	16.7	2.26	1.833	38.5	19.93	153.8	35.65	2.78
		XX-Stg.	.875	6.875	72.42	16.1	2.26	1.800	37.1	21.30	162.0	37.56	2.76
		160	.906	6.813	74.70	15.8	2.26	1.784	36.4	21.97	165.9	38.48	2.76

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(Continued). Properties of Pipe

Nominal Pipe Size	Nominal Outside Diam. D	Designation	Wall Thickness	Inside Diam. d	Weight per Foot	Wt. of Water per Ft. of Pipe	Sq. Ft. Outside Surface per Ft.	Sq. Ft. Inside Surface per Ft.	Transverse Area in.² a	Area of Metal in.² A	Moment of Inertia in.⁴ I	Section Modulus in.³ Z	Radius of Gyration in. R
10	10.750	12 Ga.	.104	10.542	11.83	37.8	2.81	2.76	87.3	3.48	49.3	9.16	3.76
		10 Ga.	.134	10.482	15.21	37.4	2.81	2.74	86.3	4.47	63.0	11.71	3.75
		8 Ga.	.164	10.422	18.56	37.0	2.81	2.73	85.3	5.45	76.4	14.22	3.74
		10S	.165	10.420	18.65	36.9	2.81	2.73	85.3	5.50	76.8	14.29	3.74
		API	.188	10.374	21.12	36.7	2.81	2.72	84.5	6.20	86.5	16.10	3.74
		6 Ga.	.194	10.362	21.89	36.6	2.81	2.71	84.3	6.43	89.7	16.68	3.73
		API	.203	10.344	22.86	36.5	2.81	2.71	84.0	6.71	93.3	17.35	3.73
		API	.219	10.310	24.60	36.2	2.81	2.70	83.4	7.24	100.5	18.70	3.72
		3 Ga.	.239	10.272	28.05	35.9	2.81	2.69	82.9	7.89	109.2	20.32	3.72
		20	.250	10.250	28.03	35.9	2.81	2.68	82.6	8.26	113.6	21.12	3.71
		API	.279	10.192	31.20	35.3	2.81	2.66	81.6	9.18	125.9	23.42	3.70
		30	.307	10.136	34.24	35.0	2.81	2.65	80.7	10.07	137.4	25.57	3.69
		API	.344	10.062	38.26	34.5	2.81	2.63	79.5	11.25	152.3	28.33	3.68
		Std.	.365	10.020	40.48	34.1	2.81	2.62	78.9	11.91	160.7	29.90	3.67
		API	.438	9.874	48.28	33.2	2.81	2.58	76.6	14.19	188.8	35.13	3.65
		X-Stg.	.500	9.750	54.74	32.3	2.81	2.55	74.7	16.10	212.0	39.43	3.63
		80	.594	9.562	64.40	31.1	2.81	2.50	71.8	18.91	244.9	45.56	3.60
		100	.719	9.312	77.00	29.5	2.81	2.44	68.1	22.62	286.2	53.25	3.56
		- -	.750	9.250	80.10	29.1	2.81	2.42	67.2	23.56	296.2	55.10	3.54
		120	.844	9.062	89.20	27.9	2.81	2.37	64.5	26.23	324.3	60.34	3.51
		140	1.000	8.750	104.20	26.1	2.81	2.29	60.1	30.63	367.8	68.43	3.46
		160	1.125	8.500	116.00	24.6	2.81	2.22	56.7	34.01	399.4	74.31	3.43
12	12.750	12 Ga.	.104	12.542	14.1	53.6	3.34	3.28	123.5	4.13	82.6	12.9	4.47
		10 Ga.	.134	12.482	18.1	53.0	3.34	3.27	122.4	5.31	105.7	16.6	4.46
		8 Ga.	.164	12.422	22.1	52.5	3.34	3.25	121.2	6.48	128.4	20.1	4.45
		10S	.180	12.390	24.2	52.2	3.34	3.24	120.6	7.11	140.4	22.0	4.44
		6 Ga.	.194	12.362	26.0	52.0	3.34	3.23	120.0	7.65	150.9	23.7	4.44
		API	.203	12.344	27.2	52.0	3.34	3.23	119.9	7.99	157.2	24.7	4.43
		API	.219	12.312	29.3	51.7	3.34	3.22	119.1	8.52	167.6	26.3	4.43
		3 Ga.	.239	12.272	32.0	51.3	3.34	3.21	118.3	9.39	183.8	28.8	4.42
		20	.250	12.250	33.4	51.3	3.34	3.12	118.0	9.84	192.3	30.2	4.42
		API	.281	12.188	37.4	50.6	3.34	3.19	116.7	11.01	214.1	33.6	4.41
		API	.312	12.126	41.5	50.1	3.34	3.17	115.5	12.19	236.0	37.0	4.40
		30	.330	12.090	43.8	49.7	3.34	3.16	114.8	12.88	248.5	39.0	4.39
		API	.344	12.062	45.5	49.7	3.34	3.16	114.5	13.46	259.0	40.7	4.38
		Std.	.375	12.000	49.6	48.9	3.34	3.14	113.1	14.58	279.3	43.8	4.37
		40	.406	11.938	53.6	48.5	3.34	3.13	111.9	15.74	300.3	47.1	4.37
		API	.438	11.874	57.5	48.2	3.34	3.11	111.0	16.95	321.0	50.4	4.35
		X-Stg.	.500	11.750	65.4	46.9	3.34	3.08	108.4	19.24	361.5	56.7	4.33
		60	.562	11.626	73.2	46.0	3.34	3.04	106.2	21.52	400.5	62.8	4.31
		- -	.625	11.500	80.9	44.9	3.34	3.01	103.8	23.81	438.7	68.8	4.29
		80	.688	11.374	88.6	44.0	3.34	2.98	101.6	26.03	475.2	74.6	4.27
		- -	.750	11.250	96.2	43.1	3.34	2.94	99.4	28.27	510.7	80.1	4.25
		100	.844	11.062	108.0	41.6	3.34	2.90	96.1	31.53	561.8	88.1	4.22
		- -	.875	11.000	110.9	41.1	3.34	2.88	95.0	32.64	578.5	90.7	4.21
		120	1.000	10.750	125.5	39.3	3.34	2.81	90.8	36.91	641.7	100.7	4.17
		140	1.125	10.500	140.0	37.5	3.34	2.75	86.6	41.08	700.7	109.9	4.13
		- -	1.250	10.250	153.6	35.8	3.34	2.68	82.5	45.16	755.5	118.5	4.09
		160	1.312	10.126	161.0	34.9	3.34	2.65	80.5	47.14	781.3	122.6	4.07
		- -	1.375	10.000	167.2	34.0	3.34	2.62	78.5	49.14	807.2	126.6	4.05
		- -	1.500	9.750	180.4	32.4	3.34	2.55	74.7	53.01	853.8	133.9	4.01

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(Continued). Properties of Pipe

Nominal Pipe Size	Nominal Outside Diam. D	Designation	Wall Thickness	Inside Diam. d	Weight per Foot	Wt. of Water per Ft. of Pipe	Sq. Ft. Outside Surface per Ft.	Sq. Ft. Inside Surface per Ft.	Transverse Area in.² a	Area of Metal in.² A	Moment of Inertia in.⁴ I	Section Modulus in.³ Z	Radius of Gyration in. R
14	14.000	10 Ga.	.134	13.732	20	64.2	3.67	3.59	148.1	5.84	140.4	20.1	4.90
		8 Ga.	.164	13.672	24	63.6	3.67	3.58	146.8	7.13	170.7	24.4	4.89
		6 Ga.	.194	13.612	29	63.1	3.67	3.56	145.5	8.41	200.6	28.7	4.88
		API	.210	13.580	31	62.8	3.67	3.55	144.8	9.10	216.2	30.9	4.87
		API	.219	13.562	32	62.6	3.67	3.55	144.5	9.48	225.1	32.2	4.87
		3 Ga.	.239	13.522	35	62.3	3.67	3.54	143.6	10.33	244.9	35.0	4.87
		10	.250	13.500	37	62.1	3.67	3.54	143.0	10.82	256.0	36.6	4.86
		API	.281	13.438	41	61.5	3.67	3.52	141.8	12.11	285.2	40.7	4.85
		20	.312	13.375	46	60.8	3.67	3.50	140.5	13.44	314.9	45.0	4.84
		API	.344	13.312	50	60.3	3.67	3.48	139.2	14.76	344.3	49.2	4.83
		Std.	.375	13.250	55	59.7	3.67	3.47	137.9	16.05	372.8	53.2	4.82
		40	.438	13.124	63	58.5	3.67	3.44	135.3	18.66	429.6	61.4	4.80
		X-Stg.	.500	13.000	72	57.4	3.67	3.40	132.7	21.21	483.8	69.1	4.78
		60	.594	12.812	85	55.9	3.67	3.35	129.0	24.98	562.4	80.3	4.74
		- -	.625	12.750	89	55.3	3.67	3.34	127.7	26.26	588.5	84.1	4.73
		80	.750	12.500	107	51.2	3.67	3.27	122.7	31.22	687.5	98.2	4.69
		- -	.875	12.250	123	51.1	3.67	3.21	117.9	36.08	780.1	111.4	4.65
		100	.938	12.124	131	50.0	3.67	3.17	115.5	38.47	820.5	117.2	4.63
		- -	1.000	12.000	139	49.0	3.67	3.14	113.1	40.84	868.0	124.0	4.61
		120	1.094	11.812	151	47.5	3.67	3.09	109.6	44.32	929.8	132.8	4.58
		- -	1.125	11.750	155	47.0	3.67	3.08	108.4	45.50	950.3	135.8	4.57
		140	1.250	11.500	171	45.0	3.67	3.01	103.9	50.07	1027.5	146.8	4.53
		- -	1.375	11.250	186	43.1	3.67	2.94	99.4	54.54	1099.5	157.1	4.49
		160	1.406	11.188	190	42.6	3.67	2.93	98.3	55.63	1116.9	159.6	4.48
		- -	1.500	11.000	200	41.2	3.67	2.88	95.0	58.90	1166.5	166.6	4.45
16	16.000	10 Ga.	.134	15.732	23	84.3	4.19	4.12	194.4	6.68	210	26.3	5.61
		8 Ga.	.164	15.672	28	83.6	4.19	4.10	192.9	8.16	256	32.0	5.60
		- -	.188	15.624	32	83.3	4.19	4.09	192.0	9.39	294	36.7	5.59
		6 Ga.	.194	15.612	33	83.0	4.19	4.09	191.4	9.63	301	37.6	5.59
		API	.219	15.562	37	82.5	4.19	4.07	190.2	10.86	338	42.3	5.58
		3 Ga.	.239	15.522	40	82.0	4.19	4.06	189.2	11.83	368	45.9	5.57
		10	.250	15.500	42	82.1	4.19	4.06	189.0	12.40	385	48.1	5.57
		API	.281	15.438	47	81.2	4.19	4.04	187.0	13.90	430	53.8	5.56
		20	.312	15.375	52	80.1	4.19	4.03	185.6	15.40	474	59.2	5.55
		API	.344	15.312	57	80.0	4.19	4.01	184.1	16.94	519	64.9	5.54
		Std.	.375	15.250	63	79.1	4.19	4.00	182.6	18.41	562	70.3	5.53
		API	.438	15.124	73	78.2	4.19	3.96	180.0	21.42	650	81.2	5.51
		X-Stg.	.500	15.000	83	76.5	4.19	3.93	176.7	24.35	732	91.5	5.48
		- -	.625	14.750	103	74.1	4.19	3.86	170.9	30.19	893	111.7	5.44
		60	.656	14.688	108	73.4	4.19	3.85	169.4	31.62	933	116.6	5.43
		- -	.750	14.500	122	71.5	4.19	3.80	165.1	35.93	1047	130.9	5.40
		80	.844	14.312	137	69.7	4.19	3.75	160.9	40.14	1157	144.6	5.37
		- -	.875	14.250	141	69.1	4.19	3.73	159.5	41.58	1192	149.0	5.35
		- -	1.000	14.000	160	66.7	4.19	3.66	153.9	47.12	1331	166.4	5.31
		100	1.031	13.938	165	66.0	4.19	3.65	152.6	48.49	1366	170.7	5.30
		- -	1.125	13.750	179	64.4	4.19	3.60	148.5	52.57	1463	182.9	5.27
		120	1.219	13.562	193	62.6	4.19	3.55	144.5	56.56	1556	194.5	5.24
		- -	1.250	13.500	197	62.1	4.19	3.53	143.1	57.92	1586	198.3	5.23
		- -	1.375	13.250	215	59.8	4.19	3.47	137.9	63.17	1704	213.0	5.19
		140	1.438	13.124	224	58.6	4.19	3.44	135.3	65.79	1761	220.1	5.17
		- -	1.500	13.000	232	57.4	4.19	3.40	132.7	68.33	1816	227.0	5.15
		160	1.594	12.812	245	55.9	4.19	3.35	129.0	72.10	1893	236.6	5.12

A-16.
(Continued). Properties of Pipe

Nominal Pipe Size	Nominal Outside Diam. D	Designation	Wall Thickness	Inside Diam. d	Weight per Foot	Wt. of Water per Ft. of Pipe	Sq. Ft. Outside Surface per Ft.	Sq. Ft. Inside Surface per Ft.	Transverse Area in.² a	Area of Metal in.² A	Moment of Inertia in.⁴ I	Section Modulus in.³ Z	Radius of Gyration in. R
18	18.000	10 Ga.	.134	17.732	26	107.1	4.71	4.64	246.9	7.52	300	33.4	6.32
		8 Ga.	.164	17.672	31	106.3	4.71	4.63	245.3	9.19	366	40.6	6.31
		6 Ga.	.194	17.612	37	105.6	4.71	4.61	243.6	10.85	430	47.8	6.29
		3 Ga.	.239	17.522	45	104.5	4.71	4.59	241.1	13.34	526	58.4	6.28
		10	.250	17.500	47	104.6	4.71	4.58	241.0	13.96	550	61.1	6.28
		API	.281	17.438	49	104.0	4.71	4.56	240.0	14.49	570	63.4	6.27
		20	.312	17.375	59	102.5	4.71	4.55	237.1	17.36	679	75.5	6.25
		API	.344	17.312	65	102.0	4.71	4.53	235.4	19.08	744	82.6	6.24
		Std.	.375	17.250	71	101.2	4.71	4.51	233.7	20.76	807	89.6	6.23
		API	.406	17.188	76	100.6	4.71	4.50	232.0	22.44	869	96.6	6.22
		30	.438	17.124	82	99.5	4.71	4.48	229.5	24.95	963	107.0	6.21
		X-Stg.	.500	17.000	93	98.2	4.71	4.45	227.0	27.49	1053	117.0	6.19
		40	.562	16.876	105	97.2	4.71	4.42	224.0	30.85	1177	130.9	6.17
		- -	.625	16.750	116	95.8	4.71	4.39	220.5	34.15	1290	143.2	6.14
		60	.750	16.500	138	92.5	4.71	4.32	213.8	40.64	1515	168.3	6.10
		- -	.875	16.250	160	89.9	4.71	4.25	207.4	47.07	1730	192.3	6.06
		80	.938	16.124	171	88.5	4.71	4.22	204.2	50.23	1834	203.8	6.04
		- -	1.000	16.000	182	87.2	4.71	4.19	201.1	53.41	1935	215.0	6.02
		- -	1.125	15.750	203	84.5	4.71	4.12	194.8	59.64	2133	237.0	5.98
		100	1.156	15.688	208	83.7	4.71	4.11	193.3	61.18	2182	242.3	5.97
		- -	1.250	15.500	224	81.8	4.71	4.06	188.7	65.78	2319	257.7	5.94
		120	1.375	15.250	244	79.2	4.71	3.99	182.7	71.82	2498	277.5	5.90
		- -	1.500	15.000	265	76.6	4.71	3.93	176.7	77.75	2668	296.5	5.86
		140	1.562	14.876	275	75.3	4.71	3.89	173.8	80.66	2750	305.5	5.84
		160	1.781	14.438	309	71.0	4.71	3.78	163.7	90.75	3020	335.5	5.77
20	20.000	10 Ga.	.134	19.732	28	132.6	5.24	5.17	305.8	8.36	413	41.3	7.02
		8 Ga.	.164	19.672	35	131.8	5.24	5.15	303.9	10.22	503	50.3	7.01
		6 Ga.	.194	19.612	41	131.0	5.24	5.13	302.1	12.07	592	59.2	7.00
		3 Ga.	.239	19.522	50	129.8	5.24	5.11	299.3	14.84	725	72.5	6.99
		10	.250	19.500	53	130.0	5.24	5.11	299.0	15.52	759	75.9	6.98
		API	.281	19.438	59	128.6	5.24	5.09	296.8	17.41	846	84.6	6.97
		API	.312	19.374	66	128.1	5.24	5.08	295.0	19.36	937	93.7	6.95
		API	.344	19.312	72	127.0	5.24	5.06	292.9	21.24	1026	102.6	6.95
		Std.	.375	19.250	79	126.0	5.24	5.04	291.1	23.12	1113	111.3	6.94
		API	.406	19.188	85	125.4	5.24	5.02	289.2	24.99	1200	120.0	6.93
		API	.438	19.124	92	125.1	5.24	5.01	288.0	26.95	1290	129.0	6.92
		X-Stg.	.500	19.000	105	122.8	5.24	4.97	283.5	30.63	1457	145.7	6.90
		40	.594	18.812	123	120.4	5.24	4.93	278.0	36.15	1704	170.4	6.86
		- -	.625	18.750	129	119.5	5.24	4.91	276.1	38.04	1787	178.7	6.85
		60	.812	18.376	167	114.9	5.24	4.81	265.2	48.95	2257	225.7	6.79
		- -	.875	18.250	179	113.2	5.24	4.78	261.6	52.57	2409	240.9	6.77
		- -	1.000	18.000	203	110.3	5.24	4.71	254.5	59.69	2702	270.2	6.73
		80	1.031	17.938	209	109.4	5.24	4.80	252.7	61.44	2771	277.1	6.72
		- -	1.125	17.750	227	107.3	5.24	4.65	247.4	66.71	2981	298.1	6.68
		- -	1.250	17.500	250	104.3	5.24	4.58	240.5	73.63	3249	324.9	6.64
		100	1.281	17.438	256	103.4	5.24	4.56	238.8	75.34	3317	331.7	6.63
		- -	1.375	17.250	274	101.3	5.24	4.52	233.7	80.45	3508	350.8	6.60
		120	1.500	17.000	297	98.3	5.24	4.45	227.0	87.18	3755	375.5	6.56
		140	1.750	16.500	342	92.6	5.24	4.32	213.8	100.33	4217	421.7	6.48
		160	1.969	16.062	379	87.9	5.24	4.21	202.7	111.49	4586	458.6	6.41

A-16.

(Continued). Properties of Pipe

Nominal Pipe Size	Nominal Outside Diam. D	Designation	Wall Thickness	Inside Diam. d	Weight per Foot	Wt. of Water per Ft. of Pipe	Sq. Ft. Outside Surface per Ft.	Sq. Ft. Inside Surface per Ft.	Transverse Area in.² a	Area of Metal in.² A	Moment of Inertia in.⁴ I	Section Modulus in.³ Z	Radius of Gyration in. R
22	22.000	8 Ga.	.164	21.672	38	159.9	5.76	5.67	368.9	11.25	671	61.0	7.72
		6 Ga.	.194	21.612	45	159.0	5.76	5.66	366.8	13.29	790	71.8	7.71
		3 Ga.	.239	21.522	56	157.7	5.76	5.63	363.8	16.34	967	87.9	7.69
		API	.250	21.500	58	157.4	5.76	5.63	363.1	17.18	1010	91.8	7.69
		API	.281	21.438	65	156.5	5.76	5.61	361.0	19.17	1131	102.8	7.68
		API	.312	21.376	72	155.6	5.76	5.60	358.9	21.26	1250	113.6	7.67
		API	.344	21.312	80	154.7	5.76	5.58	356.7	23.40	1373	124.8	7.66
		API	.375	21.250	87	153.7	5.76	5.56	354.7	25.48	1490	135.4	7.65
		API	.406	21.188	94	152.9	5.76	5.55	352.6	27.54	1607	146.1	7.64
		API	.438	21.124	101	151.9	5.76	5.53	350.5	29.67	1725	156.8	7.62
		API	.500	21.000	115	150.2	5.76	5.50	346.4	33.77	1953	177.5	7.61
		- -	.625	20.750	143	146.6	5.76	5.43	338.2	41.97	2400	218.2	7.56
		- -	.750	20.500	170	143.1	5.76	5.37	330.1	50.07	2829	257.2	7.52
		- -	.875	20.250	198	139.6	5.76	5.30	322.1	58.07	3245	295.0	7.47
		- -	1.000	20.000	224	136.2	5.76	5.24	314.2	65.97	3645	331.4	7.43
		- -	1.125	19.750	251	132.8	5.76	5.17	306.4	73.78	4029	366.3	7.39
		- -	1.250	19.500	277	129.5	5.76	5.10	298.6	81.48	4400	400.0	7.35
		- -	1.375	19.250	303	126.2	5.76	5.04	291.0	89.09	4758	432.6	7.31
		- -	1.500	19.000	329	122.9	5.76	4.97	283.5	96.60	5103	463.9	7.27
24	24.000	8 Ga.	.164	23.672	42	190.8	6.28	6.20	440.1	12.28	872	72.7	8.43
		6 Ga.	.194	23.612	49	189.8	6.28	6.18	437.9	14.51	1028	85.7	8.42
		3 Ga.	.239	23.522	61	188.4	6.28	6.16	434.5	17.84	1260	105.0	8.40
		10	.250	23.500	63	189.0	6.28	6.15	435.0	18.67	1320	110.0	8.40
		API	.281	23.438	71	187.0	6.28	6.14	431.5	20.94	1472	122.7	8.38
		API	.312	23.376	79	186.9	6.28	6.12	430.0	23.20	1630	136.0	8.38
		API	.344	23.312	87	185.0	6.28	6.10	426.8	25.57	1789	149.1	8.36
		Std.	.375	23.250	95	183.8	6.28	6.09	424.6	27.83	1942	161.9	8.35
		API	.406	23.188	102	183.1	6.28	6.07	422.3	30.09	2095	174.6	8.34
		API	.438	23.124	110	182.1	6.28	6.05	420.0	32.42	2252	187.7	8.33
		X-Stg.	.500	23.000	125	181.0	6.28	6.02	416.0	36.90	2550	213.0	8.31
		30	.562	22.876	141	178.5	6.28	5.99	411.0	41.40	2840	237.0	8.28
		- -	.625	22.750	156	175.9	6.28	5.96	406.5	45.90	3137	261.4	8.27
		40	.688	22.624	171	174.2	6.28	5.92	402.1	50.30	3422	285.2	8.25
		- -	.750	22.500	186	172.1	6.28	5.89	397.6	54.78	3705	308.8	8.22
		- -	.875	22.250	216	168.6	6.28	5.82	388.8	63.57	4257	354.7	8.18
		60	.969	22.062	238	165.8	6.28	5.78	382.3	70.04	4652	387.7	8.15
		- -	1.000	22.000	246	164.8	6.28	5.76	380.1	72.26	4788	399.0	8.14
		- -	1.125	21.750	275	161.1	6.28	5.69	371.5	80.85	5302	441.8	8.10
		80	1.219	21.562	297	158.2	6.28	5.65	365.2	87.17	5673	472.8	8.07
		- -	1.250	21.500	304	157.4	6.28	5.63	363.1	89.34	5797	483.0	8.05
		- -	1.375	21.250	332	153.8	6.28	5.56	354.7	97.73	6275	522.9	8.01
		- -	1.500	21.000	361	150.2	6.28	5.50	346.4	106.03	6740	561.7	7.97
		100	1.531	20.938	367	149.3	6.28	5.48	344.3	108.07	6847	570.6	7.96
		120	1.812	20.376	429	141.4	6.28	5.33	326.1	126.30	7823	651.9	7.87
		140	2.062	19.876	484	134.4	6.28	5.20	310.3	142.10	8627	718.9	7.79
		160	2.344	19.312	542	127.0	6.28	5.06	293.1	159.40	9457	788.1	7.70
26	26.000	8 Ga.	.164	25.672	45	224.4	6.81	6.72	517.6	13.31	1111	85.4	9.13
		6 Ga.	.194	25.612	54	223.4	6.81	6.70	515.2	15.73	1310	100.7	9.12
		3 Ga.	.239	25.522	66	221.8	6.81	6.68	511.6	19.34	1605	123.4	9.11
		API	.250	25.500	67	221.4	6.81	6.68	510.7	19.85	1646	126.6	9.10
		API	.281	25.438	77	220.3	6.81	6.66	508.2	22.70	1877	144.4	9.09
		API	.312	25.376	84	219.2	6.81	6.64	505.8	25.18	2076	159.7	9.08

A-16.
(Continued). Properties of Pipe

Nominal Pipe Size	Nominal Outside Diam. D	Designation	Wall Thickness	Inside Diam. d	Weight per Foot	Wt. of Water per Ft. of Pipe	Sq. Ft. Outside Surface per Ft.	Sq. Ft. Inside Surface per Ft.	Transverse Area in.² a	Area of Metal in.² A	Moment of Inertia in.⁴ I	Section Modulus in.³ Z	Radius of Gyration in. R
26 cont.	26.000	API	.344	25.312	94	218.2	6.81	6.63	503.2	27.73	2280	175.4	9.07
		API	.375	25.250	103	217.1	6.81	6.61	500.7	30.19	2478	190.6	9.06
		API	.406	25.188	111	216.0	6.81	6.59	498.3	32.64	2673	205.6	9.05
		API	.438	25.124	120	214.9	6.81	6.58	495.8	35.17	2674	221.1	9.04
		API	.500	25.000	136	212.8	6.81	6.54	490.9	40.06	3259	250.7	9.02
		- -	.625	24.750	169	208.6	6.81	6.48	481.1	49.82	4013	308.7	8.98
		- -	.750	24.500	202	204.4	6.81	6.41	471.4	59.49	4744	364.9	8.93
		- -	.875	24.250	235	200.2	6.81	6.35	461.9	69.07	5458	419.9	8.89
		- -	1.000	24.000	267	196.1	6.81	6.28	452.4	78.54	6149	473.0	8.85
		- -	1.125	23.750	299	192.1	6.81	6.22	443.0	87.91	6813	524.1	8.80
		- -	1.375	23.250	362	184.1	6.81	6.09	424.6	106.37	8088	622.2	8.72
		- -	1.500	23.000	393	180.1	6.81	6.02	415.5	115.45	8695	668.8	8.68
30	30.000	8 Ga.	.164	29.672	52	299.9	7.85	7.77	691.4	15.37	1711	114.0	10.55
		6 Ga.	.194	29.612	62	298.6	7.85	7.75	688.6	18.17	2017	134.4	10.53
		3 Ga.	.239	29.522	76	296.7	7.85	7.73	684.4	22.35	2474	165.0	10.52
		API	.250	29.500	79	296.3	7.85	7.72	683.4	23.37	2585	172.3	10.52
		API	.281	29.438	89	295.1	7.85	7.70	680.5	26.24	2897	193.1	10.51
		10	.312	29.376	99	293.7	7.85	7.69	677.8	29.19	3201	213.4	10.50
		API	.344	29.312	109	292.6	7.85	7.67	674.8	32.04	3524	235.0	10.49
		API	.375	29.250	119	291.2	7.85	7.66	672.0	34.90	3823	254.8	10.48
		API	.406	29.188	130	290.7	7.85	7.64	669.0	37.75	4132	275.5	10.46
		API	.438	29.124	138	288.8	7.85	7.62	666.1	40.68	4442	296.2	10.45
		20	.500	29.000	158	286.2	7.85	7.59	660.5	46.34	5033	335.5	10.43
		30	.625	28.750	196	281.3	7.85	7.53	649.2	57.68	6213	414.2	10.39
		- -	.750	28.500	234	276.6	7.85	7.46	637.9	68.92	7371	491.4	10.34
		- -	.875	28.250	272	271.8	7.85	7.39	620.7	80.06	8494	566.2	10.30
		- -	1.000	28.000	310	267.0	7.85	7.33	615.7	91.11	9591	639.4	10.26
		- -	1.125	27.750	347	262.2	7.85	7.26	604.7	102.05	10653	710.2	10.22
		- -	1.250	27.500	384	257.5	7.85	7.20	593.9	112.90	11682	778.8	10.17
		- -	1.375	27.250	421	252.9	7.85	7.13	583.1	123.65	12694	846.2	10.13
		- -	1.500	27.000	457	248.2	7.85	7.07	572.5	134.30	13673	911.5	10.09
32	32.000	API	.250	31.500	85	337.8	8.38	8.25	779.2	24.93	3141	196.3	11.22
		API	.281	31.438	95	336.5	8.38	8.23	776.2	28.04	3525	220.3	11.21
		API	.312	31.376	106	335.2	8.38	8.21	773.2	31.02	3891	243.2	11.20
		API	.344	31.312	116	333.8	8.38	8.20	770.0	34.24	4287	268.0	11.19
		API	.375	31.250	127	332.5	8.38	8.18	766.9	37.25	4656	291.0	11.18
		API	.406	31.188	137	331.2	8.38	8.16	764.0	40.29	5025	314.1	11.17
		API	.438	31.124	148	329.8	8.38	8.15	760.8	43.43	5407	337.9	11.16
		API	.500	31.000	168	327.2	8.38	8.11	754.7	49.48	6140	383.8	11.14
		- -	.625	30.750	209	321.9	8.38	8.05	742.5	61.59	7578	473.6	11.09
		- -	.750	30.500	250	316.7	8.38	7.98	730.5	73.63	8990	561.9	11.05
		- -	.875	30.250	291	311.5	8.38	7.92	718.6	85.53	10368	648.0	11.01
		- -	1.000	30.000	331	306.4	8.38	7.85	706.8	97.38	11680	730.0	10.95
		- -	1.125	29.750	371	301.3	8.38	7.79	695.0	109.0	13003	812.7	10.92
		- -	1.250	29.500	410	296.3	8.38	7.72	680.5	120.7	14398	899.9	10.88
		- -	1.375	29.250	450	291.2	8.38	7.66	671.9	132.2	15526	970.4	10.84
		- -	1.500	29.000	489	286.3	8.38	7.59	660.5	143.7	16752	1047.0	10.80

A-16.
(Concluded). Properties of Pipe

Nominal Pipe Size	Nominal Outside Diam. D	Designation	Wall Thickness	Inside Diam. d	Weight per Foot	Wt. of Water per Ft. of Pipe	Sq. Ft. Outside Surface per Ft.	Sq. Ft. Inside Surface per Ft.	Transverse Area in.² a	Area of Metal in.² A	Moment of Inertia in.⁴ I	Section Modulus in.³ Z	Radius of Gyration in. R
34	34.000	API	.250	33.500	90	382.0	8.90	8.77	881.2	26.50	3773	221.9	11.93
		API	.281	33.438	101	380.7	8.90	8.75	878.2	29.77	4230	248.8	11.92
		API	.312	33.376	112	379.3	8.90	8.74	874.9	32.99	4680	275.3	11.91
		API	.344	33.312	124	377.8	8.90	8.72	871.6	36.36	5147	302.8	11.90
		API	.375	33.250	135	376.2	8.90	8.70	867.8	39.61	5597	329.2	11.89
		API	.406	33.188	146	375.0	8.90	8.69	865.0	42.88	6047	355.7	11.87
		API	.438	33.124	157	373.6	8.90	8.67	861.7	46.18	6501	382.4	11.86
		API	.500	33.000	179	370.8	8.90	8.64	855.3	52.62	7385	434.4	11.85
		- -	.625	32.750	223	365.0	8.90	8.57	841.9	65.53	9124	536.7	11.80
		- -	.750	32.500	266	359.5	8.90	8.51	829.3	78.34	10829	637.0	11.76
		- -	.875	32.250	308	354.1	8.90	8.44	816.8	90.66	12442	731.9	11.71
		- -	1.000	32.000	353	348.6	8.90	8.38	804.2	103.6	14114	830.2	11.67
		- -	1.125	31.750	395	343.2	8.90	8.31	791.6	116.1	15703	923.7	11.63
		- -	1.250	31.500	437	337.8	8.90	8.25	779.2	128.5	17246	1014.5	11.58
		- -	1.375	31.250	479	332.4	8.90	8.18	766.9	140.9	18770	1104.1	11.54
		- -	1.500	31.000	521	327.2	8.90	8.11	754.7	153.1	20247	1191.0	11.50
36	36.000	- -	.164	35.672	63	433.2	9.42	9.34	999.3	18.53	2975	165.3	12.67
		- -	.194	35.612	74	431.8	9.42	9.32	996.0	21.83	3499	194.4	12.66
		- -	.239	35.522	91	429.6	9.42	9.30	991.0	26.86	4293	238.5	12.64
		API	.250	35.500	96	429.1	9.42	9.29	989.7	28.11	4491	249.5	12.64
		API	.281	35.438	107	427.6	9.42	9.28	986.4	31.49	5023	279.1	12.63
		API	.312	35.376	119	426.1	9.42	9.26	982.9	34.95	5565	309.1	12.62
		API	.344	35.312	131	424.6	9.42	9.24	979.3	38.56	6127	340.4	12.60
		API	.375	35.250	143	423.1	9.42	9.23	975.8	42.01	6664	370.2	12.59
		API	.406	35.188	154	421.6	9.42	9.21	972.5	45.40	7191	399.5	12.58
		API	.438	35.124	166	420.1	9.42	9.19	968.9	48.93	7737	429.9	12.57
		API	.500	35.000	190	417.1	9.42	9.16	962.1	55.76	8785	488.1	12.55
		- -	.625	34.750	236	411.1	9.42	9.10	948.3	69.50	10872	604.0	12.51
		- -	.750	34.500	282	405.3	9.42	9.03	934.7	83.01	12898	716.5	12.46
		- -	.875	34.250	329	399.4	9.42	8.97	921.2	96.60	14906	828.1	12.42
		- -	1.000	34.000	374	393.6	9.42	8.90	907.9	109.9	16851	936.2	12.38
		- -	1.125	33.750	419	387.8	9.42	8.83	894.5	123.3	18766	1042.6	12.34
		- -	1.250	33.500	464	382.1	9.42	8.77	881.3	136.5	20624	1145.8	12.29
		- -	1.375	33.250	509	376.4	9.42	8.70	868.2	149.6	22451	1247.3	12.25
		- -	1.500	33.000	553	370.8	9.42	8.64	855.3	162.6	24237	1346.5	12.21
42	42.000	- -	.250	41.500	112	586.4	10.99	10.86	1352.6	32.82	7126	339.3	14.73
		- -	.375	41.250	167	579.3	10.99	10.80	1336.3	49.08	10627	506.1	14.71
		- -	.500	41.000	222	572.3	10.99	10.73	1320.2	65.18	14037	668.4	14.67
		- -	.625	40.750	276	565.4	10.99	10.67	1304.1	81.28	17373	827.3	14.62
		- -	.750	40.500	331	558.4	10.99	10.60	1288.2	97.23	20689	985.2	14.59
		- -	.875	40.250	385	551.6	10.99	10.54	1272.3	113.0	23896	1137.9	14.54
		- -	1.000	40.000	438	544.8	10.99	10.47	1256.6	128.8	27080	1289.5	14.50
		- -	1.125	39.750	492	537.9	10.99	10.41	1240.9	144.5	30193	1437.8	14.45
		- -	1.250	39.500	544	531.2	10.99	10.34	1225.3	160.0	33233	1582.5	14.41
		- -	1.375	39.250	597	524.4	10.99	10.27	1209.9	175.5	36240	1725.7	14.37
		- -	1.500	39.000	649	517.9	10.99	10.21	1194.5	190.8	39181	1865.7	14.33

A-17.
Equation of Pipes

The table below gives the number of pipes of one size required to equal in delivery other larger pipes of same length and under same conditions. The upper portion above the diagonal line of stars pertains to "standard" steam and gas pipes, while the lower portion is for pipes of the ACTUAL internal diameters given. The figures given in the table opposite the intersection of any two sizes is the number of the smaller-sized pipes required to equal one of the larger. Thus, it requires 29 standard 2-inch pipes to equal one standard 7-inch pipe.

STANDARD STEAM AND GAS PIPES

Dia.	½	¾	1	1½	2	2½	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	Dia.
½	* * *	2.27	4.88	15.8	31.7	52.9	96.9	205	377	620	918	1292	1767	2488	3014	3786	4904	5927	7321	8535	9717	½
¾	2.60	* * *	2.05	6.97	11.0	23.3	42.5	90.4	166	273	405	569	779	1096	1328	1668	2161	2615	3226	3761	4282	¾
1	7.55	2.90	* * *	3.45	6.82	11.4	20.9	44.1	81.1	133	198	278	380	536	649	815	1070	1263	1576	1837	2092	1
1½	24.2	9.30	3.20	* * *	1.26	3.34	6.13	13.0	23.8	39.2	58.1	81.7	112	157	190	239	310	375	463	539	614	1½
2	54.8	21.0	7.25	2.26	* * *	1.67	3.06	6.47	11.9	19.6	29.0	40.8	55.8	78.5	95.1	119	155	187	231	269	307	2
2½	102	39.4	13.6	4.23	1.87	* * *	1.83	3.87	7.12	11.7	17.4	24.4	33.4	47.0	56.9	71.5	92.6	112	138	161	184	2½
3	170	65.4	22.6	7.03	3.11	1.66	* * *	2.12	3.89	6.39	9.48	13.3	20.9	23.7	31.2	39.1	50.6	61.1	75.5	88.0	100	3
4	376	144	49.8	15.5	6.87	3.67	2.21	* * *	1.84	3.02	4.48	6.30	8.61	12.1	14.7	18.5	23.9	28.9	35.7	41.6	47.4	4
5	686	263	90.9	28.3	12.5	6.70	4.03	1.83	* * *	1.65	2.44	3.43	4.69	6.60	8.00	10.0	13.0	15.7	19.4	22.6	25.8	5
6	1116	429	148	46.0	20.4	10.9	6.56	2.97	1.63	* * *	1.48	2.09	2.85	4.02	4.86	6.11	7.91	9.56	11.8	13.8	15.6	6
7	1707	656	226	70.5	31.2	16.6	10.0	4.54	2.49	1.51	* * *	1.41	1.93	2.71	3.28	4.12	5.34	6.45	7.97	9.31	10.6	7
8	2435	936	322	101	44.5	23.8	14.3	6.48	3.54	2.18	1.43	* * *	1.35	1.93	2.33	2.92	3.79	4.57	5.67	6.60	7.52	8
9	3335	1281	440	137	60.8	32.5	19.5	8.85	4.85	2.98	1.95	1.37	* * *	1.41	1.71	2.14	2.77	3.35	4.14	4.83	5.50	9
10	4393	1688	582	181	80.4	42.9	25.8	11.7	6.40	3.93	2.57	1.80	1.32	* * *	1.21	1.52	1.97	2.38	2.94	3.43	3.91	10
11	5642	2168	747	233	103	55.1	33.1	15.0	8.22	5.05	3.31	2.32	1.70	1.28	* * *	1.26	1.63	1.88	2.43	2.83	3.22	11
12	7087	2723	938	293	129	69.2	41.6	18.8	10.3	6.34	4.15	2.91	2.13	1.61	1.26	* * *	1.30	1.57	1.93	2.26	2.58	12
13	8657	3326	1146	358	158	84.5	50.7	23.0	12.6	7.75	5.07	3.56	2.60	1.98	1.53	1.22	* * *	1.21	1.49	1.74	1.98	13
14	10600	4070	1403	438	193	103	62.2	28.2	15.4	9.48	6.21	4.35	3.18	2.41	1.88	1.50	1.22	* * *	1.24	1.44	1.64	14
15	12824	4927	1698	530	234	125	75.3	34.1	18.7	11.5	7.52	5.27	3.85	2.92	2.27	1.81	1.48	1.21	* * *	1.17	1.35	15
16	14978	5758	1984	619	274	146	88.0	39.9	21.8	13.4	8.78	6.15	4.51	3.41	2.66	2.12	1.73	1.42	1.18	* * *	1.14	16
17	17537	6738	2322	724	320	171	103	46.6	25.6	15.7	10.3	7.20	5.27	3.99	3.11	2.47	2.03	1.66	1.37	1.17	* * *	17
18	20327	7810	2691	840	317	198	119	54.1	29.6	18.2	11.9	8.35	6.11	4.63	3.60	2.87	2.35	1.92	1.59	1.36	1.16	18
20	26676	10249	3532	1102	487	260	157	70.9	38.9	23.9	15.6	10.9	8.02	6.07	4.73	3.76	3.08	2.52	2.08	1.78	1.52	20
24	42624	16376	5644	1761	778	416	250	113	62.1	38.2	25.0	17.5	12.8	9.70	7.55	6.01	4.92	4.02	3.32	2.84	2.43	24
30	75453	28990	9990	3117	1378	736	443	201	110	67.6	44.2	31.0	22.7	17.2	13.4	10.7	8.72	7.14	5.88	5.03	4.30	30
36	120100	46143	15902	4961	2193	1172	705	319	175	108	70.4	49.3	36.1	27.3	21.3	16.9	13.9	11.3	9.37	8.01	6.85	36
42	177724	68282	23531	7341	3245	1734	1044	473	259	159	104	73.0	53.4	40.5	31.5	25.1	20.5	16.8	13.9	11.9	10.1	42
48	249351	95818	33020	10301	4554	2434	1465	663	363	223	146	102	75.0	56.8	44.2	35.2	28.8	23.5	19.4	16.6	14.2	48
Dia.	½	¾	1	1½	2	2½	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	

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(Continued). Circumferences and Areas of Circles (Advancing by eighths)

Dia.	Circum	Area*
29⅝	93.070	689.30
¾	93.462	695.13
⅞	93.855	700.98
30.	94.248	706.86
⅛	94.640	712.76
¼	95.033	718.69
⅜	95.426	724.64
½	95.819	730.62
⅝	96.211	736.62
¾	96.604	742.64
⅞	96.997	748.69
31.	97.389	754.77
⅛	97.782	760.87
¼	98.175	766.99
⅜	98.567	773.14
½	98.960	779.31
⅝	99.353	785.51
¾	99.746	791.73
⅞	100.138	797.98
32.	100.531	804.25
⅛	100.924	810.54
¼	101.316	816.86
⅜	101.709	823.21
½	102.102	829.58
⅝	102.494	835.97
¾	102.887	842.39
⅞	103.280	848.83
33.	103.673	855.30
⅛	104.065	861.79
¼	104.458	868.31
⅜	104.851	874.85
½	105.243	881.41
⅝	105.636	888.00
¾	106.029	894.62
⅞	106.421	901.26
34.	106.814	907.92
⅛	107.207	914.61
¼	107.600	921.32
⅜	107.992	928.06
½	108.385	934.82
⅝	108.778	941.61
¾	109.170	948.42
⅞	109.563	955.25
35.	109.956	962.11
⅛	110.348	969.00
¼	110.741	975.91
⅜	111.134	982.84
½	111.527	989.80
⅝	111.919	996.78
¾	112.312	1003.8
⅞	112.705	1010.8

Dia.	Circum.	Area*
36.	113.097	1017.9
⅛	113.490	1025.0
¼	113.883	1032.1
⅜	114.275	1039.2
½	114.668	1046.3
⅝	115.061	1053.5
¾	115.454	1060.7
⅞	115.846	1068.0
37.	116.239	1075.2
⅛	116.632	1082.5
¼	117.024	1089.8
⅜	117.417	1097.1
½	117.810	1104.5
⅝	118.202	1111.8
¾	118.596	1119.2
⅞	118.988	1126.7
38.	119.381	1134.1
⅛	119.773	1141.6
¼	120.166	1149.1
⅜	120.559	1156.6
½	120.951	1164.2
⅝	121.344	1171.7
¾	121.737	1179.3
⅞	122.129	1186.9
39.	122.522	1194.6
⅛	122.915	1202.3
¼	123.308	1210.6
⅜	123.700	1217.7
½	124.093	1225.4
⅝	124.486	1233.2
¾	124.878	1241.0
⅞	125.271	1248.8
40.	125.664	1256.6
⅛	126.056	1264.5
¼	126.449	1272.4
⅜	126.842	1280.3
½	127.235	1288.2
⅝	127.627	1296.2
¾	128.020	1304.2
⅞	128.413	1312.2
41.	128.805	1320.3
⅛	129.198	1328.3
¼	129.591	1336.4
⅜	129.983	1344.5
½	130.376	1352.7
⅝	130.769	1360.8
¾	131.161	1369.0
⅞	131.554	1377.2
42.	131.947	1385.4
⅛	132.340	1393.7

Dia.	Circum.	Area*
42¼	132.732	1402.0
⅜	133.125	1410.3
½	133.518	1418.6
⅝	133.910	1427.0
¾	134.303	1435.4
⅞	134.696	1443.8
43.	135.088	1452.2
⅛	135.481	1460.7
¼	135.874	1469.1
⅜	136.267	1477.6
½	136.659	1486.2
⅝	137.052	1494.7
¾	137.445	1503.3
⅞	137.837	1511.9
44.	138.230	1520.5
⅛	138.623	1529.2
¼	139.015	1537.9
⅜	139.408	1546.6
½	139.801	1555.3
⅝	140.194	1564.0
¾	140.586	1572.8
⅞	140.979	1581.6
45.	141.372	1590.4
⅛	141.764	1599.3
¼	142.157	1608.2
⅜	142.550	1617.0
½	142.942	1626.0
⅝	143.335	1634.9
¾	143.728	1643.9
⅞	144.121	1652.9
46.	144.513	1661.9
⅛	144.906	1670.9
¼	145.299	1680.0
⅜	145.691	1689.1
½	146.084	1698.2
⅝	146.477	1707.4
¾	146.869	1716.5
⅞	147.262	1725.7
47.	147.655	1734.9
⅛	148.048	1744.2
¼	148.440	1753.5
⅜	148.833	1762.7
½	149.226	1772.1
⅝	149.618	1781.4
¾	150.011	1790.8
⅞	150.404	1800.1
48.	150.796	1809.6
⅛	151.189	1819.0
¼	151.582	1828.5
⅜	151.975	1837.9
½	152.367	1847.5

*Approximate area sufficiently accurate for practical purposes, including estimating.

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(Continued). Circumferences and Areas of Circles (Advancing by eighths)

Dia.	Circum.	Area*
48⅝	152.760	1857.0
¾	153.153	1866.5
⅞	153.545	1876.1
49.	153.938	1885.7
⅛	154.331	1895.4
¼	154.723	1905.0
⅜	155.116	1914.7
½	155.509	1924.4
⅝	155.902	1934.2
¾	156.294	1943.9
⅞	156.687	1953.7
50.	157.080	1963.5
⅛	157.472	1973.3
¼	157.865	1983.2
⅜	158.258	1993.1
½	158.650	2003.0
⅝	159.043	2012.9
¾	159.436	2022.8
⅞	159.829	2032.8
51.	160.221	2042.8
⅛	160.614	2052.8
¼	161.007	2062.9
⅜	161.399	2073.0
½	161.792	2083.1
⅝	162.185	2093.2
¾	162.577	2103.3
⅞	162.970	2113.5
52.	163.363	2123.7
⅛	163.756	2133.9
¼	164.148	2144.2
⅜	164.541	2154.5
½	164.934	2164.8
⅝	165.326	2175.1
¾	165.719	2185.4
⅞	166.112	2195.8
53.	166.504	2206.2
⅛	166.897	2216.6
¼	167.290	2227.0
⅜	167.683	2237.5
½	168.075	2248.0
⅝	168.468	2258.5
¾	168.861	2269.1
⅞	169.253	2279.6
54.	169.646	2290.2
⅛	170.039	2300.8
¼	170.431	2311.5
⅜	170.824	2322.1
½	171.217	2332.8
⅝	171.609	2343.5
¾	172.002	2354.3
⅞	172.395	2365.0

Dia.	Circum.	Area*
55.	172.788	2375.8
⅛	173.180	2386.6
¼	173.573	2397.5
⅜	173.966	2408.3
½	174.358	2419.2
⅝	174.751	2430.1
¾	175.144	2441.1
⅞	175.536	2452.0
56.	175.929	2463.0
⅛	176.322	2474.0
¼	176.715	2485.0
⅜	177.107	2496.1
½	177.500	2507.2
⅝	177.893	2518.3
¾	178.285	2529.4
⅞	178.678	2540.6
57.	179.071	2551.8
⅛	179.463	2563.0
¼	179.856	2574.2
⅜	180.249	2585.4
½	180.642	2596.7
⅝	181.034	2608.0
¾	181.427	2619.4
⅞	181.820	2630.7
58.	182.212	2642.1
⅛	182.605	2653.5
¼	182.998	2664.9
⅜	183.390	2676.4
½	183.783	2687.8
⅝	184.176	2699.3
¾	184.569	2710.9
⅞	184.961	2722.4
59.	185.354	2734.0
⅛	185.747	2745.6
¼	186.139	2757.2
⅜	186.532	2768.8
½	186.925	2780.5
⅝	187.317	2792.2
¾	187.710	2803.9
⅞	188.103	2815.7
60.	188.496	2827.4
⅛	188.888	2839.2
¼	189.281	2851.0
⅜	189.674	2862.9
½	190.066	2874.8
⅝	190.459	2886.6
¾	190.852	2898.6
⅞	191.244	2910.5
61.	191.637	2922.5
⅛	192.030	2934.5

Dia.	Circum.	Area*
61¼	192.423	2946.5
⅜	192.815	2958.5
½	193.208	2970.6
⅝	193.601	2982.7
¾	193.993	2994.8
⅞	194.386	3006.9
62.	194.779	3019.1
⅛	195.171	3031.3
¼	195.564	3043.5
⅜	195.957	3055.7
½	196.350	3068.0
⅝	196.742	3080.3
¾	197.135	3092.6
⅞	197.528	3104.9
63.	197.920	3117.2
⅛	198.313	3129.6
¼	198.706	3142.0
⅜	199.098	3154.5
½	199.491	3166.9
⅝	199.884	3179.4
¾	200.277	3191.9
⅞	200.669	3204.4
64.	201.062	3217.0
⅛	201.455	3229.6
¼	201.847	3242.2
⅜	202.240	3254.8
½	202.633	3267.5
⅝	203.025	3280.1
¾	203.418	3292.8
⅞	203.811	3305.6
65.	204.204	3318.3
⅛	204.596	3331.1
¼	204.989	3343.9
⅜	205.382	3356.7
½	205.774	3369.6
⅝	206.167	3382.4
¾	206.560	3395.3
⅞	206.952	3408.2
66.	207.345	3421.2
⅛	207.738	3434.2
¼	208.131	3447.2
⅜	208.523	3460.2
½	208.916	3473.2
⅝	209.309	3486.3
¾	209.701	3499.4
⅞	210.094	3512.5
67.	210.487	3525.7
⅛	210.879	3538.8
¼	211.272	3552.0
⅜	211.665	3565.2

*Approximate area, sufficiently accurate for practical purposes, including estimating.

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Circumferences and Areas of Circles
(Advancing by eighths)

Dia.	Circum.	Area*
1/64	.04909	.00019
1/32	.09818	.00077
3/64	.14726	.00173
1/16	.19635	.00307
3/32	.29452	.00690
1/8	.39270	.01227
5/32	.49087	.01917
3/16	.58905	.02761
7/32	.68722	.03758
1/4	.78540	.04909
9/32	.88357	.06213
5/16	.98175	.07670
11/32	1.0799	.09281
3/8	1.1781	.11045
13/32	1.2763	.12962
7/16	1.3744	.15033
15/32	1.4726	.17257
1/2	1.5708	.19635
17/32	1.6690	.22166
9/16	1.7671	.24850
19/32	1.8653	.27688
5/8	1.9635	.30680
21/32	2.0617	.33824
11/16	2.1598	.37122
23/32	2.2580	.40574
3/4	2.3562	.44179
25/32	2.4544	.47937
13/16	2.5525	.51849
27/32	2.6507	.55914
7/8	2.7489	.60132
29/32	2.8471	.64504
15/16	2.9452	.69029
31/32	3.0434	.73708
1.	3.1416	.7854
1/16	3.3379	.8866
1/8	3.5343	.9940
3/16	3.7306	1.1075
1/4	3.9270	1.2272
5/16	4.1233	1.3530
3/8	4.3197	1.4849
7/16	4.5160	1.6230
1/2	4.7124	1.7671
9/16	4.9087	1.9175
5/8	5.1051	2.0739
11/16	5.3014	2.2365
3/4	5.4978	2.4053
13/16	5.6941	2.5802
7/8	5.8905	2.7612
15/16	6.0868	2.9483
2.	6.2832	3.1416
1/16	6.4795	3.3410

Dia.	Circum.	Area*
2 1/8	6.6759	3.5466
3/16	6.8722	3.7583
1/4	7.0686	3.9761
5/16	7.2649	4.2000
3/8	7.4613	4.4301
7/16	7.6576	4.6664
1/2	7.8540	4.9087
9/16	8.0503	5.1572
5/8	8.2467	5.4119
11/16	8.4430	5.6727
3/4	8.6394	5.9396
13/16	8.8357	6.2126
7/8	9.0321	6.4918
15/16	9.2284	6.7771
3.	9.4248	7.0686
1/16	9.6211	7.3662
1/8	9.8175	7.6699
3/16	10.014	7.9798
1/4	10.210	8.2958
5/16	10.407	8.6179
3/8	10.603	8.9462
7/16	10.799	9.2806
1/2	10.996	9.6211
9/16	11.192	9.9678
5/8	11.388	10.321
11/16	11.585	10.680
3/4	11.781	11.045
13/16	11.977	11.416
7/8	12.174	11.793
15/16	12.370	12.177
4.	12.566	12.566
1/16	12.763	12.962
1/8	12.959	13.364
3/16	13.155	13.772
1/4	13.352	14.186
5/16	13.548	14.607
3/8	13.744	15.033
7/16	13.941	15.466
1/2	14.137	15.904
9/16	14.334	16.349
5/8	14.530	16.800
11/16	14.726	17.257
3/4	14.923	17.728
13/16	15.119	18.190
7/8	15.315	18.665
15/16	15.512	19.147
5.	15.708	19.635
1/16	15.904	20.129
1/8	16.101	20.629
3/16	16.297	21.135
1/4	16.493	21.648
5/16	16.690	22.166

Dia.	Circum.	Area*
5 3/8	16.886	22.691
7/16	17.082	23.221
1/2	17.279	23.758
9/16	17.475	24.301
5/8	17.671	24.850
11/16	17.868	25.406
3/4	18.064	25.967
13/16	18.261	26.535
7/8	18.457	27.109
15/16	18.653	27.688
6.	18.850	28.274
1/8	19.242	29.465
1/4	19.635	30.680
3/8	20.028	31.919
1/2	20.420	33.183
5/8	20.813	34.472
3/4	21.206	35.785
7/8	21.598	37.122
7.	21.991	38.485
1/8	22.384	39.871
1/4	22.776	41.282
3/8	23.169	42.718
1/2	23.562	44.179
5/8	23.955	45.664
3/4	24.347	47.173
7/8	24.740	48.707
8.	25.133	50.265
1/8	25.525	51.849
1/4	25.918	53.456
3/8	26.311	55.088
1/2	26.704	56.745
5/8	27.096	58.426
3/4	27.489	60.132
7/8	27.882	61.862
9.	28.274	63.617
1/8	28.667	65.397
1/4	29.060	67.201
3/8	29.452	69.029
1/2	29.845	70.882
5/8	30.238	72.760
3/4	30.631	74.662
7/8	31.023	76.589
10.	31.416	78.540
1/8	31.809	80.516
1/4	32.201	82.516
3/8	32.594	84.541
1/2	32.987	86.590
5/8	33.379	88.664
3/4	33.772	90.763
7/8	34.165	92.886

*Approximate area, sufficiently accurate for practical purposes, including estimating.

By permission, Buffalo Tank Div., Bethlehem Steel Corp.

A-18.

(Continued). Circumferences and Areas of Circles
(Advancing by eighths)

Dia.	Circum.	Area*
11.	34.558	95.033
1/8	34.950	97.205
1/4	35.343	99.402
3/8	35.736	101.62
1/2	36.128	103.87
5/8	36.521	106.14
3/4	36.914	108.43
7/8	37.306	110.75
12.	37.699	113.10
1/8	38.092	115.47
1/4	38.485	117.86
3/8	38.877	120.28
1/2	39.270	122.72
5/8	39.663	125.19
3/4	40.055	127.68
7/8	40.448	130.19
13.	40.841	132.73
1/8	41.233	135.30
1/4	41.626	137.89
3/8	42.019	140.50
1/2	42.412	143.14
5/8	42.804	145.80
3/4	43.197	148.49
7/8	43.590	151.20
14.	43.982	153.94
1/8	44.375	156.70
1/4	44.768	159.48
3/8	45.160	162.30
1/2	45.553	165.13
5/8	45.946	167.99
3/4	46.338	170.87
7/8	46.731	173.78
15.	47.124	176.71
1/8	47.517	179.67
1/4	47.909	182.65
3/8	48.302	185.66
1/2	48.695	188.69
5/8	49.087	191.75
3/4	49.480	194.83
7/8	49.873	197.93
16.	50.265	201.06
1/8	50.658	204.22
1/4	51.051	207.39
3/8	51.444	210.60
1/2	51.836	213.82
5/8	52.229	217.08
3/4	52.622	220.35
7/8	53.014	223.65
17.	53.407	226.98
1/8	53.800	230.33

Dia.	Circum.	Area*
17 1/4	54.192	233.71
3/8	54.585	237.10
1/2	54.978	240.53
5/8	55.371	243.98
3/4	55.763	247.45
7/8	56.156	250.95
18.	56.549	254.47
1/8	56.941	258.02
1/4	57.334	261.59
3/8	57.727	265.18
1/2	58.119	268.80
5/8	58.512	272.45
3/4	58.905	276.12
7/8	59.298	279.81
19.	59.690	283.53
1/8	60.083	287.27
1/4	60.476	291.04
3/8	60.868	294.83
1/2	61.261	298.65
5/8	61.654	302.49
3/4	62.046	306.35
7/8	62.439	310.24
20.	62.832	314.16
1/8	63.225	318.10
1/4	63.617	322.06
3/8	64.010	326.05
1/2	64.403	330.06
5/8	64.795	334.10
3/4	65.188	338.16
7/8	65.581	342.25
21.	65.973	346.36
1/8	66.366	350.50
1/4	66.759	354.66
3/8	67.152	358.84
1/2	67.544	363.05
5/8	67.937	367.28
3/4	68.330	371.54
7/8	68.722	375.83
22.	69.115	380.13
1/8	69.508	384.46
1/4	69.900	388.82
3/8	70.293	393.20
1/2	70.686	397.61
5/8	71.079	402.04
3/4	71.471	406.49
7/8	71.864	410.97
23.	72.257	415.48
1/8	72.649	420.00
1/4	73.042	424.56
3/8	73.435	429.13

Dia.	Circum.	Area*
23 1/2	73.827	433.74
5/8	74.220	438.36
3/4	74.613	443.01
7/8	75.006	447.69
24.	75.398	452.39
1/8	75.791	457.11
1/4	76.184	461.86
3/8	76.576	466.64
1/2	76.969	471.44
5/8	77.362	476.26
3/4	77.754	481.11
7/8	78.147	485.98
25.	78.540	490.87
1/8	78.933	495.79
1/4	79.325	500.74
3/8	79.718	505.71
1/2	80.111	510.71
5/8	80.503	515.72
3/4	80.896	520.77
7/8	81.289	525.84
26.	81.681	530.93
1/8	82.074	536.05
1/4	82.467	541.19
3/8	82.860	546.35
1/2	83.252	551.55
5/8	83.645	556.76
3/4	84.038	562.00
7/8	84.430	567.27
27.	84.823	572.56
1/8	85.216	577.87
1/4	85.608	583.21
3/8	86.001	588.57
1/2	86.394	593.96
5/8	86.786	599.37
3/4	87.179	604.81
7/8	87.572	610.27
28.	87.965	615.75
1/8	88.357	621.26
1/4	88.750	626.80
3/8	89.143	632.36
1/2	89.535	637.94
5/8	89.928	643.55
3/4	90.321	649.18
7/8	90.713	654.84
29.	91.106	660.52
1/8	91.499	666.23
1/4	91.892	671.96
3/8	92.284	677.71
1/2	92.677	683.49

*Approximate area, sufficiently accurate for practical purposes, including estimating.

A-18.
(Continued). Circumferences and Areas of Circles (Advancing by eighths)

Dia.	Circum.	Area*
67½	212.058	3578.5
⅝	212.450	3591.7
¾	212.843	3605.0
⅞	213.236	3618.3
68.	213.628	3631.7
⅛	214.021	3645.0
¼	214.414	3658.4
⅜	214.806	3671.8
½	215.199	3685.3
⅝	215.592	3698.7
¾	215.984	3712.2
⅞	216.377	3725.7
69.	216.770	3739.3
⅛	217.163	3752.8
¼	217.555	3766.4
⅜	217.948	3780.0
½	218.341	3793.7
⅝	218.733	3807.3
¾	219.126	3821.0
⅞	219.519	3834.7
70.	219.911	3848.5
⅛	220.304	3862.2
¼	220.697	3876.0
⅜	221.090	3889.8
½	221.482	3903.6
⅝	221.875	3917.5
¾	222.268	3931.4
⅞	222.660	3945.3
71.	223.053	3959.2
⅛	223.446	3973.1
¼	223.838	3987.1
⅜	224.231	4001.1
½	224.624	4015.2
⅝	225.017	4029.2
¾	225.409	4043.3
⅞	225.802	4057.4
72.	226.195	4071.5
⅛	226.587	4085.7
¼	226.980	4099.8
⅜	227.373	4114.0
½	227.765	4128.2
⅝	228.158	4142.5
¾	228.551	4156.8
⅞	228.944	4171.1
73.	229.336	4185.4
⅛	229.729	4199.7
¼	230.122	4214.1
⅜	230.514	4228.5
½	230.907	4242.9
⅝	231.300	4257.4
73¾	231.692	4271.8
⅞	232.085	4286.3
74.	232.478	4300.8
⅛	232.871	4315.4
¼	233.263	4329.9
⅜	233.656	4344.5
½	234.049	4359.2
⅝	234.441	4373.8
¾	234.834	4388.5
⅞	235.227	4403.1
75.	235.619	4417.9
⅛	236.012	4432.6
¼	236.405	4447.4
⅜	236.798	4462.2
½	237.190	4477.0
⅝	237.583	4491.8
¾	237.976	4506.7
⅞	238.368	4521.5
76.	238.761	4536.5
⅛	239.154	4551.4
¼	239.546	4566.4
⅜	239.939	4581.3
½	240.332	4596.3
⅝	240.725	4611.4
¾	241.117	4626.4
⅞	241.510	4641.5
77.	241.903	4656.6
⅛	242.295	4671.8
¼	242.688	4686.9
⅜	243.081	4702.1
½	243.473	4717.3
⅝	243.866	4732.5
¾	244.259	4747.8
⅞	244.652	4763.1
78.	245.044	4778.4
⅛	245.437	4793.7
¼	245.830	4809.0
⅜	246.222	4824.4
½	246.615	4839.8
⅝	247.008	4855.2
¾	247.400	4870.7
⅞	247.793	4886.2
79.	248.186	4901.7
⅛	248.579	4917.2
¼	248.971	4932.7
⅜	249.364	4948.3
½	249.757	4963.9
⅝	250.149	4979.5
¾	250.542	4995.2
⅞	250.935	5010.9
80.	251.327	5026.5
⅛	251.720	5042.3
¼	252.113	5058.0
⅜	252.506	5073.8
½	252.898	5089.6
⅝	253.291	5105.4
¾	253.684	5121.2
⅞	254.076	5137.1
81.	254.469	5153.0
⅛	254.862	5168.9
¼	255.254	5184.9
⅜	255.647	5200.8
½	256.040	5216.8
⅝	256.433	5232.8
¾	256.825	5248.9
⅞	257.218	5264.9
82.	257.611	5281.0
⅛	258.003	5297.1
¼	258.396	5313.3
⅜	258.789	5329.4
½	259.181	5345.6
⅝	259.574	5361.8
¾	259.967	5378.1
⅞	260.359	5394.3
83.	260.752	5410.6
⅛	261.145	5426.9
¼	261.538	5443.3
⅜	261.930	5459.6
½	262.323	5476.0
⅝	262.716	5492.4
¾	263.108	5508.8
⅞	263.501	5525.3
84.	263.894	5541.8
⅛	264.286	5558.3
¼	264.679	5574.8
⅜	265.072	5591.4
½	265.465	5607.9
⅝	265.857	5624.5
¾	266.250	5641.2
⅞	266.643	5657.8
85.	267.035	5674.5
⅛	267.428	5691.2
¼	267.821	5707.9
⅜	268.213	5724.7
½	268.606	5741.5
⅝	268.999	5758.3
¾	269.392	5775.1
⅞	269.784	5791.9
86.	270.177	5808.8
⅛	270.570	5825.7

*Approximate area, sufficiently accurate for practical purposes, including estimating.

A-18.
(Continued). Circumferences and Areas of Circles (Advancing by eighths)

Dia.	Circum.	Area*
86¼	270.962	5842.6
⅜	271.355	5859.6
½	271.748	5876.5
⅝	272.140	5893.5
¾	272.533	5910.6
⅞	272.926	5927.6
87.	273.319	5944.7
⅛	273.711	5961.8
¼	274.104	5978.9
⅜	274.497	5996.0
½	274.889	6013.2
⅝	275.282	6030.4
¾	275.675	6047.6
⅞	276.067	6064.9
88.	276.460	6082.1
⅛	276.853	6099.4
¼	277.246	6116.7
⅜	277.638	6134.1
½	278.031	6151.4
⅝	278.424	6168.8
¾	278.816	6186.2
⅞	279.209	6203.7
89.	279.602	6221.1
⅛	279.994	6238.6
¼	280.387	6256.1
⅜	280.780	6273.7
½	281.173	6291.2
⅝	281.565	6308.8
¾	281.958	6326.4
⅞	282.351	6344.1
90.	282.743	6361.7
⅛	283.136	6379.4
¼	283.529	6397.1
⅜	283.921	6414.9
½	284.314	6432.6
⅝	284.707	6450.4
¾	285.100	6468.2
⅞	285.492	6486.0
91.	285.885	6503.9
⅛	286.278	6521.8
¼	286.670	6539.7
⅜	287.063	6557.6
½	287.456	6575.5
⅝	287.848	6593.5
¾	288.241	6611.5
⅞	288.634	6629.6
92.	289.027	6647.6
⅛	289.419	6665.7
¼	289.812	6683.8
⅜	290.205	6701.9
92½	290.597	6720.1
⅝	290.990	6738.2
¾	291.383	6756.4
⅞	291.775	6774.7
93.	292.168	6792.9
⅛	292.561	6811.2
¼	292.954	6829.5
⅜	293.346	6847.8
½	293.739	6866.1
⅝	294.132	6884.5
¾	294.524	6902.9
⅞	294.917	6921.3
94.	295.310	6939.8
⅛	295.702	6958.2
¼	296.095	6976.7
⅜	296.488	6995.3
½	296.881	7013.8
⅝	297.273	7032.4
¾	297.666	7051.0
⅞	298.059	7069.6
95.	298.451	7088.2
⅛	298.844	7106.9
¼	299.237	7125.6
⅜	299.629	7144.3
½	300.022	7163.0
⅝	300.415	7181.8
¾	300.807	7200.6
⅞	301.200	7219.4
96.	301.593	7238.2
⅛	301.986	7257.1
¼	302.378	7276.0
⅜	302.771	7294.9
½	303.164	7313.8
⅝	303.556	7332.8
¾	303.949	7351.8
⅞	304.342	7370.8
97.	304.734	7389.8
⅛	305.127	7408.9
¼	305.520	7428.0
⅜	305.913	7447.1
½	306.305	7466.2
⅝	306.698	7485.3
¾	307.091	7504.5
⅞	307.483	7523.7
98.	307.876	7543.0
⅛	308.269	7562.2
¼	308.661	7581.5
⅜	309.054	7600.8
½	309.447	7620.1
⅝	309.840	7639.5
98¾	310.232	7658.9
⅞	310.625	7678.3
99.	311.018	7697.7
⅛	311.410	7717.1
¼	311.803	7736.6
⅜	312.196	7756.1
½	312.588	7775.6
⅝	312.981	7795.2
¾	313.374	7814.8
⅞	313.767	7834.4
100.	314.16	7854
⅛	314.55	7873
¼	314.95	7893
⅜	315.34	7913
½	315.73	7933
⅝	316.12	7952
¾	316.52	7972
⅞	316.91	7992
101.	317.30	8012
⅛	317.69	8032
¼	318.09	8052
⅜	318.48	8071
½	318.87	8091
⅝	319.27	8111
¾	319.66	8131
⅞	320.05	8151
102.	320.44	8171
⅛	320.84	8191
¼	321.23	8211
⅜	321.62	8231
½	322.01	8252
⅝	322.41	8272
¾	322.80	8292
⅞	323.19	8312
103.	323.59	8332
⅛	323.98	8352
¼	324.37	8372
⅜	324.76	8393
½	325.16	8413
⅝	325.55	8434
¾	325.94	8454
⅞	326.33	8474
104.	326.73	8495
⅛	327.12	8515
¼	327.51	8536
⅜	327.91	8556
½	328.30	8577
⅝	328.69	8597
¾	329.08	8618
⅞	329.48	8638

*Approximate area, sufficiently accurate for practical purposes, including estimating.

A-18.
(Continued). Circumferences and Areas of Circles
(Advancing by eighths)

Dia.	Circum.	Area*
105.	329.87	8659
⅛	330.26	8679
¼	330.65	8700
⅜	331.05	8721
½	331.44	8741
⅝	331.83	8762
¾	332.22	8783
⅞	332.62	8804
106.	333.01	8825
⅛	333.40	8845
¼	333.80	8866
⅜	334.19	8887
½	334.58	8908
⅝	334.97	8929
¾	335.37	8950
⅞	335.76	8971
107.	336.15	8992
⅛	336.54	9014
¼	336.94	9035
⅜	337.33	9056
½	337.72	9077
⅝	338.12	9098
¾	338.51	9119
⅞	338.90	9140
108.	339.29	9161
⅛	339.69	9183
¼	340.08	9204
⅜	340.47	9225
½	340.86	9246
⅝	341.26	9268
¾	341.65	9289
⅞	342.04	9310
109.	342.43	9331
⅛	342.83	9353
¼	343.22	9374
⅜	343.61	9396
½	344.01	9417
⅝	344.40	9439
¾	344.79	9460
⅞	345.18	9481
110.	345.58	9503
⅛	345.97	9525
¼	346.36	9546
⅜	346.75	9568
½	347.15	9589
⅝	347.54	9611
¾	347.93	9633
⅞	348.33	9655
111.	348.72	9677
⅛	349.11	9698

Dia.	Circum.	Area*
111¼	349.50	9720
⅜	349.90	9742
½	350.29	9764
⅝	350.68	9786
¾	351.07	9808
⅞	351.47	9830
112.	351.86	9852
⅛	352.25	9874
¼	352.65	9897
⅜	353.04	9919
½	353.43	9941
⅝	353.82	9963
¾	354.22	9985
⅞	354.61	10007
113.	355.00	10029
⅛	355.39	10052
¼	355.79	10074
⅜	356.18	10097
½	356.57	10119
⅝	356.96	10141
¾	357.36	10163
⅞	357.75	10185
114.	358.14	10207
⅛	358.54	10230
¼	358.93	10252
⅜	359.32	10275
½	359.71	10297
⅝	360.11	10320
¾	360.50	10342
⅞	360.89	10365
115.	361.28	10387
⅛	361.68	10410
¼	362.07	10432
⅜	362.46	10455
½	362.86	10477
⅝	363.25	10500
¾	363.64	10522
⅞	364.03	10545
116.	364.43	10568
⅛	364.82	10590
¼	365.21	10613
⅜	365.60	10636
½	366.00	10659
⅝	366.39	10682
¾	366.78	10705
⅞	367.18	10728
117.	367.57	10751
⅛	367.96	10774
¼	368.35	10798
⅜	368.75	10821

Dia.	Circum.	Area*
117½	369.14	10844
⅝	369.53	10867
¾	369.92	10890
⅞	370.32	10913
118.	370.71	10936
⅛	371.11	10960
¼	371.49	10983
⅜	371.89	11007
½	372.28	11030
⅝	372.67	11053
¾	373.07	11076
⅞	373.46	11099
119.	373.85	11122
⅛	374.24	11146
¼	374.64	11169
⅜	375.03	11193
½	375.42	11216
⅝	375.81	11240
¾	376.21	11263
⅞	376.60	11287
120.	376.99	11310
⅛	377.39	11334
¼	377.78	11357
⅜	378.17	11381
½	378.56	11404
⅝	378.96	11428
¾	379.35	11451
⅞	379.74	11475
121.	380.13	11499
⅛	380.53	11522
¼	380.92	11546
⅜	381.31	11570
½	381.70	11594
⅝	382.10	11618
¾	382.49	11642
⅞	382.88	11666
122.	383.28	11690
⅛	383.67	11714
¼	384.06	11738
⅜	384.45	11762
½	384.85	11786
⅝	385.24	11810
¾	385.63	11834
⅞	386.02	11858
123.	386.42	11882
⅛	386.81	11907
¼	387.20	11931
⅜	387.60	11956
½	387.99	11980

*Approximate area, sufficiently accurate for practical purposes, including estimating.

A-18.
(Continued). Circumferences and Areas of Circles
(Advancing by eighths)

Dia.	Circum.	Area*
123⅝	388.38	12004
¾	388.77	12028
⅞	389.17	12052
124.	389.56	12076
⅛	389.95	12101
¼	390.34	12125
⅜	390.74	12150
½	391.13	12174
⅝	391.52	12199
¾	391.92	12223
⅞	392.31	12248
125.	392.70	12272
⅛	393.09	12297
¼	393.49	12321
⅜	393.88	12346
½	394.27	12370
⅝	394.66	12395
¾	395.06	12419
⅞	395.45	12444
126.	395.84	12469
⅛	396.23	12494
¼	396.63	12518
⅜	397.02	12543
½	397.41	12568
⅝	397.81	12593
¾	398.20	12618
⅞	398.59	12643
127.	398.98	12668
⅛	399.38	12693
¼	399.77	12718
⅜	400.16	12743
½	400.55	12768
⅝	400.95	12793
¾	401.34	12818
⅞	401.73	12843
128.	402.13	12868
⅛	402.52	12893
¼	402.91	12919
⅜	403.30	12944
½	403.70	12970
⅝	404.09	12995
¾	404.48	13020
⅞	404.87	13045
129.	405.27	13070
⅛	405.66	13096
¼	406.05	13121
⅜	406.44	13147
½	406.84	13172
⅝	407.23	13198
¾	407.62	13223
⅞	408.02	13248

Dia.	Circum.	Area*
130.	408.41	13273
⅛	408.80	13299
¼	409.19	13324
⅜	409.59	13350
½	409.98	13375
⅝	410.37	13401
¾	410.76	13426
⅞	411.16	13452
131.	411.55	13478
⅛	411.94	13504
¼	412.34	13529
⅜	412.73	13555
½	413.12	13581
⅝	413.51	13607
¾	413.91	13633
⅞	414.30	13659
132.	414.69	13685
⅛	415.08	13711
¼	415.48	13737
⅜	415.87	13763
½	416.26	13789
⅝	416.66	13815
¾	417.05	13841
⅞	417.44	13867
133.	417.83	13893
⅛	418.23	13919
¼	418.62	13946
⅜	419.01	13972
½	419.40	13999
⅝	419.80	14025
¾	420.19	14051
⅞	420.58	14077
134.	420.97	14103
⅛	421.37	14130
¼	421.76	14156
⅜	422.15	14183
½	422.55	14209
⅝	422.94	14236
¾	423.33	14262
⅞	423.72	14288
135.	424.12	14314
⅛	424.51	14341
¼	424.90	14367
⅜	425.29	14394
½	425.69	14420
⅝	426.08	14447
¾	426.47	14473
⅞	426.87	14500
136.	427.26	14527
⅛	427.65	14553

Dia.	Circum.	Area*
136¼	428.04	14580
⅜	428.44	14607
½	428.83	14633
⅝	429.22	14660
¾	429.61	14687
⅞	430.01	14714
137.	430.40	14741
⅛	430.79	14768
¼	431.19	14795
⅜	431.58	14822
½	431.97	14849
⅝	432.36	14876
¾	432.76	14903
⅞	433.15	14930
138.	433.54	14957
⅛	433.93	14984
¼	434.33	15012
⅜	434.72	15039
½	435.11	15067
⅝	435.50	15094
¾	435.90	15121
⅞	436.29	15148
139.	436.68	15175
⅛	437.08	15203
¼	437.47	15230
⅜	437.86	15258
½	438.25	15285
⅝	438.65	15313
¾	439.04	15340
⅞	439.43	15367
140.	439.82	15394
⅛	440.22	15422
¼	440.61	15449
⅜	441.00	15477
½	441.40	15504
⅝	441.79	15532
¾	442.18	15559
⅞	442.57	15587
141.	442.97	15615
⅛	443.36	15642
¼	443.75	15670
⅜	444.14	15697
½	444.54	15725
⅝	444.93	15753
¾	445.32	15781
⅞	445.72	15809
142.	446.11	15837
⅛	446.50	15865
¼	446.89	15893
⅜	447.29	15921

*Approximate area, sufficiently accurate for practical purposes, including estimating.

A-18.
(Continued). Circumferences and Areas of Circles
(Advancing by eighths)

Dia.	Circum.	Area*
142½	447.68	15949
⅝	448.07	15977
¾	448.46	16005
⅞	448.86	16033
143.	449.25	16061
⅛	449.64	16089
¼	450.03	16117
⅜	450.43	16145
½	450.82	16173
⅝	451.21	16201
¾	451.61	16229
⅞	452.00	16258
144.	452.39	16286
⅛	452.78	16314
¼	453.18	16342
⅜	453.57	16371
½	453.96	16399
⅝	454.35	16428
¾	454.75	16456
⅞	455.14	16485
145.	455.53	16513
⅛	455.93	16542
¼	456.32	16570
⅜	456.71	16599
½	457.10	16627
⅝	457.50	16656
¾	457.89	16684
⅞	458.28	16713
146.	458.67	16742
⅛	459.07	16770
¼	459.46	16799
⅜	459.85	16827
½	460.24	16856
⅝	460.64	16885
¾	461.03	16914
⅞	461.42	16943
147.	461.82	16972
⅛	462.21	17000
¼	462.60	17029
⅜	462.99	17058
½	463.39	17087
⅝	463.78	17116
¾	464.17	17145
⅞	464.56	17174
148.	464.96	17203
⅛	465.35	17232
¼	465.74	17262
⅜	466.14	17291
½	466.53	17321
⅝	466.92	17350

Dia.	Circum.	Area*
148¾	467.31	17379
⅞	467.71	17408
149.	468.10	17437
⅛	468.49	17466
¼	468.88	17496
⅜	469.28	17525
½	469.67	17555
⅝	470.06	17584
¾	470.46	17614
⅞	470.85	17643
150.	471.24	17672
⅛	471.63	17702
¼	472.03	17731
⅜	472.42	17761
½	472.81	17790
⅝	473.20	17820
¾	473.60	17849
⅞	473.99	17879
151.	474.38	17908
⅛	474.77	17938
¼	475.17	17967
⅜	475.56	17997
½	475.95	18026
⅝	476.35	18056
¾	476.74	18086
⅞	477.13	18116
152.	477.52	18146
⅛	477.92	18175
¼	478.31	18205
⅜	478.70	18235
½	479.09	18265
⅝	479.49	18295
¾	479.88	18325
⅞	480.27	18355
153.	480.67	18385
⅛	481.06	18415
¼	481.45	18446
⅜	481.84	18476
½	482.24	18507
⅝	482.63	18537
¾	483.02	18567
⅞	483.41	18597
154.	483.81	18627
⅛	484.20	18658
¼	484.59	18688
⅜	484.99	18719
½	485.38	18749
⅝	485.77	18779
¾	486.16	18809
⅞	486.56	18839

Dia.	Circum.	Area*
155.	486.95	18869
⅛	487.34	18900
¼	487.73	18930
⅜	488.13	18961
½	488.52	18991
⅝	488.91	19022
¾	489.30	19052
⅞	489.70	19083
156.	490.09	19113
⅛	490.48	19144
¼	490.88	19174
⅜	491.27	19205
½	491.66	19235
⅝	492.05	19266
¾	492.45	19297
⅞	492.84	19328
157.	493.23	19359
⅛	493.62	19390
¼	494.02	19421
⅜	494.41	19452
½	494.80	19483
⅝	495.20	19514
¾	495.59	19545
⅞	495.98	19576
158.	496.37	19607
⅛	496.77	19638
¼	497.16	19669
⅜	497.55	19701
½	497.94	19732
⅝	498.34	19763
¾	498.73	19794
⅞	499.12	19825
159.	499.51	19856
⅛	499.91	19887
¼	500.30	19919
⅜	500.69	19950
½	501.09	19982
⅝	501.48	20013
¾	501.87	20044
⅞	502.26	20075
160.	502.66	20106
⅛	503.05	20138
¼	503.44	20169
⅜	503.83	20201
½	504.23	20232
⅝	504.62	20264
¾	505.01	20295
⅞	505.41	20327
161.	505.80	20358
⅛	506.19	20390

*Approximate area, sufficiently accurate for practical purposes, including estimating.

A-18.
(Continued). Circumferences and Areas of Circles
(Advancing by eighths)

Dia.	Circum.	Area*
161¼	506.58	20421
⅜	506.98	20453
½	507.37	20484
⅝	507.76	20516
¾	508.15	20548
⅞	508.55	20580
162.	508.94	20612
⅛	509.33	20644
¼	509.73	20675
⅜	510.12	20707
½	510.51	20739
⅝	510.90	20771
¾	511.30	20803
⅞	511.69	20835
163.	512.08	20867
⅛	512.47	20899
¼	512.87	20931
⅜	513.26	20964
½	513.65	20996
⅝	514.04	21028
¾	514.44	21060
⅞	514.83	21092
164.	515.22	21124
⅛	515.62	21157
¼	516.01	21189
⅜	516.40	21222
½	516.7[illegible]	21254
⅝	517.[illegible]	21287
¾	517.58	21319
⅞	517.97	21351
165.	518.36	21383
⅛	518.76	21416
¼	519.15	21448
⅜	519.54	21481
½	519.94	21513
⅝	520.33	21546
¾	520.72	21578
⅞	521.11	21610
166.	521.51	21642
⅛	521.90	21675
¼	522.29	21707
⅜	522.68	21740
½	523.08	21772
⅝	523.47	21805
¾	523.86	21838
⅞	524.26	21871
167.	524.65	21904
⅛	525.04	21937
¼	525.43	21969
⅜	525.83	22002

Dia.	Circum.	Area*
167½	526.22	22035
⅝	526.61	22068
¾	527.00	22101
⅞	527.40	22134
168.	527.79	22167
⅛	528.18	22200
¼	528.57	22233
⅜	528.97	22266
½	529.36	22299
⅝	529.75	22332
¾	530.15	22366
⅞	530.54	22399
169.	530.93	22432
⅛	531.32	22465
¼	531.72	22499
⅜	532.11	22532
½	532.50	22566
⅝	532.89	22599
¾	533.29	22632
⅞	533.68	22665
170.	534.07	22698
⅛	534.47	22731
¼	534.86	22765
⅜	535.25	22798
½	535.64	22832
⅝	536.04	22865
¾	536.43	22899
⅞	536.82	22932
171.	537.21	22966
⅛	537.61	22999
¼	538.00	23033
⅜	538.39	23066
½	538.78	23100
⅝	539.18	23133
¾	539.57	23167
⅞	539.96	23201
172.	540.36	23235
⅛	540.75	23268
¼	541.14	23302
⅜	541.53	23336
½	541.93	23370
⅝	542.32	23404
¾	542.71	23438
⅞	543.10	23472
173.	543.50	23506
⅛	543.89	23540
¼	544.28	23575
⅜	544.68	23609
½	545.07	23643
⅝	545.46	23677

Dia.	Circum.	Area*
173¾	545.85	23711
⅞	546.25	23745
174.	546.64	23779
⅛	547.03	23813
¼	547.42	23848
⅜	547.82	23882
½	548.21	23917
⅝	548.60	23951
¾	549.00	23985
⅞	549.39	24019
175.	549.78	24053
⅛	550.17	24087
¼	550.57	24122
⅜	550.96	24156
½	551.35	24191
⅝	551.74	24225
¾	552.14	24260
⅞	552.53	24294
176.	552.92	24329
⅛	553.31	24363
¼	553.71	24398
⅜	554.10	24432
½	554.49	24467
⅝	554.89	24501
¾	555.28	24536
⅞	555.67	24571
177.	556.06	24606
⅛	556.46	24640
¼	556.85	24675
⅜	557.24	24710
½	557.63	24745
⅝	558.03	24780
¾	558.42	24815
⅞	558.81	24850
178.	559.21	24885
⅛	559.60	24920
¼	559.99	24955
⅜	560.38	24990
½	560.78	25025
⅝	561.17	25060
¾	561.56	25095
⅞	561.95	25130
179.	562.35	25165
⅛	562.74	25200
¼	563.13	25236
⅜	563.53	25271
½	563.92	25307
⅝	564.31	25342
¾	564.70	25377
⅞	565.10	25412

*Approximate area, sufficiently accurate for practical purposes, including estimating.

A-18.

(Continued). Circumferences and Areas of Circles (Advancing by eighths)

Dia.	Circum.	Area*
180.	565.49	25447
1/8	565.88	25482
1/4	566.27	25518
3/8	566.67	25553
1/2	567.06	25589
5/8	567.45	25624
3/4	567.84	25660
7/8	568.24	25695
181.	568.63	25730
1/8	569.02	25765
1/4	569.42	25801
3/8	569.81	25836
1/2	570.20	25872
5/8	570.59	25908
3/4	570.99	25944
7/8	571.38	25980
182.	571.77	26016
1/8	572.16	26051
1/4	572.56	26087
3/8	572.95	26122
1/2	573.34	26158
5/8	573.74	26194
3/4	574.13	26230
7/8	574.52	26266
183.	574.91	26302
1/8	575.31	26338
1/4	575.70	26374
3/8	576.09	26410
1/2	576.48	26446
5/8	576.88	26482
3/4	577.27	26518
7/8	577.66	26554
184.	578.05	26590
1/8	578.45	26626
1/4	578.84	26663
3/8	579.23	26699
1/2	579.63	26736
5/8	580.02	26772
3/4	580.41	26808
7/8	580.80	26844
185.	581.20	26880
1/8	581.59	26916
1/4	581.98	26953
3/8	582.37	26989
1/2	582.77	27026
5/8	583.16	27062
3/4	583.55	27099
7/8	583.95	27135
186.	584.34	27172
1/8	584.73	27208

Dia.	Circum.	Area*
186 1/4	585.12	27245
3/8	585.52	27281
1/2	585.91	27318
5/8	586.30	27354
3/4	586.59	27391
7/8	587.09	27428
187.	587.48	27465
1/8	587.87	27501
1/4	588.27	27538
3/8	588.66	27574
1/2	589.05	27611
5/8	589.44	27648
3/4	589.84	27685
7/8	590.23	27722
188.	590.62	27759
1/8	591.01	27796
1/4	591.41	27833
3/8	591.80	27870
1/2	592.19	27907
5/8	592.58	27944
3/4	592.98	27981
7/8	593.37	28018
189.	593.76	28055
1/8	594.16	28092
1/4	594.55	28130
3/8	594.94	28167
1/2	595.33	28205
5/8	595.73	28242
3/4	596.12	28279
7/8	596.51	28316
190.	596.90	28353
1/8	597.29	28390
1/4	597.68	28428
3/8	598.08	28465
1/2	598.47	28503
5/8	598.86	28540
3/4	599.25	28578
7/8	599.64	28615
191.	600.04	28652
1/8	600.44	28689
1/4	600.83	28727
3/8	601.22	28764
1/2	601.62	28802
5/8	602.01	28839
3/4	602.40	28877
7/8	602.79	28915
192.	603.19	28953
1/8	603.58	28990
1/4	603.97	29028
3/8	604.36	29065

Dia.	Circum.	Area*
192 1/2	604.76	29103
5/8	605.15	29141
3/4	605.54	29179
7/8	605.94	29217
193.	606.33	29255
1/8	606.72	29293
1/4	607.11	29331
3/8	607.51	29369
1/2	607.90	29407
5/8	608.29	29445
3/4	608.58	29483
7/8	609.08	29521
194.	609.47	29559
1/8	609.86	29597
1/4	610.26	29636
3/8	610.65	29674
1/2	611.05	29713
5/8	611.43	29751
3/4	611.83	29789
7/8	612.29	29827
195.	612.61	29865
1/8	613.00	29903
1/4	613.40	29942
3/8	613.79	29980
1/2	614.18	30019
5/8	614.57	30057
3/4	614.97	30096
7/8	615.36	30134
196.	615.75	30172
1/8	616.15	30210
1/4	616.54	30249
3/8	616.93	30287
1/2	617.32	30326
5/8	617.72	30364
3/4	618.11	30403
7/8	618.50	30442
197.	618.89	30481
1/8	619.29	30519
1/4	619.68	30558
3/8	620.08	30596
1/2	620.47	30635
5/8	620.86	30674
3/4	621.25	30713
7/8	621.64	30752
198.	622.04	30791
1/8	622.44	30830
1/4	622.83	30869
3/8	623.22	30908
1/2	623.62	30947
5/8	624.01	30986

*Approximate area, sufficiently accurate for practical purposes, including estimating.

A-18.

(Concluded). Circumferences and Areas of Circles (Advancing by eighths)

Dia.	Circum.	Area*
198 3/4	624.40	31025
7/8	624.79	31064
199.	625.18	31103
1/8	625.58	31142
1/4	625.97	31181
3/8	626.36	31220
1/2	626.76	31260
5/8	627.15	31299
3/4	627.54	31338
7/8	627.94	31377
200.	628.32	31416
1/8	628.72	31455
1/4	629.11	31495
3/8	629.51	31534
1/2	629.90	31574
5/8	630.29	31613
3/4	630.58	31653
7/8	631.08	31692
201.	631.46	31731
1/8	631.86	31770
1/4	632.26	31810
3/8	632.65	31849
1/2	633.05	31889
5/8	633.43	31928
3/4	633.83	31968
7/8	634.29	32007
202.	634.60	32047
1/8	635.00	32086
1/4	635.40	32126
3/8	635.79	32166
1/2	636.18	32206
5/8	636.57	32246
3/4	636.97	32286
7/8	637.36	32326
203.	637.74	32366
1/8	638.15	32405
1/4	638.54	32445
3/8	638.93	32485
1/2	639.32	32525
5/8	639.72	32565
3/4	640.11	32605
7/8	640.50	32645
204.	640.88	32685
1/8	641.28	32725
1/4	641.67	32766
3/8	642.07	32806
1/2	642.46	32846
5/8	642.85	32886
3/4	643.24	32926

Dia.	Circum.	Area*
204 7/8	643.63	32966
205.	644.03	33006
1/8	644.43	33046
1/4	644.82	33087
3/8	645.21	33127
1/2	645.61	33168
5/8	646.00	33208
3/4	646.39	33249
7/8	646.78	33289
206.	647.17	33329
1/8	647.57	33369
1/4	647.96	33410
3/8	648.35	33450
1/2	648.75	33491
5/8	649.14	33531
3/4	649.53	33572
7/8	649.93	33613
207.	650.31	33654
1/8	650.71	33694
1/4	651.10	33735
3/8	651.50	33775
1/2	651.89	33816
5/8	652.28	33857
3/4	652.57	33898
7/8	653 07	33939
208.	653.45	33980
1/8	653.85	34020
1/4	654.25	34061
3/8	654.64	34102
1/2	655.04	34143
5/8	655.42	34184
3/4	655.82	34225
7/8	656.28	34266
209.	656.59	34307
1/8	656.99	34348
1/4	657.39	34389
3/8	657.78	34431
1/2	658.17	34472
5/8	658.56	34513
3/4	658.96	34554
7/8	659.35	34595
210.	659.73	34636
1/8	660.14	34677
1/4	660.53	34719
3/8	660.92	34760
1/2	661.31	34802
5/8	661.71	34843
3/4	662.10	34885
7/8	662.49	34926

Dia.	Circum.	Area*
211.	662.88	34967
1/8	663.28	35008
1/4	663.67	35050
3/8	664.07	35091
1/2	664.46	35133
5/8	664.85	35174
3/4	665.24	35216
7/8	665.63	35257
212.	666.02	35299
1/8	666.43	35340
1/4	666.82	35382
3/8	667.21	35423
1/2	667.61	35465
5/8	668.00	35507
3/4	668.39	35549
7/8	688.78	35591
213.	669.16	35633
1/8	669.57	35674
1/4	669.96	35716
3/8	670.35	35758
1/2	670.75	35800
5/8	671.14	35842
3/4	671.53	35884
7/8	671.93	35926
214.	672.30	35968
1/8	672.71	36010
1/4	673.10	36052
3/8	673.50	36094
1/2	673.89	36137
5/8	674.28	36179
3/4	674.57	36221
7/8	675.07	36263
215.	675.44	36305
1/8	675.85	36347
1/4	676.25	36390
3/8	676.64	36432
1/2	677.04	36475
5/8	677.42	36517
3/4	677.82	36560
7/8	678.28	36602
216.	678.58	36644
1/8	678.99	36686
1/4	679.39	36729
3/8	679.78	36771
1/2	680.17	36814
5/8	680.56	36856
3/4	680.96	36899
7/8	681.36	36941

*Approximate area, sufficiently accurate for practical purposes, including estimating.

A-19.

Capacities of Cylinders and Spheres

Diam. in Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.
1/64	.0002	.00143	.000034	.00077	.000002
1/32	.0008	.00574	.000137	.00307	.000016
1/16	.0031	.02295	.000546	.01227	.000128
3/32	.0069	.05164	.00123	.02761	.000431
1/8	.0123	.09180	.00219	.04909	.00102
5/32	.0192	.14344	.00342	.07670	.00200
3/16	.0276	.20655	.00492	.11045	.00345
7/32	.0376	.28114	.00669	.15033	.00548
1/4	.0491	.36720	.00874	.19635	.00818
9/32	.0621	.46474	.01107	.24850	.01165
5/16	.0767	.57375	.01366	.30680	.01598
11/32	.0928	.69424	.01653	.37122	.02127
3/8	.1104	.82620	.01967	.44179	.02761
13/32	.1296	.96964	.02309	.51849	.03511
7/16	.1503	1.1245	.02677	.60132	.04385
15/32	.1726	1.2909	.03074	.69029	.05393
1/2	.1963	1.4688	.03497	.78540	.06545
17/32	.2217	1.6581	.03948	.88664	.07850
9/16	.2485	1.8589	.04426	.99402	.09319
19/32	.2769	2.0712	.04932	1.1075	.10960
5/8	.3068	2.2950	.05464	1.2272	.12783
21/32	.3382	2.5302	.06024	1.3530	.14798
11/16	.3712	2.7769	.06612	1.4849	.17014
23/32	.4057	3.0351	.07227	1.6230	.19442
3/4	.4418	3.3048	.07869	1.7671	.22089
25/32	.4794	3.5859	.08538	1.9175	.24967
13/16	.5185	3.8785	.09235	2.0739	.28085
27/32	.5591	4.1826	.09959	2.2365	.31451
7/8	.6013	4.4982	.10710	2.4053	.35077
29/32	.6450	4.8252	.11489	2.5802	.38971
15/16	.6903	5.1637	.12295	2.7612	.43143
31/32	.7371	5.5137	.13128	2.9483	.47603
1	.7854	5.8752	.13989	3.1416	.52360
1 1/16	.8866	6.6325	.15792	3.5466	.62804
1 1/8	.9940	7.4358	.17704	3.9761	.74551
1 3/16	1.1075	8.2849	.19726	4.4301	.87680
1 1/4	1.2272	9.1800	.21857	4.9087	1.0227
1 5/16	1.3530	10.121	.24097	5.4119	1.1838
1 3/8	1.4849	11.108	.26447	5.9396	1.3612
1 7/16	1.6230	12.141	.28906	6.4918	1.5553
1 1/2	1.7671	13.219	.31474	7.0686	1.7671
1 9/16	1.9175	14.344	.34152	7.6699	1.9974
1 5/8	2.0739	15.514	.36938	8.2958	2.2468
1 11/16	2.2365	16.731	.39835	8.9462	2.5161
1 3/4	2.4053	17.993	.42840	9.6211	2.8062
1 13/16	2.5802	19.301	.45955	10.321	3.1177
1 7/8	2.7612	20.655	.49178	11.045	3.4515
1 15/16	2.9483	22.055	.52512	11.793	3.8082
2	3.1416	23.501	.55954	12.566	4.1888
2 1/16	3.3410	24.992	.59506	13.364	4.5939
2 1/8	3.5466	26.530	.63167	14.186	5.0243
2 3/16	3.7583	28.114	.66937	15.033	5.4808
2 1/4	3.9761	29.743	.70817	15.904	5.9641
2 5/16	4.2000	31.418	.74806	16.800	6.4751
2 3/8	4.4301	33.140	.78904	17.721	7.0144
2 7/16	4.6664	34.907	.83112	18.665	7.5829

Diam. in Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.
2 1/2	4.9087	36.720	.87428	19.635	8.1812
2 9/16	5.1572	38.579	.91854	20.629	8.8103
2 5/8	5.4119	40.484	.96390	21.648	9.4708
2 11/16	5.6727	42.434	1.0103	22.691	10.164
2 3/4	5.9396	44.431	1.0578	23.758	10.889
2 13/16	6.2126	46.474	1.1065	24.850	11.649
2 7/8	6.4918	48.562	1.1562	25.967	12.443
2 15/16	6.7771	50.696	1.2071	27.109	13.272
3	7.0686	52.877	1.2590	28.274	14.137
3 1/16	7.3662	55.103	1.3120	29.465	15.039
3 1/8	7.6699	57.375	1.3661	30.680	15.979
3 3/16	7.9798	59.693	1.4213	31.919	16.957
3 1/4	8.2958	62.057	1.4775	33.183	17.974
3 5/16	8.6179	64.466	1.5349	34.472	19.031
3 3/8	8.9462	66.922	1.5934	35.785	20.129
3 7/16	9.2806	69.424	1.6529	37.122	21.268
3 1/2	9.6211	71.971	1.7136	38.485	22.449
3 5/8	10.321	77.204	1.8382	41.282	24.942
3 3/4	11.045	82.620	1.9671	44.179	27.612
3 7/8	11.793	88.220	2.1005	47.173	30.466
4	12.566	94.003	2.2382	50.265	33.510
4 1/8	13.364	99.970	2.3802	53.456	36.751
4 1/4	14.186	106.12	2.5267	56.745	40.194
4 3/8	15.033	112.45	2.6775	60.132	43.846
4 1/2	15.904	118.97	2.8327	63.617	47.713
4 5/8	16.800	125.67	2.9922	67.201	51.800
4 3/4	17.721	132.56	3.1562	70.882	56.115
4 7/8	18.665	139.63	3.3245	74.662	60.663
5	19.635	146.88	3.4971	78.540	65.450
5 1/8	20.629	154.32	3.6742	82.516	70.482
5 1/4	21.648	161.93	3.8556	86.590	75.766
5 3/8	22.691	169.74	4.0414	90.763	81.308
5 1/2	23.758	177.72	4.2315	95.033	87.114
5 5/8	24.850	185.89	4.4261	99.402	93.189
5 3/4	25.967	194.25	4.6250	103.87	99.541
5 7/8	27.109	202.79	4.8282	108.43	106.17
6	28.274	211.51	5.0359	113.10	113.10
6 1/8	29.465	220.41	5.2479	117.86	120.31
6 1/4	30.680	229.50	5.4643	122.72	127.83
6 3/8	31.919	238.77	5.6850	127.68	135.66
6 1/2	33.183	248.23	5.9102	132.73	143.79
6 5/8	34.472	257.87	6.1397	137.89	152.25
6 3/4	35.785	267.69	6.3735	143.14	161.03
6 7/8	37.122	277.69	6.6118	148.49	170.14
7	38.485	287.88	6.8544	153.94	179.59
7 1/8	39.871	298.26	7.1014	159.48	189.39
7 1/4	41.282	308.81	7.3527	165.13	199.53
7 3/8	42.718	319.56	7.6085	170.87	210.03
7 1/2	44.179	330.48	7.8686	176.71	220.89
7 5/8	45.664	341.59	8.1330	182.65	232.12
7 3/4	47.173	352.88	8.4019	188.69	243.73
7 7/8	48.707	364.35	8.6751	194.83	255.71

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(Continued). Capacities of Cylinders and Spheres

Diam. in Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.
8	50.265	376.01	8.9527	201.06	268.08
8 1/8	51.849	387.85	9.2346	207.39	280.85
8 1/4	53.456	399.88	9.5209	213.82	294.01
8 3/8	55.088	412.09	9.8116	220.35	307.58
8 1/2	56.745	424.48	10.107	226.98	321.56
8 5/8	58.426	437.06	10.406	233.71	335.95
8 3/4	60.132	449.82	10.710	240.53	350.77
8 7/8	61.862	462.76	11.018	247.45	366.02
9	63.617	475.89	11.331	254.47	381.70
9 1/8	65.397	489.20	11.648	261.59	397.83
9 1/4	67.201	502.70	11.969	268.80	414.40
9 3/8	69.029	516.37	12.295	276.12	431.43
9 1/2	70.882	530.24	12.625	283.53	448.92
9 5/8	72.760	544.28	12.959	291.04	466.88
9 3/4	74.662	558.51	13.298	298.65	485.30
9 7/8	76.589	572.92	13.641	306.35	504.21
10	78.540	587.52	13.989	314.16	523.60
10 1/4	82.516	617.26	14.697	330.06	563.86
10 1/2	86.590	647.74	15.422	346.36	606.13
10 3/4	90.763	678.95	16.166	363.05	650.47
11	95.033	710.90	16.926	380.13	696.91
11 1/4	99.402	743.58	17.704	397.61	745.51
11 1/2	103.87	776.99	18.500	415.48	796.33
11 3/4	108.43	811.14	19.313	433.74	849.40
12	113.10	846.03	20.143	452.39	904.78
12 1/4	117.86	881.65	20.992	471.44	962.51
12 1/2	122.72	918.00	21.857	490.87	1022.7
12 3/4	127.68	955.08	22.740	510.71	1085.2
13	132.73	992.91	23.641	530.93	1150.3
13 1/4	137.89	1031.5	24.559	551.55	1218.0
13 1/2	143.14	1070.8	25.494	572.56	1288.2
13 3/4	148.49	1110.8	26.447	593.96	1361.2
14	153.94	1151.5	27.418	615.75	1436.8
14 1/4	159.48	1193.0	28.405	637.94	1515.1
14 1/2	165.13	1235.3	29.411	660.52	1596.3
14 3/4	170.87	1278.2	30.434	683.49	1680.3
15	176.71	1321.9	31.474	706.86	1767.1
15 1/4	182.65	1366.3	32.532	730.62	1857.0
15 1/2	188.69	1411.5	33.607	754.77	1949.8
15 3/4	194.83	1457.4	34.700	779.31	2045.7
16	201.06	1504.0	35.811	804.25	2144.7
16 1/4	207.39	1551.4	36.938	829.58	2246.8
16 1/2	213.82	1599.5	38.084	855.30	2352.1
16 3/4	220.35	1648.4	39.247	881.41	2460.6
17	226.98	1697.9	40.427	907.92	2572.4
17 1/4	233.71	1748.2	41.625	934.82	2687.6
17 1/2	240.53	1799.3	42.840	962.11	2806.2
17 3/4	247.45	1851.1	44.073	989.80	2928.2

Diam. in Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.
18	254.47	1903.6	45.323	1017.9	3053.6
18 1/4	261.59	1956.8	46.591	1046.3	3182.6
18 1/2	268.80	2010.8	47.876	1075.2	3315.2
18 3/4	276.12	2065.5	49.178	1104.5	3451.5
19	283.53	2120.9	50.499	1134.1	3591.4
19 1/4	291.04	2177.1	51.836	1164.2	3735.0
19 1/2	298.65	2234.0	53.191	1194.6	3882.4
19 3/4	306.35	2291.7	54.564	1225.4	4033.7
20	314.16	2350.1	55.954	1256.6	4188.8
20 1/4	322.06	2409.2	57.362	1288.2	4347.8
20 1/2	330.06	2469.0	58.787	1320.3	4510.9
20 3/4	338.16	2529.6	60.229	1352.7	4677.9
21	346.36	2591.0	61.689	1385.4	4849.0
21 1/4	354.66	2653.0	63.167	1418.6	5024.3
21 1/2	363.05	2715.8	64.662	1452.2	5203.7
21 3/4	371.54	2779.3	66.175	1486.2	5387.4
22	380.13	2843.6	67.705	1520.5	5575.3
22 1/4	388.82	2908.6	69.252	1555.3	5767.5
22 1/2	397.61	2974.3	70.817	1590.4	5964.1
22 3/4	406.49	3040.8	72.399	1626.0	6165.1
23	415.48	3108.0	73.999	1661.9	6370.6
23 1/4	424.56	3175.9	75.617	1698.2	6580.6
23 1/2	433.74	3244.6	77.252	1734.9	6795.2
23 3/4	443.01	3314.0	78.904	1772.1	7014.4
24	452.39	3384.1	80.574	1809.6	7238.2
24 1/4	461.86	3455.0	82.261	1847.5	7466.8
24 1/2	471.44	3526.6	83.966	1885.7	7700.1
24 3/4	481.11	3598.9	85.689	1924.4	7938.2
25	490.87	3672.0	87.428	1963.5	8181.2
25 1/4	500.74	3745.8	89.186	2003.0	8429.1
25 1/2	510.71	3820.3	90.960	2042.8	8682.0
25 3/4	520.77	3895.6	92.753	2083.1	8939.9
26	530.93	3971.6	94.563	2123.7	9202.8
26 1/4	541.19	4048.4	96.390	2164.8	9470.8
26 1/2	551.55	4125.8	98.235	2206.2	9744.0
26 3/4	562.00	4204.1	100.10	2248.0	10022
27	572.56	4283.0	101.98	2290.2	10306
27 1/4	583.21	4362.7	103.87	2332.8	10595
27 1/2	593.96	4443.1	105.79	2375.8	10889
27 3/4	604.81	4524.3	107.72	2419.2	11189
28	615.75	4606.1	109.67	2463.0	11494
28 1/4	626.80	4688.8	111.64	2507.2	11805
28 1/2	637.94	4772.1	113.62	2551.8	12121
28 3/4	649.18	4856.2	115.62	2596.7	12443
29	660.52	4941.0	117.64	2642.1	12770
29 1/4	671.96	5026.6	119.68	2687.8	13103
29 1/2	683.49	5112.9	121.74	2734.0	13442
29 3/4	695.13	5199.9	123.81	2780.5	13787

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(Continued). Capacities of Cylinders and Spheres

Diam. in Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.	Diam. in Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.
30	706.86	5287.7	125.90	2827.4	14137	42	1385.4	10364	246.76	5541.8	38792
30¼	718.69	5376.2	128.00	2874.8	14494	42¼	1402.0	10488	249.70	5607.9	39489
30½	730.62	5465.4	130.13	2922.5	14856	42½	1418.6	10612	252.67	5674.5	40194
30¾	742.64	5555.4	132.27	2970.6	15224	42¾	1435.4	10737	255.65	5714.5	40908
31	754.77	5646.1	134.43	3019.1	15599	43	1452.2	10863	258.65	5808.8	41630
31¼	766.99	5737.5	136.61	3068.0	15979	43¼	1469.1	10990	261.66	5876.5	42360
31½	779.31	5829.7	138.80	3117.2	16366	43½	1486.2	11117	264.70	5944.7	43099
31¾	791.73	5922.6	141.01	3166.9	16758	43¾	1503.3	11245	267.75	6013.2	43846
32	804.25	6016.2	143.24	3217.0	17157	44	1520.5	11374	270.82	6082.1	44602
32¼	816.86	6110.6	145.49	3267.5	17563	44¼	1537.9	11504	273.90	6151.4	45367
32½	829.58	6205.7	147.75	3318.3	17974	44½	1555.3	11634	277.01	6221.1	46140
32¾	842.39	6301.5	150.04	3369.6	18392	44¾	1572.8	11765	280.13	6291.2	46922
33	855.30	6398.1	152.34	3421.2	18817	45	1590.4	11897	283.27	6361.7	47713
33¼	868.31	6495.4	154.65	3473.2	19247	45¼	1608.2	12030	286.42	6432.6	48513
33½	881.41	6593.4	156.99	3525.7	19685	45½	1626.0	12163	289.60	6503.9	49321
33¾	894.62	6692.2	159.34	3578.5	20129	45¾	1643.9	12297	292.79	6575.5	50139
34	907.92	6791.7	161.71	3631.7	20580	46	1661.9	12432	296.00	6647.6	50965
34¼	921.32	6892.0	164.09	3685.3	21037	46¼	1680.0	12567	299.22	6720.1	51800
34½	934.82	6992.9	166.50	3739.3	21501	46½	1698.2	12704	302.47	6792.9	52645
34¾	948.42	7094.7	168.92	3793.7	21972	46¾	1716.5	12841	305.73	6866.1	53499
35	962.11	7197.1	171.36	3848.5	22449	47	1734.9	12978	309.01	6939.8	54362
35¼	975.91	7300.3	173.82	3903.6	22934	47¼	1753.5	13117	312.30	7013.8	55234
35½	989.80	7404.2	176.29	3959.2	23425	47½	1772.1	13256	315.62	7088.2	56115
35¾	1003.8	7508.9	178.78	4015.2	23924	47¾	1790.8	13396	318.95	7163.0	57006
36	1017.9	7614.2	181.29	4071.5	24429	48	1809.6	13536	322.30	7238.2	57906
36¼	1032.1	7720.4	183.82	4128.2	24942	48¼	1828.5	13678	325.66	7313.8	58815
36½	1046.3	7827.2	186.36	4185.4	25461	48½	1847.5	13820	329.05	7389.8	59734
36¾	1060.7	7934.8	188.92	4242.9	25988	48¾	1866.5	13963	332.45	7466.2	60663
37	1075.2	8043.1	191.50	4300.8	26522	49	1885.7	14106	335.86	7543.0	61601
37¼	1089.8	8152.2	194.10	4359.2	27063	49¼	1905.0	14251	339.30	7620.1	62549
37½	1104.5	8262.0	196.71	4417.9	27612	49½	1924.4	14396	342.75	7697.7	63506
37¾	1119.2	8372.5	199.35	4477.0	28168	49¾	1943.9	14541	346.23	7775.6	64473
38	1134.1	8483.8	201.99	4536.5	28731	50	1963.5	14688	349.71	7854.0	65450
38¼	1149.1	8595.8	204.66	4596.3	29302	50¼	1983.2	14835	353.22	7932.7	66437
38½	1164.2	8708.5	207.35	4656.6	29880	50½	2003.0	14983	356.74	8011.8	67433
38¾	1179.3	8822.0	210.05	4717.3	30466	50¾	2022.8	15132	360.28	8091.4	68439
39	1194.6	8936.2	212.77	4778.4	31059	51	2042.8	15281	363.84	8171.3	69456
39¼	1210.0	9051.1	215.50	4839.8	31660	51¼	2062.9	15432	367.42	8251.6	70482
39½	1225.4	9166.8	218.26	4901.7	32269	51½	2083.1	15582	371.01	8332.3	71519
39¾	1241.0	9283.2	221.03	4963.9	32886	51¾	2103.3	15734	374.62	8413.4	72565
40	1256.6	9400.3	223.82	5026.5	33510	52	2123.7	15887	378.25	8494.9	73622
40¼	1272.4	9518.2	226.62	5089.6	34143	52¼	2144.2	16040	381.90	8576.7	74689
40½	1288.2	9636.8	229.45	5153.0	34783	52½	2164.8	16193	385.56	8659.0	75766
40¾	1304.2	9756.1	232.29	5216.8	35431	52¾	2185.4	16348	389.24	8741.7	76854
41	1320.3	9876.2	235.15	5281.0	36087	53	2206.2	16503	392.94	8824.7	77952
41¼	1336.4	9997.0	238.02	5345.6	36751	53¼	2227.0	16659	396.65	8908.2	79060
41½	1352.7	10119.	240.92	5410.6	37423	53½	2248.0	16816	400.39	8992.0	80179
41¾	1369.0	10241.	243.83	5476.0	38104	53¾	2269.1	16974	404.14	9076.3	81308

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(Continued). Capacities of Cylinders and Spheres

Diam. in Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.	Diam. in Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.
54	2290.2	17132	407.91	9160.9	82448	66	3421.2	25592	609.34	13685	150533
54¼	2311.5	17291	411.69	9245.9	83598	66¼	3447.2	25787	613.97	13789	152250
54½	2332.8	17451	415.49	9331.3	84759	66½	3473.2	25982	618.61	13893	153980
54¾	2354.3	17611	419.32	9417.1	85931	66¾	3499.4	26177	623.27	13998	155723
55	2375.8	17772	423.15	9503.3	87114	67	3525.7	26374	627.95	14103	157479
55¼	2397.5	17934	427.01	9589.9	88307	67¼	3552.0	26571	632.64	14208	159249
55½	2419.2	18097	430.88	9676.9	89511	67½	3578.5	26769	637.35	14314	161031
55¾	2441.1	18260	434.77	9764.3	90726	67¾	3605.0	26967	642.08	14420	162827
56	2463.0	18245	438.68	9852.0	91952	68	3631.7	27167	646.83	14527	164636
56¼	2485.0	18589	442.61	9940.2	93189	68¼	3658.4	27367	651.59	14634	166459
56½	2507.2	18755	446.55	10029	94437	68½	3685.3	27568	656.38	14741	168295
56¾	2529.4	18921	450.51	10118	95697	68¾	3712.2	27769	661.18	14849	170144
57	2551.8	19088	454.49	10207	96967	69	3739.3	27972	665.99	14957	172007
57¼	2574.2	19256	458.48	10297	98248	69¼	3766.4	28175	670.83	15066	173883
57½	2596.7	19425	462.50	10387	99541	69½	3793.7	28379	675.68	15175	175773
57¾	2619.4	19594	466.53	10477	100845	69¾	3821.0	28583	680.55	15284	177677
58	2642.1	19764	470.57	10568	102160	70	3848.5	28788	685.44	15394	179594
58¼	2664.9	19935	474.64	10660	103487	70¼	3876.0	28994	690.34	15504	181525
58½	2687.8	20106	478.72	10751	104825	70½	3903.6	29201	695.27	15615	183470
58¾	2710.9	20279	482.82	10843	106175	70¾	3931.4	29409	700.21	15725	185429
59	2734.0	20452	486.94	10936	107536	71	3959.2	29617	705.16	15837	187402
59¼	2757.2	20625	491.08	11029	108909	71¼	3987.1	29826	710.14	15948	189388
59½	2780.5	20800	495.23	11122	110293	71½	4015.2	30035	715.13	16061	191389
59¾	2803.9	20975	499.40	11216	111690	71¾	4043.3	30246	720.14	16173	193404
60	2827.4	21151	503.59	11310	113097	72	4071.5	30457	725.17	16286	195432
60¼	2851.0	21327	507.79	11404	114517	72¼	4099.8	30669	730.21	16399	197475
60½	2874.8	21505	512.02	11499	115948	72½	4128.2	30881	735.27	16513	199532
60¾	2898.6	21683	516.26	11594	117392	72¾	4156.8	31095	740.35	16627	201603
61	2922.5	21862	520.51	11690	118847	73	4185.4	31309	745.45	16742	203689
61¼	2946.5	22041	524.79	11786	120314	73¼	4214.1	31524	750.56	16856	205789
61½	2970.6	22221	529.08	11882	121793	73½	4242.9	31739	755.70	16972	207903
61¾	2994.8	22402	533.39	11979	123285	73¾	4271.8	31956	760.85	17087	210032
62	3019.1	22584	537.72	12076	124788	74	4300.8	32173	766.01	17203	212175
62¼	3043.5	22767	542.06	12174	126304	74¼	4329.9	32390	771.20	17320	214332
62½	3068.0	22950	546.43	12272	127832	74½	4359.2	32609	776.40	17437	216505
62¾	3092.6	23134	550.81	12370	129372	74¾	4388.5	32828	781.62	17554	218692
63	3117.2	23319	555.21	12469	130924	75	4417.9	33048	786.86	17671	220893
63¼	3142.0	23504	559.62	12568	132489	75¼	4447.4	33269	792.11	17789	223110
63½	3166.9	23690	564.05	12668	134066	75½	4477.0	33490	797.38	17908	225341
63¾	3191.9	23877	568.50	12768	135656	75¾	4506 7	33712	802.67	18027	227587
64	3217.0	24065	572.97	12868	137258	76	4536.5	33935	807.98	18146	229847
64¼	3242.2	24253	577.46	12969	138873	76¼	4566.4	34159	813.30	18265	232123
64½	3267.5	24442	581.96	13070	140500	76½	4596.3	34383	818.64	18385	234414
64¾	3292.8	24632	586.48	13171	142141	76¾	4626.4	34608	824.00	18506	236719
65	3318.3	24823	591.02	13273	143793	77	4656.6	34834	829.38	18627	239040
65¼	3343.9	25014	595.57	13376	145459	77¼	4686.9	35061	834.77	18748	241376
65½	3369.6	25206	600.14	13478	147137	77½	4717.3	35288	840.19	18869	243727
65¾	3395.3	25399	604.73	13581	148828	77¾	4747.8	35516	845.62	18991	246093

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(Continued). Capacities of Cylinders and Spheres

Diam. in Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.	Diam. in Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.
78	4778.4	35745	851.06	19113	248475	90	6361.7	47589	1133.1	25447	381704
78¼	4809.0	35974	856.53	19236	250872	90¼	6397.1	47854	1139.4	25588	384893
78½	4839.8	36204	862.01	19359	253284	90½	6432.6	48119	1145.7	25730	388101
78¾	4870.7	36435	867.51	19483	255712	90¾	6468.2	48385	1152.0	25873	391326
79	4901.7	36667	873.02	19607	258155	91	6503.9	48652	1158.4	26016	394569
79¼	4932.7	36899	878.56	19731	260613	91¼	6539.7	48920	1164.8	26159	397830
79½	4963.9	37133	884.11	19856	263087	91½	6575.5	49189	1171.2	26302	401109
79¾	4995.2	37367	889.68	19981	265577	91¾	6611.5	49458	1177.6	26446	404405
80	5026.5	37601	895.27	20106	268083	92	6647.6	49728	1184.0	26590	407720
80¼	5058.0	37837	900.87	20232	270604	92¼	6683.8	49998	1190.4	26735	411053
80½	5089.6	38073	906.49	20358	273141	92½	6720.1	50270	1196.9	26880	414404
80¾	5121.2	38310	912.13	20485	275693	92¾	6756.4	50542	1203.4	27026	417773
81	5153.0	38547	917.79	20612	278262	93	6792.9	50814	1209.9	27172	421160
81¼	5184.9	38785	923.46	20739	280846	93¼	6829.5	51088	1216.4	27318	424566
81½	5216.8	39024	929.15	20867	283447	93½	6866.1	51362	1222.9	27465	427990
81¾	5248.9	39264	934.86	20995	286063	93¾	6902.9	51637	1229.5	27612	431432
82	5281.0	39505	940.59	21124	288696	94	6939.8	51913	1236.0	27759	434893
82¼	5313.3	39746	946.33	21253	291344	94¼	6976.7	52190	1242.6	27907	438372
82½	5345.6	39988	952.09	21382	294009	94½	7013.8	52467	1249.2	28055	441870
82¾	5378.1	40231	957.87	21512	296690	94¾	7051.0	52745	1255.8	28204	445386
83	5410.6	40474	963.67	21642	299387	95	7088.2	53024	1262.5	28353	448920
83¼	5443.3	40718	969.48	21773	302100	95¼	7125.6	53303	1269.1	28502	452474
83½	5476.0	40963	975.32	21904	304830	95½	7163.0	53583	1275.8	28652	456046
83¾	5508.8	41209	981.16	22035	307576	95¾	7200.6	53864	1282.5	28802	459637
84	5541.8	41455	987.03	22167	310339	96	7238.2	54146	1289.2	28953	463247
84¼	5574.8	41702	992.92	22299	313118	96¼	7276.0	54428	1295.9	29104	466875
84½	5607.9	41950	998.82	22432	315914	96½	7313.8	54711	1302.6	29255	470523
84¾	5641.2	42199	1004.7	22565	318726	96¾	7351.8	54995	1309.4	29407	474189
85	5674.5	42448	1010.7	22698	321555	97	7389.8	55280	1316.2	29559	477874
85¼	5707.9	42698	1016.6	22832	324401	97¼	7428.0	55565	1323.0	29712	481579
85½	5741.5	42949	1022.6	22966	327263	97½	7466.2	55851	1329.8	29865	485302
85¾	5775.1	43201	1028.6	23100	330142	97¾	7504.5	56138	1336.6	30018	489045
86	5808.8	43453	1034.6	23235	333038	98	7543.0	56425	1343.5	30172	492807
86¼	5842.6	43706	1040.6	23371	335951	98¼	7581.5	56714	1350.3	30326	496588
86½	5876.5	43960	1046.7	23506	338881	98½	7620.1	57003	1357.2	30481	500388
86¾	5910.6	44214	1052.7	23642	341828	98¾	7658.9	57292	1364.1	30635	504208
87	5944.7	44469	1058.8	23779	344791	99	7697.7	57583	1371.0	30791	508047
87¼	5978.9	44725	1064.9	23916	347772	99¼	7736.6	57874	1377.9	30946	511906
87½	6013.2	44982	1071.0	24053	350770	99½	7775.6	58166	1384.9	31103	515784
87¾	6047.6	45239	1077.1	24190	353785	99¾	7814.8	58458	1391.9	31259	519682
88	6082.1	45497	1083.3	24328	356818	100	7854.0	58752	1398.9	31416	523599
88¼	6116.7	45756	1089.4	24467	359868	100¼	7893.3	59046	1405.9	31573	527536
88½	6151.4	46016	1095.6	24606	362935	100½	7932.7	59341	1412.9	31731	531492
88¾	6186.2	46276	1101.8	24745	366019	100¾	7972.2	59636	1419.9	31889	535468
89	6221.1	46537	1108.0	24885	369121	101	8011.8	59933	1427.0	32047	539464
89¼	6256.1	46799	1114.3	25025	372240	101¼	8051.6	60230	1434.0	32206	543480
89½	6291.2	47062	1120.5	25165	375377	101½	8091.4	60528	1441.1	32365	547516
89¾	6326.4	47325	1126.8	25306	378531	101¾	8131.3	60826	1448.2	32525	551572

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(Continued). Capacities of Cylinders and Spheres

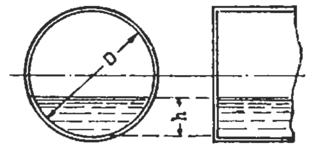
Diam. in Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.	Diam. in Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.
102	8171.3	61125	1455.4	32685	555647	114	10207	76354	1818.0	40828	775735
102¼	8211.4	61425	1462.5	32846	559743	114¼	10252	76689	1825.9	41007	780849
102½	8251.6	61726	1469.7	33006	563859	114½	10297	77025	1833.9	41187	785986
102¾	8291.9	62028	1476.8	33168	567994	114¾	10342	77362	1841.9	41367	791146
103	8332.3	62330	1484.0	33329	572151	115	10387	77699	1850.0	41548	796328
103¼	8372.8	62633	1491.3	33491	576327	115¼	10432	78038	1858.0	41728	801533
103½	8413.4	62936	1498.5	33654	580523	115½	10477	78376	1866.1	41910	806760
103¾	8454.1	63241	1505.7	33816	584740	115¾	10523	78716	1874.2	42091	812010
104	8494.9	63546	1513.0	33979	588977	116	10568	79057	1882.3	42273	817283
104¼	8535.8	63852	1520.3	34143	593235	116¼	10614	79398	1890.4	42456	822579
104½	8576.7	64159	1527.6	34307	597513	116½	10660	79739	1898.6	42638	827897
104¾	8617.8	64466	1534.9	34471	601812	116¾	10705	80082	1906.7	42822	833238
105	8659.0	64774	1542.2	34636	606131	117	10751	80425	1914.9	43005	838603
105¼	8700.3	65083	1549.6	34801	610471	117¼	10797	80769	1923.1	43189	843990
105½	8741.7	65392	1557.0	34967	614831	117½	10843	81114	1931.3	43374	849400
105¾	8783.2	65703	1564.3	35133	619213	117¾	10890	81460	1939.5	43558	854833
106	8824.7	66014	1571.8	35299	623615	118	10936	81806	1947.8	43744	860290
106¼	8866.4	66325	1579.2	35466	628037	118¼	10982	82153	1956.0	43929	865769
106½	8908.2	66638	1586.6	35633	632481	118½	11029	82501	1964.3	44115	871272
106¾	8950.1	66951	1594.1	35800	636945	118¾	11075	82849	1972.6	44301	876798
107	8992.0	67265	1601.5	35968	641431	119	11122	83199	1980.9	44488	882347
107¼	9034.1	67580	1609.0	36136	645938	119¼	11169	83548	1989.2	44675	887920
107½	9076.3	67895	1616.6	36305	650465	119½	11216	83899	1997.6	44863	893516
107¾	9118.5	68211	1624.1	36474	655014	119¾	11263	84251	2006.0	45051	899136
108	9160.9	68528	1631.6	36644	659584	120	11310	84603	2014.3	45239	904779
108¼	9203.3	68846	1639.2	36813	664175	120¼	11357	84956	2022.8	45428	910445
108½	9245.9	69164	1646.8	36984	668787	120½	11404	85309	2031.2	45617	916136
108¾	9288.6	69483	1654.4	37154	673421	120¾	11452	85664	2039.6	45806	921850
109	9331.3	69803	1662.0	37325	678076	121	11499	86019	2048.1	45996	927587
109¼	9374.2	70124	1669.6	37497	682752	121¼	11547	86374	2056.5	46186	933349
109½	9417.1	70445	1677.3	37668	687450	121½	11594	86731	2065.0	46377	939134
109¾	9460.2	70767	1684.9	37841	692169	121¾	11642	87088	2073.5	46568	944943
110	9503.3	71090	1692.6	38013	696910	122	11690	87446	2082.1	46759	950776
110¼	9546.6	71413	1700.3	38186	701672	122¼	11738	87805	2090.6	46951	956633
110½	9589.9	71737	1708.0	38360	706457	122½	11786	88165	2099.2	47144	962514
110¾	9633.4	72062	1715.8	38533	711262	122¾	11834	88525	2107.7	47336	968419
111	9676.9	72388	1723.5	38708	716090	123	11882	88886	2116.3	47529	974348
111¼	9720.5	72715	1731.3	38882	720939	123¼	11931	89247	2124.9	47723	980301
111½	9764.3	73042	1739.1	39057	725810	123½	11979	89610	2133.6	47916	986278
111¾	9808.1	73370	1746.9	39232	730704	123¾	12028	89973	2142.2	48111	992280
112	9852.0	73698	1754.7	39408	735619	124	12076	90337	2150.9	48305	998306
112¼	9896.1	74028	1762.6	39584	740556	124¼	12125	90701	2159.6	48500	1004356
112½	9940.2	74358	1770.4	39761	745515	124½	12174	91067	2168.3	48695	1010431
112¾	9984.4	74689	1778.3	39938	750496	124¾	12223	91433	2177.0	48891	1016530
113	10029	75020	1786.2	40115	755499	125	12272	91800	2185.7	49087	1022654
113¼	10073	75353	1794.1	40293	760525	125¼	12321	92167	2194.5	49284	1028802
113½	10118	75686	1802.0	40471	765572	125½	12370	92536	2203.2	49481	1034975
113¾	10162	76019	1810.0	40649	770642	125¾	12420	92905	2212.0	49678	1041172

A-19.
(Concluded). Capacities of Cylinders and Spheres

Diam. in Feet	Ft. Cu. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.	Diam. in Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.
126	12469	93274	2220.8	49876	1047394	138	14957	111887	2664.0	59828	1376055
126¼	12519	93645	2229.6	50074	1053641	138¼	15011	112293	2673.6	60045	1383547
126½	12568	94016	2238.5	50273	1059913	138½	15066	112699	2683.3	60263	1391067
126¾	12618	94388	2247.3	50471	1066209	138¾	15120	113107	2693.0	60481	1398613
127	12668	94761	2256.2	50671	1072531	139	15175	113514	2702.7	60699	1406187
127¼	12718	95134	2265.1	50870	1078877	139¼	15229	113923	2712.5	60917	1413788
127½	12768	95508	2274.0	51071	1085248	139½	15284	114333	2722.2	61136	1421416
127¾	12818	95883	2282.9	51271	1091645	139¾	15339	114743	2732.0	61356	1429072
128	12868	96259	2291.9	51472	1098066	140	15394	115154	2741.8	61575	1436755
128¼	12918	96635	2300.8	51673	1104513	140¼	15449	115565	2751.6	61795	1444466
128½	12969	97013	2309.8	51875	1110985	140½	15504	115978	2761.4	62016	1452204
128¾	13019	97390	2318.8	52077	1117481	140¾	15559	116391	2771.2	62237	1459970
129	13070	97769	2327.8	52279	1124004	141	15615	116805	2781.1	62458	1467763
129¼	13121	98148	2336.9	52482	1130551	141¼	15670	117219	2790.9	62680	1475584
129½	13171	98528	2345.9	52685	1137124	141½	15725	117634	2800.8	62902	1483433
129¾	13222	98909	2355.0	52889	1143723	141¾	15781	118050	2810.7	63124	1491310
130	13273	99291	2364.1	53093	1150347	142	15837	118467	2820.6	63347	1499214
130¼	13324	99673	2373.2	53297	1156996	142¼	15893	118885	2830.6	63570	1507146
130½	13376	100056	2382.3	53502	1163671	142½	15948	119303	2840.5	63794	1515107
130¾	13427	100440	2391.4	53707	1170371	142¾	16005	119722	2850.5	64018	1523095
131	13478	100824	2400.6	53913	1177098	143	16061	120142	2860.5	64242	1531111
131¼	13530	101209	2409.7	54119	1183850	143¼	16117	120562	2870.5	64467	1539156
131½	13581	101595	2418.9	54325	1190627	143½	16173	120983	2880.6	64692	1547228
131¾	13633	101982	2428.1	54532	1197431	143¾	16230	121405	2890.6	64918	1555329
132	13685	102369	2437.4	54739	1204260	144	16286	121828	2900.7	65144	1563458
132¼	13737	102757	2446.6	54947	1211116	144¼	16343	122251	2910.7	65370	1571615
132½	13789	103146	2455.9	55155	1217997	144½	16399	122675	2920.8	65597	1579800
132¾	13841	103536	2465.1	55363	1224904	144¾	16456	123100	2931.0	65824	1588014
133	13893	103926	2474.4	55572	1231838	145	16513	123526	2941.1	66052	1596256
133¼	13945	104317	2483.7	55781	1238797	145¼	16570	123952	2951.2	66280	1604527
133½	13998	104709	2493.1	55990	1245783	145½	16627	124379	2961.4	66508	1612826
133¾	14050	105102	2502.4	56200	1252795	145¾	16684	124807	2971.6	66737	1621154
134	14103	105495	2511.8	56410	1259833	146	16742	125235	2981.8	66966	1629511
134¼	14155	105889	2521.2	56621	1266898	146¼	16799	125665	2992.0	67196	1637896
134½	14208	106284	2530.6	56832	1273988	146½	16856	126095	3002.3	67426	1646310
134¾	14261	106679	2540.0	57044	1281106	146¾	16914	126525	3012.5	67656	1654752
135	14314	107075	2549.4	57256	1288249	147	16972	126957	3022.8	67887	1663224
135¼	14367	107472	2558.9	57468	1295420	147¼	17029	127389	3033.1	68118	1671724
135½	14420	107870	2568.3	57680	1302616	147½	17087	127822	3043.4	68349	1680253
135¾	14473	108268	2577.8	57893	1309840	147¾	17145	128256	3053.7	68581	1688811
136	14527	108667	2587.3	58107	1317090	148	17203	128690	3064.0	68813	1697398
136¼	14580	109067	2596.8	58321	1324366	148¼	17262	129125	3074.4	69046	1706015
136½	14634	109468	2606.4	58535	1331670	148½	17320	129561	3084.8	69279	1714660
136¾	14687	109869	2615.9	58750	1339000	148¾	17378	129998	3095.2	69513	1723334
137	14741	110271	2625.5	58965	1346357	149	17437	130435	3105.6	69746	1732038
137¼	14795	110674	2635.1	59180	1353741	149¼	17495	130873	3116.0	69981	1740771
137½	14849	111078	2644.7	59396	1361152	149½	17554	131312	3126.5	70215	1749533
137¾	14903	111482	2654.3	59612	1368590	149¾	17613	131751	3136.9	70450	1758325
						150	17671	132192	3147.4	70686	1767146

A-20.

Tank Capacities, Horizontal Cylindrical— Contents of Tanks with Flat Ends When Filled to Various Depths



Diameter of tank inches	Full tank	Depth of liquid, in inches = h: 3″	6″	9″	12″	15″	18″	21″	24″	27″	30″	33″	36″	39″	42″	45″	48″	51″	54″	57″	60″
12″	5.88	1.15	2.94	.....	.....																
18″	13.22	1.45	3.86	6.61	.....																
24″	23.50	1.70	4.60	8.05	11.75																
30″	36.72	1.91	5.23	9.27	13.72	18.36	...	.....													
36″	52.88	2.12	5.79	10.34	15.43	20.85	26.44	...													
42″	71.97	2.28	6.31	11.31	16.97	23.07	29.47	35.99													
48″	94.01	2.45	6.78	12.20	18.38	25.10	32.20	39.54	47.60	....	......										
54″	118.98	2.60	7.22	13.04	19.68	26.97	34.72	42.80	51.08	59.49	......										
60″	146.89	2.75	7.64	13.82	20.91	28.72	37.06	45.82	54.87	64.11	73.44										
66″	177.73	2.89	8.04	14.56	22.07	30.37	39.28	48.65	58.39	68.41	78.59	88.86	....	......							
72″	211.52	3.02	8.42	15.26	23.17	31.92	41.36	51.32	61.71	72.45	83.41	94.54	105.76	......							
78″	248.24	3.15	8.78	15.94	24.21	33.41	43.34	53.86	64.87	76.27	87.97	99.90	111.97	124.13							
84″	287.90	3.26	9.12	16.57	25.24	34.85	45.24	56.29	67.87	79.91	92.30	104.98	117.85	130.87	143.95	......	......				
90″	330.49	3.43	9.46	17.20	26.20	36.21	47.05	58.61	70.75	83.39	96.43	109.81	123.45	137.28	151.23	165.25	......				
96″	376.02	3.50	9.79	17.80	27.13	37.52	48.81	60.84	73.52	86.73	100.39	114.44	128.79	143.40	158.17	173.06	188.01				
102″	424.50	3.61	10.10	18.37	28.01	39.00	50.49	62.99	76.18	89.94	104.20	118.89	133.92	149.25	164.81	180.53	196.37	212.25	......	......	......
108″	476.10	3.71	10.39	18.94	28.90	40.03	52.14	65.09	78.74	93.04	107.87	123.17	138.87	154.89	171.19	187.71	204.37	221.14	238.05	......	......
114″	530.25	3.78	10.74	19.49	29.75	41.22	53.73	67.10	81.24	96.05	111.43	127.31	143.63	160.33	177.33	194.60	212.05	229.65	247.37	265.13	......
120″	587.54	3.91	10.98	20.02	30.57	42.39	55.26	69.06	83.65	98.95	114.87	131.32	148.25	165.58	183.27	201.24	219.46	237.87	256.43	275.08	293.77

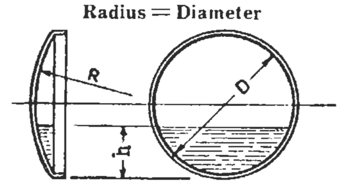
To ascertain the contents of a tank over one-half full: Let h = depth of unfilled portion. Find from the table the quantity corresponding to a depth h. Subtract this quantity from the contents of a full tank.

Contents in U.S. gallons per 1 foot of length.
By permission, The Permutit Co., Inc., Data Book, 1953.

A-21.

Tank Capacities, Horizontal Cylindrical— Contents of Standard Dished Heads When Filled to Various Depths

Radius = Diameter



Diameter of head inches	Full head	Depth of liquid, in inches = h: 3″	6″	9″	12″	15″	18″	21″	24″	27″	30″	33″	36″	39″	42″	45″	48″	51″	54″	57″	60″
12″	0.49	0.05	0.20	.....	.....																
18″	1.36	0.07	0.32	0.68	.....																
24″	3.22	0.08	0.41	0.95	1.61																
30″	6.30	0.10	0.49	1.18	2.10	3.15	.....	.....													
36″	10.88	0.11	0.56	1.39	2.54	3.92	5.44	.....													
42″	17.28	0.12	0.63	1.59	2.94	4.64	6.57	8.64													
48″	25.79	0.13	0.68	1.75	3.31	5.29	7.62	10.19	12.89	.....	......										
54″	36.72	0.14	0.74	1.90	3.64	5.91	8.60	11.65	14.95	18.36	......										
60″	50.37	0.14	0.82	2.07	3.98	6.49	9.54	13.03	16.87	20.96	25.18										
66″	67.01	0.15	0.83	2.19	4.25	6.98	10.35	14.30	18.68	23.43	28.42	33.52	......	......							
72″	87.04	0.16	0.88	2.32	4.52	7.47	11.15	15.48	20.38	25.74	31.46	37.43	43.52	......							
78″	110.66	0.17	0.93	2.44	4.79	7.97	11.94	16.65	22.02	27.97	34.39	41.16	48.20	55.33							
84″	138.22	0.18	0.98	2.59	5.07	8.44	12.69	17.78	23.60	30.11	37.19	44.75	52.67	60.83	69.11	......	......				
90″	170.01	0.18	1.00	2.68	5.33	8.91	13.44	18.86	25.12	32.18	39.90	48.22	56.99	66.14	75.52	85.00	.....				
96″	206.32	0.20	1.07	2.83	5.59	9.36	14.14	19.90	26.60	34.17	42.52	51.53	61.13	71.22	81.66	92.34	103.16				
102″	247.48	0.22	1.14	3.01	5.89	9.87	14.92	21.01	28.11	36.18	45.19	54.91	65.31	76.29	87.73	99.56	111.59	123.74	......	......	......
108″	293.77	0.20	1.13	3.03	6.04	10.21	15.50	21.93	29.47	38.03	47.56	57.97	69.14	81.05	93.53	106.47	119.76	133.26	146.88	......	......
114″	345.51	0.21	1.16	3.12	6.25	10.55	16.06	22.80	30.70	39.73	49.81	60.88	72.85	85.61	99.05	113.07	127.56	142.41	157.51	172.75	.....
120″	402.27	0.21	1.19	3.23	6.47	10.93	16.68	23.70	31.96	41.43	52.04	63.73	76.40	89.95	104.32	119.39	135.04	151.15	167.62	184.32	201.13

To ascertain the contents of a head over one-half full: Let h = depth of unfilled portion. Find from the table the quantity corresponding to a depth h. Subtract this quantity from the contents of a full head.

Contents in U.S. gallons for one head only. This table is only approximate, but close enough for practical use.
By permission, The Permutit Co., Inc., Data Book, 1953.

A-22.

Miscellaneous Formulas
(Courtesy of Chicago Bridge and Iron Co.)

1. Area of Roofs.

Umbrella Roofs:

D = diameter of tank in feet.

Surface area in square feet $\left\{ \begin{array}{l} = 0.842\ D^2 \text{ (when radius = diameter)} \\ = 0.882\ D^2 \text{ (when radius = 0.8 diameter)} \end{array} \right.$

Conical Roofs:

Surface area in square feet $\left\{ \begin{array}{l} = 0.787\ D^2 \text{ (when pitch is } \tfrac{3}{4} \text{ in 12)} \\ = 0.792\ D^2 \text{ (when pitch is } 1\tfrac{1}{2} \text{ in 12)} \end{array} \right.$

2. Average weights.

Steel —490 pounds per cubic foot—specific gravity 7.85

Wrought iron —485 pounds per cubic foot—specific gravity 7.77

Cast iron —450 pounds per cubic foot—specific gravity 7.21

1 cubic foot air or gas at 32° F., 760 m.m. barometer = molecular weight x 0.0027855 pounds.

3. Expansion in steel pipe = 0.78 inch per 100 lineal feet per 100 degrees Fahr. change in temperature = 0.412 inch per mile per degree Fahr. temperature change.

4. Linear coefficients of expansion per degree increase in temperature:

	Per Degree Fahrenheit	Per Degree Centigrade
STRUCTURAL STEEL—A-7		
70° to 200° F.	0.0000067	—
21.1° to 93° C.	—	0.0000121
STAINLESS STEEL - TYPE 304		
32° to 932° F.	0.0000102	—
0° to 500° C.	—	0.0000184
ALUMINUM		
−76° to 68° F.	0.0000120	—
−60° to 20° C.	—	0.0000216

5. To determine the net thickness of shells for horizontal cylindrical pressure tanks:

$$T = \frac{6\ PD}{S}$$

P = working pressure in pounds per square inch

D = diameter of cylinder in feet

S = allowable unit working stress in pounds per square inch

T = Net thickness in inches

Resulting net thickness must be corrected to gross or actual thickness by dividing by joint efficiency.

6. To determine the net thickness of heads for cylindrical pressure tanks:

(6a) Ellipsoidal or Bumped Heads:

$$T = \frac{6\ PD}{S}$$

T, P and D as in formula 5

(6b) Dished or Basket Heads:

$$T = \frac{10.6P(MR)}{S}$$

T, S and P as in formula 5

MR = principal radius of head in feet

Resulting net thickness of heads is both net and gross thickness if one piece seamless heads are used, otherwise net thickness must be corrected to gross thickness as above.

Formulas 5 and 6 must often be modified to comply with various engineering codes, and state and municipal regulations. Calculated gross plate thicknesses are sometimes arbitrarily increased to provide an additional allowance for corrosion.

7. Heads for Horizontal Cylindrical Tanks:

Hemi-ellipsoidal Heads have an ellipsoidal cross section, usually with minor axis equal to one half the major axis—that is, depth = ¼ D, or more.

Dished or Basket Heads consist of a spherical segment normally dished to a radius equal to the inside diameter of the tank cylinder (or within a range of 6 inches plus or minus) and connected to the straight cylindrical flange by a "knuckle" whose inside radius is usually not less than 6 per cent of the inside diameter of the cylinder nor less than 3 times the thickness of the head plate. Basket heads closely approximate hemi-ellipsoidal heads.

Bumped Heads consist of a spherical segment joining the tank cylinder directly without the transition "knuckle." The radius = D, or less. This type of head is used only for pressures of 10 pounds per square inch or less, excepting where a compression ring is placed at the junction of head and shell.

Surface Area of Heads:

(7a) Hemi-ellipsoidal Heads:

$$S = \pi R^2 [1 + K^2(2-K)]$$

S = surface area in square feet

R = radius of cylinder in feet

K = ratio of the depth of the head (not including the straight flange) to the radius of the cylinder

The above formula is not exact but is within limits of practical accuracy.

(7b) Dished or Basket Heads:
Formula (7a) gives surface area within practical limits.

(7c) Bumped Heads:

$$S = \pi R^2 (1 + K^2)$$

S, R, and K as in formula (7a)

Volume of Heads:

(7d) Hemi-ellipsoidal Heads:

$$V = \tfrac{2}{3} \pi K R^3$$

R = radius of cylinder in feet

K = ratio of the depth of the head (not including the straight flange) to the radius of the cylinder

(7e) Dished or Basket Heads:
Formula (7d) gives volume within practical limits.

(7f) Bumped Heads:

$$V = \tfrac{1}{2} \pi K R^3 (1 + \tfrac{1}{3} K^2)$$

V, K and R as in formula (7d)

Note: K in above formulas may be determined as follows:
Hemi-ellipsoidal heads—K is known

Dished Heads—$K = M - \sqrt{(M-1)(M+1-2m)}$

Bumped Heads—$K = [M - \sqrt{M^2-1}]$

MR = principal radius of head in feet

mR = radius of knuckle in feet

R = radius of cylinder in feet

$$M = \frac{MR}{R} \qquad m = \frac{mR}{R}$$

For bumped heads m = o

8. Total volume or length of shell in cylindrical tank with ellipsiodal or hemispherical heads:

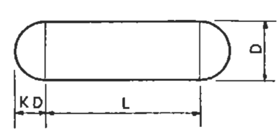
V = Total volume

L = Length of cylindrical shell

KD = Depth of head

$$V = \frac{\pi D^2}{4} (L + 1\tfrac{1}{3}\ KD)$$

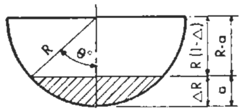
$$L = (V \div \frac{\pi D^2}{4}) - 1\tfrac{1}{3}\ KD$$



A-22.
(Continued). Miscellaneous Formulas

9. Volume or contents of partially filled horizontal cylindrical tanks:

(9a) Tank cylinder or shell (straight portion only)



$$Q = R^2L\left[\left(\frac{\pi\ \Theta^\circ}{180}\right) - \sin\Theta\cos\Theta\right]$$

Q = partially filled volume or contents in cubic feet

R = radius of cylinder in feet

L = length of straight portion of cylinder in feet

The straight portion or flange of the heads must be considered a part of the cylinder. The length of flange depends upon the diameter of tank and thickness of head but ranges usually between 2 and 4 inches.

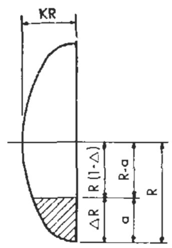
$a = \Delta R$ = depth of liquid in feet

$\Delta = \frac{a}{R}$ = a ratio

$\cos\Theta = 1 - \Delta$, or $\frac{R-a}{R}$

Θ = degrees

(9b) Hemi-ellipsoidal Heads:



$Q = \frac{3}{4} V \Delta^2 (1 - \frac{1}{3}\Delta)$

Q = partially filled volume or contents in cubic feet

V = total volume of one head per formula (7d)

$\Delta = \frac{a}{R}$ = a ratio

$a = \Delta R$ = depth of liquid in feet

R = radius of cylinder in feet

(9c) Dished or Basket Heads:
Formula (9b) gives partially filled volume within practical limits, and formula (7d) gives V within practical limits.

(9d) Bumped Heads:
Formula (9b) gives partially filled volume within practical limits, and formula (7f) gives V.

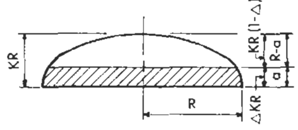
Note: To obtain the volume or quantity of liquid in partially filled tanks, add the volume per formula (9a) for the cylinder or straight portion to twice (for 2 heads) the volume per formula (9b), (9c) or (9d) for the type of head concerned.

10. Volume or contents of partially filled hemi-ellipsoidal heads with major axis vertical:

Q = Partially filled volume or contents in cubic feet

V = Total volume of one head per formula (7d)

R = Radius of cylinder in feet

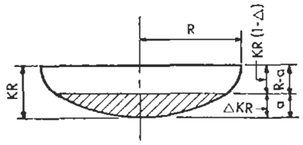


(10a) Upper Head:

$Q = 1\frac{1}{2} V \Delta (1 - \frac{1}{3}\Delta^2)$

$\Delta = \frac{a}{KR}$ = a ratio

$a = \Delta KR$ = depth of liquid in feet



(10b) Lower Head:

$Q = 1\frac{1}{2} V \Delta^2 (1 - \frac{1}{3}\Delta)$

$\Delta = \frac{a}{KR}$ = a ratio

$a = \Delta KR$ = depth of liquid in feet

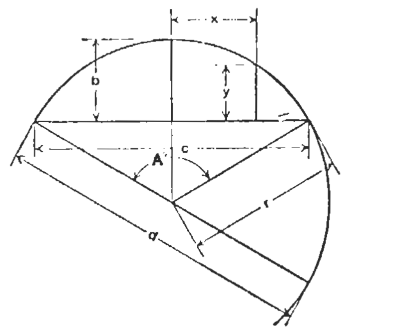
A-23.
Decimal Equivalents in Inches, Feet and Millimeters

In. Equiv. for Decimal of In.	*Decimals*	*Millimeter Equiv. for Decimal of In.*	*In. Equiv. for Decimal of Ft.*
1/64	.0156	0.397	3/16
1/32	.0313	0.794	3/8
3/64	.0469	1.191	9/16
1/16	.0625	1.588	3/4
5/64	.0781	1.984	15/16
3/32	.0938	2.381	1 1/8
7/64	.1094	2.778	1 5/16
1/8	.1250	3.175	1 1/2
9/64	.1406	3.572	1 11/16
5/32	.1563	3.969	1 7/8
11/64	.1719	4.366	2 1/16
3/16	.1875	4.763	2 1/4
13/64	.2031	5.159	2 7/16
7/32	.2188	5.556	2 5/8
15/64	.2344	5.953	2 13/16
1/4	.2500	6.350	3
17/64	.2656	6.747	3 3/16
9/32	.2813	7.144	3 3/8
19/64	.2969	7.541	3 9/16
5/16	.3125	7.938	3 3/4
21/64	.3281	8.334	3 15/16
11/32	.3438	8.731	4 1/8
23/64	.3594	9.128	4 5/16
3/8	.3750	9.525	4 1/2
25/64	.3906	9.922	4 11/16
13/32	.4063	10.319	4 7/8
27/64	.4219	10.716	5 1/16
7/16	.4375	11.113	5 1/4
29/64	.4531	11.509	5 7/16
15/32	.4688	11.906	5 5/8
31/64	.4844	12.303	5 13/16
1/2	.5000	12.700	6
33/64	.5156	13.097	6 3/16
17/32	.5313	13.494	6 3/8
35/64	.5469	13.891	6 9/16
9/16	.5625	14.288	6 3/4
37/64	.5781	14.684	6 15/16
19/32	.5938	15.081	7 1/8
39/64	.6094	15.478	7 5/16
5/8	.6250	15.875	7 1/2
41/64	.6406	16.272	7 11/16
21/32	.6563	16.669	7 7/8
43/64	.6719	17.066	8 1/16
11/16	.6875	17.463	8 1/4
45/64	.7031	17.859	8 7/16
23/32	.7188	18.256	8 5/8
47/64	.7344	18.653	8 13/16
3/4	.7500	19.050	9
49/64	.7656	19.447	9 3/16
25/32	.7813	19.844	9 3/8
51/64	.7969	20.241	9 9/16
13/16	.8125	20.638	9 3/4
53/64	.8281	21.034	9 15/16
27/32	.8438	21.431	10 1/8
55/64	.8594	21.828	10 5/16
7/8	.8750	22.225	10 1/2
57/64	.8906	22.622	10 11/16
29/32	.9063	23.019	10 7/8
59/64	.9219	23.416	11 1/16
15/16	.9375	23.813	11 1/4
61/64	.9531	24.209	11 7/16
31/32	.9688	24.606	11 5/8
63/64	.9844	25.003	11 13/16
1	1.0000	25.400	12

A-24.

(By permission of Buffalo Tank Div., Bethlehem Steel Corp.)

PROPERTIES OF THE CIRCLE



Circumference $= 6.28318\ r = 3.14159\ d$

Diameter $= 0.31831$ circumference

Area $= 3.14159\ r^2$

Arc $a = \dfrac{\pi r A^\circ}{180^\circ} = 0.017453\ r\ A^\circ$

Angle $A^\circ = \dfrac{180^\circ a}{\pi r} = 57.29578\ \dfrac{a}{r}$

Radius $r = \dfrac{4b^2 + c^2}{8b}$

Chord $c = 2\sqrt{2br - b^2} = 2\ r \sin\dfrac{A}{2}$

Rise $b = r - \tfrac{1}{2}\sqrt{4r^2 - c^2} = \dfrac{c}{2}\tan\dfrac{A}{4}$

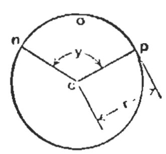
$= 2\ r \sin^2\dfrac{A}{4} = r + y - \sqrt{r^2 - x^2}$

$y = b - r + \sqrt{r^2 - x^2}$

$x = \sqrt{r^2 - (r + y - b)^2}$

Diameter of circle of equal periphery as square = 1.27324 side of square
Side of square of equal periphery as circle = 0.78540 diameter of circle
Diameter of circle circumscribed about square = 1.41421 side of square
Side of square inscribed in circle = 0.70711 diameter of circle

CIRCULAR SECTOR



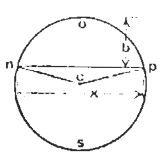
r = radius of circle y = angle ncp in degrees

Area of Sector ncpo = ½ (length of arc nop × r)

= Area of Circle $\times \dfrac{y}{360}$

$= 0.0087266 \times r^2 \times y$

CIRCULAR SEGMENT



r = radius of circle x = chord b = rise

Area of Segment nop = Area of Sector ncpo — Area of triangle ncp

$= \dfrac{(\text{Length of arc nop} \times r) - x\,(r - b)}{2}$

Area of Segment nsp = Area of Circle — Area of Segment nop

VALUES FOR FUNCTIONS OF π

$\pi = 3.14159265359$, log $= 0.4971499$

$\pi^2 = 9.8696044$, log $= 0.9942998$	$\dfrac{1}{\pi} = 0.3183099$, log $= \bar{1}.5028501$	$\sqrt{\dfrac{1}{\pi}} = 0.5641896$, log $= \bar{1}.7514251$
$\pi^3 = 31.0062767$, log $= 1.4914497$	$\dfrac{1}{\pi^2} = 0.1013212$, log $= \bar{1}.0057002$	$\dfrac{\pi}{180} = 0.0174533$, log $= \bar{2}.2418774$
$\sqrt{\pi} = 1.7724539$, log $= 0.2485749$	$\dfrac{1}{\pi^3} = 0.0322515$, log $= \bar{2}.5085503$	$\dfrac{180}{\pi} = 57.2957795$, log $= 1.7581226$

A-24.

(Continued).

AREA OF PLANE FIGURES

Triangle: Base x ½ perpendicular height.

$\sqrt{s(s-a)\,(s-b)\,(s-c)}$,
$s = \tfrac{1}{2}$ sum of the three sides a, b and c.

Trapezium: Sum of area of the two triangles.

Trapezoid: ½ sum of parallel sides x perpendicular height.

Parallelogram: Base x perpendicular height.

Regular Polygon: ½ sum of sides x inside radius.

Circle: $\pi r^2 = 0.78540 \times \text{dia.}^2 = 0.07958 \times \text{circumference}^2$

Sector of Circle: $\dfrac{\pi r^2 A^\circ}{360} = 0.0087266 r^2 A^\circ$ = arc x ½ radius.

Segment of Circle: $\dfrac{r^2}{2}\left(\dfrac{\pi A^\circ}{180} - \sin A^\circ\right)$

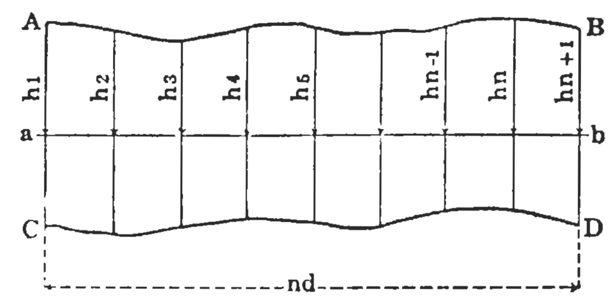
Circle of same area as square: diameter = side x 1.12838

Square of same area as circle: side = diameter x 0.88623

Ellipse: Long diameter x short diameter x 0.78540

Parabola: Base x ⅔ perpendicular height.

Irregular plane surface

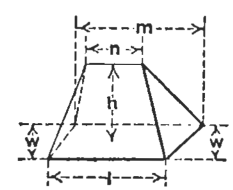


Divide any plane surface A, B, C, D, along a line a-b into an even number, n, of parallel and sufficiently small strips, d, whose ordinates are $h_1, h_2, h_3, h_4, h_5 \ldots h_{n-1}, h_n, h_{n+1}$, and considering contours between three ordinates as parabolic curves, then for section ABCD,

$$\text{Area} = \frac{d}{3}\left[h_1 + h_{n+1} + 4(h_2 + h_4 + h_6 \ldots + h_n) + 2(h_3 + h_5 + h_7 \ldots + h_{n-1})\right]$$

or, approximately, Area = Sum of ordinates x width, d.

VOLUME OF A WEDGE

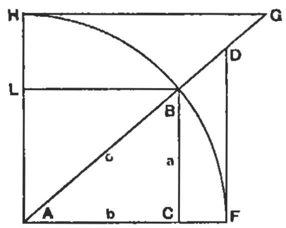


This formula is useful in obtaining the contents of special, wedge-shaped, tank bottoms.

$$\text{Volume} = \frac{wh}{6}\,(l + m + n)$$

TRIGONOMETRIC FORMULAS

TRIGONOMETRIC FUNCTIONS



Radius AF $= 1$
$= \sin^2 A + \cos^2 A = \sin A \operatorname{cosec} A$
$= \cos A \sec A = \tan A \cot A$

Sine A $= \dfrac{\cos A}{\cot A} = \dfrac{1}{\operatorname{cosec} A} = \cos A \tan A = \sqrt{1-\cos^2 A} =$ BC

Cosine A $= \dfrac{\sin A}{\tan A} = \dfrac{1}{\sec A} = \sin A \cot A = \sqrt{1-\sin^2 A} =$ AC

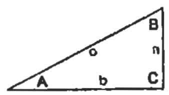
Tangent A $= \dfrac{\sin A}{\cos A} = \dfrac{1}{\cot A} = \sin A \sec A$ = FD

Cotangent A $= \dfrac{\cos A}{\sin A} = \dfrac{1}{\tan A} = \cos A \operatorname{cosec} A$ = HG

Secant A $= \dfrac{\tan A}{\sin A} = \dfrac{1}{\cos A}$ = AD

Cosecant A $= \dfrac{\cot A}{\cos A} = \dfrac{1}{\sin A}$ = AG

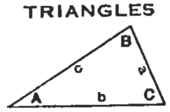
RIGHT ANGLED TRIANGLES



$a^2 = c^2 - b^2$
$b^2 = c^2 - a^2$
$c^2 = a^2 + b^2$

Known	Required: A	B	a	b	c	Area
a, b	$\tan A = \frac{a}{b}$	$\tan B = \frac{b}{a}$			$\sqrt{a^2+b^2}$	$\frac{ab}{2}$
a, c	$\sin A = \frac{a}{c}$	$\cos B = \frac{a}{c}$		$\sqrt{c^2-a^2}$		$\frac{a\sqrt{c^2-a^2}}{2}$
A, a		$90° - A$		$a \cot A$	$\frac{a}{\sin A}$	$\frac{a^2 \cot A}{2}$
A, b		$90° - A$	$b \tan A$		$\frac{b}{\cos A}$	$\frac{b^2 \tan A}{2}$
A, c		$90° - A$	$c \sin A$	$c \cos A$		$\frac{c^2 \sin 2A}{4}$

OBLIQUE ANGLED TRIANGLES



$$s = \frac{a+b+c}{2}$$

$a^2 = b^2 + c^2 - 2\,bc \cos A$
$b^2 = a^2 + c^2 - 2\,ac \cos B$
$c^2 = a^2 + b^2 - 2\,ab \cos C$

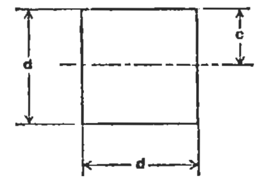
Known	Required: A	B	C	b	c	Area
a, b, c	$\cos \frac{1}{2} A = \sqrt{\frac{s(s-a)}{bc}}$	$\cos \frac{1}{2} B = \sqrt{\frac{s(s-b)}{ac}}$	$\cos \frac{1}{2} C = \sqrt{\frac{s(s-c)}{ab}}$			$\sqrt{s(s-a)(s-b)(s-c)}$
a, A, B			$180° - (A+B)$	$\frac{a \sin B}{\sin A}$	$\frac{a \sin C}{\sin A}$	
a, b, A		$\sin B = \frac{b \sin A}{a}$			$\frac{b \sin C}{\sin B}$	
a, b, C	$\tan A = \frac{a \sin C}{b - a \cos C}$				$\sqrt{a^2+b^2-2ab\cos C}$	$\frac{ab \sin C}{2}$

A-24.
(Continued).

PROPERTIES OF SECTIONS

SQUARE

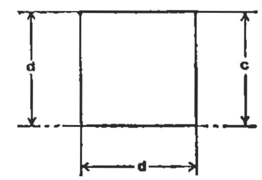
Axis of moments through center



$A = d^2$
$c = \dfrac{d}{2}$
$I = \dfrac{d^4}{12}$
$S = \dfrac{d^3}{6}$
$r = \dfrac{d}{\sqrt{12}} = .288675\ d$

SQUARE

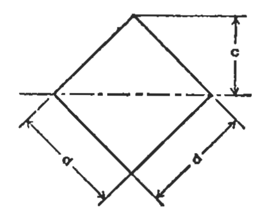
Axis of moments on base



$A = d^2$
$c = d$
$I = \dfrac{d^4}{3}$
$S = \dfrac{d^3}{3}$
$r = \dfrac{d}{\sqrt{3}} = .577350\ d$

SQUARE

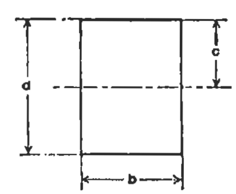
Axis of moments on diagonal



$A = d^2$
$c = \dfrac{d}{\sqrt{2}} = .707107\ d$
$I = \dfrac{d^4}{12}$
$S = \dfrac{d^3}{6\sqrt{2}} = .117851\ d^3$
$r = \dfrac{d}{\sqrt{12}} = .288675\ d$

RECTANGLE

Axis of moments through center

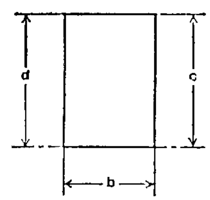


$A = bd$
$c = \dfrac{d}{2}$
$I = \dfrac{bd^3}{12}$
$S = \dfrac{bd^2}{6}$
$r = \dfrac{d}{\sqrt{12}} = .288675\ d$

A-24.
(Continued).

PROPERTIES OF SECTIONS

RECTANGLE
Axis of moments on base



$A = bd$

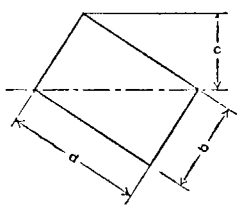
$c = d$

$I = \frac{bd^3}{3}$

$S = \frac{bd^2}{3}$

$r = \frac{d}{\sqrt{3}} = .577350\ d$

RECTANGLE
Axis of moments on diagonal



$A = bd$

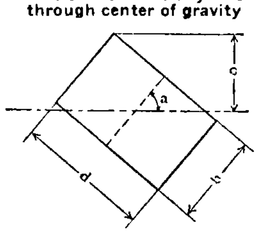
$c = \frac{bd}{\sqrt{b^2 + d^2}}$

$I = \frac{b^3d^3}{6\,(b^2 + d^2)}$

$S = \frac{b^2d^2}{6\sqrt{b^2 + d^2}}$

$r = \frac{bd}{\sqrt{6\,(b^2 + d^2)}}$

RECTANGLE
Axis of moments any line through center of gravity



$A = bd$

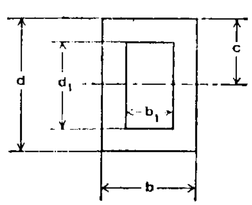
$c = \frac{b \sin a + d \cos a}{2}$

$I = \frac{bd\,(b^2 \sin^2 a + d^2 \cos^2 a)}{12}$

$S = \frac{bd\,(b^2 \sin^2 a + d^2 \cos^2 a)}{6\,(b \sin a + d \cos a)}$

$r = \sqrt{\frac{b^2 \sin^2 a + d^2 \cos^2 a}{12}}$

HOLLOW RECTANGLE
Axis of moments through center



$A = bd - b_1d_1$

$c = \frac{d}{2}$

$I = \frac{bd^3 - b_1d_1^3}{12}$

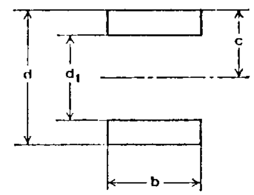
$S = \frac{bd^3 - b_1d_1^3}{6d}$

$r = \sqrt{\frac{bd^3 - b_1d_1^3}{12\,A}}$

A-24.
(Continued).

PROPERTIES OF SECTIONS

EQUAL RECTANGLES
Axis of moments through center of gravity



$A = b\,(d - d_1)$

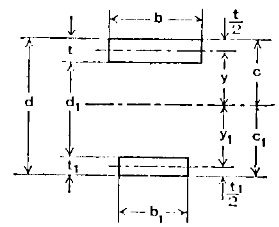
$c = \frac{d}{2}$

$I = \frac{b\,(d^3 - d_1^3)}{12}$

$S = \frac{b\,(d^3 - d_1^3)}{6d}$

$r = \sqrt{\frac{d^3 - d_1^3}{12(d - d_1)}}$

UNEQUAL RECTANGLES
Axis of moments through center of gravity



$A = bt + b_1t_1$

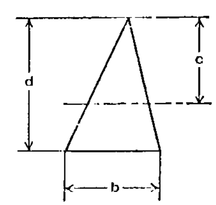
$c = \frac{\frac{1}{2} bt^2 + b_1t_1\,(d - \frac{1}{2} t_1)}{A}$

$I = \frac{bt^3}{12} + bty^2 + \frac{b_1t_1^3}{12} + b_1t_1y_1^2$

$S = \frac{I}{c}$ $\quad S_1 = \frac{I}{c_1}$

$r = \sqrt{\frac{I}{A}}$

TRIANGLE
Axis of moments through center of gravity



$A = \frac{bd}{2}$

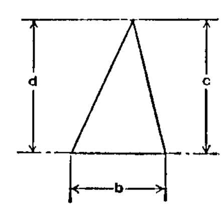
$c = \frac{2d}{3}$

$I = \frac{bd^3}{36}$

$S = \frac{bd^2}{24}$

$r = \frac{d}{\sqrt{18}} = .235702\ d$

TRIANGLE
Axis of moments on base



$A = \frac{bd}{2}$

$c = d$

$I = \frac{bd^3}{12}$

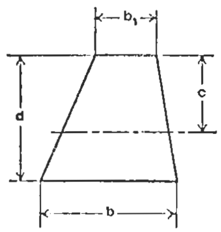
$S = \frac{bd^2}{12}$

$r = \frac{d}{\sqrt{6}} = .408248\ d$

PROPERTIES OF SECTIONS

TRAPEZOID

Axis of moments through center of gravity



$$A = \frac{d(b + b_1)}{2}$$

$$c = \frac{d(2b + b_1)}{3(b + b_1)}$$

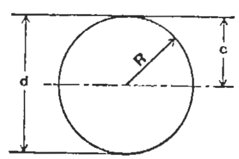
$$I = \frac{d^3 (b^2 + 4\,bb_1 + b_1^2)}{36\,(b + b_1)}$$

$$S = \frac{d^2 (b^2 + 4\,bb_1 + b_1^2)}{12\,(2b + b_1)}$$

$$r = \frac{d}{6(b + b_1)} \sqrt{2\,(b^2 + 4\,bb_1 + b_1^2)}$$

CIRCLE

Axis of moments through center



$$A = \frac{\pi d^2}{4} = \pi R^2 = .785398\ d^2 = 3.141593\ R^2$$

$$c = \frac{d}{2} = R$$

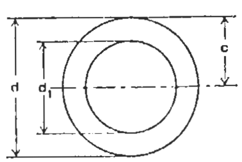
$$I = \frac{\pi d^4}{64} = \frac{\pi R^4}{4} = .049087\ d^4 = .785398\ R^4$$

$$S = \frac{\pi d^3}{32} = \frac{\pi R^3}{4} = .098175\ d^3 = .785398\ R^3$$

$$r = \frac{d}{4} = \frac{R}{2}$$

HOLLOW CIRCLE

Axis of moments through center



$$A = \frac{\pi(d^2 - d_1^2)}{4} = .785398\ (d^2 - d_1^2)$$

$$c = \frac{d}{2}$$

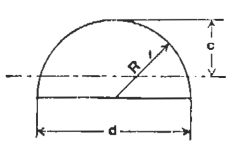
$$I = \frac{\pi(d^4 - d_1^4)}{64} = .049087\ (d^4 - d_1^4)$$

$$S = \frac{\pi(d^4 - d_1^4)}{32d} = .098175\ \frac{d^4 - d_1^4}{d}$$

$$r = \frac{\sqrt{d^2 + d_1^2}}{4}$$

HALF CIRCLE

Axis of moments through center of gravity



$$A = \frac{\pi R^2}{2} = 1.570796\ R^2$$

$$c = R\left(1 - \frac{4}{3\pi}\right) = .575587\ R$$

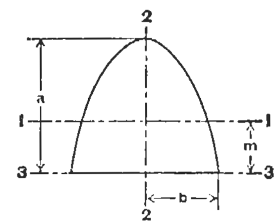
$$I = R^4\left(\frac{\pi}{8} - \frac{8}{9\pi}\right) = .109757\ R^4$$

$$S = \frac{R^3}{24}\ \frac{(9\pi^2 - 64)}{(3\pi - 4)} = .190687\ R^3$$

$$r = R\ \frac{\sqrt{9\pi^2 - 64}}{6\pi} = .264336\ R$$

PROPERTIES OF SECTIONS

PARABOLA



$$A = \frac{4}{3}\,ab$$

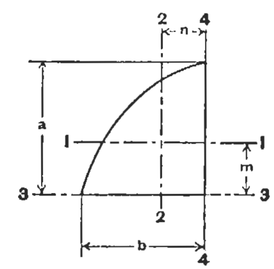
$$m = \frac{2}{5}\,a$$

$$I_1 = \frac{16}{175}\,a^3b$$

$$I_2 = \frac{4}{15}\,ab^3$$

$$I_3 = \frac{32}{105}\,a^3b$$

HALF PARABOLA



$$A = \frac{2}{3}\,ab$$

$$m = \frac{2}{5}\,a$$

$$n = \frac{3}{8}\,b$$

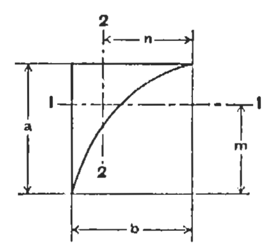
$$I_1 = \frac{8}{175}\,a^3b$$

$$I_2 = \frac{19}{480}\,ab^3$$

$$I_3 = \frac{16}{105}\,a^3b$$

$$I_4 = \frac{2}{15}\,ab^3$$

COMPLEMENT OF HALF PARABOLA



$$A = \frac{1}{3}\,ab$$

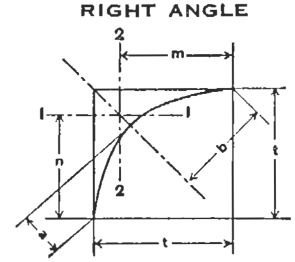
$$m = \frac{7}{10}\,a$$

$$n = \frac{3}{4}\,b$$

$$I_1 = \frac{37}{2100}\,a^3b$$

$$I_2 = \frac{1}{80}\,ab^3$$

PARABOLIC FILLET IN RIGHT ANGLE



$$a = \frac{t}{2\sqrt{2}}$$

$$b = \frac{t}{\sqrt{2}}$$

$$A = \frac{1}{6}\,t^2$$

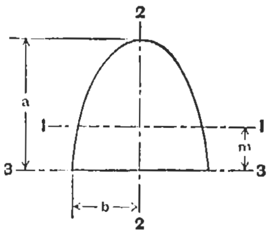
$$m = n = \frac{4}{5}\,t$$

$$I_1 = I_2 = \frac{11}{2100}\,t^4$$

A-24. (Continued).

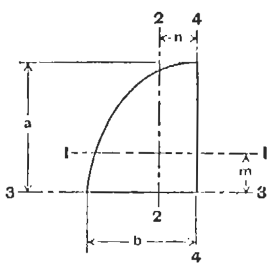
PROPERTIES OF SECTIONS

* HALF ELLIPSE



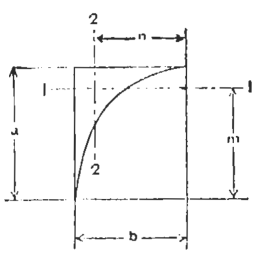
$$A = \frac{1}{2}\pi ab$$
$$m = \frac{4a}{3\pi}$$
$$I_1 = a^3b\left(\frac{\pi}{8} - \frac{8}{9\pi}\right)$$
$$I_2 = \frac{1}{8}\pi ab^3$$
$$I_3 = \frac{1}{8}\pi a^3b$$

* QUARTER ELLIPSE



$$A = \frac{1}{4}\pi ab$$
$$m = \frac{4a}{3\pi}$$
$$n = \frac{4b}{3\pi}$$
$$I_1 = a^3b\left(\frac{\pi}{16} - \frac{4}{9\pi}\right)$$
$$I_2 = ab^3\left(\frac{\pi}{16} - \frac{4}{9\pi}\right)$$
$$I_3 = \frac{1}{16}\pi a^3b$$
$$I_4 = \frac{1}{16}\pi ab^3$$

* ELLIPTIC COMPLEMENT



$$A = ab\left(1 - \frac{\pi}{4}\right)$$
$$m = \frac{a}{6\left(1 - \frac{\pi}{4}\right)}$$
$$n = \frac{b}{6\left(1 - \frac{\pi}{4}\right)}$$
$$I_1 = a^3b\left(\frac{1}{3} - \frac{\pi}{16} - \frac{1}{36\left(1 - \frac{\pi}{4}\right)}\right)$$
$$I_2 = ab^3\left(\frac{1}{3} - \frac{\pi}{16} - \frac{1}{36\left(1 - \frac{\pi}{4}\right)}\right)$$

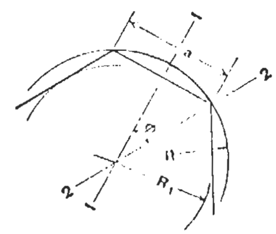
* To obtain properties of half circle, quarter circle and circular complement substitute a = b = R.

A-24. (Concluded).

PROPERTIES OF SECTIONS

REGULAR POLYGON

Axis of moments through center

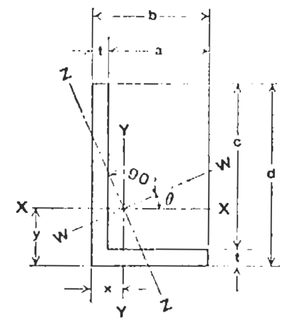


n = Number of sides

$$\phi = \frac{180°}{n}$$
$$a = 2\sqrt{R^2 - R_1^2}$$
$$R = \frac{a}{2\sin\phi}$$
$$R_1 = \frac{a}{2\tan\phi}$$
$$A = \frac{1}{4}na^2\cot\phi = \frac{1}{2}nR^2\sin 2\phi = nR_1^2\tan\phi$$
$$I_1 = I_2 = \frac{A(6R^2 - a^2)}{24} = \frac{A(12R_1^2 + a^2)}{48}$$
$$r_1 = r_2 = \sqrt{\frac{6R^2 - a^2}{24}} = \sqrt{\frac{12R_1^2 + a^2}{48}}$$

ANGLE

Axis of moments through center of gravity



Z-Z is axis of minimum I

$$\tan 2\theta = \frac{2K}{I_Y - I_X}$$
$$A = t(b + c) \quad x = \frac{b^2 + ct}{2(b + c)} \quad y = \frac{d^2 + at}{2(b + c)}$$

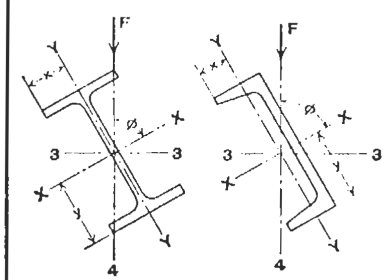
K = Product of Inertia about X-X & Y-Y

$$= \mp \frac{abcdt}{4(b + c)}$$
$$I_X = \frac{1}{3}\left(t(d - y)^3 + by^3 - a(y - t)^3\right)$$
$$I_Y = \frac{1}{3}\left(t(b - x)^3 + dx^3 - c(x - t)^3\right)$$
$$I_Z = I_X\sin^2\theta + I_Y\cos^2\theta + K\sin 2\theta$$
$$I_W = I_X\cos^2\theta + I_Y\sin^2\theta - K\sin 2\theta$$

K is negative when heel of angle, with respect to c. g., is in 1st or 3rd quadrant, positive when in 2nd or 4th quadrant.

BEAMS AND CHANNELS

Transverse force oblique through center of gravity



$$I_3 = I_X\sin^2\phi + I_Y\cos^2\phi$$
$$I_4 = I_X\cos^2\phi + I_Y\sin^2\phi$$
$$f = M\left(\frac{y}{I_X}\sin\phi + \frac{x}{I_Y}\cos\phi\right)$$

where M is bending moment due to force F. Extreme fiber assumed same as for case $\phi = 0$. If not, locate extreme fiber and find f by usual method.

A-25.
Wind Chill Equivalent Temperatures on Exposed Flesh at Varying Velocity

	WIND VELOCITY (MILES PER HOUR)										
	45	35	25	20	15	10	5	3	2	1	0
Temperature, F	90	89.5	89	88.5	88	88.75	87.5	87	86	84.5	83
	82	81	80.5	80	79.5	78	76	74	72.5	70	60
	72	71	69.5	68	67	65	60	57	53.5	47.5	23
	63	61	59	57	55	52	44.5	39	34.5	20	−11
	51	49	47	45	42.5	38	28	18.5	11	0	−27
	41	39	36	34	30.5	25	11	0	−9	−23.5	−38
	30	28	25	23	18	11	−5	−16.5	−40	Below −40	Below −40
	20	18	14	11	6	−2	−19	−40	Below −40	do	do
	10	7.5	3	0	−6	−15	−35	Below −40	do	do	do
	0	−2.5	−8	−12	−18	−29	Below −40	do	do	do	do
	−11	−14	−18	−23	−30	Below −40	do	do	do	do	do
	−21	−24	−30	−35	Below −40	do	do	do	do	do	do
	−32	−35	−40	−40				do	do	do	

Instructions for use of the table:

(1) First obtain the temperature and wind velocity forecast data.
(2) Locate the number at the top corresponding to the expected wind speed (or the number closest to this).
(3) Read down this column until the number corresponding to the expected temperature (or the number closest to this) is reached.
(4) From this point follow across to the right on the same line until the last number is reached under the column marked zero (0) wind speed.
(5) This is the equivalent temperature reading. Example: weather information gives the expected temperature (at a given time, such as midnight) to be 35°F, and the expected wind speed (at the same time, midnight) to be 20 miles per hour (mph). Locate the 20 mph column at the top, follow down this column to the number nearest to 35°F. The nearest number is 34°F. From this point, move all the way to the right on the same line and find the last number, which is −38°F. This means that with a temperature of 35°F, and a windspeed of 20 mph the rate of cooling of all exposed flesh is the same as −38°F, with no wind.

do means ditto.

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A-26.
Impurities in Water

U. S. Systems of Expressing Impurities

1 grain per gallon .... = 1 grain calcium carbonate ($CaCO_3$) per U. S. gallon of water
1 part per million .... = 1 part calcium carbonate ($CaCO_3$) per 1,000,000 parts of water
1 part per hundred thousand .. = 1 part calcium carbonate ($CaCO_3$) per 100,000 parts of water

Foreign Systems of Expressing Impurities

1 English degree (or °Clark) .. = 1 grain calcium carbonate ($CaCO_3$) per British Imperial gal. of water
1 French degree .... = 1 part calcium carbonate ($CaCO_3$) per 100,000 parts of water
1 German degree .... = 1 part calcium oxide (CaO) per 100,000 parts of water

Conversions

CONVERSION TABLE (Expressed to 3 Significant Figures)	Parts CaCo₃ per Million (ppm)	Parts CaCo₃ per Hundred Thousand (Pts./100,000)	Grains CaCo₃ per U.S. Gallon (gpg)	English Degrees or ° Clark	French Degrees — °French	German Degrees — °German	Milli-equivalents per Liter or Equivalents per Million
1 Part per Million	1.	.1	.0583	.07	.1	.0560	.020
1 Part per Hundred Thousand	10.0	1.	.583	.7	1.	.560	.20
1 Grain per U. S. Gallon	17.1	1.71	1.	1.2	1.71	.958	.343
1 English or Clark Degree	14.3	1.43	.833	1.	1.43	.800	.286
1 French Degree	10.	1.	.583	.7	1.	.560	.20
1 German Degree	17.9	1.79	1.04	1.24	1.79	1.	.357
1 Milli-equivalent per Liter or 1 Equivalent per Million	50.	5.	2.92	3.50		2.80	1.

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A-27.
Water Analysis Conversions for Units Employed: Equivalents

WATER ANALYSIS UNITS CONVERSION TABLE (Expressed to 3 Significant Figures)	Parts per Million (ppm)	Milligrams per Liter (mgm/L)	Grams per Liter (grms/L)	Parts per Hundred Thousand (Pts./100,000)	Grains U.S. Gallon (grs/U.S. gal)	Grains per British Imp. Gallon	Kilograins per Cubic Foot (Kgr/cu. ft.)
1 Part per Million	1.	1.	.001	.1	.0583	.07	.0004
1 Milligram per Liter	1.	1.	.001	.1	.0583	.07	.0004
1 Gram per Liter	1000.	1000.	1.	100.	58.3	70.	.436
1 Part per Hundred Thousand	10.	10.	.01	1.	.583	.7	.00436
1 Grain per U.S. Gallon	17.1	17.1	.017	1.71	1.	1.2	.0075
1 Grain per British Imp. Gallon	14.3	14.3	.014	1.43	.833	1.	.0062
1 Kilograin per Cubic Foot	2294.	2294.	2.294	229.4	134.	161.	1.

NOTE: In practice, water analysis samples are measured by volume, not by weight and corrections for variations in specific gravity are practically never made. Therefore, parts per million are assumed to be the same as milligrams per liter and hence the above relationships are, for practical purposes, true.

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A-28.
Parts Per Million to Grains Per U. S. Gallon

A. To convert parts per million of hardness to grains per U. S. gallon, divide by the factor 17.1.

B. To convert grains per U. S. gallons to parts per million of hardness, multiply by the factor 17.1.

Example:

1. $\frac{242 \text{ parts/million}}{17.1} = 14.1$ grains/U. S. gallon
2. 24.3 grains/U. S. gallon $\times$ 17.1 = 416 parts/million

Equivalents

Water analyses may also be expressed as:

(1) Equivalents per million (epm) .... $= \dfrac{\text{No. of ppm of substance present}}{\text{Equivalent weight of substance}}$

(2) Milli equivalents per liter (meq/l) .... = Equivalents per million

(3) Parts per million expressed as $CaCO_3$........... = No. of ppm $CaCO_3$ equivalent to No. of ppm of substance present

(4) Fiftieths of equivalents per million (epm/50) $= \dfrac{\text{No. of ppm of substance present} \times 50}{\text{Equivalent weight of substance}}$

NOTES: Numerically (1) and (2) are equal.
Numerically (3) and (4) are equal.

Section XXIII contains equivalent weights of a number of substances.
Section XXIII contains factors for converting various substances to $CaCO_3$.
Section XXIII contains factors for various chemical conversions.

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A-29.

Formulas, Molecular and Equivalent Weights, and Conversion Factors to $CaCO_3$ of Substances Frequently Appearing in the Chemistry of Water Softening

Substance	Formula	Molecular weight	Equivalent weight	Multiplying Factor (Considering molecular wt. of $CaCO_3$ as 100.) Substance to $CaCO_3$ equivalent	$CaCO_3$ equivalent to substance
Aluminum	Al	27.0	9.0	5.56	0.18
Aluminum Chloride	$AlCl_3$	133.	44.4	1.13	0.89
Aluminum Chloride	$AlCl_3 \cdot 6H_2O$	241.	80.5	0.62	1.61
Aluminum Sulfate	$Al_2(SO_4)_3 \cdot 18H_2O$	666.4	111.1	0.45	2.22
Aluminum Sulfate	$Al_2(SO_4)_3$ (anhydrous)	342.1	57.0	0.88	1.14
Aluminum Hydrate	$Al(OH)_3$	78.0	26.0	1.92	0.52
Alumina	Al_2O_3	101.9	17.0	2.94	0.34
Sodium Aluminate	$Na_2Al_2O_4$	163.9	27.3	1.83	0.55
Ammonium Alum	$Al_2(SO_4)_3 (NH_4)_2SO_4 \cdot 24H_2O$	906.6	151.1	0.33	3.02
Potassium Alum	$Al_2(SO_4)_3K_2SO_4 \cdot 24H_2O$	948.8	156.1	0.32	3.12
Ammonia	NH_3	17.0	17.0	2.94	0.34
Ammonium (Ion)	NH_4	18.0	18.0	2.78	0.36
Ammonium Chloride	NH_4Cl	53.5	53.5	0.94	1.07
Ammonium Hydroxide	NH_4OH	35.1	35.1	1.43	0.70
Ammonium Sulfate	$(NH_4)_2SO_4$	132.	66.1	0.76	1.32
Barium	Ba	137.4	68.7	0.73	1.37
Barium Carbonate	$BaCO_3$	197.4	98.7	0.51	1.97
Barium Chloride	$BaCl_2 \cdot 2H_2O$	244.3	122.2	0.41	2.44
Barium Hydroxide	$Ba(OH)_2$	171.	85.7	0.59	1.71
Barium Oxide	BaO	153.	76.7	0.65	1.53
Barium Sulfate	$BaSO_4$	233.4	116.7	0.43	2.33
Calcium	Ca	40.1	20.0	2.50	0.40
Calcium Bicarbonate	$Ca(HCO_3)_2$	162.1	81.1	0.62	1.62
Calcium Carbonate	$CaCO_3$	100.08	50.1	1.00	1.00
Calcium Chloride	$CaCl_2$	111.0	55.5	0.90	1.11
Calcium Hydrate	$Ca(OH)_2$	74.1	37.1	1.35	0.74
Calcium Hypochlorite	$Ca(ClO)_2$	143.1	35.8	0.70	1.43
Calcium Oxide	CaO	56.1	28.0	1.79	0.56
Calcium Sulfate	$CaSO_4$ (anhydrous)	136.1	68.1	0.74	1.36
Calcium Sulfate	$CaSO_4 \cdot 2H_2O$ (gypsum)	172.2	86.1	0.58	1.72
Calcium Nitrate	$Ca(NO_3)_2$	164.1	82.1	0.61	1.64
Calcium Phosphate	$Ca_3(PO_4)_2$	310.3	51.7	0.97	1.03
Carbon	C	12.0	3.00	16.67	0.06
Chlorine (Ion)	Cl	35.5	35.5	1.41	0.71
Copper (Cupric)	Cu	63.6	31.8	1.57	0.64
Copper Sulfate (Cupric)	$CuSO_4$	160.	80.0	0.63	1.60
Copper Sulfate (Cupric)	$CuSO_4 \cdot 5H_2O$	250.	125.	0.40	2.50
Iron (Ferrous)	Fe''	55.8	27.9	1.79	0.56
Iron (Ferric)	Fe'''	55.8	18.6	2.69	0.37
Ferrous Carbonate	$FeCO_3$	116.	57.9	0.86	1.16
Ferrous Hydroxide	$Fe(OH)_2$	89.9	44.9	1.11	0.90
Ferrous Oxide	FeO	71.8	35.9	1.39	0.72
Ferrous Sulfate	$FeSO_4$ (anhydrous)	151.9	76.0	0.66	1.52
Ferrous Sulfate	$FeSO_4 \cdot 7H_2O$	278.0	139.0	0.36	2.78
Ferrous Sulfate	$FeSO_4$ (anhydrous)	151.9	151.9	oxidation	
Ferric Chloride	$FeCl_3$	162.	54.1	0.93	1.08
Ferric Chloride	$FeCl_3 \cdot 6H_2O$	270.	90.1	0.56	1.80
Ferric Hydroxide	$Fe(OH)_3$	107.	35.6	1.41	0.71
Ferric Oxide	Fe_2O_3	160.	26.6	1.88	0.53
Ferric Sulfate (Ferrisul)	$Fe_2(SO_4)_3$	399.9	66.7	0.75	1.33
Ferrous or Ferric	Fe or Fe	55.8	55.8	oxidation	
Ferrous Sulfate	$FeSO_4$	151.9	151.9	oxidation	
Fluorine	F	19.0	19.0	2.66	0.38
Hydrogen (Ion)	H	1.01	1.01	50.0	0.02
Iodine	I	127.	127.	0.40	2.54
Lead	Pb	207.	104.	0.48	2.08
Magnesium	Mg	24.3	12.2	4.10	0.24
Magnesium Oxide	MgO	40.3	20.2	2.48	0.40
Magnesium Bicarbonate	$Mg(HCO_3)_2$	146.3	73.2	0.68	1.46
Magnesium Carbonate	$MgCO_3$	84.3	42.2	1.19	0.84
Magnesium Chloride	$MgCl_2$	95.2	47.6	1.05	0.95
Magnesium Hydrate	$Mg(OH)_2$	58.3	29.2	1.71	0.58
Magnesium Nitrate	$Mg(NO_3)_2$	148.3	74.2	0.67	1.48
Magnesium Phosphate	$Mg_3(PO_4)_2$	262.9	43.8	1.14	0.88
Magnesium Sulfate	$MgSO_4$	120.4	60.2	0.83	1.20
Manganese (Manganous)	Mn''	54.9	27.5	1.82	0.55
Manganese (Manganic)	Mn'''	54.9	18.3	2.73	0.37
Manganese Chloride	$MnCl_2$	125.8	62.9	0.80	1.26
Manganese Dioxide	MnO_2	86.9	21.7	2.30	0.43
Manganese Hydrate	$Mn(OH)_2$	89.0	44.4	1.13	0.89
Manganic Oxide	Mn_3O_4	158.	26.3	1.90	0.53
Manganous Oxide	MnO	70.9	35.5	1.41	0.71

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(continued on next page)

A-29.

(Concluded). Formulas, Molecular and Equivalent Weights, and Conversion Factors to $CaCO_3$ of Substances Frequently Appearing in the Chemistry of Water Softening

				Multiplying Factor Considering molecular wt. of $CaCO_3$ as 100.	
Substance	*Formula*	*Molecular weight*	*Equivalent weight*	*Substance to $CaCO_3$ equivalent*	*$CaCO_3$ equivalent to substance*
Nitrate (Ion)	NO_3	62.0	62.0	0.81	1.24
Nitric Acid	HNO_3	63.0	63.0	0.79	1.26
Nitrogen (Valence 3)	N'''	14.0	4.67	10.8	0.093
Nitrogen (Valence 5)	N'''''	14.0	2.80	17.9	0.056
Oxygen	O	16.0	8.00	6.25	0.16
Phosphorus (Valence 3)	P'''	31.0	10.3	4.76	0.21
Phosphorus (Valence 5)	P'''''	31.0	6.20	8.33	0.12
Potassium	K	39.1	39.1	1.28	0.78
Potassium Carbonate	K_2CO_3	138.	69.1	0.72	1.38
Potassium Chloride	KCl	74.6	74.6	0.67	1.49
Potassium Hydroxide	KOH	56.1	56.1	0.88	1.12
Silver Chloride	$AgCl$	143.3	143.3	0.35	2.87
Silver Nitrate	$AgNO_3$	169.9	169.9	0.29	3.40
Silica	SiO_2	60.1	30.0	1.67	0.60
Silicon	Si	28.1	7.03	7.14	0.14
Sodium	Na	23.0	23.0	2.18	0.46
Sodium Bicarbonate	$NaHCO_3$	84.0	84.0	0.60	1.68
Sodium Bisulfate	$NaHSO_4$	120.			
Sodium Bisulfite	$NaHSO_3$	104.			
Sodium Carbonate	Na_2CO_3	106.	53.0	0.94	1.06
Sodium Carbonate	$Na_2CO_3 \cdot 10H_2O$	286.	143.	0.35	2.86
Sodium Chloride	$NaCl$	58.5	58.5	0.85	1.17
Sodium Hypochlorite	$NaClO$	74.5	37.3	0.67	1.49
Sodium Hydrate	$NaOH$	40.0	40.0	1.25	0.80
Sodium Nitrate	$NaNO_3$	85.0	85.0	0.59	1.70
Sodium Nitrite	$NaNO_2$	69.0	34.5	0.73	1.38
Sodium Oxide	Na_2O	62.0	31.0	1.61	0.62
Tri-sodium Phosphate	$Na_3PO_4 \cdot 12H_2O$ (18.7% P_2O_5)	380.2	126.7	0.40	2.53
Tri-sodium Phos. (anhydrous)	Na_3PO_4(43.2% P_2O_5)	164.0	54.7	0.91	1.09
Di-sodium Phosphate	$Na_2HPO_4 \cdot 12H_2O$ (19.8% P_2O_5)	358.2	119.4	0.42	2.39
Di-sodium Phos. (anhydrous)	Na_2HPO_4 (50% P_2O_5)	142.0	47.3	1.06	0.95
Mono-sodium Phosphate	$NaH_2PO_4 \cdot H_2O$ (51.4% P_2O_5)	138.1	46.0	1.09	0.92
Mono-sod. phos. (anhydrous)	NaH_2PO_4 (59.1% P_2O_5)	120.0	40.0	1.25	0.80
Meta-Phosphate (Hagan)	$NaPO_3$ (69% P_2O_5)	102.0	34.0	1.47	0.68
Sodium Sulfate	$Na_2SO_4 \cdot 10H_2O$	322.1	161.1	0.31	3.22
Sodium Sulfate	Na_2SO_4	142.1	71.0	0.70	1.42
Sodium Thiosulfate	$Na_2S_2O_3$	158.1	158.1	0.63	1.59
Sodium Tetrathionate	$Na_2S_4O_6$	270.2	135.1	0.37	2.71
Sodium Sulfite	Na_2SO_3	126.1	83.0	0.79	1.27
Sulfur (Valence 2)	S''	32.1	16.0	3.13	0.32
Sulfur (Valence 4)	S''''	32.1	8.02	6.25	0.16
Sulfur (Valence 6)	S''''''	32.1	5.34	9.10	0.11
Sulfur Dioxide	SO_2	64.1	32.0		
Tin	Sn	119.			
Water	H_2O	18.0	9.00	5.56	0.18
Zinc	Zn	65.4	32.7	1.54	0.65
ACID RADICALS					
Bicarbonate	HCO_3	61.0	61.0	0.82	1.22
Carbonate	CO_3	60.0	30.0	1.67	.60
Carbon Dioxide	CO_2	44.0	22.0	2.27	.44
Chloride	Cl	35.5	35.5	1.41	.71
Iodide	I	126.9	126.9	0.40	2.54
Nitrate	NO_3	62.0	62.0	0.81	1.24
Hydrate	OH	17.0	17.0	2.94	0.34
Phosphate	PO_4	95.0	31.7	1.58	0.63
Phosphorous Oxide	P_2O_5	142.0	23.7	2.11	0.47
Sulfide	S	32.1	16.0	3.11	0.32
Sulfate	SO_4	96.1	48.0	1.04	0.96
Sulfur Trioxide	SO_3	80.1	40.0	1.25	0.80
ACIDS					
Hydrogen	H	1.0	1.0	50.00	0.02
Acetic Acid	$HC_2H_3O_2$	60.1	60.1	0.83	1.20
Carbonic Acid	H_2CO_3	62.0	31.0	1.61	0.62
Hydrochloric Acid	HCl	36.5	36.5	1.37	0.73
Phosphoric Acid	H_3PO_4	98.0	32.7	1.53	0.65
Sulfurous Acid	H_2SO_3	82.1	41.1	1.22	0.82
Sulfuric Acid	H_2SO_4	98.1	49.0	1.02	0.98
Hydrogen Sulfide	H_2S	—	—	—	—
Manganous Acid	H_2MnO_3	104.9	52.5	0.95	1.05

A-30.
Grains Per U.S. Gallons—Pounds Per 1000 Gallons

A. To convert grains per U. S. gallons to pounds per 1000 gallons multiply by the factor 0.143.

B. To convert pounds per 1000 gallons to grains per U. S. gallons multiply by the factor 7.0.

Example:

1. 4.5 grains/U. S. gallon × 0.143 = 0.644 lbs./1000 gals.
2. 0.5 lbs./1000 gallons × 7.0 = 3.5 grains/U. S. gal.

A-31.
Parts Per Million—Pounds Per 1000 Gallons

A. To convert parts per million to pounds per 1000 gallons divide by the factor 120.

B. To convert pounds per 1000 gallons to parts per million multiply by the factor 120.

Example:

1. $\frac{39 \text{ parts/million}}{120} = 0.325$ lbs./1000 gals.
2. 0.167 lbs./1000 gals. × 120 = 20 parts/million

A-32.
Coagulant, Acid, and Sulfate—1 ppm Equivalents

1 Ppm Name of Chemical	1 Ppm. Formula of Chemical	ppm Alkalinity Reduction	ppm SO_3 as $CaCO_3$ Increase	ppm Na_2SO_4 Increase	ppm CO_2 Increase	ppm Total Solids Increase
Filter Alum	$Al_2(SO_4)_3 \cdot 18H_2O$	0.45	0.45	0.64	0.40	0.16
Ammonia Alum	$Al_2(SO_4)_3 \cdot (NH_4)_2SO_4 \cdot 24H_2O$	0.33	0.44	0.63	0.29	0.27
Potash Alum	$Al_2(SO_4)_3 \cdot K_2SO_4 \cdot 24H_2O$	0.32	0.43	0.60	0.28	0.30
Copperas (ferrous sulfate)	$FeSO_4 \cdot 7H_2O$	0.36	0.36	0.51	0.31	0.13
Chlorinated Copperas	$FeSO_4 \cdot 7H_2O + (\frac{1}{2}Cl_2)$	0.54	0.36	0.51	0.48	0.18
Ferric Sulfate (100% $Fe_2(SO_4)_3$)	$Fe_2(SO_4)_3$	0.75	0.75	1.07	0.66	0.27
Sulfuric Acid—98%	H_2SO_4	1.00	1.00	1.42	0.88	0.36
Sulfuric Acid—93.2% (66° Be)	H_2SO_4	0.95	0.95	1.35	0.84	0.34
Sulfuric Acid—77.7% (60° Be)	H_2SO_4	0.79	0.79	1.13	0.70	0.28
Salt Cake—95%	Na_2SO_4	—	0.66	0.95	—	1.00

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A-33.
Alkali and Lime—1 ppm Equivalents

Name 1 Ppm	Formula 1 Ppm	Alkalinity A Increase ppm	Free CO_2 Reduction ppm	T.H. as $CaCO_3$ Increase ppm
Sodium Bicarbonate	$NaHCO_3$	0.60	—	—
Soda Ash (56% Na_2O = 99.16% Na_2CO_3)	Na_2CO_3	0.94	0.41	—
Caustic Soda (76% Na_2O = 98.06% NaOH)	NaOH	1.23	1.08	—
Chemical Lime (Quicklime—Usually 90% CaO)	CaO	1.61	1.41	1.61
Hydrated Lime (Usually 93% $Ca(OH)_2$)	$Ca(OH)_2$	1.26	1.11	1.26

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A-34.
Sulfuric, Hydrochloric Acid Equivalent

Name		Formula	Specific Gravity 60°/60°F.	Concentration	Grams/Liter	$CaCO_3$ Equivalent to one lb. Acid	
						Lbs.	Grains
Sulfuric Acid	60° Be	H_2SO_4	1.7059	77.67%	1325	.7926	5548
Sulfuric Acid	66° Be	H_2SO_4	1.8354	93.19%	1710	.9509	6657
Sulfuric Acid	98%	H_2SO_4	1.8407	98.00%	1804	1.0000	7000
Hydrochloric Acid	18% Be	HCl	1.1417	27.92%	319	.3831	2682

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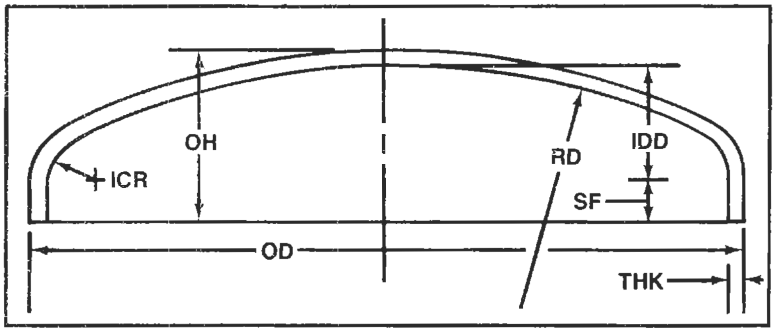
A-35.



For "Overall Height" add length of straight flange to IDD given, plus thickness of material.

ASME FLANGED AND DISHED HEADS IDD CHART

OD - Outside Diameter
THK - Thickness
OH - Overall Height
SF - Straight Flange
RD - Radius of Dish
ICR - Inside Corner Radius
IDD - Inside Depth of Dish

Use when **RD EQUALS DIAMETER**

OD	ICR \ THK	3/16"	1/4"	5/16"	3/8"	7/16"	1/2"	9/16"	5/8"	11/16"	3/4"	7/8"	1"
12	3/4	1.95	1.92	2.00	2.09	On Application							
14	7/8	2.29	2.26	2.27	2.35								
16	1	2.63	2.60	2.57	2.62								
18	1 1/8	2.97	2.94	2.91	2.88								
20	1 1/4	3.31	3.28	3.25	3.22								
22	1 1/2	3.73	3.75	3.67	3.64								
24	1 1/2	4.00	3.97	3.93	3.90								
26	2 1/4	4.72	4.69	4.66	4.63								
28	2 1/4	4.99	4.96	4.92	4.89								
30	2 1/4	5.25	5.22	5.19	5.16								
32	2 1/4	5.52	5.48	5.45	5.42								
34	2 1/4	5.78	5.75	5.72	5.69								
36	2 1/4	6.05	6.02	5.98	5.95								
38	3	6.78	6.74	6.71	6.66	6.65	6.62	6.59	6.56	6.53	6.50	6.44	6.38
40	3	7.04	7.01	6.98	6.95	6.91	6.88	6.85	6.82	6.79	6.76	6.70	6.64
42	3	7.31	7.27	7.24	7.21	7.18	7.15	7.12	7.08	7.05	7.02	6.96	6.90
44	3	7.57	7.54	7.51	7.47	7.44	7.41	7.38	7.35	7.32	7.29	7.22	7.16
46	3	7.84	7.80	7.77	7.74	7.71	7.68	7.64	7.61	7.58	7.55	7.49	7.43
48	3	8.10	8.07	8.04	8.00	7.97	7.94	7.91	7.88	7.85	7.81	7.75	7.69
50	3	8.37	8.34	8.30	8.27	8.24	8.21	8.17	8.14	8.11	8.08	8.02	7.95
52	3 3/4	9.09	9.06	9.03	8.99	8.96	8.93	8.90	8.87	8.84	8.80	8.74	8.68
54	3 3/4	9.35	9.32	9.29	9.26	9.23	9.19	9.16	9.13	9.10	9.07	9.01	8.94
56	3 3/4	9.62	9.59	9.56	9.52	9.49	9.46	9.43	9.40	9.36	9.33	9.27	9.21
58	3 3/4	9.89	9.85	9.82	9.79	9.76	9.72	9.69	9.66	9.63	9.60	9.53	9.47
60	3 3/4	10.15	10.12	10.09	10.05	10.02	9.99	9.96	9.93	9.89	9.86	9.80	9.74
62	3 3/4	10.42	10.39	10.35	10.32	10.29	10.26	10.22	10.19	10.16	10.13	10.06	10.00
64	4 1/4	10.99	10.95	10.92	10.89	10.86	10.83	10.79	10.76	10.73	10.70	10.64	10.57
66	4 1/4	11.25	11.22	11.19	11.16	11.12	11.09	11.06	11.03	10.99	10.96	10.90	10.84
68	4 1/4	11.52	11.49	11.45	11.42	11.39	11.36	11.32	11.29	11.26	11.23	11.16	11.10
70	4 1/4	11.78	11.75	11.72	11.69	11.65	11.62	11.59	11.56	11.53	11.49	11.43	11.37
72	4 3/4	12.35	12.32	12.29	12.26	12.22	12.19	12.16	12.13	12.10	12.06	12.00	11.94
74	4 3/4	12.62	12.59	12.55	12.52	12.49	12.46	12.43	12.39	12.36	12.33	12.27	12.20
76	4 3/4	12.89	12.85	12.82	12.79	12.76	12.72	12.69	12.66	12.63	12.59	12.53	12.47
78	4 3/4	13.15	13.12	13.09	13.05	13.02	12.99	12.96	12.92	12.89	12.86	12.80	12.73
80	5	13.57	13.54	13.50	13.47	13.44	13.41	13.37	13.34	13.31	13.28	13.21	13.15
82	5	13.84	13.80	13.77	13.74	13.70	13.67	13.64	13.61	13.57	13.54	13.48	13.41
84	5 3/4	14.56	14.52	14.49	14.46	14.43	14.39	14.36	14.33	14.30	14.27	14.20	14.14
86	5 3/4	14.82	14.79	14.76	14.72	14.69	14.66	14.63	14.60	14.56	14.53	14.47	14.40
88	5 3/4	15.09	15.05	15.02	14.99	14.96	14.92	14.89	14.86	14.83	14.80	14.73	14.67
90	5 3/4	15.35	15.32	15.29	15.26	15.22	15.19	15.16	15.13	15.09	15.06	15.00	14.93
92	5 3/4	15.62	15.59	15.55	15.52	15.49	15.46	15.42	15.39	15.36	15.33	15.26	15.20
94	5 3/4	15.89	15.85	15.82	15.79	15.75	15.72	15.69	15.66	15.62	15.59	15.53	15.46
96	6 1/2	16.61	16.57	16.54	16.51	16.48	16.44	16.41	16.38	16.35	16.32	16.25	16.19
98	6 1/2	16.87	16.84	16.81	16.77	16.74	16.71	16.68	16.64	16.61	16.58	16.52	16.45
100	6 1/2	17.14	17.10	17.07	17.04	17.01	16.97	16.94	16.91	16.88	16.85	16.78	16.72
102	6 1/2	17.40	17.37	17.34	17.31	17.27	17.24	17.21	17.18	17.14	17.11	17.05	16.98
104	6 1/2	17.67	17.64	17.60	17.57	17.54	17.51	17.47	17.44	17.41	17.38	17.31	17.25
106	6 1/2	17.94	17.90	17.87	17.84	17.80	17.77	17.74	17.71	17.67	17.64	17.58	17.51
108	6 1/2	18.20	18.17	18.14	18.10	18.07	18.04	18.00	17.97	17.94	17.91	17.84	17.78
110	7 1/4	18.92	18.89	18.86	18.82	18.79	18.76	18.73	18.69	18.66	18.63	18.57	18.50
112	7 1/4	19.19	19.15	19.12	19.09	19.06	19.02	18.99	18.96	18.93	18.89	18.83	18.77
114	7 1/4	19.45	19.42	19.39	19.36	19.32	19.29	19.26	19.23	19.19	19.16	19.10	19.03
116	7 1/4	19.72	19.69	19.65	19.62	19.59	19.56	19.52	19.49	19.46	19.43	19.36	19.30
118	7 1/4	19.99	19.95	19.92	19.89	19.85	19.82	19.79	19.76	19.72	19.69	19.63	19.56
120	7 1/4	20.25	20.22	20.19	20.15	20.12	20.09	20.05	20.02	19.99	19.96	19.89	19.83

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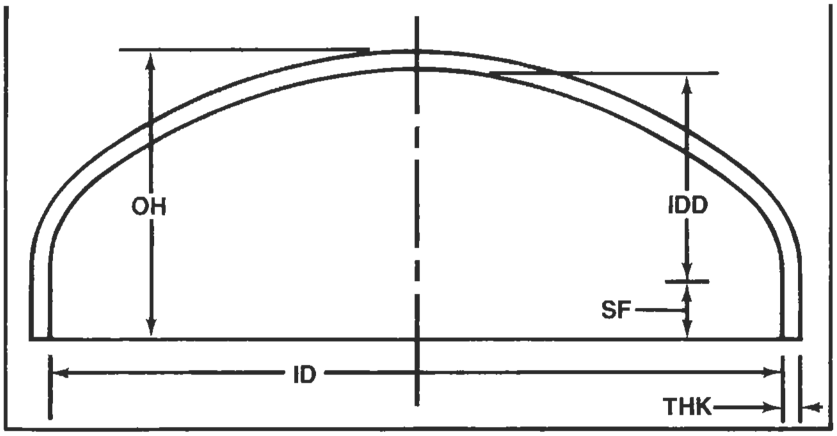
A-35.



ELLIPTICAL HEADS (2:1 RATIO)

ID - Inside Diameter
THK - Thickness
OH - Overall Height
SF - Straight Flange
IDD - Inside Depth of Dish

X - STANDARD
I - INQUIRE

SIZES AND THICKNESSES OF HEADS

ID \ THK	3/16	1/4	5/16	3/8, 7/16	1/2, 9/16, 5/8	11/16, 3/4, 13/16	7/8, 1	1 1/8, 1 1/4	1 3/8	1 1/2, 1 5/8, 1 3/4	1 7/8	2	2 1/4, 2 1/2, 2 3/4	3	3 1/4	3 1/2	3 3/4	4, 4 1/4, 4 1/2	4 3/4	5, 5 1/4	5 1/2	5 3/4	6	6 1/2	7	8	THK / ID
6	X	X	X	X	X	X	X	I	I	I																	6
8	X	X	X	X	X	X	X	X	X	I																	8
10	X	X	X	X	X	X	X	X	X	X	X	X															10
12	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I												12
14	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I												14
16	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I												16
18	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I												18
20	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I												20
22	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X												22
24	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I										24
30	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I										30
36	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X									36
42	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X							42
48	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I						48
54	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I	I				54
60	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I	I	60
66		X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I	I	66
72		X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I	I	72
78			X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I	I	78
84				X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I	I	84
90				X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I	I	90
96				X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I	I	96
102				X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I	I	102
108				X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I	I	108
114				X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I	I	114
120				X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	I	I	120

Elliptical (2:1 Ratio)

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A-35.

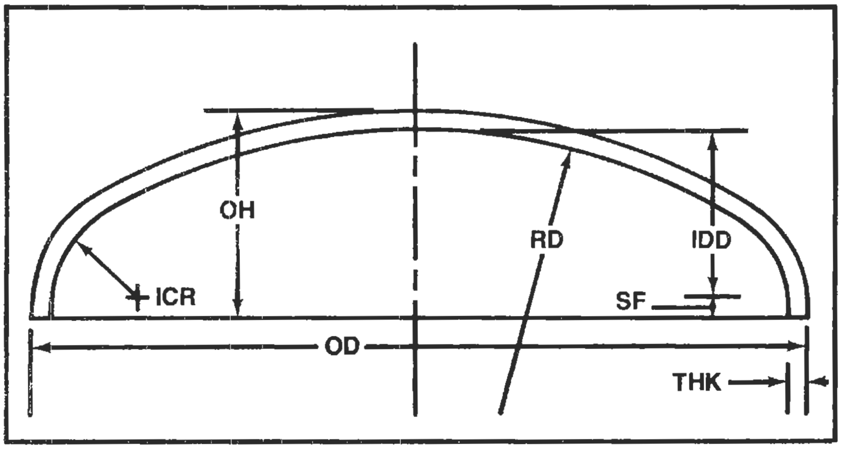


Figure 1.

INTERNAL PRESSURE COMPARISON

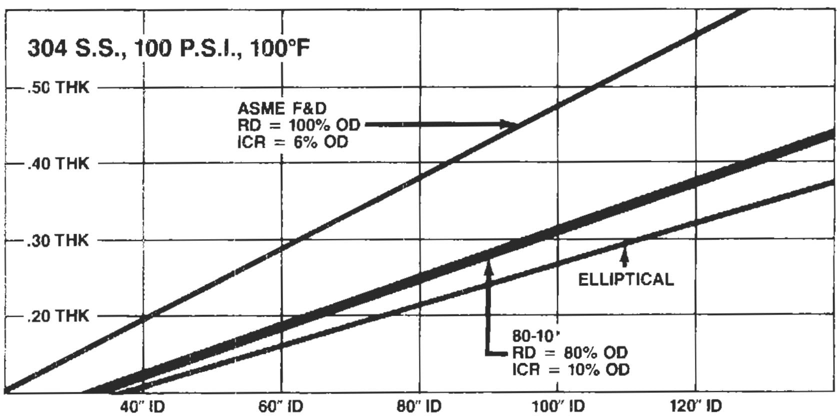
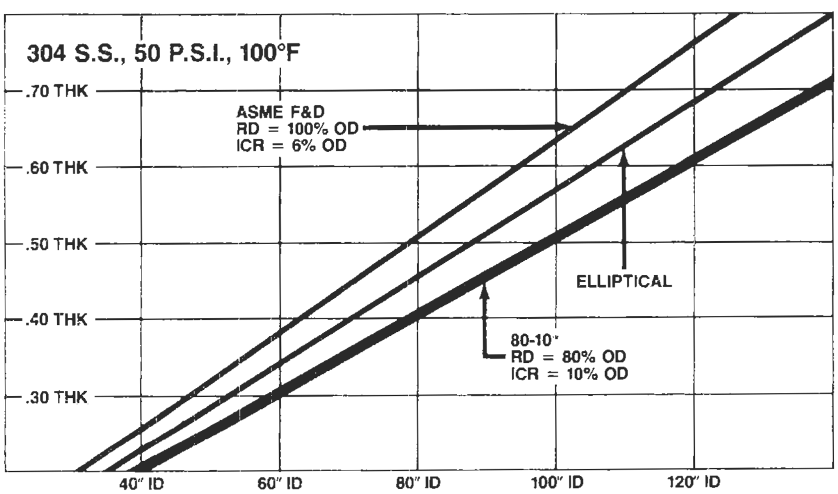


Figure 2.

EXTERNAL PRESSURE COMPARISON



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80-10® HEADS

OD — Outside Diameter
THK — Thickness
OH — Overall Height
SF — Straight Flange
RD — Radius of Dish
ICR — Inside Corner Radius
IDD — Inside Depth of Dish

Meeting all A.S.M.E. Unfired Pressure Vessel Code requirements, the 80-10® Head permits significantly higher pressures than other configurations selected for the same service. The 80-10® Head is named for its unique dimensions—the dish radius equals 80% of the head diameter and the inside corner radius equals 10% of the head diameter. These dimensions compare to 100% and 6% respectively for A.S.M.E. F&D Heads.

Index

Applied Process Design for Chemical and Petrochemical Plants

Volume 1 Third Edition

Completely revised and updated throughout, this new edition of the all-time best-selling process industry classic is now more valuable than ever. Volume 1 of this three-volume set is the definitive guide for process engineers and designers. It covers a complete range of basic day-to-day operation topics such as the latest ASME Code requirements; design and assembly of piping systems; selection of mechanical equipment; manpower requirements; pump types, ratings, and selections; separation of liquid and solid particles from vapor; and methods for liquid and gas flow and pumping of liquids.

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Ernest E. Ludwig, a registered professional chemical engineer in Louisiana, is a Fellow of the American Institute of Chemical Engineers and a past member of the American Society of Mechanical Engineers. He is a consulting engineer in Baton Rouge, specializing in plant design, operations, industrial fires and explosions, and management in the chemical and petrochemical industry.

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