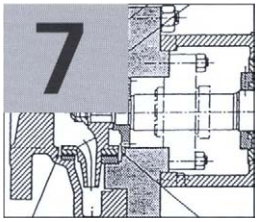


7 Root cause analysis techniques

(Improving component function knowledge base)

- Introduction
- Component function
- Component failure causes
- Component condition monitoring
- Examples of knowledge base enhancement

Introduction

The purpose of this chapter is to present important information concerning component and supporting system function, failure causes and condition monitoring requirements to increase your knowledge base for use in the root cause analysis procedure.

A sound component knowledge base is an essential tool for use in effective predictive maintenance and root cause analysis programs. Next to obtaining all the pertinent facts, knowledge base is a corner stone in the root cause analysis process. No matter what your level of knowledge, it can be improved. And, with process design improvements being made frequently, machinery design requirements and component functions are changing so that continuous improvement is a necessity.

A recent root cause analysis that I was connected with is an example. The apparent failure was a steam turbine rotor severe rub to the inner casing of a high pressure condensing turbine application. Lack of specific knowledge, on my part, regarding the thermal expansion characteristics of the inner casing was leading to false conclusions in the RCA process. I requested a design audit meeting with the machinery supplier and reviewed the finite element analysis (FEA) of the inner casing. Once that knowledge or awareness of the design basis was obtained, the possible causes of the failure were easily identified.

Figure 7.1 presents this important fact.

Component function

'If you don't know *what* the component is supposed to do and *how* it does it, effective analysis is impossible'

Figure 7.1 Component function

As previously stated in Chapter 4, regardless of the type of machinery, there are five (5) major components and systems:

- The rotor
- The bearing(s) – radial
- The thrust bearing
- The seal(s)
- The auxiliary systems

It is assumed that the reader has previous experience with rotating equipment. Most likely, it will be experience with the use, assembly, disassembly, predictive or preventive maintenance of the machinery. Unless you have worked for a machinery supplier at one point in your career, your knowledge base regarding the design basis of the components and their supporting systems will be limited.

Therefore, the objective of this Chapter is to 'stretch' and 'improve' your component function knowledge base for those machinery components that typically have the lowest reliability and availability.

The important components and systems that are present in all types of machinery are shown in Figure 7.2.

Machinery component/system knowledge base examples

1. Centrifugal rotors – effect of the process and fluid head concepts
2. Radial bearing(s)
3. Thrust bearings and balance drums
4. Seals
5. Auxiliary systems – lubrication, seal, buffer and cooling

Figure 7.2 Machinery component/system knowledge base examples

The method used to determine your present component knowledge base and expand it is outlined in Figure 7.3. Information concerning the design basis for each of the components and systems noted in Figure 7.2 is included in this chapter.

Component knowledge base enhancement procedure

For each selected component:

I. Answer the following questions:

1. Component function – 'What it does'
2. Component operation – 'How it does it'
3. List failure causes for the component
4. List component monitoring parameter required to detect component condition changes (refer to Chapter 4)

II. Then review the reference information included in this chapter and modify the answers in item I as required.

III. Continue to modify the answers in item I as you acquire additional knowledge and keep the information handy for use in all RCA procedures.

Figure 7.3 Component knowledge base enhancement procedure

For each of the components and systems noted in Figure 7.2, you will be asked to complete the following activities:

1. Component function exercise
2. Component failure cause exercise
3. Component monitoring parameter exercise

You are encouraged to make time to complete these exercises for each of the five items contained in this chapter. Keep in mind that every machinery type you have or will ever encounter contains these components and systems! Having a sound knowledge base of component and system function, failure causes and monitoring parameters will greatly improve your RCA skills.

These exercises should be scheduled over time but completed as soon as possible. Once the information is listed, keep it available and update it after very RCA, article you have read, conference you have attended, supplier technical presentation etc. Your machinery component knowledge base is really the foundation of your troubleshooting skills! And the foundation must be checked and fortified periodically to support your efforts. This will be an ongoing process and should never stop as long as you are associated with rotating machinery predictive maintenance and problem analysis.

Regardless of your experience level, it is suggested that you complete these exercises for each of the five items, before reading the supporting information contained in this chapter or before obtaining information from other sources.

Then review the information carefully and make any required changes. It is suggested that you then concentrate on one item at a time and read the information contained in this chapter and modify your initial answers accordingly. After you have completed this step, discuss your answers with associates if possible and be sure to include all available disciplines when discussing these items operations, maintenance process engineering, reliability engineering etc.). Finally, consult supplier instruction books, available books on the subject, magazine articles and of course the web for additional information. Update your information and continue to do so.

Actually, this process is no different than the action the machinery suppliers take! They have an established experience data base and continually update it with new information to produce a maximum data base of correct information that will enable them to design new equipment and solve reliability issues. You may ask, why then do I need to obtain this information? My answer is ... because your company's objective and the suppliers objective are the same, to maximize profit, but the methods to achieve that objective are directly opposed! Your company maximizes profits by operating the unspared (critical) machinery in the plant 24/7 and identifying potential reliability issues through effective predictive maintenance to meet that objective.

The suppliers maximize profit by manufacturing the machinery to meet your specifications at the lowest cost to assure that the equipment will be reliable for the warrantee period of the equipment. Certainly, this is not to say that planned obsolescence is a supplier design objective, but suppliers cannot stay competitive and in business if the equipment is designed for the life of the plant (30 years).

The reference information in this chapter constitutes the minimum that you will need to be an effective problem solver. Increase this information by reading the supplier instruction manuals in your area as a minimum.

Component function

Figure 7.4 requires you to define 'what' the component system or item is supposed to do and 'how' this task is accomplished. Complete this exercise for the selected component or system. It is suggested that you perform this exercise and the following failure cause and monitoring parameter exercises at one time for each selected component or system.

Function exercise

Component ____________ or system _________

I. 'What' the function of ______________________________

_________ is to ______________________________

____

____

____

II. 'How' the function of ______________________________

_________ is accomplished ______________________________

by ______________________________

____

____

____

____

Figure 7.4 Function exercise

The following example in Figure 7.5 for a hydrodynamic thrust bearing, demonstrates the requirements of the exercise.

Function exercise

Component: *hydrodynamic thrust bearing*

I. 'What' The function of a *hydrodynamic thrust bearing* is to *support the rotor in the axial direction.*

II. 'How' The function of a *thrust bearing* is accomplished by *having sufficient bearing area to support the maximum anticipated thrust loads, be properly installed and be provided with clean, cool lubricating oil of the correct viscosity and flowrate.*

Figure 7.5 Function exercise

Component failure causes

Figure 7.6 requires you to list all of the failure causes you can think of for your assigned component or system. In the case of the thrust bearing example above, failure causes are:

1. Insufficient bearing area
2. More load than anticipated
3. Incorrect installation
4. Unclean oil
5. Hot oil
6. Incorrect oil viscosity used or incorrect oil temperature
7. Incorrect oil flowrate

As can be seen from the above example, the key is to have a concise function definition. Once that is achieved, the remainder of the exercise is relatively easy. Once the failure causes are defined, the monitoring parameters to review for trends can be defined.

Do not forget however that while the use of the function and supporting system definitions aid greatly in the effectiveness of the RCA procedure, all of the failure classifications noted in Chapter 2 must be considered. As an example, the cause of 'More load than anticipated' for the thrust bearing example above could be in 4 of the 5 failure classifications:

- The effects of the process – higher discharge pressure
- Assembly/disassembly errors – insufficient clearance
- Operation errors – closing the discharge valve

- Design (thrust balance) and/or manufacturing errors – supplier balance drum sizing error.

Component failure causes

Component ___________ or system ________

List all possible failure causes. (Note: examine each of the failure classifications – in preparing this list).

Reference
Number

1. ____________________
2. ____________________
3. ____________________
4. ____________________
5. ____________________
6. ____________________
7. ____________________

Figure 7.6 Component failure causes

For your reference, we have included the failure classifications in Figure 7.7.

Failure classifications

- Process condition changes
- Improper installation/maintenance assembly
- Improper operating procedure
- Design problems
- Component wearout

Figure 7.7 Failure classifications

Remember to use *all* of the knowledge available for these exercises! The information gathered for these exercises will give you a 'running start' in your RCA procedures whether they will be used for predictive maintenance analysis or root cause failure analysis. Once you have completed these exercises for each component and system consult associates in your plant for their input and go outside your immediate group. It is a fact that many RCA procedures stay in the discipline of the RCA leader. Most RCA exercises are confined to the maintenance department. The root causes of failure can and will be found in all areas of the plant and every discipline should be consulted and can contribute. Reliability (and root cause analysis) is everyone's responsibility!

Component condition monitoring

Figure 7.8 presents the monitoring parameter exercise. Complete this exercise for your selected component or system now. For each failure cause listed in Figure 7.6, note in the corresponding number reference in Figure 7.8 the parameters (instruments or instruments required to be used) to monitor to determine if that failure cause is present.

For the hydrodynamic thrust bearing example given in Figure 7.5, the monitoring parameters for each failure cause are:

1. More load than anticipated – Axial displacement, thrust pad temperature, oil supply and drain temperature, process conditions (flow, fluid composition (MW or SG), suction and discharge pressure), machine speed or guide vane angle
2. Insufficient bearing area – Axial displacement, thrust pad temperature, oil supply and drain temperature
3. Unclean oil – Oil particle analysis
4. Hot oil – Oil supply temperature
5. Incorrect oil viscosity used or incorrect oil temperature – Oil analysis and oil temperature
6. Incorrect oil flowrate – Install temporary ultrasonic flowmeter and possibly confirm by calculating flow thru supply oil valve using valve position, valve differential pressure, oil specific gravity and valve trim information (valve Cv). Also view thrust bearing sight glass for a flow indication.

The parameter monitoring points noted above were obtained from the component condition monitoring lists in Chapter 4.

Failure cause monitoring parameters

Component ____________ or system ________

List all parameters (instruments) to be monitored or used for each failure cause listed in Figure 7.6. (Note: use corresponding failure cause reference number (same reference number as in Figure 7.6).

Reference
Number

1. ______________________________
2. ______________________________
3. ______________________________
4. ______________________________
5. ______________________________
6. ______________________________
7. ______________________________

Figure 7.8 Failure cause monitoring parameters

Once this information is obtained, condition monitoring trends to be reviewed can be easily identified and checked to define the causes of the component change of failure.

For the hydrodynamic thrust bearing example, axial displacement and thrust pad temperature trends may show when the thrust bearing load increased. Examination of process information system trends for the same time period (PI system, Aspen etc.) may show the specific cause of increased thrust bearing loading.

The remainder of this chapter contains the reference material for the major specific component or systems. It is strongly recommended that you read this information after you have first attempted to complete all exercises for each of the five major component and systems items mentioned in this chapter.

You are also encouraged to continuously improve your 'component knowledge' by:

- Obtaining articles
- Surfing the net
- Reading vendor publications

- Network with experienced associates
- Attend industry conferences
- Through additional site training courses (including supplier sponsored courses)

The answers for each of the five major components and systems in rotating machinery are contained at the end of this chapter.

Examples of knowledge base enhancement

In this section, I have included material that will provide a 'start' to your major component knowledge base for

- The rotor
- Thrust bearings
- Auxiliary systems
- Journal bearings
- Seals

The effect of the process on machinery reliability
(Rotor component knowledge)

- Introduction
- The major machinery components

Introduction

The effect of the process on machinery reliability is often neglected as a root cause of machinery failure. It is a fact that process condition changes can cause damage and/or failure to every major machinery component. For this discussion, the most common type of driven equipment – pumps will be used.

There are two (2) major classifications of pumps, positive displacement and kinetic, centrifugal types being the most common. A positive displacement pump is shown in Figure 7.9. A centrifugal pump is shown in Figure 7.10.

It is most important to remember that all driven equipment (pumps, compressors, fans, etc.) react to the process system requirements. They do only what the process requires. This fact is noted in Figure 7.11 for pumps.

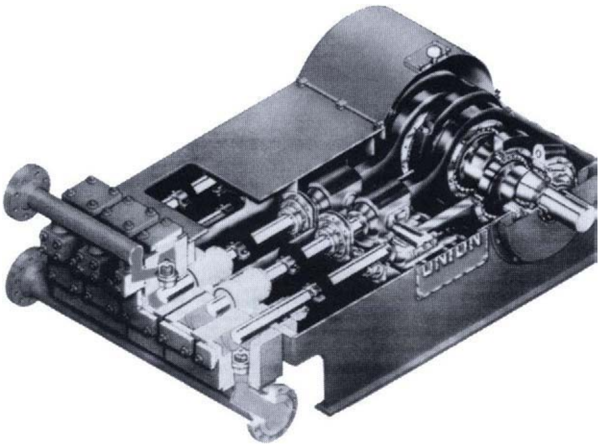


Figure 7.9 Positive displacement plunger pump (Courtesy of Union Pump Company)

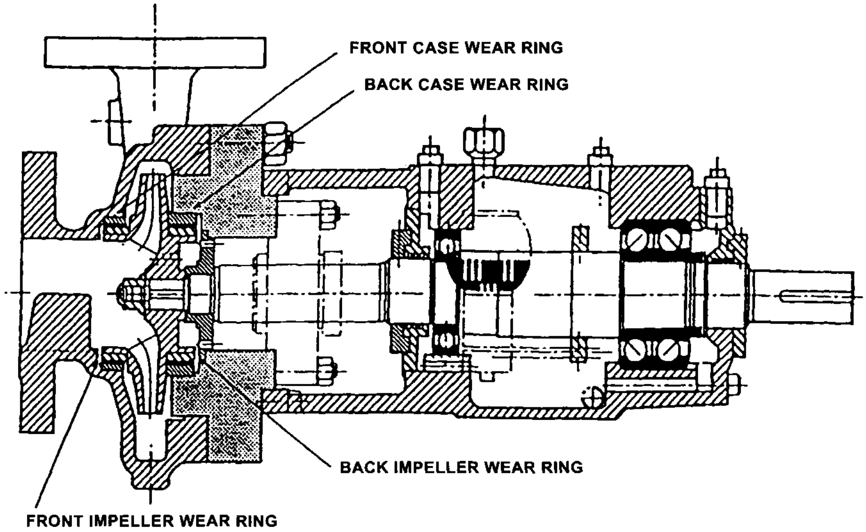


Figure 7.10 Centrifugal pump

Pump performance

- Pumps produce the pressure required by the process
- The flow rate for the required pressure is dependent on the pump's characteristics

Figure 7.11 Pump performance

Centrifugal (kinetic) pumps and their drivers

Centrifugal pumps increase the pressure of the liquid by using rotating blades to increase the velocity of a liquid and then reduce the velocity of the liquid in the volute. Refer again to Figure 7.10.

A good analogy to this procedure is a football (soccer) game. When the ball (liquid molecule) is kicked, the leg (vane) increases its velocity. When the goal tender (volute), hopefully, catches the ball, its velocity is significantly reduced and the pressure in the ball (molecule) is increased. If an instant replay 'freeze shot' picture is taken of the ball at this instant, the volume of the ball is reduced and the pressure is increased.

The characteristics of any centrifugal pump then are significantly different from positive displacement pumps and are noted in Figure 7.12.

Centrifugal pump characteristics

- Variable flow
- Fixed differential pressure produced *for a specific flow**
- Does not require a pressure limiting device
- Flow varies with differential pressure (P_1–P_2) and/or specific gravity

*assuring specific gravity is constant

Figure 7.12 Centrifugal pump characteristics

Refer again to Figure 7.11 and note that all pumps react to the process requirements.

Based on the characteristics of centrifugal pumps noted in Figure 7.12, the flow rate of all types of centrifugal pumps is affected by the Process System. This fact is shown in Figure 7.13.

Therefore, the flow rate of any centrifugal pump is affected by the process system. A typical process system with a centrifugal pump installed, is shown in Figure 7.14.

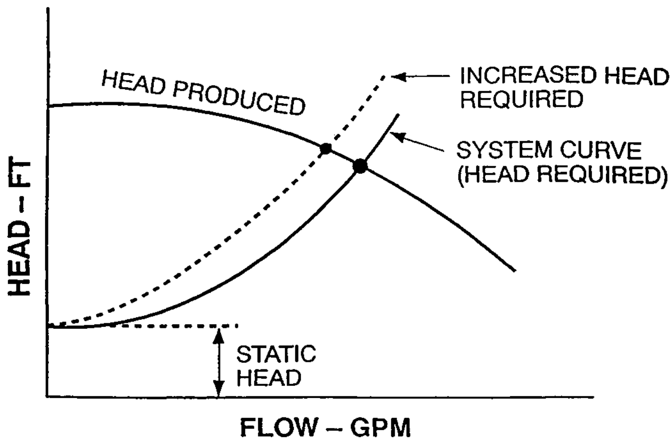


Figure 7.13 A centrifugal pump in a process system

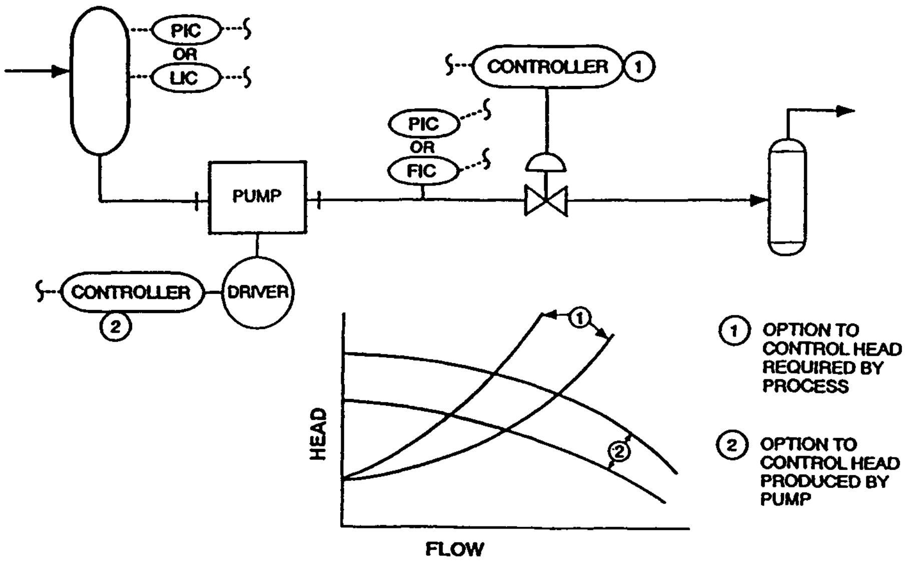


Figure 7.14 Centrifugal pump control options

The differential pressure required (proportional to head) by any process system is the result of the pressure and liquid level in the suction and discharge vessel and the system resistance (pressure drop) in the suction and discharge piping.

Therefore, the differential pressure required by the process can be changed by adjusting a control valve in the discharge line. Any of the following process variables (P.V.) shown in Figure 7.14, can be controlled:

- Level
- Pressure
- Flow

As shown in Figure 7.13, changing the head required by the process (differential pressure divided by specific gravity), will change the flow rate of any centrifugal pump!

Refer to Figure 7.15 and it can be observed that all types of mechanical failures can occur based on *where the pump is operating based on the process requirements.*

Since greater than 95% of the pumps used in this refinery are centrifugal, their operating flow will be affected by the process. Please

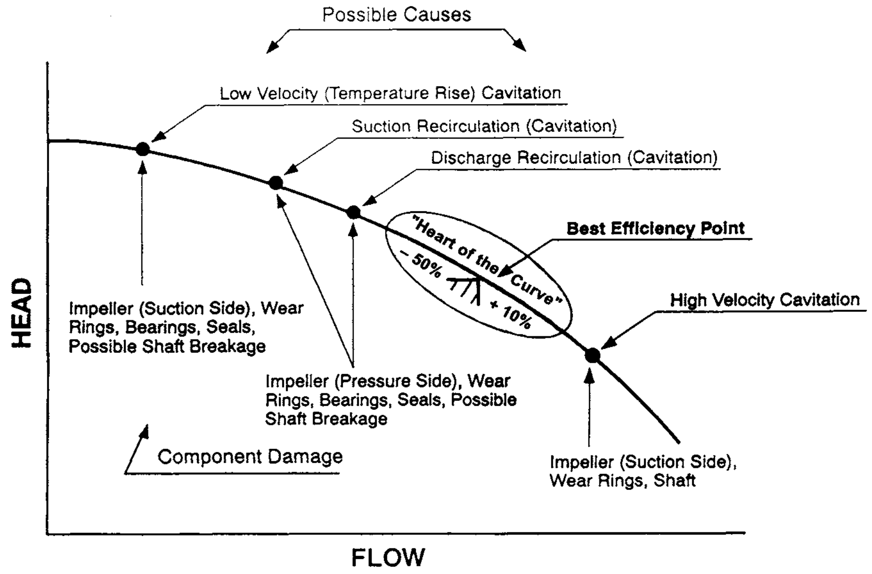


Figure 7.15 Centrifugal pump component damage and causes as a function of operating point

refer to Figure 7.16 which shows centrifugal pump reliability and flow rate is affected by process system changes.

Centrifugal pump reliability

- Is affected by process system changes (system resistance and S.G.)
- It is not affected by the operators!
- Increased differential pressure (P_2–P_1) means reduced flow rate
- Decreased differential pressure (P_2–P_1) means increased flow rate

Figure 7.16 Centrifugal pump reliability

At this point it should be easy to see how we can condition monitor the centrifugal pump operating point. Refer to Figure 7.17.

Centrifugal pump practical condition monitoring

- Monitor flow and check with reliability unit (RERU) for significant changes
- Flow can also be monitored by:
  - Control valve position
  - Motor amps
  - Steam turbine valve position

Figure 7.17 Centrifugal pump practical condition monitoring

Driver reliability (motors, steam turbine and diesel engines) can also be affected by the process when centrifugal driven equipment (pumps, compressor and fans) are used.

Refer to Figure 7.18 and observe a typical centrifugal pump curve.

Since the flow rate will be determined by the process requirements, the power (BHP) required by the driver will also be affected. What would occur if an 8½" diameter impeller were used and the head (differential pressure) required by the process was low? Answer: Since the pressure differential required is low, the flow rate will increase and for the 8½" diameter impeller, the power required by the drier (BHP) will increase.

Therefore, a motor can trip out on overload, a steam turbine's speed can reduce or a diesel engine can trip on high engine temperature.

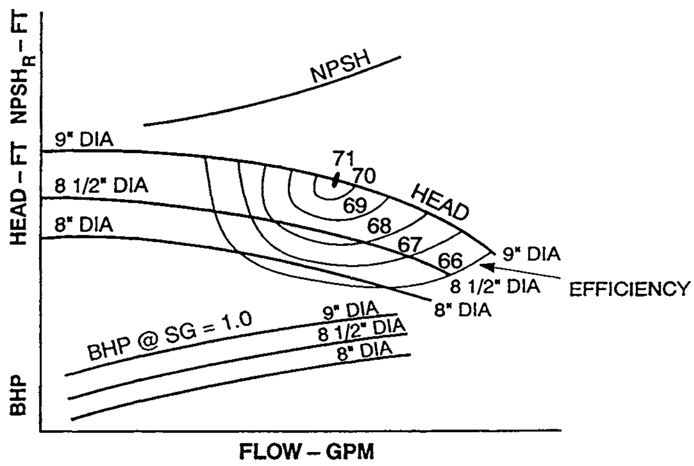


Figure 7.18 A typical centrifugal pump performance curve

These facts are shown in Figure 7.19.

Effect of the process on drivers

- Motors can trip on overload
- Steam turbines can reduce speed
- Diesel engines can trip on high engine temperature

Figure 7.19 Effect of the process on drivers

Auxiliary system reliability is also affected by process changes. Auxiliary systems support the equipment and their components by providing ... clean, cold fluid to the components at the correct differential pressure, temperature and flow rate.

Typical auxiliary systems are:

- Lube oil systems
- Seal flush system
- Seal steam quench system
- Cooling water system

The reliability of machinery components (bearings, seals, etc.) is directly related to the reliability of the auxiliary system. In many cases, the root cause of the component failure is found in the supporting auxiliary system.

As an example, changes in auxiliary system supply temperature, resulting from cooling water temperature or ambient air temperature changes, can be the root cause of component failure. Figure 7.20 presents these facts.

Component (bearing and seal) reliability

- Is directly related to auxiliary system reliability
- Auxiliary system reliability is affected by process condition changes.
- 'Root causes' of component failure are often found in the auxiliary system.

Figure 7.20 Component (bearing and seal) reliability

As a result, the condition of all the auxiliary systems supporting a piece of equipment must be monitored. Please refer to Figure 7.21.

Always 'think system'

- Monitor auxiliary system condition
- Inspect auxiliary system during component replacement

Figure 7.21 Always 'think system'

The major machinery components

Please refer again to Figure 7.15 which shows how process condition changes can cause damage and/or failure to any pump component.

Regardless of the type of machinery, the major component classifications are the same. The major machinery components and their systems are shown in Figure 7.22.

Major machinery components and systems

- Rotor
- Radial bearing
- Thrust bearing
- Seal
- Auxiliary systems

Figure 7.22 Major machinery components and systems

Regardless of the type of machinery, if we monitor the condition of each major component and its associated system, we will know the condition of the machine!

Refer to Figure 7.23 and define the major components and their associated systems.

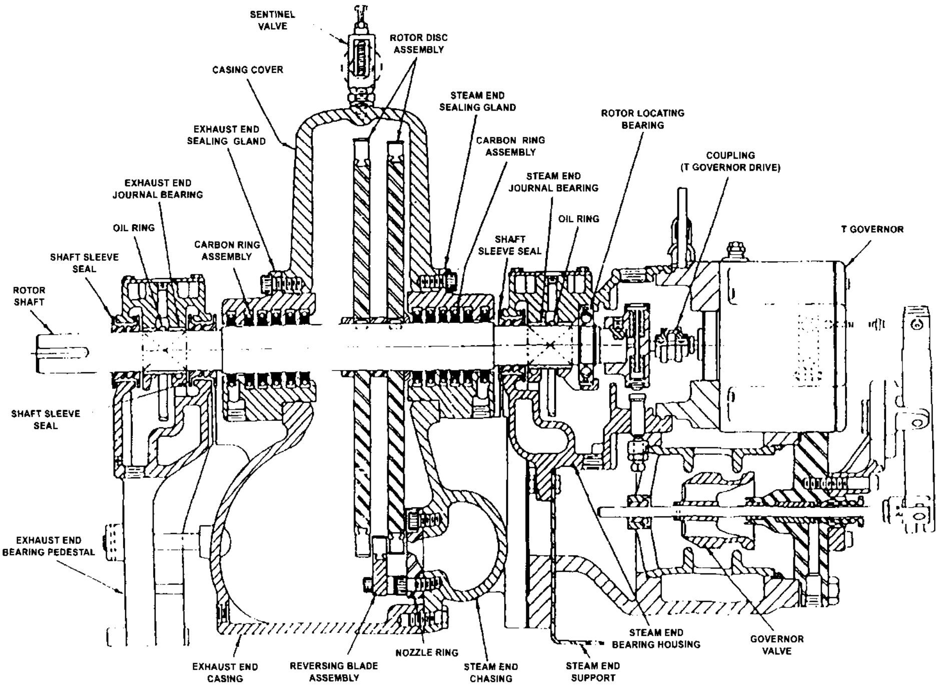


Figure 7.23 Typical general purpose (pump drive) turbine assembly (Courtesy of Elliott Company)

The concept of fluid head
(Rotor component knowledge)

- Introduction
- Definition
- Paths of compression
- The different types of gas head
- Dynamic compressor curve format

Introduction

Without a doubt one of the most confused principles of turbo-compressor design in my experience has been that of fluid head. I have found that understanding this concept can best be achieved by recognizing fluid head is the energy required to achieve specific process requirements.

Head is the energy in foot-pounds force required to compress and deliver one pound of a given fluid from one energy level to another. One of the confusing things about this concept is that the industry persists in defining head in feet. Head should be expressed in foot-pounds force per pound mass or British Thermal Units per pound. A British Thermal Unit per pound of fluid is equal to exactly 778 foot-pounds force per pound mass of that fluid.

Remember, when we deal with fluid head, a fluid can be either a liquid or a gas depending upon the conditions of the fluid at that time. Ethylene for instance, can be either a liquid or a gas depending on it's pressure and temperature. If it is a liquid, an ethylene pump will be used and the energy required to increase the pressure of the liquid from P_1 to P_2 will be defined as head in foot-pound force per pound mass. Conversely, if the conditions render it a vapor, an ethylene compressor would be used to achieve the same purpose. We will see in this section that the amount of energy required to compress a liquid or a gas the same amount will be significantly higher in favor of the gas because the gas is at a much lower density than that of a liquid. Understanding a Mollier Diagram is an important aid to understanding the concept of fluid head. Every fluid can have a Mollier Diagram drawing which expresses energy on the X axis and pressure on the Y axis for various temperatures. Increasing the pressure of a vapor will result in increased energy required.

Having defined the concept of head as that of energy, one must now investigate the different types of ideal (reversible) gas heads, namely isothermal, isentropic or adiabatic and polytropic. All of these types of fluid head simply describe the path that the gas takes in being compressed. It must be remembered that any type of head can be used to describe a reversible compressor path as long as the Vendor uses the appropriate head and efficiency in his data reduction calculations. In this section we will show the assumptions for various kinds of heads and the relative difference in their values. In addition, the definitions of each type will be stated.

Definition

The definition of head required by any fluid compression process is presented in Figure 7.24.

In any compression process, the amount of energy required to compress one pound of mass of a specific gas, at a given temperature from compressor suction flange (P_1) to the discharge flange (P_2) is defined as head required.

Figure 7.25 demonstrates how the density of a fluid significantly affects the amount of energy (head) required in a compression process.

Water with a density of 62.4 lbs/ft^3 requires only 231 ft-lb force per lb mass to compress the liquid 100 PSI. Note also, the equation for head

HEAD REQUIRED *is the energy in foot pounds required to compress and deliver one pound of a given fluid from one energy level to another.*

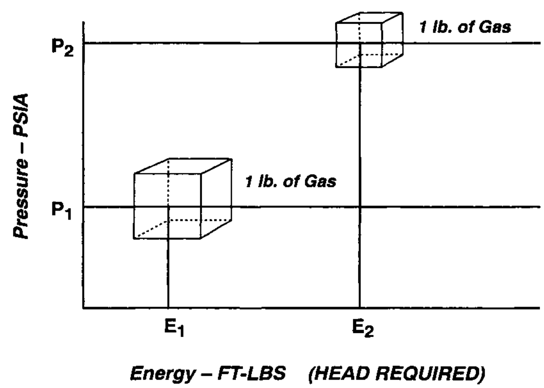


Figure 7.24 Head required definition

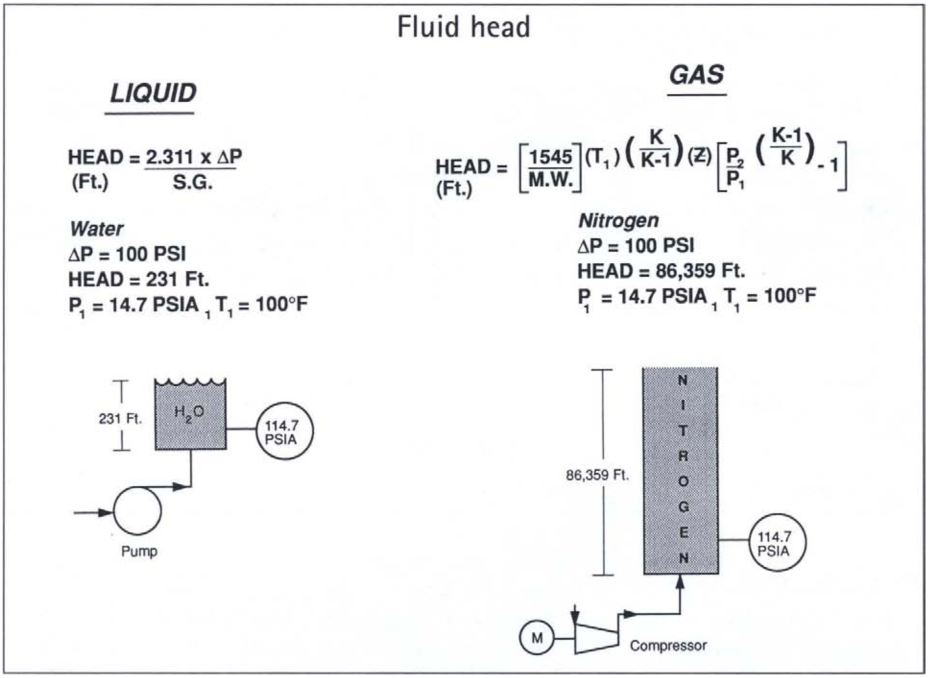


Figure 7.25 Fluid head definition

required by a liquid is independent of temperature. On the other hand, nitrogen with a density of 0.07 lbs/ft³ requires approximately 350 times the energy! This is because in both the case of water and nitrogen, the process requires that the fluid be compressed 100 PSI. However, since the mass of nitrogen is only 0.1% of the mass of water, a much greater amount of energy is required to compress the gas.

Figure 7.26 shows the head produced characteristics of positive displacement and dynamic compressors.

Note that regardless the amount of head required by the process, the flow rate of a positive displacement compressor is not affected. On the other hand, a dynamic compressor's flow rate is significantly affected by changes in the head required by the process. This is because the characteristic of any dynamic compressor is that it can only produce a greater amount of energy at a lower flow rate. The reason for this characteristic will be explained in a subsequent module. Therefore, any increase in the head required by a process will reduce the flow rate of a dynamic compressor. This is an extremely important fact because reduced flow rate in a dynamic compressor can lead to extreme, long term mechanical damage to the compressor unit. Also note in Figure 7.26 that the flatter the head produced curve a dynamic compressor possesses, the greater the effect of head required upon flow rate.

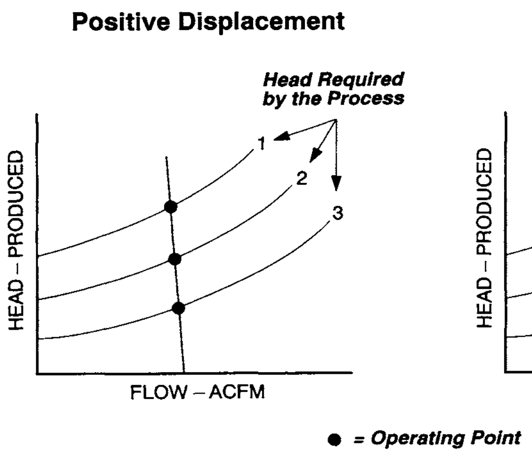


Figure 7.26 Compressor head produced characteristics

Head required

Thorough understanding of the concepts of head required by the process and head produced by the compressor is absolutely essential if dynamic compressor operation is to be understood.

It has been my experience that a lack of understanding exists in the area of dynamic compressor performance and often leads to a much greater emphasis upon the compressor's mechanical components (impellers, labyrinths, seals, bearings and shafts). In many cases, the root cause of dynamic compressor mechanical damage is that the head required by the process system exceeded the capability of the dynamic compressor.

Figure 7.27 presents the factors that determine the head (energy) required by any process.

Note that the head required by the process is inversely proportional to the gas density. If the gas density decreases, the head required by the process will increase. Gas density will decrease if gas temperature increases, inlet pressure decreases or molecular weight decreases. If the head required by the process increases, the flow rate of any dynamic compressor will decrease as shown in Figure 7.26. If the gas density increases, the head required by the process will decrease and dynamic compressor flow rate will increase.

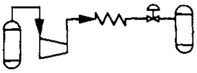
Head produced

In Figure 7.28, the factors that determine the head produced by a dynamic compressor are presented.

Simply stated, for a given impeller vane shape, head produced by a dynamic compressor is a function of impeller diameter and impeller

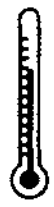
Head (energy) required

- Head (Energy) Required is Determined by:
  - System Resistance (P_2 / P_1)



  - Gas Composition (MW,Z,K)

  - Inlet Temperature (T_1)



- Head Required Increases If:
  - (P_2 / P_1) Increases
  - MW Decreases
  - (T_1) Increases
- Head Required Decreases If:
  - (P_2 / P_1) Decreases
  - MW Increases
  - (T_1) Decreases

Figure 7.27 Head (energy) required

speed. Once the impeller is designed, it will produce only one value of head for a given shaft speed and flow rate.

The only factor that will cause a lower value of head to be produced than stated by the compressor performance curve is if the compressor has experienced mechanical damage or if it is fouled.

Head (energy) produced

- Head (Energy) Produced is Determined by:
  - Compressor Impeller Design

- Head Produced by the Impeller,
  - Increases with Tip Speed
    - Impeller Diameter
    - Compressor Speed (RPM)
  - Increases with Decreasing Flow

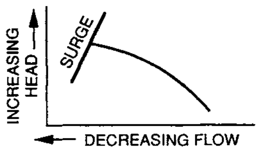
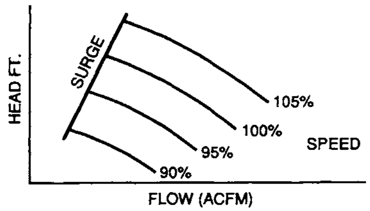


Figure 7.28 Head (energy) produced

The turbo-compressor curve

- The Turbo-Compressor Head (Energy) vs. Flow Curve is *Fixed by Compressor Design*



- For a Given Speed and Flow, If an Operating (Test) Point *Does Not Fall on the Curve,** the Unit Requires Maintenance

* *Within ± 10% (Accounting for Instrument Calibration Error)*

Figure 7.29 The turbo-compressor curve

Figure 7.29 shows how the need for dynamic compressor inspection can be determined.

If for a given flow rate and shaft speed, the head produced falls below the value predicted by greater than 10%, the compressor should be inspected at the first opportunity. Having explained the concept of head required by the process, the method of calculating the head required by a process system needs to be discussed.

Paths of compression

Figure 7.30 presents a typical Mollier Diagram plotted pressure vs energy.

A Mollier Diagram can be drawn for any pure fluid or fluid mixture. Usually, Mollier Diagrams are prepared only for pure fluids since any change in fluid mixture will require a new Mollier Diagram to be prepared. The Mollier Diagram can be used to determine the head required by the process system for any liquid, saturated vapor or vapor compression process. Observe that for liquid compression, the amount of energy required to increase the pressure from P_1 to P_2 is very small. However, for compression of a vapor, the amount of energy required to compress from P_1 to P_2 is very large as previously explained. Refer back to Figure 7.26 of this module and study the equations that are used to

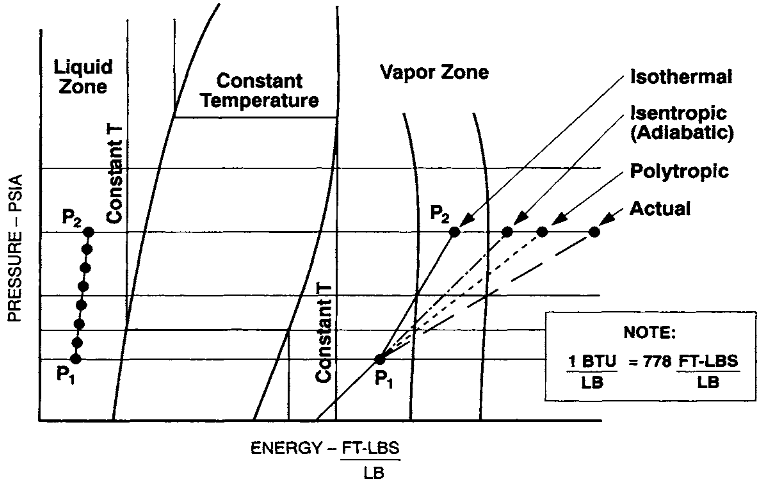


Figure 7.30 Ideal paths of compression

determine the head required for a liquid and a vapor. In addition to many more parameters being required for calculation of head required for a vapor, certain ideal assumptions must be made.

The different types of gas head

A vapor can be ideally compressed by any one of the following reversible thermodynamic paths:

- Isothermal – constant temperature
- Isentropic (adiabatic) – no heat loss
- Polytropic – temperature not constant and heat lost

The actual path that any compressor follows in compressing a vapor from P_1 to P_2 is equal to the reversible path divided by the compressor's corresponding path efficiency. Therefore, we can write the following equation:

$$\textit{Actual Head} = \frac{\textit{Head Isothermal}}{\textit{Eff'y. Isothermal}} = \frac{\textit{Head Isentroic}}{\textit{Eff'y. Isentropic}} = \frac{\textit{Head Polytropic}}{\textit{Eff'y. Polytropic}}$$

When evaluating compressor bids from different vendors quoting different types of reversible heads the above equation proves useful. If the head produced by one vendor divided by the corresponding

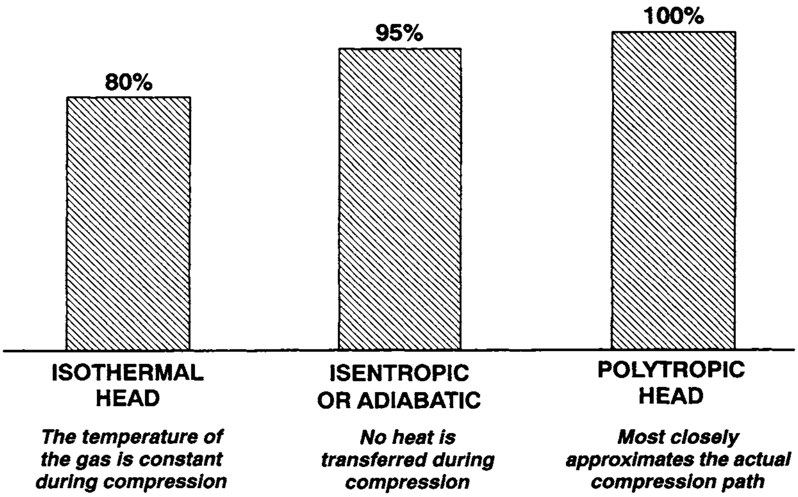


Figure 7.31 Ideal compression heads – relative values

efficiency is not equal to the corresponding values quoted by the competition . . . Better start asking why!

Figure 7.31 defines the different types of ideal heads and shows the relative difference between their values as compared to polytropic head.

The definition of polytropic head is confusing and difficult to understand if not investigated further.

The ideal gas head equations are described in Figure 7.32.

Note that the only difference between isentropic and polytropic head is the values;

$$\frac{K-1}{K} \quad \text{and} \quad \frac{n-1}{n}$$

Also note that $\dfrac{n-1}{n} = \dfrac{\frac{K-1}{K}}{\eta \text{ poly}}$

Now if η poly = 100%, $\dfrac{n-1}{n} = \dfrac{K-1}{K}$ or

Polytropic Head = Isentropic Head

Therefore, I think of $\dfrac{n-1}{n}$ as a correction

Factor to $\dfrac{K-1}{K}$ that will most closely approximate the actual

compressor path for a given compressor (refer back to Figure 7.30)

Ideal gas head equations

Isothermal	Isentropic (Adiabatic)	Polytropic
$HD = (\frac{1546}{M.W})(T_1)(Z_{AVG})\left[L_N(\frac{P_2}{P_1})\right]$	$HD = (\frac{1545}{M.W})(T_1)(\frac{K}{K-1})(Z_{AVG})\left[(\frac{P_2}{P_1})^{\frac{(K-1)}{K}} - 1\right]$	$HD = (\frac{1545}{M.W})(T_1)(\frac{n}{n-1})(Z_{AVG})\left[(\frac{P_2}{p1})^{\frac{(n-1)}{n}} - 1\right]$

Where:

$\frac{1545}{M.W}$ =	Gas constant "R"	P_2 =	Discharge Pressure – PSIA
T_1 =	Inlet temp. – °R	P_1 =	Inlet pressure – PSIA
°R =	460 + °F	K =	Ratio of specific heats C_p/C_v
Z_{AVG} =	Average compressibility $\frac{(Z_1 + Z_2)}{2}$	n =	Polytropic exponent
		$\frac{(n-1)}{n}$ =	$\left(\frac{K-1}{K}\right)\left(\frac{1}{\eta_{POLY}}\right)$
Ln =	Log to base e	η_{POLY} =	Polytropic efficiency

Figure 7.32 Ideal gas head equations

Remember, polytropic head is an ideal reversible compression path. Today, most compressor vendors have adopted polytropic head as their standard for multistage compressors. The main reason is that polytropic head, since it contains an efficiency term, allows individual impeller (stage) heads to be added.

Dynamic compressor curves format

Finally, Figure 7.33 presents the different ways that compressor compression performance can be formatted.

Head vs flow is always preferred because the head produced by a dynamic compressor is not significantly affected by gas density. However, compression ratio or discharge pressure are! That is, compression ratio and discharge pressure curves are invalid if the inlet gas temperature, inlet pressure or molecular weight changes! This may seem confusing at first, but refer back to Figure 7.28 which shows how head is produced by a dynamic compressor. It is a function only of impeller diameter and shaft speed. Gas density influences the head required by the process (refer to Figure 7.27).

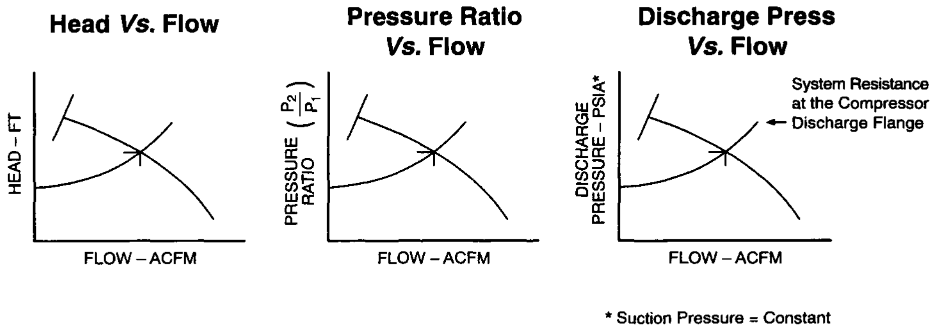


Figure 7.33 Dynamic compressor curves format

Radial bearing design
(Journal bearing component knowledge)

- Introduction
- Anti-friction bearings
- Hydrodynamic bearings
- Hydrodynamic bearing types
- Condition monitoring
- Vibration instabilities

Introduction

Radial bearings fall into two major categories:

- Anti-friction
- Hydrodynamic

Anti-friction bearings rely on rolling elements to carry the load of the equipment and reduce the power losses or friction.

Hydrodynamic bearings rely on a liquid film, usually lubricating oil, to carry the load of the equipment and minimize friction. In general, anti-friction bearings are used for equipment of low horsepower. Hydrodynamic bearings are generally used for all rotating equipment above approximately 500 horsepower. During this section we will concentrate on the subject of hydrodynamic bearings since they are the principle bearing type used in turbo-compressor operation.

A bearing in the radial position is responsible for mainly supporting the weight of the shaft and any dynamic loads that are present in the system. It can be generally stated that the dynamic loads are in the order of 20% of the static loads on the bearing journal. In the case of gears however, the radial load component is made up principally of the meshing force of the gear teeth and the load will vary from zero to maximum torque. One must be careful in this design to assure that the various load angles occur in zones of the bearing that can support these loads.

We will examine loads on hydrodynamic bearings and present an example of the bearing sizing based on static and dynamic loads of a rotor system. We will see that there are specific oil film pressure limits which dictate the bearing dimensions (length and diameter).

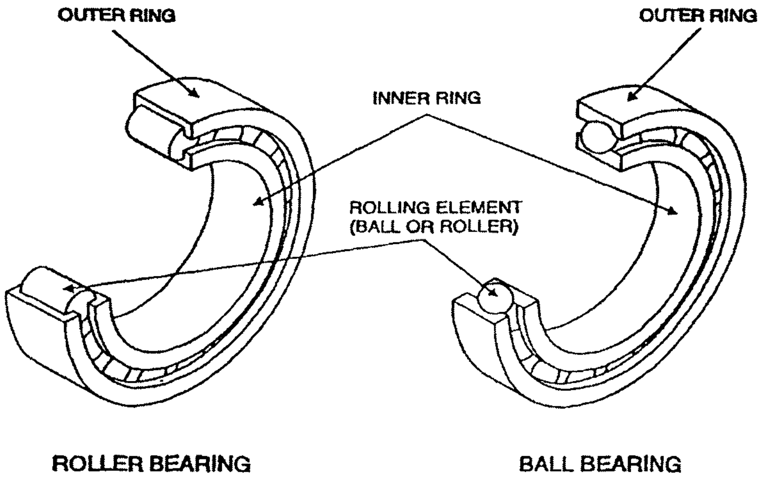


Figure 7.34 Anti-friction bearings

The types of hydrodynamic bearings will be reviewed; plain, stationary anti-whirl types and tilt pads.

We will conclude this section by discussing the condition monitoring requirements for radial bearings and briefly discuss shaft vibration and vibration troubleshooting (diagnostics).

Anti-friction bearings

Anti-friction radial bearings support the rotor using rolling elements to reduce friction losses. They are used in low horsepower applications (below 500 H.P.) and in aero-derivative gas turbines. Examples of roller and ball type anti-friction bearings are shown in Figure 7.34.

As previously mentioned, all bearings are designed on the basis of sufficient bearing area to support all the forces acting on the bearing. That is:

$$P = \frac{F}{A}$$

Where: P = Pressure on the bearing elements – P.S.I.
F = The total of all static and dynamic forces acting on the bearing
A = Contact area

For anti-friction bearing applications, the pressure, P is the point contact or 'Hertzian' stress on the bearing elements and rings or 'races'. For an anti-friction bearing to be properly designed, its D-N number and bearing life must be determined. Figure 7.35 presents the definition of D-N number and its uses.

D-N Number

- Is a measure of the rotational speed of the anti-friction bearing elements
- D-N number = bearing bore (millimeters) x speed (RPM)
- Approximate lubrication ranges

Lubrication type	**D-N Range**
Sealed	below 100,000
Regreaseable	100,000-300,000
Oil lube (unpressurized)	300,000-500,000
Oil lube (pressurized)	above 500,000

Figure 7.35 D-N number

Each type of anti-friction bearing has a maximum operating D-N number. If this value is exceeded, rapid bearing failure can occur. In addition, D-N numbers are typically used to determine the type of lubrication required for bearings. A common practice in the turbo-machinery industry has been to use hydrodynamic bearings when the D-N number exceeds approximately 500,000.

The exception to this rule is aircraft gas turbines which can have D-N numbers in excess of 2,000,000. In these applications, hydrodynamic bearings are not used since the size and weight of the required lubrication system would be prohibitive.

Anti-friction bearings possess a finite life which is usually specified as 'B-10' or 'L-10' life as defined in Figure 7.36.

'B' OR 'L' – 10 Life

'B' or 'L' – 10 Life is defined as the life in hours that 9 out of 10 randomly selected bearings would exceed in a specific application.

'B' or 'L' - 10 Life =

$$\frac{16700}{N}\left[\frac{C}{F}\right]^{3}$$

Where: N = RPM
C = Load in LBS that will result in a bearing element life of 1,000,000 revolutions.
F= Actual load in LBS

Figure 7.36 'B' or 'L' – 10 life

An important fact to note is that the life of any anti-friction bearing is inversely proportional to cube of the bearing loads. As a result, a small change in the journal bearing loads can significantly reduce the bearing life. When anti-friction bearings suddenly start failing where they did not in the past, investigate all possible sources of increased bearing loads (piping forces, foundation forces, misalignment, unbalance, etc.). Anti-friction bearings are usually designed for a minimum life of 25,000 hours continuous operation.

Hydrodynamic bearings

Hydrodynamic bearings support the rotor using a liquid wedge formed by the motion of the shaft (see Figure 7.37).

Oil enters the bearing at supply pressure values of typically 15–20 psig. The shaft acts like a pump which increases the support pressure to form a wedge. The pressure of the support liquid (usually mineral oil) is determined by the area of the bearing by the relationship:

$$P = \frac{F}{A}$$

Where: P = Wedge support pressure (P.S.I.)
F = Total bearing loads (static and dynamic)
A = Projected bearing area ($A_{PROJECTED}$)

$$A_{PROJECTED} = L \times d$$

Where: L = Bearing axial length
d = Bearing diameter

As an example, a 4" diameter bearing with an axial length of 2" (L/d = .5) would have $A_{PROJECTED} = 8\ in^2$.

If the total of static and dynamic forces acting on the bearing are 1600 lbs. force, the pressure of the support wedge is:

$$P = \frac{1600\ Lb_{FORCE}}{8\ in^2} = 200\ P.S.I.$$

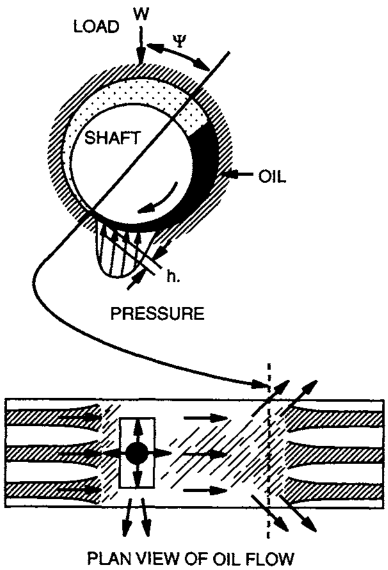


Figure 7.37 Hydrodynamic Lubrication (Courtesy of Bently Nevada Corp.)

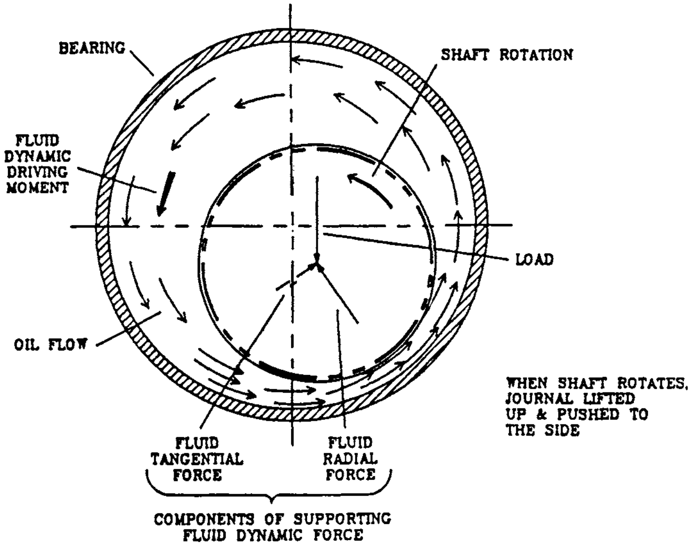


Figure 7.38 Shaft/bearing dynamics (Courtesy of Bently Nevada Corp.)

The maximum desired design wedge pressure for oil is approximately 500 psi. However, it has been common practice to limit hydrodynamic bearing loads to approximately 250 psi in compressor applications. Figure 7.38 is a side view of a simple hydrodynamic bearing showing the dynamic load forces.

The primary force is the load which acts in the vertical direction for horizontal bearings. However, the fluid tangential force can become large at high shaft speeds. The bearing load vector then is the resultant of the load force and fluid tangential force. The fluid radial force opposes the load vector and thus supports the shaft. It has been demonstrated that the average velocity of the oil flow is approximately 47–52% of the shaft velocity. The fluid tangential force is proportional to the journal oil flow velocity. If the fluid tangential force exceeds the load force, the shaft will become unstable and will be moved around the bearing shell. This phenomena is known as oil whirl.

Hydrodynamic bearing types

Regardless of the type of hydrodynamic bearing, all bearing surfaces are lined with a soft, surface material made of a composition of tin and lead. This material is known as Babbitt. Its melting temperature is above 400°F, but under load will begin to deform at approximately 320°F. Typical thickness of Babbitt over steel is 0.060 (1.5mm). Bearing embedded temperature probes are a most effective means of

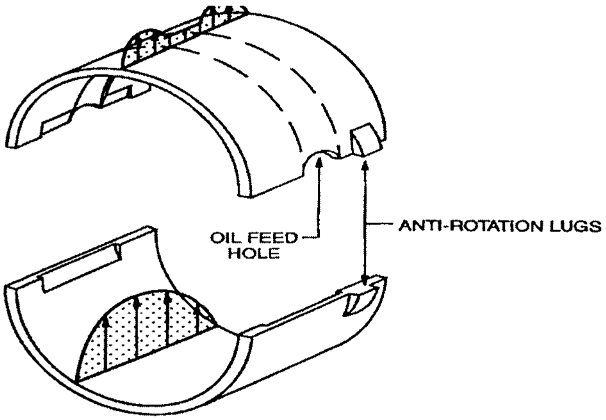


Figure 7.39 Straight sleeve bearing liner (Courtesy of Elliott Co.)

measuring bearing load point temperature and are inserted just below the Babbitt surface. RTD's or thermocouples can be used. There are many modifications available to increase the load effectiveness of hydrodynamic bearings. Among the methods available are:

- Copper backed Babbitt or 'Trimetal' – to aid in heat removal
- Back pad cooling – used on tilt pad bearings to remove heat
- Direct cooling – directing cool oil to maximum load points

A typical straight sleeve hydrodynamic journal bearing is shown in Figure 7.39.

Straight sleeve bearings are used for low shaft speeds (less than 5,000 RPM) or for older turbo-compressor designs. Frequently, they are modified to incorporate a pressure dam, in the direction of rotation. The pressure dam must be positioned in the top half of the bearing to increase the load vector (see Figure 7.38). This action assures that the tangential force vector will be small relative to the load vector thus preventing shaft instability. It should be noted that incorrectly assembling the pressure dam in the lower half of the bearing would render this type of bearing unstable. When shaft speed is high, other alternatives to prevent rotor instabilities are noted in Figure 7.40.

Shown are examples of anti-whirl bearings. The most common types of these bearings are the 3 and 4 lobe design. Elliptical and offset bearing designs do prevent instabilities but tend to increase shaft vibration if the load vector passes through the major axis of the bearing. These types of bearings may have to be rotated in the bearing brackets to prevent this occupance.

The most common hydrodynamic bearing for higher speed applications is the tilt pad journal bearing shown in Figure 7.41.

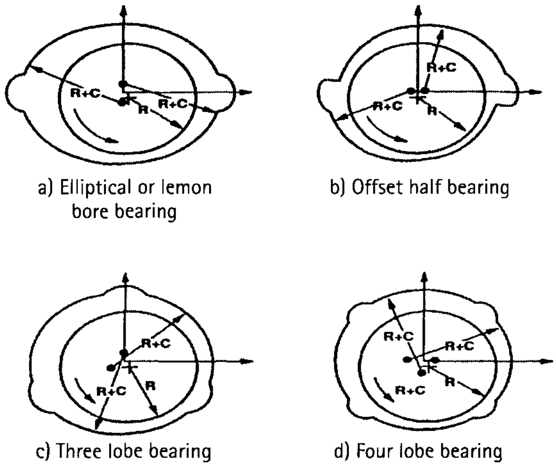


Figure 7.40 Prevention of rotor instabilities

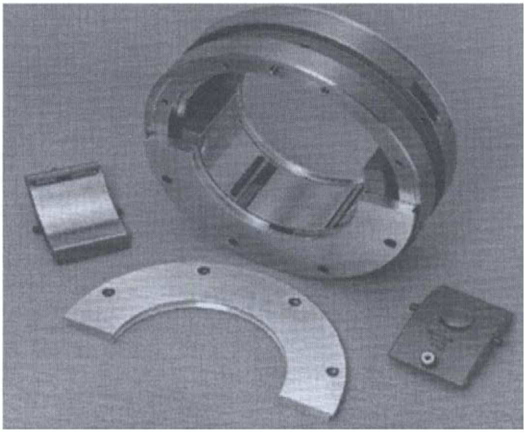


Figure 7.41 Tilting pad journal bearing assembly (Courtesy of Kingsbury, Inc.)

A tilting pad bearing offers the advantage of increased contact area since the individual pads conform to the shaft orbit. In addition, this type is also a highly effective anti-whirl bearing since the spaces between the pads prevent oil whirl.

Most end users specify tilt pad radial and thrust bearings for turbo-compressor applications.

Figure 7.42 shows the mechanical frictional losses and oil flow requirements for a tilt pad journal bearing as a function of shaft speed.

Note that the basis for horsepower loss and oil flow is an oil temperature rise of 30°F. This is the normal design ΔT for all

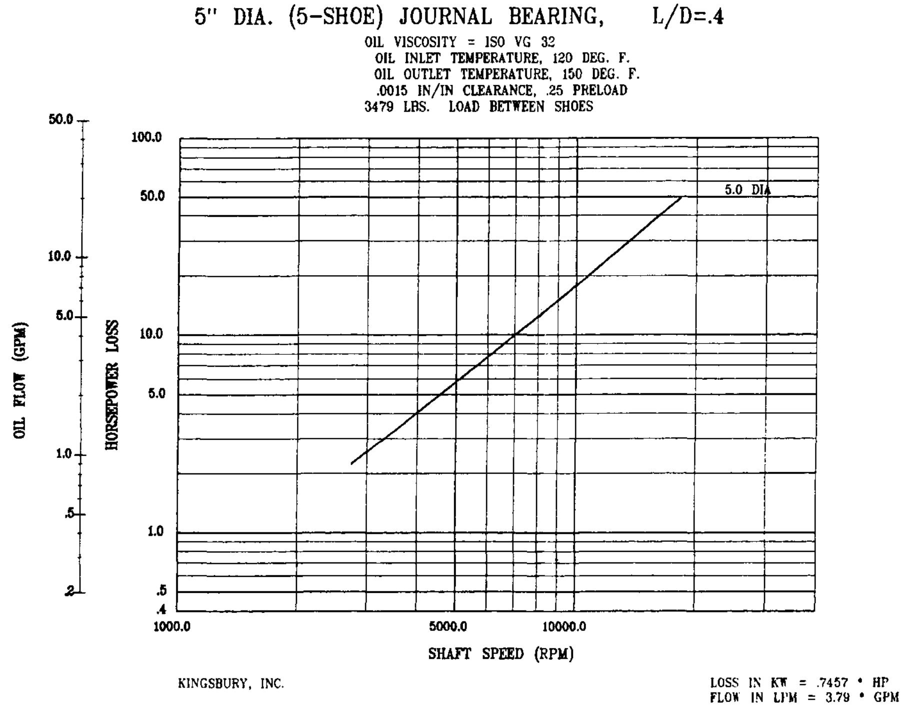


Figure 7.42 Typical journal bearing selection curve (Courtesy of Kingsbury, Inc.)

hydrodynamic bearings. Also given in this figure is the data necessary to calculate bearing pressure at the load point.

As an exercise calculate the following for this bearing:

- Projected Area

$A_{PROJECTED}$ = 5" x 2"

= 10 square inches

- Pressure

= 3479 Lb force = 10 square inches

= 347.9 psi on the oil film at load point

Condition monitoring

In order to determine the condition of any journal bearing, all the parameters that determine its condition must be monitored.

Figure 7.43 presents the eight parameters that determine the condition of a hydrodynamic journal bearing along with typical limits. Attendees are advised to consult the manufacturers instruction book for vendor recommended limits.

Parameter		Limits
1	Radial vibration (peak to peak)	2.5 mils (60 microns)
2	Bearing pad temperature	220°F (108°C)
3	Radial shaft position (except for gearboxes where greater values are normal from unloaded to loaded operation)	>30° change and/or 30% position change
4	Lube oil supply temperature	140°F (60°C)
5	Lube oil drain temperature	190°F (90°C)
6	Lube oil viscosity	Off spec 50%
7	Lube oil flash point	Below 200°F (100°C)
8	Lube oil particle size	Greater than 25 microns

Condition monitoring parameters and their alarm limits according to component:

1. Journal bearing (hydrodynamic)

Figure 7.43 The eight parameters that determine the condition of a hydrodynamic journal bearing along with typical limits

One important parameter noted in Figure 7.43 that is frequently overlooked is shaft position. Change of shaft position can only occur if the forces acting on a bearing change or if the bearing surface wears. Figure 7.44 shows how shaft position is determined using standard shaft proximity probes.

Regardless of the parameters that are condition monitored, relative change of condition determines if and when action is required. Therefore, effective condition monitoring requires the following action for each monitored condition.

- Establish baseline condition
- Record condition trend
- Establish condition limit

Figure 7.45 presents these facts for a typical hydrodynamic journal bearing.

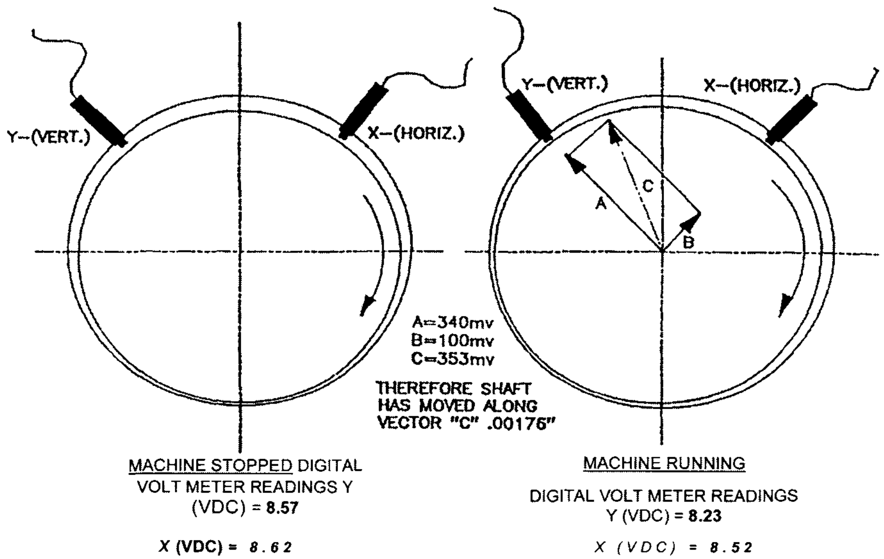


Figure 7.44 Shaft movement analysis (relative to bearing bore) (Courtesy of M.E. Crane Consultant)

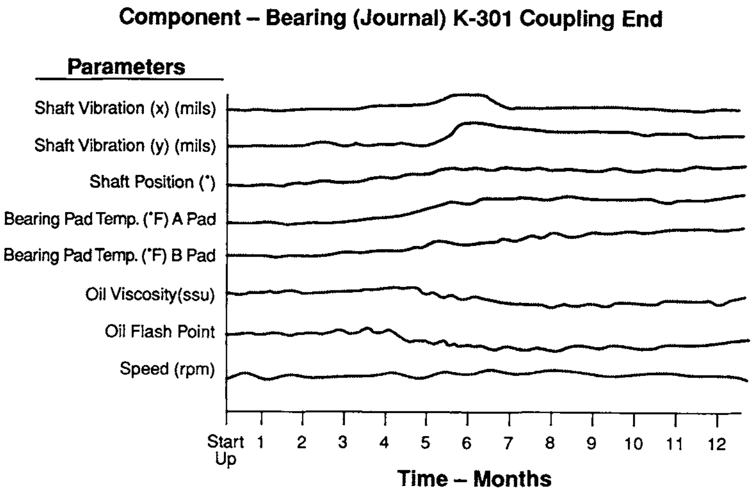


Figure 7.45 Trending data for a typical hydrodynamic journal bearing

Based on the information shown in this trend, the bearing should be inspected at the next scheduled shutdown. A change in parameters during month 6 has resulted in increased shaft position, vibration and bearing pad temperature.

Vibration instabilities

Vibration is an important condition associated with journal bearings because it can provide a wealth of diagnostic information valuable in determining the root cause of a problem.

Figure 7.46 presents important information concerning vibration.

Vibration

- Vibration is the result of a system being acted on by an excitation.
- This excitation produces a dynamic force by the relationship:

$$F_{DYNAMIC} = Ma$$

Where: M = Mass (Weight/g)
g = Acceleration due to gravity (386 IN/SEC^2)
a = Acceleration of mass M (IN/SEC^2)

- Vibration can be:
  - Lateral ________ ↕ ________
  - Axial →________________ ←
  - Torsional ____________________

Figure 7.46 Vibration

Figure 7.47 defines excitation forces with examples that can cause rotor (shaft) vibration.

Turbo-compressors generally monitor shaft vibration relative to the bearing bracket using a non-contact or 'proximity probe' system as shown in Figure 7.48. The probe generates a D.C. eddy current which continuously measures the change in gap between the probe tip and the shaft. The result is that the peak to peak unfiltered (overall) shaft vibration is read in mils or thousandth of an inch. The D.C. signal is normally calibrated for 200 milli volts per mil. Probe gaps (distance between probe and shaft) are typically 0.040 mils or 8 volts D.C. to assure the calibration curve is in the linear range. It is important to remember that this system measures shaft vibration relative to the bearing bracket and assumes the bearing bracket is fixed. Some systems incorporate an additional bearing bracket vibration monitor and thus record vibration relative to the earth or 'seismic vibration'.

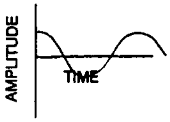
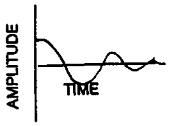
CATEGORY	EXAMPLES	EXCITATION TYPE	VIBRATION TYPE
FORCED VIBRATIONS 	UNBALANCE MISALIGNMENT PULSATION	CONSTANT CONSTANT PERIODIC	LATERAL LATERAL AND AXIAL TORSIONAL
TRANSIENT VIBRATIONS 	SYNCHRONOUS MOTOR START-UP IMPULSE (SHOCK FORCE)	RANDOM RANDOM	TORSIONAL TORSIONAL, AXIAL, RADIAL
SELF EXCITED	INTERNAL RUB OIL WHIRL GAS WHIRL	RANDOM CONSTANT CONSTANT	LATERAL LATERAL LATERAL

Figure 7.47 Excitation forces with examples

As previously discussed, vibration limits are usually defined by:

$$\text{Vibration(mils p-p)} = \sqrt{\frac{12000}{RPM}}$$

This value represents the allowable shop acceptance level. A.P.I. recommends alarm and trip shaft vibration levels be set as follows:

$$V_{ALARM} = \sqrt{\frac{24000}{RPM}}$$

$$V_{TRIP} = \sqrt{\frac{36000}{RPM}}$$

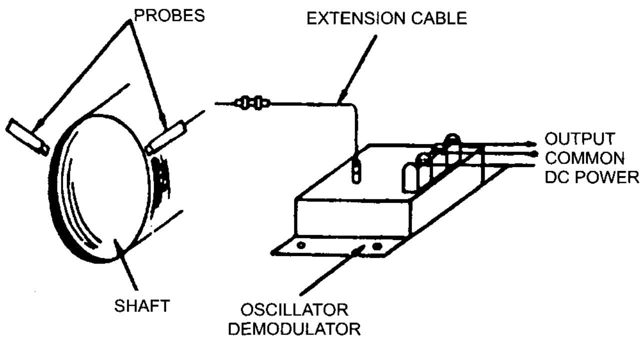


Figure 7.48 Non contact displacement measuring system

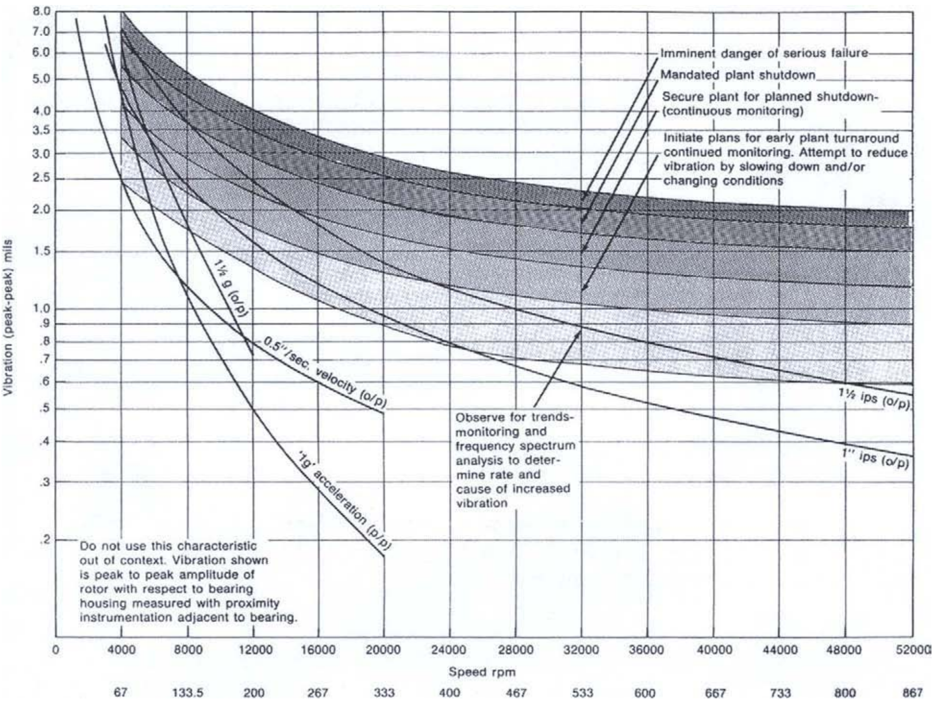


Figure 7.49 Vibration severity chart (Courtesy of Dresser-Rand and C. J. Jackson P.E.)

In the writers' opinion, shaft vibration alarm and trip levels should be based on the following parameters as a minimum and should be discussed with the machinery vendor prior to establishing levels:

- Application (critical or general purpose)
- Potential loss of revenue
- Application characteristics (prone to fouling, liquid, unbalance, etc.)
- Bearing clearance
- Speed
- Rotor actual response (Bode Plot)
- Rotor mode shapes (at critical and operating speeds)

Figure 7.49 presents a vibration severity chart with recommended action.

A schematic of a shaft vibration and shaft displacement monitor are shown in Figure 7.50.

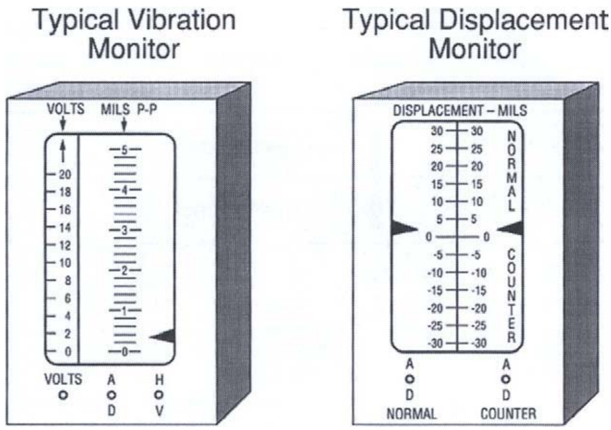
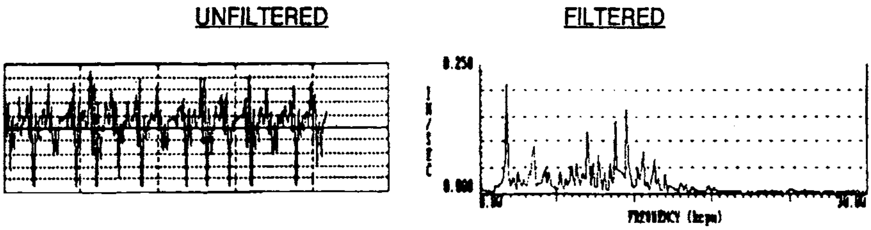


Figure 7.50 Shaft vibration and displacement

ANY VIBRATION SIGNAL IS MADE UP OF ONE OR MORE FREQUENCIES. A TYPICAL UNFILTERED (OVERALL) AND FILTERED SIGNAL ARE SHOWN BELOW:

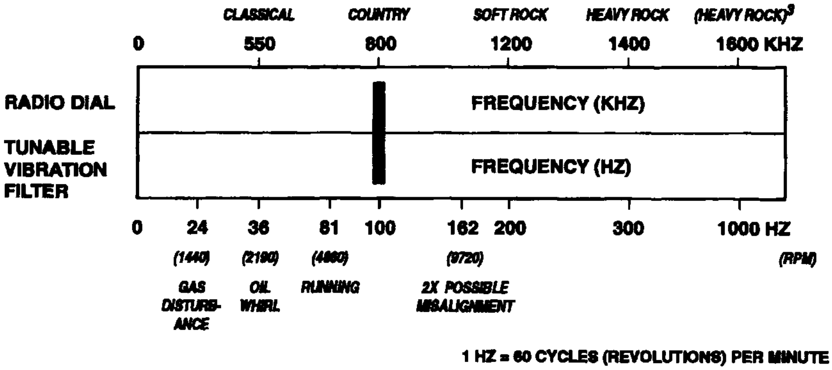


AN ANALOGY TO A FILTERED SIGNAL IS A RADIO. IN A GIVEN LOCALITY, MANY STATIONS ARE TRANSMITTING SIMULTANEOUSLY. ANY GIVEN STATION IS OBTAINED BY ADJUSTING THE TUNER TO THE CORRECT TRANSMISSION FREQUENCY.

Figure 7.51 Vibration frequency

As mentioned above, vibration is measured unfiltered or presents ‘overall vibration’. Figure 7.51 shows a vibration signal in the unfiltered and filtered conditions. All vibration diagnostic work (troubleshooting) relies heavily on filtered vibration to supply valuable information to determine the root cause of the vibration.

Figure 7.52 presents an example of a radio tuner as an analogy to a filtered vibration signal.



By Adjusting the Tuner (Filter) to a Selected Station (Frequency) the Desired Program (Vibration Frequency Signal) can be Obtained if it is "On the Air" (Present)

Figure 7.52 Radio tuner/vibration filter analogy

By observing the predominant filtered frequencies in any overall (unfiltered) vibration signal, valuable information can be gained to add in the troubleshooting procedure and thus define the root cause of the problem.

Rotor axial (thrust) forces
(Thrust bearing component knowledge)

- Introduction
- The hydrodynamic thrust bearing
- Impeller thrust forces
- Rotor thrust balance
- Thrust condition monitoring

Introduction

In every rotating machine utilizing reaction type blading, a significant thrust is developed across the rotor by the action of the impellers or blades. Also in the case of equipment incorporating higher than atmospheric suction pressure, a thrust force is exerted in the axial direction as a result of the pressure differential between the pressure in the case and atmospheric pressure.

In this section we will cover a specific rotor thrust example and calculate thrust balance for a specific case. We will see the necessity in some applications of employing an axial force balance device known as a balance drum. In many instances, the absence of this device will result in excessive axial (thrust) bearing loadings. For the case of a machine with a balance device, the maintenance of the clearances on this device are of utmost importance. In many older designs the clearances are maintained by a fixed close clearance bushing made out of babbitt which has a melting temperature of approximately 350°F, depending on the pressure differential across the balance drum. If the temperature in this region should exceed this value, the effectiveness of the balance drum would suddenly be lost and catastrophic failures can occur inside the machine. Understanding the function of this device and the potential high axial forces involved in its absence is a very important aspect of condition monitoring of turbo-compressors.

We will also examine various machine configurations including natural balanced (opposed) thrust and see how thrust values change even in the case of a balanced machine as a function of machine flow rate.

Finally, we will examine thrust system condition monitoring and discuss some of the confusion that results with monitoring these machines.

The hydrodynamic thrust bearing

A typical hydrodynamic double acting thrust bearing is pictured in Figure 7.53.

The thrust bearing assembly consists of a thrust collar mounted on the rotor and two sets of thrust pads (usually identical in capacity) supported by a base ring (Michell Type).

The Kingsbury type includes a set of leveling plates between each set of pads and the base ring. This design is shown in Figure 7.54.

Both the Michell and Kingsbury types are used. Figure 7.55 provides a view of the leveling plates providing the self-equalizing feature in the Kingsbury design. The self-equalizing feature allows the thrust pads to lie in a plane parallel to the thrust collar.

Regardless of the design features, the functions of all thrust bearings are:

- To continuously support all axial loads
- To maintain the axial position of the rotor

The first function is accomplished by designing the thrust bearing to provide sufficient thrust area to absorb all thrust loads without exceeding the support film (oil) pressure limit (approximately 500 psi).

Figure 7.56 shows what occurs when the support film pressure limit is exceeded.

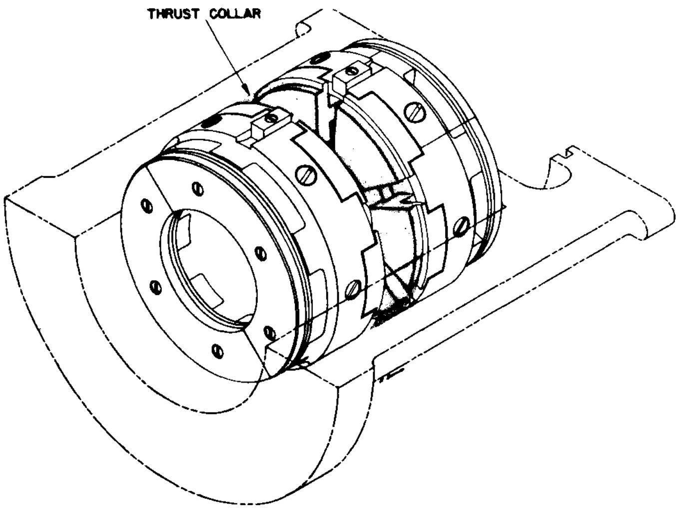


Figure 7.53 Double acting self-equalizing thrust bearing assembly (thrust collar removed) (Courtesy of Elliott Company)

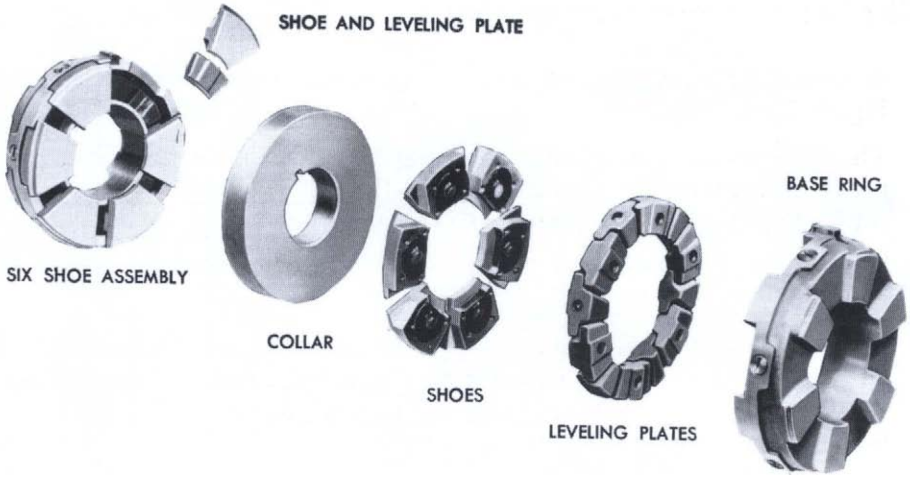


Figure 7.54 Small Kingsbury six-shoe, two direction thrust bearing. Left-hand group assembled, except for one shoe and 'upper' leveling plate. Right-hand group disassembled (Courtesy of Kingsbury, Inc.)

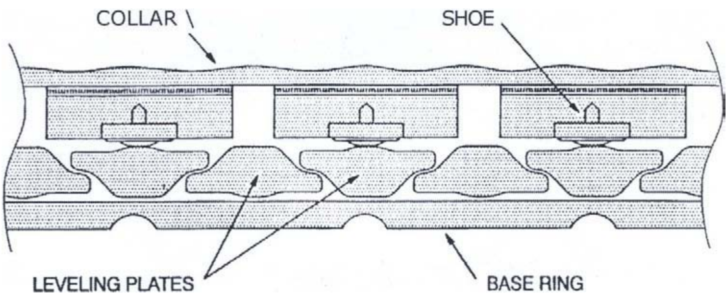


Figure 7.55 Self-equalizing tilt-pad thrust bearing (View – looking down on assembly) (Courtesy of Kingsbury, Inc.)

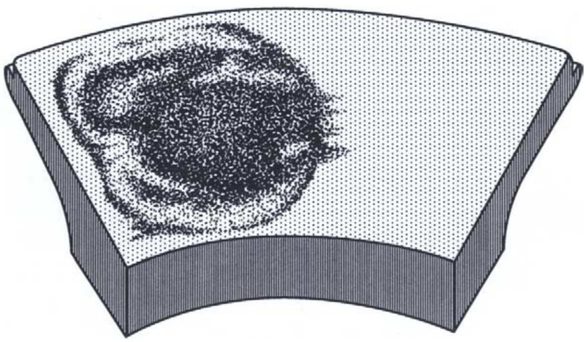


Figure 7.56 Evidence of overload on a tilt-pad self-equalizing thrust bearing pad (Courtesy of Kingsbury Corp.)

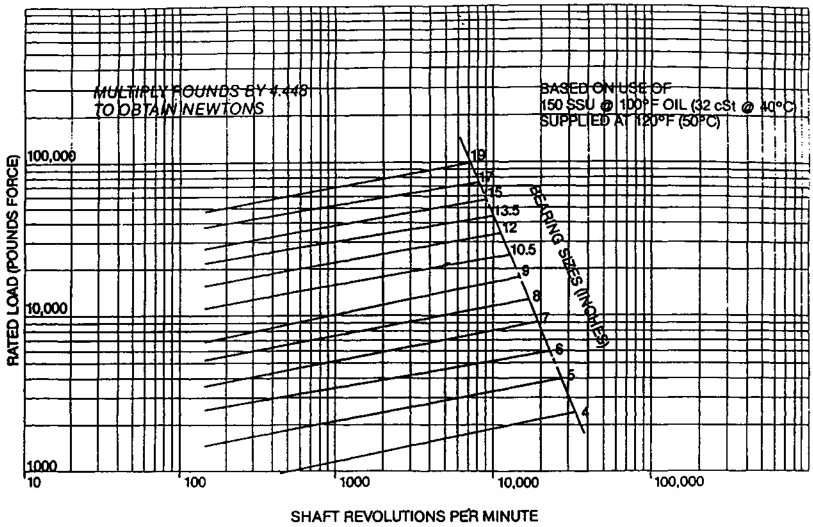


Figure 7.57 Thrust bearing rated load vs. speed (Courtesy of Kingsbury Corp.)

The oil film breaks down, thus allowing contact between the steel thrust collar and soft thrust bearing pad overlay (Babbitt). Once this thin layer (1/16") is worn away, steel to steel contact occurs resulting in significant turbo-compressor damage.

Thrust pad temperature sensors, located directly behind the babbitt at the pad maximum load point protect the compressor by tripping the unit before steel to steel contact can occur.

Figure 7.57 presents different Kingsbury bearing size rated capacities as a function of speed.

Figure 7.58 shows how thrust pad temperature and thrust load are related for a given thrust bearing size and shaft speed. Note that the greater the thrust load (P.S.I.), the smaller the oil film and the greater the effect of oil viscosity on oil flow and heat removal. Based on a maximum load of 500 psi, it can be seen from Figure 7.58 that a turbo-compressor thrust bearing pad temperature trip setting should be between 260° and 270°F.

Other than to support the rotor in an axial direction, the other function of the thrust bearing is to continuously maintain the axial position of the rotor. This is accomplished by locating stainless steel shims between the thrust bearing assembly and compressor axial bearing support plates. The most common thrust assembly clearance with the thrust shims installed is 0.011 – 0.014. These values vary with thrust bearing size. The vendor instruction book must be consulted to determine the proper clearance.

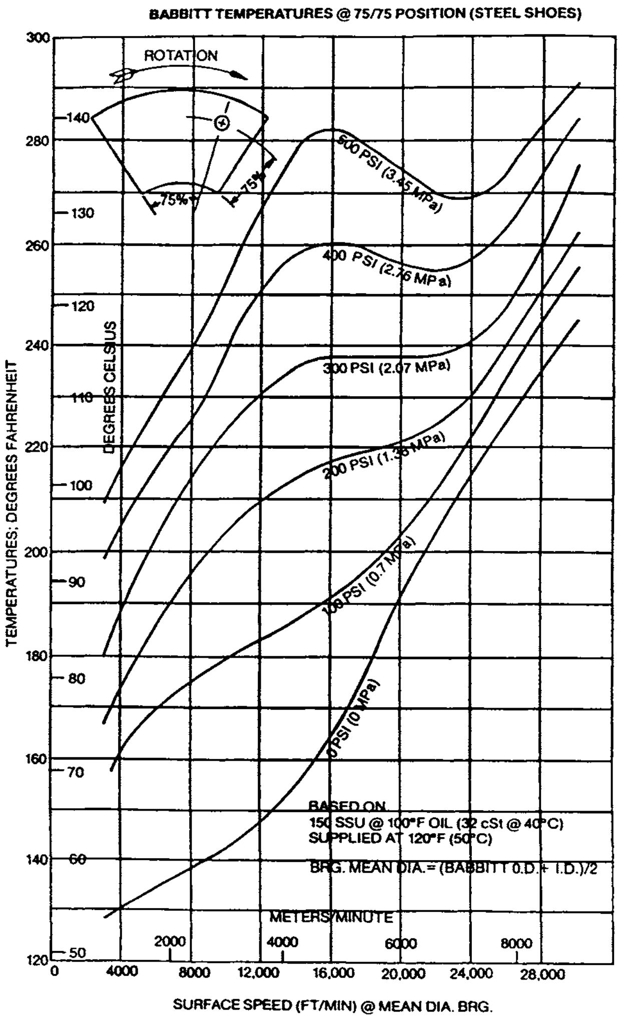


Figure 7.58 The relationship between thrust pad temperature and thrust load (Courtesy of Kingsbury, Inc.)

The following procedure is used to assure that the rotor is properly positioned in the axial direction.

1. With thrust shims removed, record total end float by pushing rotor axially in both directions (typically .250"–.500").
2. Position rotor as stated in instruction book.
3. Install minimum number of stainless steel thrust shims to limit end float to specified value.*

*An excessive number of thrust shims act as a spring resulting in a greater than specified axial clearance during full thrust load conditions.

Proper running position of the rotor is critical to obtaining optimum efficiency and preventing axial rubs during transient and upset conditions (start-up, surge, etc.)

Impeller thrust forces

Every reaction type compressor blade set or impeller produces an axial force towards the suction of the blade or impeller. Refer to Figure 7.59.

In this example, the net force towards the compressor suction is 2,000 psi for the set of conditions noted. Note that the pressure behind the impeller is essentially constant (50 psi), but the pressure on the front side of impeller varies (from 50 psi to 40 psi) because of the pressure drop across the eye labyrinth. Every impeller in a multistage compressor will produce a specific value of axial force towards it's suction at a specific flow rate, speed and gas composition. A change in any or all of these parameters will produce a corresponding change in impeller thrust.

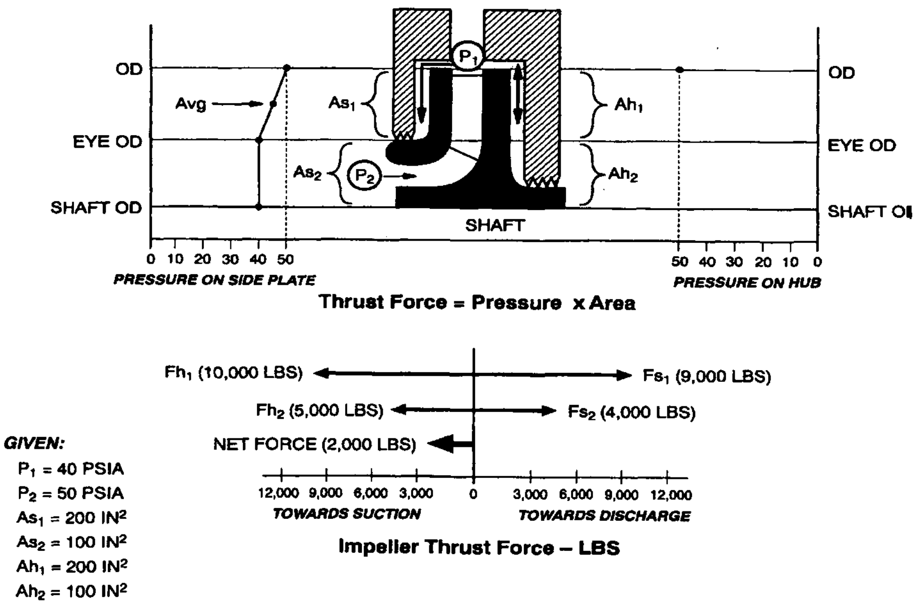
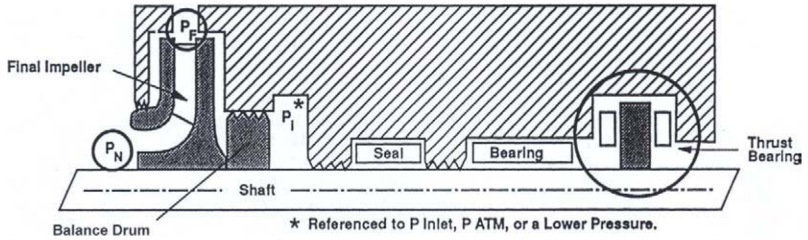


Figure 7.59 Impeller thrust force



Total Impeller Thrust (LB) = Σ Individual Impeller Thrust
Balance Drum Thrust (LB) = (P_F - P_I) x (Balance Drum Area)
Thrust Bearing Load (LB) = Total Impeller Thrust – Balance Drum Thrust

Examples of Rotor Thrust

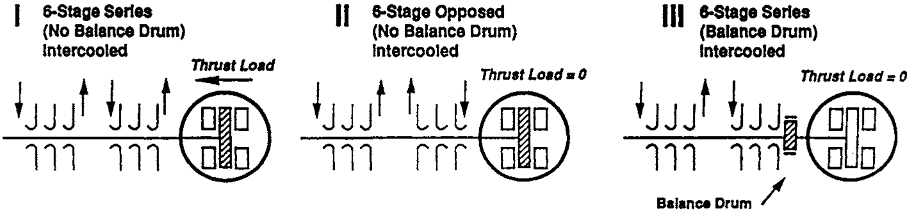


Figure 7.60 Rotor thrust force

Rotor thrust balance

Figure 7.60 shows how a balance drum or opposed impeller design reduces thrust force. The total impeller force is the sum of the forces from the individual impellers. If the suction side of the impellers is opposed, as noted in Figure 7.60, the thrust force will be significantly reduced and can approach 0. If the suction side of all impellers are the same (in series), the total impeller thrust force can be very high and may exceed the thrust bearing rating. If this is the case, a balance drum must be mounted on the rotor as shown in Figure 7.60. The balance drum face area is varied such that the opposing force generated by the balance drum reduces the thrust bearing load to an acceptable value. The opposing thrust force results from the differential between compressor discharge pressure (P_F) and compressor suction pressure (P_I) since the area behind the balance drum is usually referenced to the suction of the compressor. This is accomplished by a pipe that connects this chamber to the compressor suction. This line is typically called the 'balance line'.

It is very important to note that a balance drum is used only where the thrust bearing does not have sufficient capacity to absorb the total compressor axial load. And the effectiveness of the balance drum

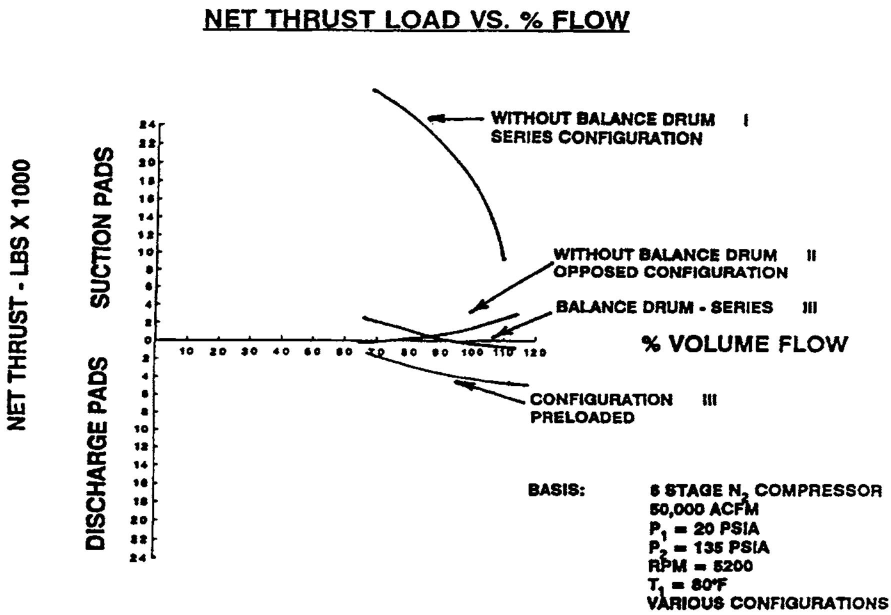


Figure 7.61 Rotor system designed four different ways

depends directly on the balance drum seal. Fail the seal, (open clearance significantly) and thrust bearing failure can result.

A common misunderstanding associated with balance drum systems is that a balance drum always reduces the rotor thrust to zero. Refer to Figure 7.61 and observe that this statement may or may not be true depending on the thrust balance system design. And even if it is, the thrust is zero only at one set of operating conditions.

Figure 7.61 shows a rotor system designed four (4) different ways. Note how the thrust **always** changes with the flow rate regardless of the design. Another misconception regarding thrust balance systems is the normal or 'active' direction of thrust. In many cases, the active thrust is assumed to always be towards the suction of the compressor. Observing Figure 7.61, it is obvious that the 'active' direction can change when the turbo-compressor has a balance drum or is an opposed design. It is recommended that the use of active thrust be avoided where possible and that axial displacement monitors be labeled to allow determination of the thrust direction at all times.

Please refer to Figure 7.62 which shows a typical thrust displacement monitor.

These monitors detect thrust position by targeting the shaft end, thrust collar or other collar on the rotor. Usually two or three probes (multiple voting arrangement) are provided to eliminate unnecessary compressor trips. The output of the probes is noted on the monitor as either + (normal) or – (counter). However, this information gives no direct indication of the axial direction of the thrust collar. The following procedure is recommended:

1. With compressor shutdown, push rotor towards the suction and note direction of displacement indicator.
2. Label indicator to show direction towards suction of compressor.

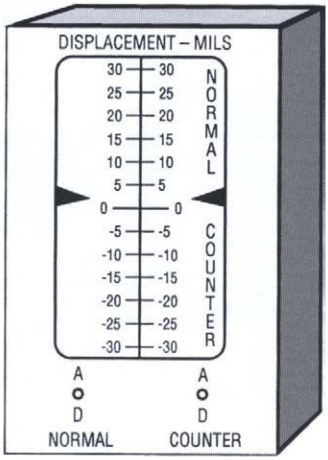


Figure 7.62 Typical axial thrust monitor

Knowing the actual direction of the thrust can be very useful during troubleshooting exercises in determining the root cause of thrust position changes.

Thrust condition monitoring

Failure of a thrust bearing can cause long term and possibly catastrophic damage to a turbo-compressor. Condition monitoring and trending of critical thrust bearing parameters will optimize turbo-compressor reliability.

The critical thrust bearing condition monitoring parameters are:

- Rotor position
- Thrust pad temperature
- Balance line ΔP

Rotor position is the most common thrust bearing condition parameter and provides useful information regarding the direction of thrust. It also provides an indication of thrust load but does not confirm that thrust load is high. Refer to Figure 7.63.

All axial displacement monitors have pre-set (adjustable) values for alarm and trip in both thrust directions. Typically, the established procedure is to record the thrust clearance (shims installed) during shutdown and set the alarm and trip settings as follows:

Alarm = $\frac{\text{Clearance}}{2}$ + 10 mils (each direction)

Trip = Alarm Setting + 5 mils (each direction)

The above procedure assumes the rotor is in the mid or zero position of the thrust clearance. An alternative method is to hand push the rotor to the assumed active position and add appropriate values for alarm and trip.

The writer personally recommends the first method since an active direction of thrust does not have to be assumed.

As noted, axial displacement monitors only indicate the quantity of thrust load. False indication of alarm or even trip settings can come from:

- Compression of thrust bearing components
- Thermal expansion of probe adaptors or bearing brackets
- Loose probes

It is strongly recommended that any alarm or trip displacement value be confirmed by thrust pad temperature if possible prior to taking action. Please refer back to Figure 7.58 of this chapter and note that the thrust pad temperature in the case of thrust pad overload is approximately 250°F. If an axial displacement alarm or trip signal is activated **observe** the corresponding thrust pad temperature. If it is below 220°F, take the following action:

- Observe thrust pads. If no evidence of high load is observed (pad

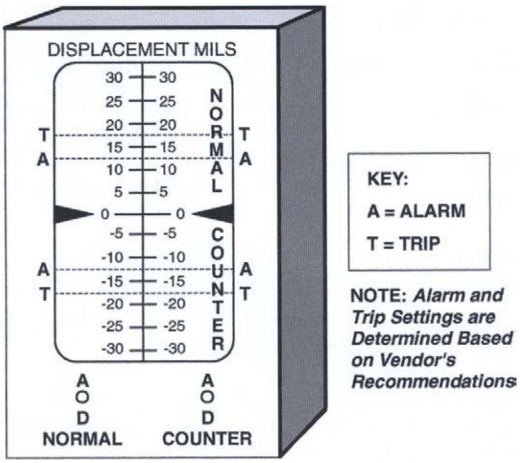


Figure 7.63 Typical axial displacement monitor

and back of pad) confirm calibration of thrust monitor and change settings if necessary.

The last condition monitoring parameter for the thrust system is balance line pressure drop. An increase of balance line ΔP will indicate increased balance drum seal leakage and will result in higher thrust bearing load. Noting the baseline ΔP of the balance line and trending this parameter will provide valuable information as to the root cause of a thrust bearing failure. In many field case histories, the end user made many thrust bearing replacements until an excessive balance drum clearance was discovered as the root cause of the thrust bearing failure. It is a good practice to always check the balance line ΔP after reported machine surge. Surging will cause high internal gas temperatures which can damage the balance drum seal.

Mechanical seals
(Pump mechanical seal component knowledge)

- Introduction
- Function of mechanical seals
- The seal system
- Controlling flush flow to the seal
- Examining some causes of seal failures
- Seal configurations
- Flush system types
- Auxiliary stuffing box and flush plans

Introduction

The part of the pump that is exposed to the atmosphere and through which passes the rotating shaft or reciprocating rod is called the stuffing box. A properly sealed stuffing box prevents the escape of pumped liquid. Mechanical seals are commonly specified for centrifugal pump applications (Refer to Figure 7.64).

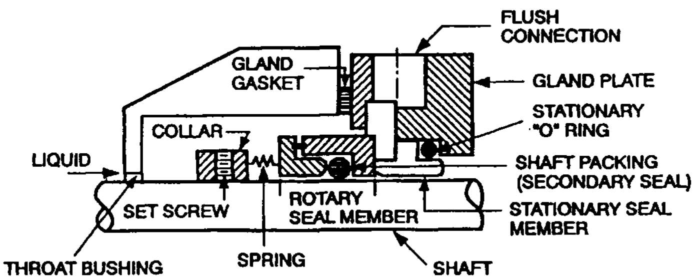


Figure 7.64 Typical single mechanical seal

Function of mechanical seals

The mechanical seal is comprised of two basic components (Refer to Figure 7.65).

Basic seal components

- Stationary member fastened to the casing
- Rotating member fastened to shaft, either direct or with shaft sleeve

Figure 7.65 Basic seal components

The mating faces of each member perform the sealing. The mating surface of each component is highly polished and they are held in contact with a spring or bellows which results in a net face loading closure force (Refer to Figure 7.64).

In order to meet the objective of seal design (prevent fluid escape to the atmosphere), additional seals are required. These seals are either 'O' rings, gaskets or packing (Refer to Figure 7.64). For high temperature applications (above 400°) the secondary seal is usually 'Graphoil' or 'Kalrez' material in a 'U' or chevron configuration. An attractive alternative is to eliminate the secondary seal entirely by using a bellows seal since the bellows replaces the springs and forms a leak tight element thus eliminating the requirement for a secondary seal (Refer to Figure 7.66).

To achieve satisfactory seal performance for extended periods of time, proper lubrication and cooling is required. The lubricant, usually the pumped product, is injected into the seal chamber and a small amount passes through the interface of the mating surfaces. Therefore, it can be stated that all seals leak and the amount of leakage depends on the pressure drop across the faces. This performance can be considered like flow through an equivalent orifice (Refer to Figure 7.67).

The amount of heat generated at the seal face is a function of the face loading and friction coefficient, which is related to material selection and lubrication. Figure 7.68 shows the equation for calculating the amount of heat which needs to be removed by the flush liquid.

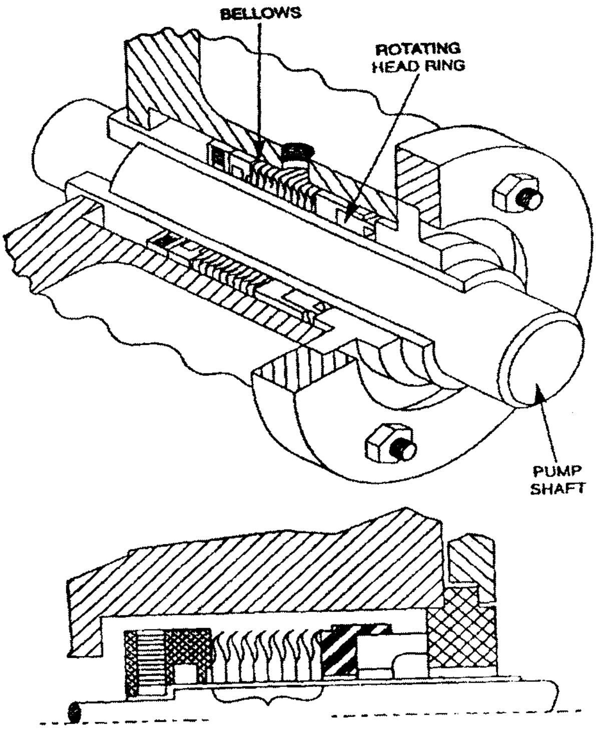


Figure 7.66 Metal bellows seal

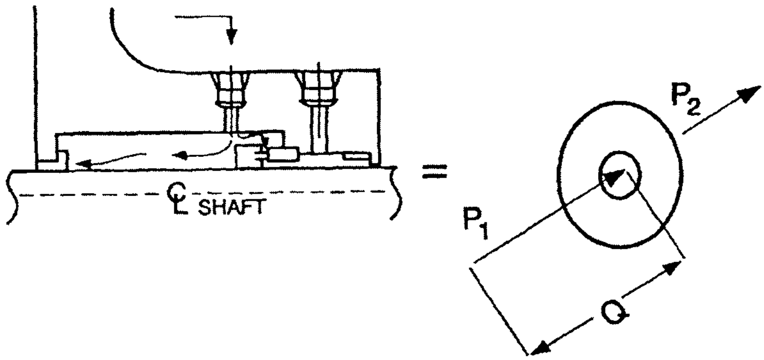


Figure 7.67 Equivalent orifice flow across seal faces

Heat generated by mechanical seal

$Q = 500 \bullet S.G. \bullet Q_{INJ} \bullet Cp \bullet \Delta T$

where: Q = heat load (BTU/HR)

S.G. = specific gravity of injection liquid

C_p= specific heat of injection liquid $\frac{BTU}{LB-°F}$

ΔT = temperature rise of injection liquid (°F)

Q_{INJ}= injection liquid flow rate (G.P.M.)

Figure 7.68 Heat generated by mechanical seal

As the lubricant flows across the interface, it is prone to vaporization. The initiation point of this vaporization is dependent upon the flush liquid pressure and its relationship to the margin of liquid vapor pressure at the liquid temperature. The closer the liquid flush pressure is to the vapor pressure of the liquid at the temperature of the liquid, the sooner vaporization will occur (Refer to Figure 7.69).

The seal system

To assure reliable trouble free operation for extended periods of time, the seal must operate in a properly controlled environment. This requires that the seal be installed correctly so that the seal faces

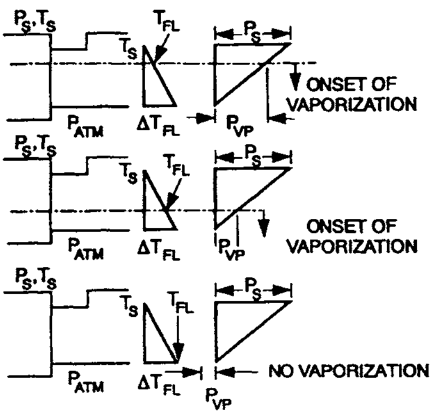


Figure 7.69 Typical seal face pressure temperature relationship to vaporization

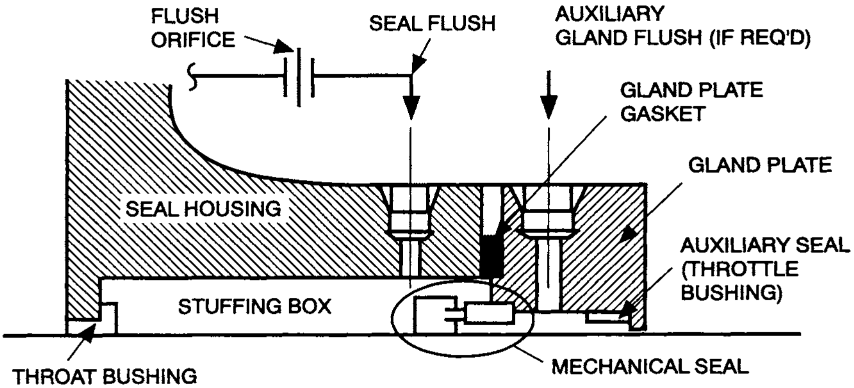


Figure 7.70 Simple seal system

maintain perfect contact and alignment and that proper lubrication and cooling be provided. A typical seal system for a simple single mechanical seal is comprised of the seal, stuffing box throat bushing, liquid flush system, auxiliary seal and auxiliary flush or barrier fluid (when required) (Refer to Figure 7.70).

The purpose of the seal is to prevent leakage of pumped product from escaping to the atmosphere. The liquid flush (normally pumped product from the discharge) is injected into the seal chamber to provide lubrication and cooling. An auxiliary seal is sometimes fitted to the gland plate on the atmospheric side of the seal chamber. Its purpose is to create a secondary containment chamber when handling flammable or toxic fluids which would be considered a safety hazard to personnel if they were to leak to atmosphere. A liquid (non-toxic) flush or barrier fluid, complete with a liquid reservoir and appropriate alarm devices can be used to assure toxic fluid does not escape to the atmosphere.

Controlling flush flow to the seal

The simple seal system shown in Figure 7.71 incorporates an orifice in the flush line from the pump discharge to the mechanical seal. Its purpose is to limit the injection flow rate to the seal and to control pressure in the seal chamber. A minimum bore diameter or 1/8" is normally specified (to minimize potential of blockage) and the orifice can either be installed between flanges or in an orifice nipple.

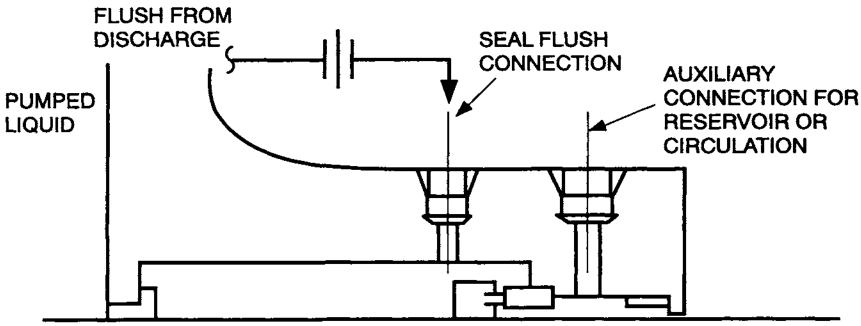


Figure 7.71 Seal flow control orifice

Examining some causes of seal failures

An indication of some causes of seal failures can be obtained while the seal is operating. When you consider the seal as an equivalent orifice, examination of 'tell tale' symptoms can indicate potential failure causes for which corrective action can be implemented or at least can provide direction of subsequent failure analysis (Refer to Figure 7.72). It should be noted that improper application, installation, and/or manufacturing errors can also result in mechanical seal failures.

Comments	Possible causes	Comments/recommendations
■ Seal squeal during operation	Insufficient amount of liquid to lubricate seal faces	Flush line may need to be enlarged and/or orifice size may need to be increased
■ Carbon dust accumulating on outside of seal area	Insufficient amount of liquid to lubricate seal faces	See above
	Liquid film vaporizing/ flashing between seal faces	Pressure in seal chamber may be too low for seal type
■ Seal spits and sputters in operation (popping)	Product vaporizing/flashing across seal faces	Corrective action is to provide proper liquid environment of the product at all times
		1. Increase seal chamber pressure if it can be achieved within operating parameters (maintain at a minimum of 25 Psig above suction pressure)

2. Check for proper seal balance with manufacturer
3. Change seal design to one not requiring as much product temperature margin (ΔT)
4. Seal flush line and/or orifice may have to be enlarged
5. Increase cooling of seal faces

Note: A review of seal balance requires accurate measurement of seal chamber pressure, temperature and product sample for vapor pressure determination

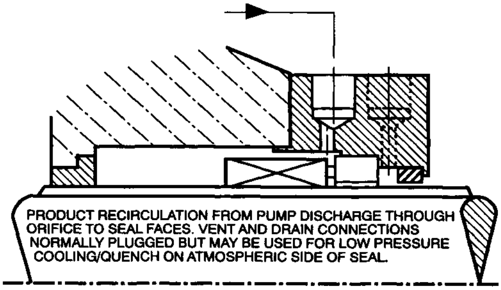
Figure 7.72 Possible causes of seal failure

Seal configurations

Mechanical seals are the predominant type of seals used today in centrifugal pumps. They are available in a variety of configurations, depending upon the application service conditions and/or the User's preference (Refer to Figures 7.73 to 7.76 for the most common arrangements used in refinery and petrochemical applications).

Single mechanical seal applications

Single mechanical seals (Refer to Figure 7.73) are the most widely used seal configuration and should be used in any application where the liquid- is non-toxic and non-flammable. As mentioned earlier in this section, many single mechanical seals are used with flammable and even toxic liquids and only rely on the auxiliary seal throttle bushing to prevent leakage to atmosphere. Since the throttle bushing does not positively contain leakage, State and Federal environmental regulations now require use of a tandem or double seal for these applications. In some plants, a dynamic type throttle bushing ('Impro' or equal) is used to virtually eliminate leakage of the pumped fluid to atmosphere in the event of a mechanical seal failure.



APPLICATIONS: NON HYDROCARBON, HYDROCARBON LIQUIDS. SPECIAL FEATURES INCORPORATED DEPENDING ON CHEMICAL CONTAMINANTS

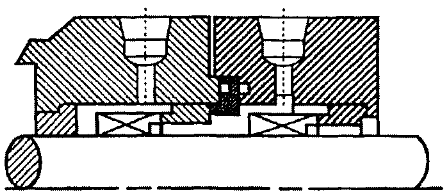
Figure 7.73 Single mechanical seal

Tandem mechanical seal applications

Tandem mechanical seals (Refer to Figure 7.74) are used in applications where the pumped fluid is toxic and/or flammable. They consist of two (2) mechanical seals (primary and back-up). The primary seal is flushed by any selected seal flush plan. The back-up seal is provided with a flush system incorporating a safe, low flash point liquid. A pressure alarm is provided to actuate on increasing stuffing box pressure between the primary and back-up seal thus indicating a primary seal failure. Since the pumped product now occupies the volume between the seals, failure of the back-up seal will result in leakage of the pumped fluid to atmosphere. In essence any time a tandem seal in alarm, it is actually a single seal and should be shut down immediately to assure that the toxic and/or flammable liquid does not leak to atmosphere.

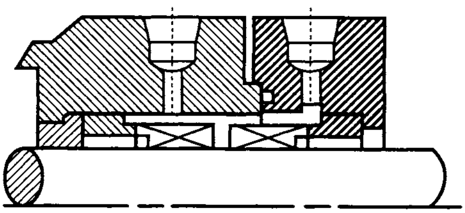
Double mechanical seal applications

Double mechanical seals (Refer to Figure 7.75) are used in applications where the pumped fluid is flammable or toxic and leakage to atmosphere cannot be tolerated under any circumstances. Typical process applications for double seals are H_2S service, Hydrofluoric acid alkylation services or sulfuric acid services.



APPLICATIONS: TOXIC, EXPLOSIVE HAZARD FROM LEAKAGE, CRYOGENIC LIQUIDS

Figure 7.74 Tandem seal arrangement



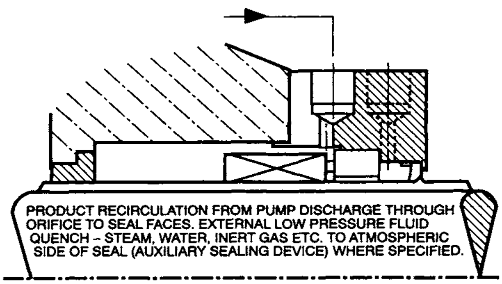
APPLICATIONS: SIMILAR TO THESE USED FOR
TANDEM SEALS

Figure 7.75 Double mechanical seal

Leakage of the pumped fluid to the atmosphere is positively prevented by providing a seal system, whose liquid is compatible with the pumped liquid, that continuously provides a safe barrier liquid at a pressure higher than the pumped fluid. The seals are usually identical in design with the exception that one seal incorporates a pumping ring to provide a continuous flow of liquid to cool the seals. Typical double seal system components are: reservoir, cooler, pressure switch and control valve.

Liquid/Gas tandem mechanical seal applications

In this configuration (Refer to Figure 7.76) a conventional single liquid mechanical seal is used as the primary and a gas seal (non-contacting faces) that can temporarily act as a liquid seal in the event of primary seal failure serves as the back-up seal. This seal configuration is used in low specific gravity applications where the pumped fluid is easily vaporized. Using a gas seal as the back-up has the advantage of eliminating the vessel, cooler and pumping ring necessary for conventional tandem liquid seals.



APPLICATIONS: CAN BE USED FOR LIQUIDS ABOVE AUTO
IGNITION TEMP. TO PREVENT FIRE IF
LEAKAGE EXPOSED TO ATMOSPHERE

Figure 7.76 Liquid/gas tandem seal combination

This application is well proven and has been used successfully for natural gas liquids, propane, ethylene, ethane and butane pump applications.

Double gas seal applications

Before leaving this subject, a relatively new application utilizes two (2) gas seals in a double seal configuration and uses N_2 or air as a buffer maintained at a higher pressure than the pumped fluid to positively prevent the leakage of pumped fluid to atmosphere. This configuration, like the tandem liquid/gas seal mentioned above eliminates the seal system required in a conventional liquid double seal arrangement. Note however, that the pumped product must be compatible with the small amount of gas introduced into the pumped fluid. This configuration cannot be used in recycle (closed loop) services.

An excellent resource for additional information covering design, selection and testing criteria, is API Standard 682.

Flush system types

Providing the proper environment for the seal is a key factor in achieving satisfactory seal operation. In conjunction with defining the seal design and materials of construction, it is necessary to decide which type of seal flush system will be selected to lubricate and cool the seal faces. The API industry has developed various systems to accommodate the requirements of almost every possible sealing arrangement (Refer to Figure 7.77 and 7.77A).

Clean product systems (Plan 11)

In this plan, product is routed from the pump discharge to the seal chamber for lubricating and cooling the seal faces. It will also vent air and/or vapors from the chamber as it passes to the pump suction through the throat bushing. NOTE: For single stage pumps without back hub pumping vanes, it is necessary to have holes in the impeller that reduce the pressure behind the impeller below the discharge pressure to allow flush flow to exit the stuffing box through the throat bushing. (Refer to Figure 7.78).

Clean product flush (Plan 13)

In this plan, the product is routed from behind the pump impeller, through the throat bushing into the stuffing box and out of the stuffing box, through an orifice back to the suction (Refer to Figure 7.79) This plan is used primarily in vertical pump applications because it provides

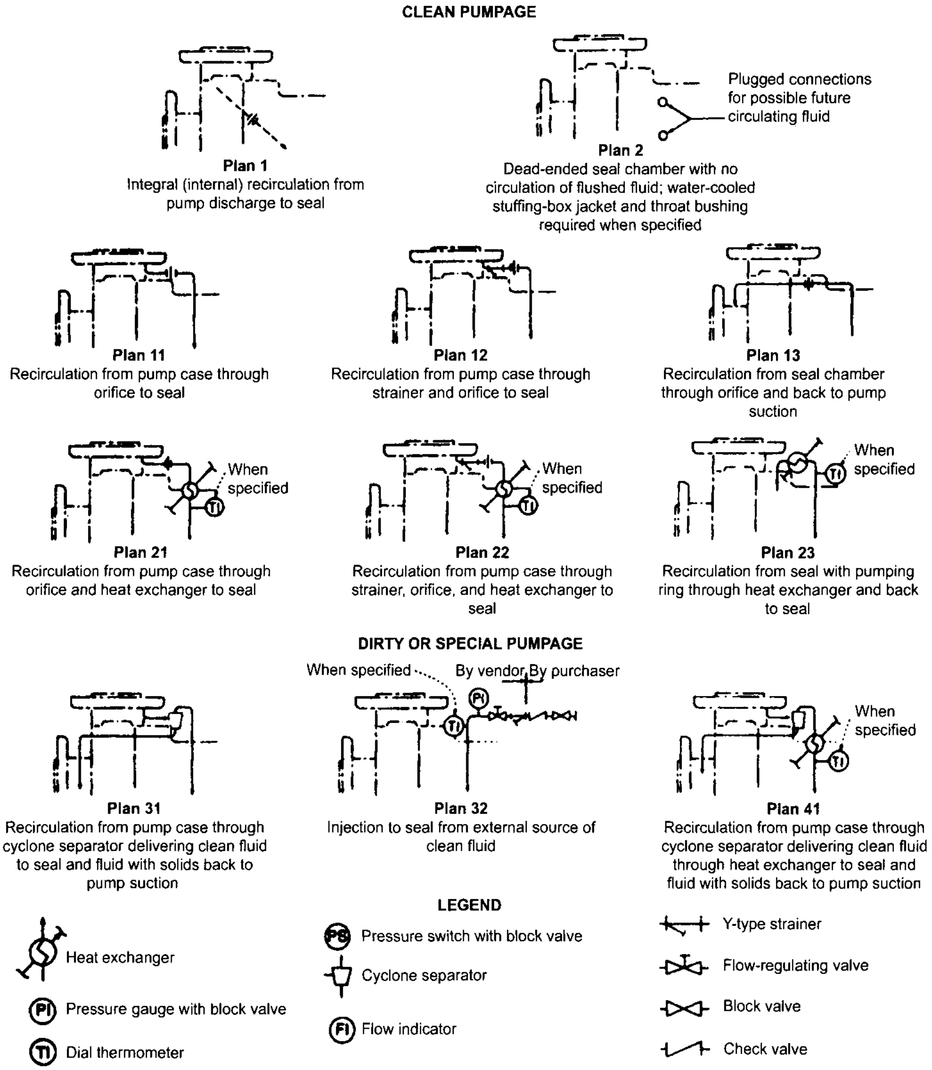
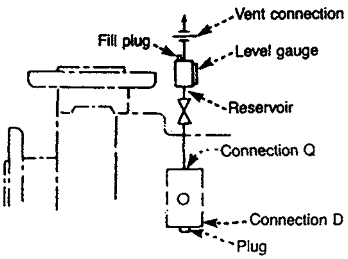


Figure 7.77 All flush plans reprinted with the permission of The American Petroleum Institute

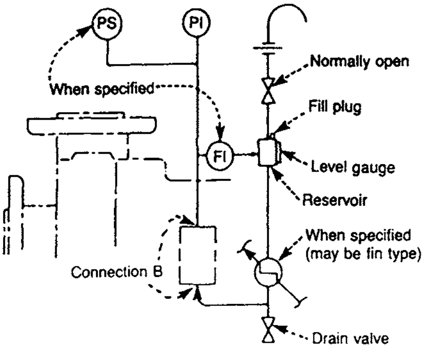
positive venting of the stuffing box. It is also used in single stage pump applications that do not employ pumping vanes or holes in the impellers.

Dirty product system (Plan 31)

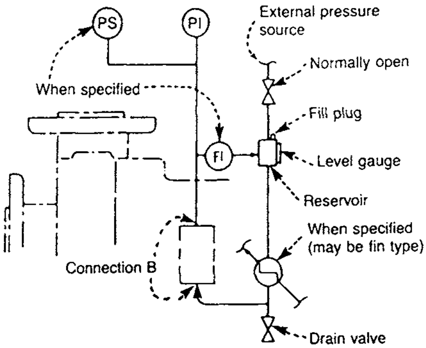
Liquid product from the pump discharge is routed to the seal chamber through a cyclone separator which is selected to optimize removal of solids across an individual pump stage (Refer to Figure 7.80 and to Figure 7.81 for guidelines covering the use of cyclones).



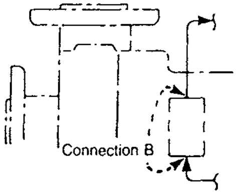
Plan 51
Dead-ended blanket (usually methanol, typically used with auxiliary sealing device (single- or double-seal arrangement



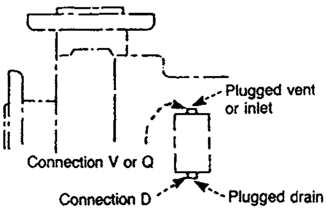
Plan 52
Nonpressurized external fluid reservoir with forced circulation; typically used with tandem-seal arrangement



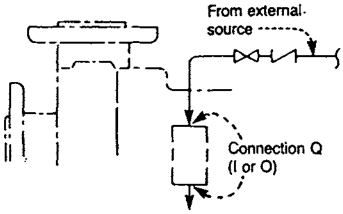
Plan 53
Pressurized external fluid reservoir with forced circulation; typically used with double-seal arrangement



Plan 54
Circulation of clean fluid from external system typically used with double-seal arrangement

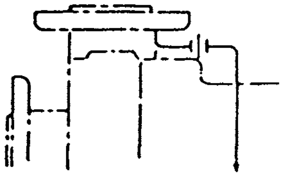


Plan 61
Tapped connections for purchaser's use; applies when purchaser is to supply fluid (steam, gas, water, etc.) to auxiliary sealing device (single- or double-seal arrangement



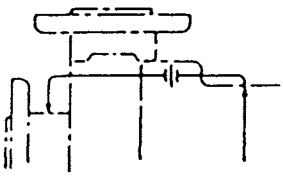
Plan 62
External fluid quench (steam, gas, water, etc.; typically used with throttle bushing or auxiliary sealing device (single- or double-seal arrangement

Figure 7.77A API Flush plans continued



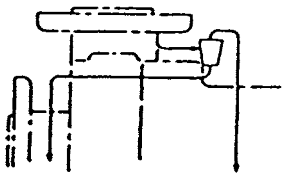
Plan 11
Recirculation from pump case through orifice to seal

Figure 7.78 API Flush plan 11



Plan 13
Recirculation from seal chamber through orifice and back to pump suction

Figure 7.79 API Flush plan 13



Plan 31
Recirculation from pump case through cyclone separator delivering clean fluid to seal and fluid with solids back to pump suction

Figure 7.80 API Flush plan 31

Guidelines for use of cyclones

- Do not use cyclones when differential pressure is less that 25 Psi
- Consider using orifice when pressure differential exceeds cyclone design differential
- Solids to be removed should have density at least twice that of the fluid
- Efficiency of separation is reduced as differential pressure across cyclone varies from design differential
- Separation efficiency drops as particle size decreases

Figure 7.81 Guidelines for use of cyclones

When a clean, cool seal flush liquid is mandated for reasons of preventing solids accumulation in the seal chamber, an external liquid flush system (Plan 32) is used. The fluid pressure is higher than that behind the impeller so that flow is always towards the pump suction,

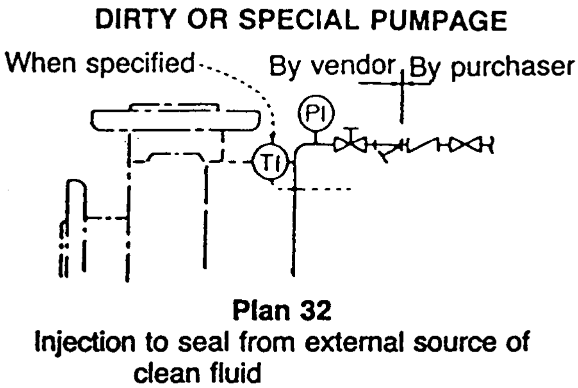


Figure 7.82 API Flush plan 32

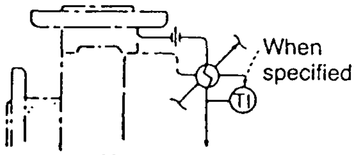
thus preventing back flow of dirty product into the seal chamber (Refer to Figure 7.82). When using Flush Plan 32, it is important to confirm that the flush fluid will not vaporize in the stuffing box.

High temperature product flush system (plan 23)

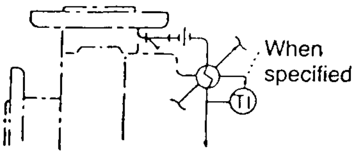
This flush plan is desirable when it is necessary to maintain the required margin between liquid vapor pressure (at seal chamber temperature) and seal chamber pressure. The feature about this plan is that the cooler only removes heat generated by the seal faces plus the heat soak through the shaft from the process. A throat bushing is installed in the seal chamber to isolate the product in this area from that in the impeller area of the pump. A circulating device (pumping ring) is mounted on the seal which circulates liquid in the seal chamber through a cooler and back to the seal chamber. It is more efficient than Plans 21 and 22 which incorporate a cooler to continuously cool the flush from the discharge. These plans are simply Flush Plan 1 1 with the addition of a cooler. Flush Plans 21, 22 or 23 should be used when the temperature of the pumped fluid is above 400°F and a bellows seal is not applied. It is important to confirm that the pumped fluid is clean when using Flush Plan 23 since there is not a constant external flush. This plan is typically used for boiler feed pump applications (Refer to Figure 7.83).

Low temperature product flush/buffer system (plan 52)

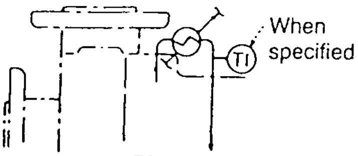
This system is well suited for low temperature applications such as ethylene, propylene and other low temperature liquids which are susceptible to forming ice on the seal faces when the atmospheric side of the seal is exposed to the atmosphere; thus separating the faces and resulting in excessive seal leakage. This plan consists of a tandem (dual) seal with a buffer liquid between them. A seal pot containing the buffer



Plan 21
Recirculation from pump case through orifice and heat exchanger to seal

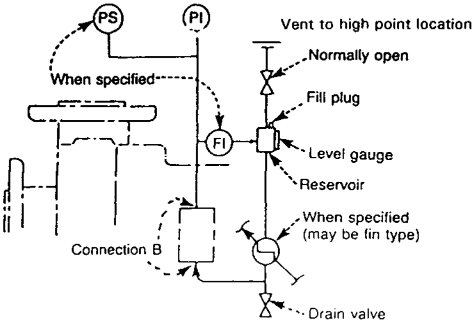


Plan 22
Recirculation from pump case through strainer, orifice, and heat exchanger to seal



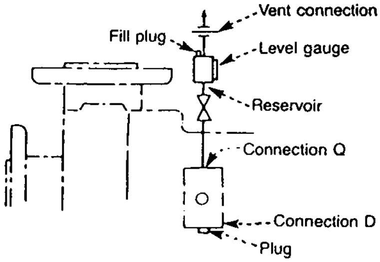
Plan 23
Recirculation from seal with pumping ring through heat exchanger and back to seal

Figure 7.83 API high temperature flush plans 21, 22, 23



Plan 52
Nonpressurized external fluid reservoir with forced circulation; typically used with tandem-seal arrangement

Figure 7.84A Plan 52

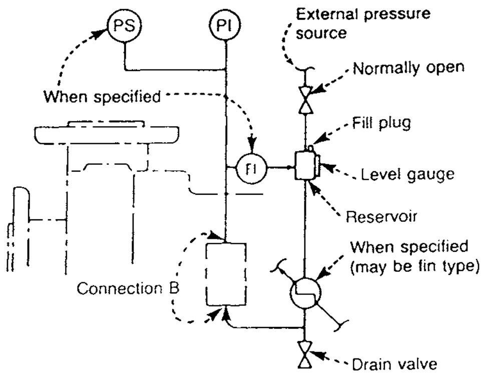


Plan 51
Dead-ended blanket (usually methanol, see Note 3); typically used with auxiliary sealing device (single- or double-seal arrangement in Figure D-1)

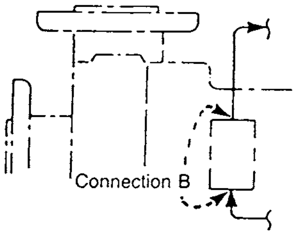
Figure 7.84B Plan 51

liquid (usually methanol – a drying agent) is vented to a lower pressure vent system. The seal pot system is usually equipped with a pressure switch to sound an alarm if the inner seal product leakage cannot be adequately carried away through the orifice vent system (Refer to Figure 7.84A). When the alarm sounds, the pump should be shut down as soon as possible since the back-up seal is now functioning as the primary seal and will leak the pumped fluid to atmosphere if it fails.

Seal flush Pan 51 may also be used when leakage to atmosphere cannot be tolerated. Plan 51 incorporates a dead ended system design and is shown in Figure 7.84B.



Plan 53
Pressurized external fluid reservoir (see Note 3) with forced circulation; typically used with double-seal arrangement in Figure D-1



Plan 54
Circulation of clean fluid from external system typically used with double-seal arrangement

Figure 7.85A & B Plans 53 and 54

Toxic or flammable product flush system (plan 53)

This system is used when leakage to the atmosphere cannot be tolerated (see Figure 7.85A). It consists of a dual seal arrangement with a barrier liquid between them. A seal pot contains the barrier liquid at a pressure higher than seal chamber pressure (usually 20–25 PSI). Inner seal leakage will always be barrier liquid leakage into the product, resulting in some product contamination. The barrier liquid should be selected on the basis of its compatibility with the product. An internal pumping device (pumping ring) is used to circulate the barrier liquid into and out of the seal chamber through the seal pot. The integrity of the system always needs to be monitored to assure that seal pot pressure level is maintained with barrier liquid.

Plan 54 is also a dual system which utilizes a pressurized barrier liquid from an external reservoir or system to supply clean cool liquid to the seal chamber. As described in the previous plan, the barrier liquid pressure level is higher than the seal chamber pressure (usually 20–25 PSI) so that inner seal leakage is always into the pump. With this plan, it is also necessary to consider the compatibility of the barrier liquid and the pumped product. This system is considered to be one the most reliable systems available. However, it is more complex and more costly than other systems (Refer to Figure 7.85B).

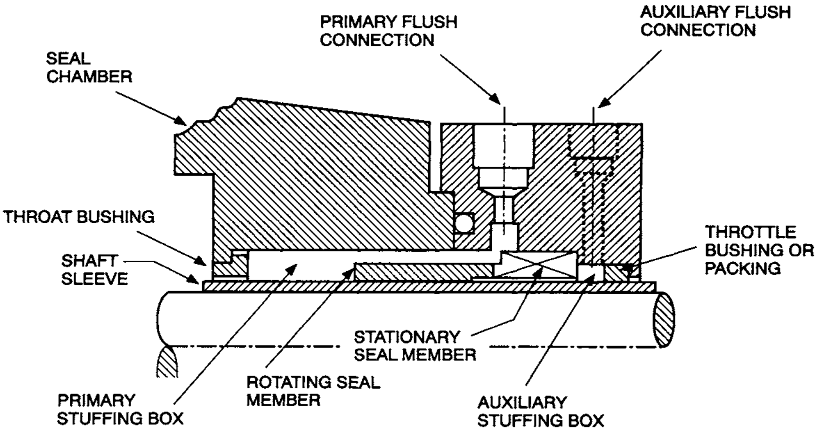


Figure 7.86 Auxiliary stuffing box

Auxiliary stuffing box and flush plans

As mentioned previously, the auxiliary stuffing box can be used as an auxiliary sealing device in the event of seal failure. (Refer to Figure 7.86). It is important to note that the auxiliary stuffing box contains a restricted flow seal (packing or close fitting throttle bushing) and does not positively contain the pumped fluid. Therefore, the auxiliary stuffing box seal device is for emergency containment of the pumped fluid only. The pump should be shut down immediately in the event of leakage observed from the auxiliary stuffing box if a quench is not supplied. It is always a good practice to require that the auxiliary stuffing box drain connection, which will come plugged, be piped to a drain system that meets environmental standards.

When the pumped fluid can vaporize and form hard deposits, the auxiliary stuffing box is used to contain a quench fluid that will dissolve (wash away) the hard deposits at the exit of the seal faces, thus eliminating seal face wear.

A typical refinery quench application is the use of low pressure (50 PSI) steam in and out of the auxiliary stuffing box to dissolve coke deposits on the seal face. This application also has the added advantage of keeping the standby pump seal warm which prevents thermal expansion problems in start-up. This arrangement uses a throttle bushing as the external seal.

Water is also used as an auxiliary stuffing box flush in caustic applications where solid deposits need to be flushed from the seal face. When a water flush is used, two (2) rows of packing are usually provided in the auxiliary stuffing box as an external seal to minimize external water leakage.

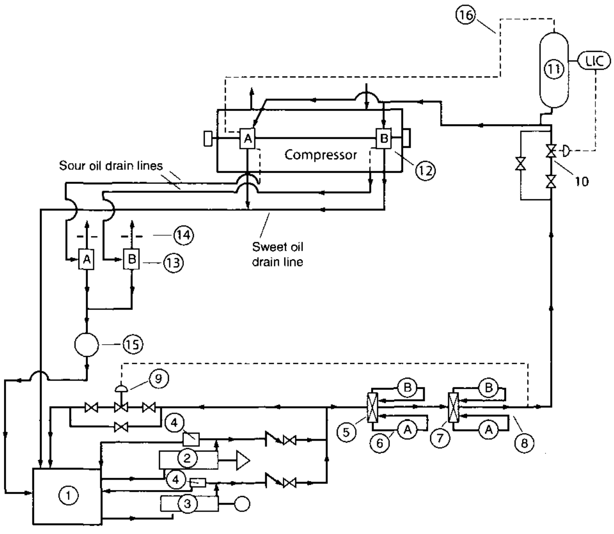
Compressor seal system overview and types
(Compressor liquid seal component knowledge)

- Introduction
- The supply system
- The seal housing system
- Seal supply systems
- Seal supply system summary
- Seal liquid leakage system

Introduction

There are numerous types of fluid seal systems since the types of seals utilized, sealing fluids and sealing pressures vary widely. Regardless of the type of seal used, the function of a critical equipment seal system is as follows: '*To continuously supply clean fluid to each specified seal interface point at the required differential pressure, temperature and flow rate*'. A typical seal system for a centrifugal compressor is shown in Figure 7.87.

The system shown is for use with clearance bushing seals. Let's examine



1. Oil reservoir
2. Main screw pump - turbine driven
3. Aux screw pump - motor driven
4. Relief valves
5. Transfer valve
6. Oil coolers (A & B)
7. Transfer valve
8. Oil filters (A & B)
9. Back pressure control valve
10. Level control valve
11. Overhead seal oil tank (typically 15 ft above compressor centerline)
12. ISO sleeve seals (also called liquid bushing seals)
13. Seal oil drainers (A & B)
14. Drainer orifice vents (A & B)
15. Degassing tank
16. Reference gas line to seal oil tank

Note: component condition instrumentation and autostarts not shown

Figure 7.87 API 614 lube/seal oil system for ISO-sleeve seals (Courtesy of Elliott Co.)

this figure by proceeding through the system from the seal oil reservoir through the compressor shaft seal and back through the reservoir. As previously discussed, the concept of sub-systems can be useful here. The seal oil system shown can be divided into four major sub-systems:

A The supply system

B The seal housing system

C The atmospheric drain system

D The seal leakage system

A The supply system

This system consists of the reservoir, pumping units, the exchangers, transfer valves, temperature control valves, and filters. The purpose of this sub-system is to continuously supply clean, cool sealing fluid to the seal interfaces at the correct differential pressure.

B The seal housing system

This system is comprised of two different seals. A gas side bushing, and an atmospheric bushing. The purpose of the seal housing system is to positively contain the fluid in the compressor and not allow leakage to the atmosphere. The seal fluid is introduced between both seal interfaces, thus constituting a double seal arrangement. Refer to Figure 7.88 for a closer examination of the seal.

The purpose of the gas side bushing seal is to constantly contain the reference fluid and minimize sour oil leakage. This bushing can be

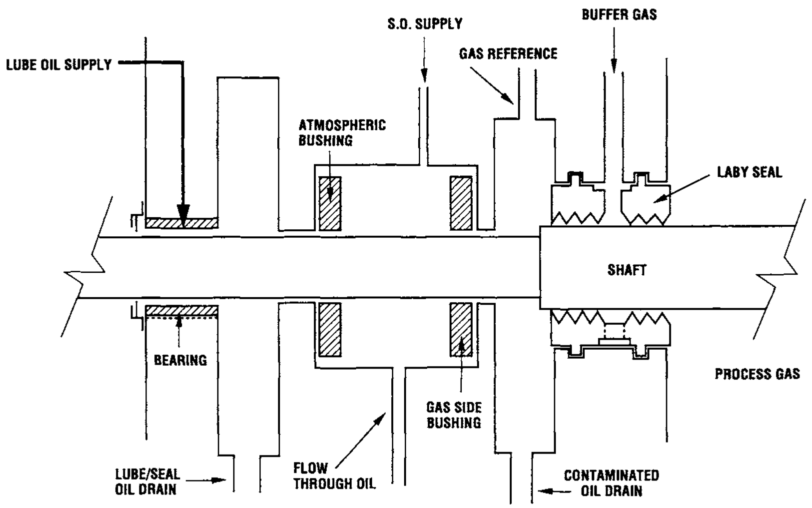


Figure 7.88 Bushing seal schematic (Courtesy of M.E. Crane Consultant)

conceived as an equivalent orifice. This concept is similar to bearings previously discussed with the exception that the referenced downstream pressure of the gas side bushing can change. In order to assure a constant flow across this 'orifice,' the differential pressure must be maintained constant. Therefore, every compressor seal system is designed to maintain a constant differential against the gas side seal. The means of obtaining this objective will be discussed as we proceed.

The other seal in the system is the atmospheric bushing whose purpose is to minimize the flow of seal liquid to an amount that will remove frictional heat from the seal. This bushing can be conceptualized as a bearing, since the downstream pressure is usually atmospheric pressure. In systems that directly feed into a bearing the atmospheric bushing downstream pressure will be constant (approximately 20 psi). However, the upstream supply pressure will vary with the pressure required by the sealing media in the compressor.

As an example, if a seal system is designed to maintain a constant differential of 5 lbs. per square inch between the compressor process gas and the seal oil supply to the gas side bushing, the supply pressure with 0 PSIG process gas pressure, would be 5 psi to both the gas side bushing and atmospheric bushing. Therefore, gas side bushing and the atmospheric bushing differential would both be equal to 5 psi. If the process gas pressure were increased to 20 psi, the seal oil system would maintain a differential of 5 psi across the gas side seal, and the supply pressure to the gas side bushing and atmospheric bushing would be 25 psi. In this case, the differential across the gas side bushing would remain constant at 5 psi, but the atmospheric bushing differential pressure would increase from 5 to 25 psi. As a result, a primary concern in any seal liquid system is the assurance that the atmospheric bushing receives proper fluid flow under all conditions. After the seal fluid exits the seal chamber, it essentially returns through two additional sub-systems.

C The atmospheric draining system

The flow from the atmospheric bushing, if it does not directly enter the bearing system, will return to the seal oil reservoir. In addition, flow from any downstream control valve will also return through the atmospheric drain system to the seal oil reservoir. Both these streams should be gas free since they should not come in contact with the process gas.

D The seal leakage system

The fluid that enters the gas side bushing is controlled to a minimum amount such that it can be either discarded or properly returned to the reservoir after it is degassed. Typically, this amount is limited to less than 20 gallons per day per seal. Since this liquid is in contact with the high speed shaft it is atomized and combines with sealing gas to enter the leakage system. This system consists of:

- An automatic drainer
- A vent system
- Degassing tank (if furnished)

The function of each component is as follows:

The drainer

The drainer contains the oil-gas mixture from the gas side seal. The liquid level under pressure in the drainer, is controlled by an internal float or external level control valve to drain oil back to the reservoir or the de-gassing tank, as required.

The vent system

The function of the seal oil drainer vent system is to assure that all gas side seal oil leakage is directed to the drainer. This is accomplished by referencing the drainer vent to a lower pressure than the pressure present at the gas side seal in the compressor. The drainer vent can be routed back to the compressor suction, suction vessel or a lower pressure source.

The degassing tank

This vessel is usually a heated tank, with ample residence time (72 hours or greater) to sufficiently de-gas all seal oil such that it will be returned to the reservoir and meet the seal oil specification. (Viscosity, flashpoint, dissolved gasses, etc.) These items will be discussed in detail later.

We will now proceed to discuss each of the major sub-systems in detail. Defining the function of each such that the total operation of a seal system can be simplified.

The supply system

Referring to definition of a seal oil system, it can be seen that the function is identical to that of a lube oil system, with one exception. The exception is that the seal fluid must be delivered to the seals at the specified differential pressure. Let's examine this requirement further.

Refer again to Figure 7.88 which shows an equivalent orifice diagram for a typical compressor shaft seal. Notice, that the atmospheric bushing downstream pressure is constant (atmospheric pressure). However, the gas side bushing pressure is referenced to the compressor process pressure. This pressure can and will vary during operation. If it were always constant, the requirement for differential pressure control would not be present in a seal system and would be identical to that of a lubricating system. Another way of visualizing the systems is to understand that the lube system utilizes differential pressure control as well, but the reference pressure (atmospheric pressure) is constant and consequently all control valves need only control lube oil pressure. However seal systems require some means of constant differential pressure control (reference gas pressure to seal oil supply pressure). This objective can be accomplished in many different ways. Referring back to Figure 7.87 it can be seen that the supply system function is identical to that of a lube oil system with the exception that the liquid is referenced to a pressure that can vary and must be controlled to maintain a constant differential between the referenced pressure and the seal system supply pressure. The sizing of the seal oil system components is also identical to that of the lube oil system components. Refer back and observe the heat load and flow required of each seal is determined in a similar way to that of the bearings. Seals are tested at various speeds and a necessary flow is determined to remove the heat of friction under various conditions. The seal oil flow requirements and corresponding heat loads, are then tabulated and pumps exchangers, filters, and control valves are sized accordingly.

The seal oil reservoir is sized exactly the same way as lube oil reservoir in our previous example. The only major difference between the component sizing of a seal and a bearing is that the seal flows across the atmospheric bushing will change with differential pressures. As previously explained, any liquid compressor seal incorporates a double seal arrangement. The gas side seal differential is held constant by system design. The atmospheric side seal differential varies with varying seal reference (process) pressure. Therefore, the total flow to the seals will vary with process pressure and must be specified for maximum and minimum values when sizing seal system components. Remembering the concept of an equivalent orifice, a compressor at atmospheric conditions will require significantly less seal oil flow than it will at high pressure (200 psi) conditions. This is true since the differential across the atmospheric seal and liquid flow will increase from a low value to a significantly higher value, while the gas side bushing differential and liquid flow will remain constant provided seal clearances remain constant.

Many seal system problems have been related to insufficient seal oil flow through the atmospheric bushing at low suction pressure conditions. Close attention to the atmospheric drain cavity temperature is recommended during any off design (low suction pressure condition) operation.

The seal housing system

Regardless of the type, the purpose of any seal is to contain the fluid in the prescribed vessel (pump, compressor, turbine, etc.). Types and designs of seals vary widely. Figure 7.89 shows a typical mechanical seal used for a pump.

Since the contained fluid is a liquid, this seal utilizes that fluid to remove the frictional heat of the seal and vaporize the liquid, thus attaining a perceived perfect seal. A small amount of vaporized liquid constantly exits the pump across the seal face. It is a fact that all seals leak. This is the major reason that many pump applications today are required to utilize seal-less pumps to prevent emission of toxic vapors. The following is a discussion of major types of seal combinations used in centrifugal compressor seal applications.

Gas seals

A typical gas seal is shown in Figure 7.90.

Gas seals have recently drawn attention since their supply systems appear to be much simpler than those of a traditional liquid seal system. Since gas seals utilize the sealed gas or a clean buffer gas, a liquid seal-system incorporating pumps, a reservoir and other components, is not required. However, one must remember that the sealing fluid still must be supplied at the proper flow rate, temperature and cleanliness. As a result, a highly efficient, reliable source of filtration, cooling, and supply must be furnished. If the system relies upon inert buffer gas for continued operation, the supply source of the buffer gas must be as reliable as the critical equipment itself. Gas seal configurations vary and will be discussed in detail in the next section. They can take the form of single, tandem (series), or multiple seal systems. The principle of operation is to maintain a fixed minimum clearance between the rotating and non-rotating face of the seal. The seal employed is

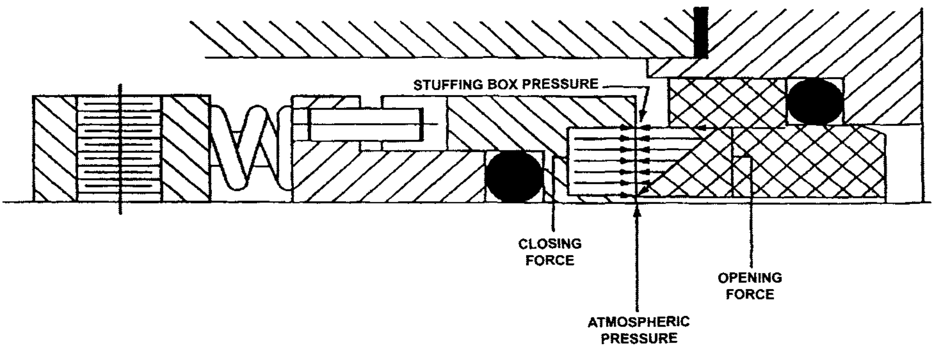
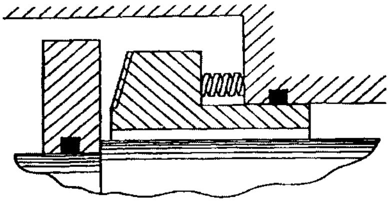
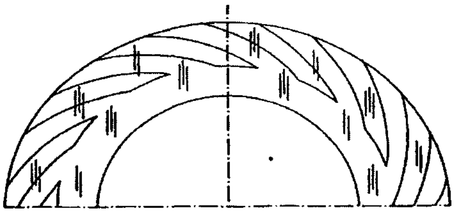


Figure 7.89 Typical pump single mechanical seal



Typical Design For Curved Face — Spiral Groove Non-contact Seal; Curvature May Alternately Be On Rotor



Typical Spiral Groove Pattern On Face Of Seal
Typical Non-contact Gas Seal

Figure 7.90 Typical gas seal (Courtesy of John Crane Co.)

essentially a contact seal with some type of lifting device to maintain a fixed minimum clearance between the rotating faces. It is essential that the gas between these surfaces be clean since any debris will quickly clog areas and reduce the effectiveness of the lifting devices, consequently resulting in rapid damage to the seal faces.

Liquid seals

Traditionally, the type of seal used in compressor service has been a liquid seal. Since the media that we are sealing against is a gas, a liquid must be introduced that will remove the frictional heat of the seal and assure proper sealing. Therefore, all compressor liquid seals take the form of a double seal. That is, they are comprised of two seals with the sealing liquid introduced between the sealing faces. Refer to Figure 7.91. To assure proper lubrication of both the gas side (inboard) and atmospheric side (outboard) seals, the equivalent 'orifices' of each seal must be properly designed such that the differential pressure present provides sufficient flow through the seal to remove the heat of friction at the maximum operating speed. The type of gas side seal used in Figure 7.91 is a contact seal similar to that used in most pump applications. This seal provides a minimum of leakage (five to ten gallons per day per seal) and provides reliable operation. (Continuous operation for 3+ years.) As will be discussed below, the specific types of seals used in the double seal (liquid) configuration can vary.

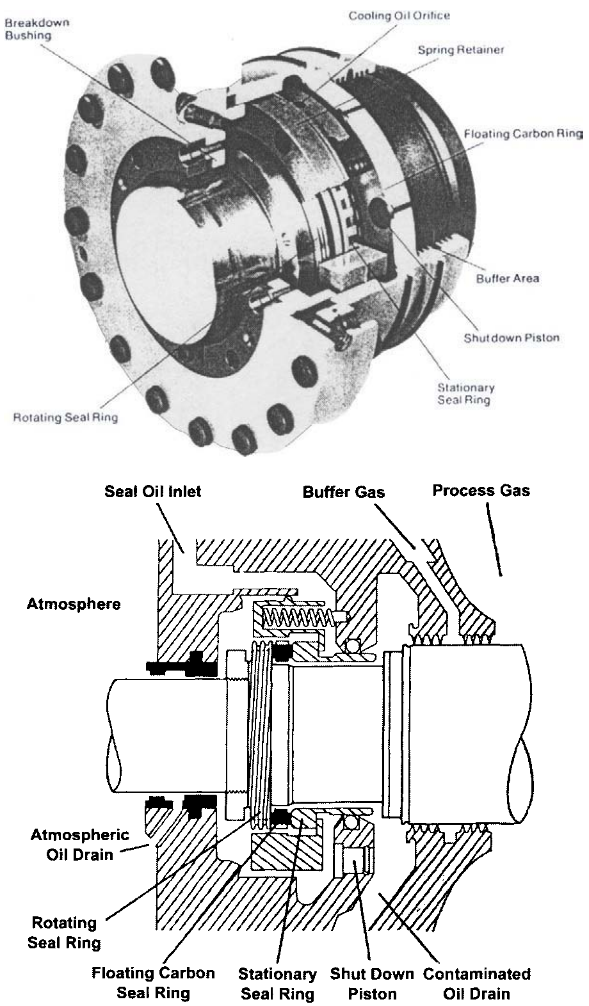
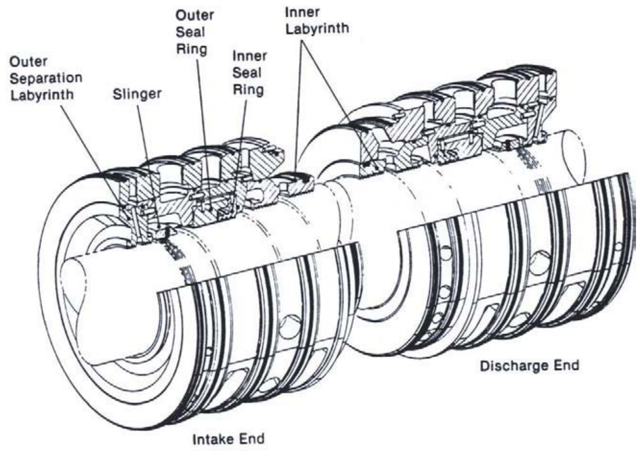


Figure 7.91 Iso carbon seal (Courtesy of Elliott Co.)

Liquid bushing seals

A liquid bushing seal can be used for either a gas side or an atmospheric side seal application. Most seals utilize a liquid bushing seal for an atmospheric bushing application. A typical bushing seal is shown in Figure 7.92.

The principle of a bushing seal is that of an orifice. That is, a minimum clearance between the shaft and the bushing surface to minimize leakage. The bushing seal is designed such that the clearance is sufficient to remove all the frictional heat at the maximum power loss condition of that bushing with the available fluid differential across the



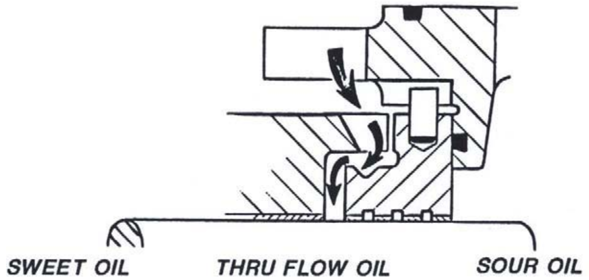


Figure 7.92 Bushing seal - top: oil film seal: bottom: seal oil flow (Courtesy of Dresser-Rand)

bushing. It is important to realize that while acting as a seal, the bushing must not act as a bearing. That is, it must have degrees of freedom (float) to assure that it does not support the load of the rotor. Since its configuration is similar to a bearing, if not allowed freedom of movement, it can act as an equipment bearing and result in a significant change to the dynamic characteristics of equipment with potential to cause damage to the critical equipment. In order to achieve the objectives of a bushing seal, clearances are on the order of 0.0005" diametrical clearance per inch of shaft diameter.

Liquid bushing seals are also used for gas side seals, however, their leakage rate will be significantly larger than that of a contact seal since they are essentially an orifice. When used as a gas side bushing, therefore, the system must be designed to minimize the differential across the bushing. As a result, the differential control system utilized must be accurate enough to maintain the specified oil/gas differential under all operating conditions. The typical design differential across a gas side bushing seal is on the order of five to ten psid. The accurate control of this differential is usually maintained by a level control system.

Referring back to Figure 7.92, one can see that functioning of the bushing seal totally depends on maintaining a liquid interface between the seal and shaft surface. Failure to achieve this results in leakage of gas outward through the seal. It must be fully understood that all bushing seals must continuously maintain this liquid interface to assure proper sealing. All systems incorporating gas side bushing seals must have the seal system in operation whenever pressurized gas is present inside the compressor case. If a liquid interface is not maintained, gas will migrate across the atmospheric bushing seal and proceed through the system returning back to the supply system. There have been cases in such system designs where failures to operate the seal system when the compressor is pressurized have resulted in effectively turning the gas side bushing into a filter for the entire process gas system! This resulted in the supply side of the seal oil system being filled with extensive debris that required lengthy flushing and system cleaning operations prior to putting the unit back into service. Remember, any system incorporating a gas side bushing seal must be designed such that the entrance of process gas into the supply system is prohibited at all time. This can be accomplished by either:

- Continuous buffer gas supply
- A check valve installed as close as possible to seals in the seal oil supply header
- Rapid venting and isolation of the compressor case on seal system failure

In the second and third cases above, supply seal oil piping must be thoroughly checked for debris prior to re-start of the compressor. It is our experience that many bushing seal system problems have resulted from improper attention to the above facts.

Contact seals

Figure 7.93 shows a typical compressor contact seal. As mentioned, these seals are similar in design to pump seals. In order to remove the heat of friction for this type of seal, a sufficient differential pressure above the reference gas must be maintained. Typical differentials for contact seals vary between 35 and 50 lbs. per square inch differential pressure. Leakage rates with a properly installed seal can be maintained between five to ten gallons per day per seal.

A limitation in the use of contact seals are shaft speed, since the contact seal operates on a surface perpendicular to the axis of rotation, the rubbing speed of the seal surface is critical. As a result, contact seals are speed limited. Typical maximum speeds are approximately 12,000 revolutions per minute. Above those speeds, bushing seals are used, since the sealing surface is maintained at a lower diameter and corres-

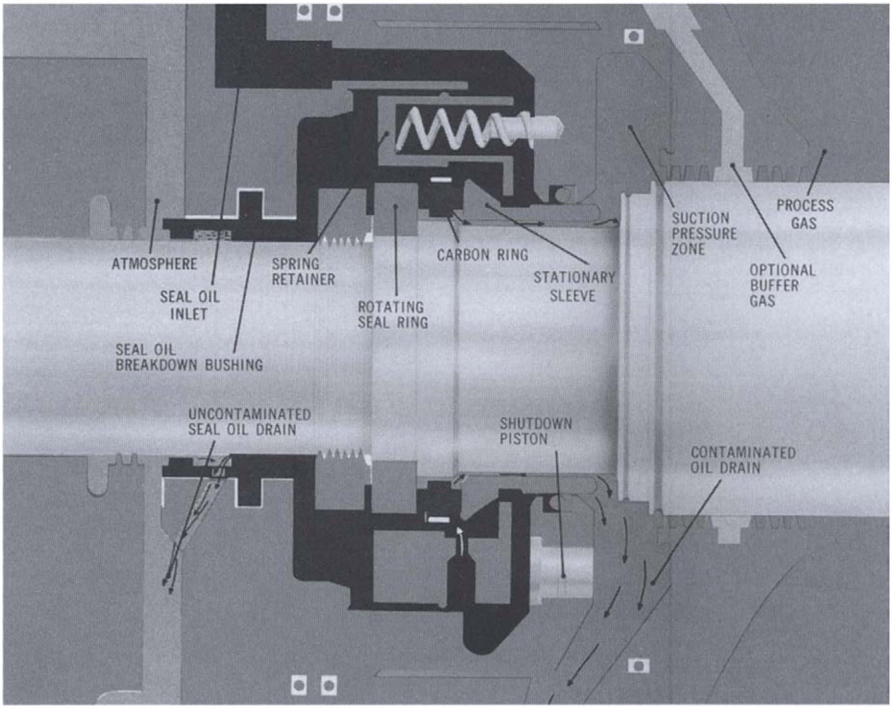


Figure 7.93 Compressor contact seal (Courtesy of Elliott Co.)

pondingly lower rubbing speed. The maximum limit of differential pressure across contact seals is controlled by the materials of construction and is approximately 200 lbs. per square inch differential. As a result, contact seals are usually used for gas side seal applications. They are very seldom utilized for atmospheric seal applications since they are differential pressure limited.

Since the differential pressure required across the seal face is relatively high as compared to a bushing seal, contact seals utilize differential pressure control as opposed to level control for most bushing seals. This fact will be discussed in the next section.

Restricted bushing seals

The last type of seal to be discussed is a restricted bushing type seal. This type of seal is shown in Figure 7.94.

This particular type of restricted bushing seal utilizes a small pumping ring in the opposite direction of bushing liquid flow to compensate for the relatively large leakage experienced with bushing seals by introducing an opposing pumping flow in the opposite direction. Seals of this type can be designed for practically zero flow leakages. However, it must be pointed out that in variable speed applications, the pumping

Compact Design — allows shorter bearing spans for higher critical speeds of the compressor rotor. **Sleeve** (impeller) with interference fit under bushing — protects shaft and simplifies assembly and disassembly. Requires only a jack/puller bolt ring. **Spacer** fit at initial assembly — no field fitting of parts.

ITEM	DESCRIPTION
1	Shaft
2	Impeller
3	Stator
4	Stepped Dual Bushing
5	Bushing Cage
6	Nut
7	Shear Ring
8	Oil/Gas Baffle
9	Spacer Ring

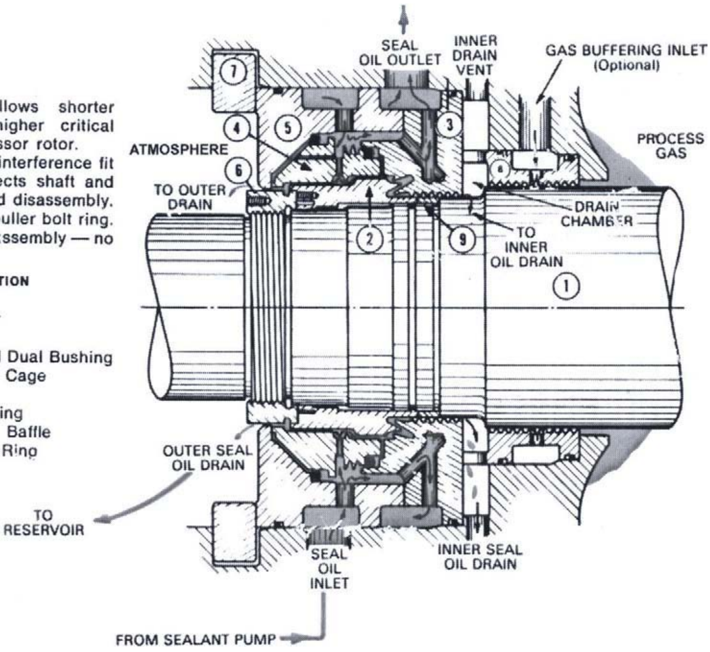


Figure 7.94 Turbo-compressor 'trapped bushing seal' (Courtesy of A.C. Compressor Corp.)

capability of the trapped seal ring must be calculated for both minimum and maximum speeds. Failure to do so can result in the actual pumping of gas from the compressor into the sealing system. It is recommended that such seals be designed to leak a small amount at maximum operating speed. Any retrofits of equipment employing this type of seal should be investigated when higher operating speeds are anticipated. A restricted bushing seal is used exclusively for gas side service.

In summary, the basic types of liquid seals used for compressor applications can be either: open bushing types, contact types, or restricted bushing types. Contact types are used primarily on the gas side. Liquid bushing types are used on either the gas or atmospheric side. Restricted bushing types are used exclusively on the gas side.

We will now investigate various seal system designs using various seal combinations employing the types of seals that have been discussed in this section.

Seal supply systems

As can be seen from the previous discussion, the type of seal system will depend on the type of seal utilized. We will now examine five different types of seal systems, each utilizing a different type of main compressor shaft seal system. As we proceed through each type, the function of each system will become clear.

Example 1: Contact type gas side seal – bushing type atmospheric side seal with cooling flow

This system incorporates a contact seal on the gas side and a bushing seal on the atmospheric side of each end of the compressor. The inlet pressure of the seal fluid on each end is referenced to the suction pressure of the compressor. It should be noted that some applications employ different reference pressures on each end of the compressor. The reference pressure should be taken off the balance drum end, or high pressure end of the compressor, to assure that the oil to gas differential pressure is always at a minimum acceptable value. Therefore, the low pressure end may experience a slightly higher oil to gas differential then the reference end of the seal. Refer to Figure 7.95.

Proceeding through the seal, the seal oil supply, which is referenced to the gas reference pressure, enters the seal chamber. The differential across the gas side contact seal is maintained by a differential pressure

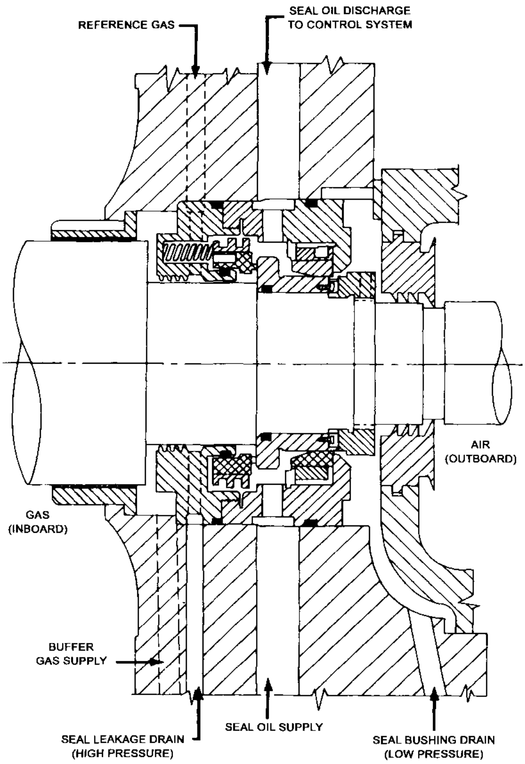


Figure 7.95 Compressor shaft seal (Courtesy of IMO Industries)

control valve located downstream of the seal. Seal oil flows in three separate directions:

- Through the seal chamber (cooling flow)
- Through the gas side contact seal (10–20 gallons/day)
- Through the atmospheric seal

Let's examine the variants of flows across the equivalent orifice of each portion of this configuration.

The gas side contact seal will experience a constant flow, that for purposes of discussion, can be assumed to be zero gallons per minute (since the maximum flow rate will usually be on the order of ten gallons per day).

The atmospheric side bushing seal flow will vary based upon the referenced gas pressure. At low suction pressure conditions, this flow will be significantly less than it will be under high pressure conditions. The seal system design must consider the maximum reference pressure to be experienced in the compressor case to assure that sufficient seal oil flow is available at maximum pressure conditions.

The seal chamber through flow in this seal design is used to remove any excess frictional heat of the seals and is regulatedby the downstream control valve. As an example, let us assume the following values were calculated for this specific seal application.

1. *Gas Side Seal Flow* = 0 GPM
2. *Atmospheric Side Seal Flow*
   Reference Pressure = 0 PSIG
   Seal Flow = 5 GPM
   Reference Pressure = 200 PSIG
   Seal flow = 12 GPM
3. *Flow Through Flow*
   Minimum = 3 GPM (occurring at high ATM bushing flow = 12 GPM)
   Maximum = 12 GPM (occurring at low ATM bushing flow = 3 GPM)
4. *Seal Oil Supply Flow* in both cases = 15 GPM

As shown in the previous example, the required seal oil supply at maximum operating speed required to remove frictional heat is 15 gallons a minute. At start-up, low suction pressure conditions, the control valve must open to allow an additional ten gallons a minute flow through to the seal chamber. At maximum operating pressure, however, the valve only passes a flow of three gallons a minute since 12 gallons a minute exit through the atmospheric bushing. This type of system is less sensitive to low suction pressure operation since flow through oil will remove frictional heat around the atmospheric bushing.

Example 2: Contact type gas side seal – bushing type atmospheric seal with orificed through flow

The only difference between this type of system and the previous system is that the back pressure is maintained constant by a permanently installed through flow orifice. As a result, the differential pressure control valve is installed on the inlet side of the system. The process gas reference is still the same as before, that is, to the highest pressure side of the compressor. Figure 7.96 shows this type of system.

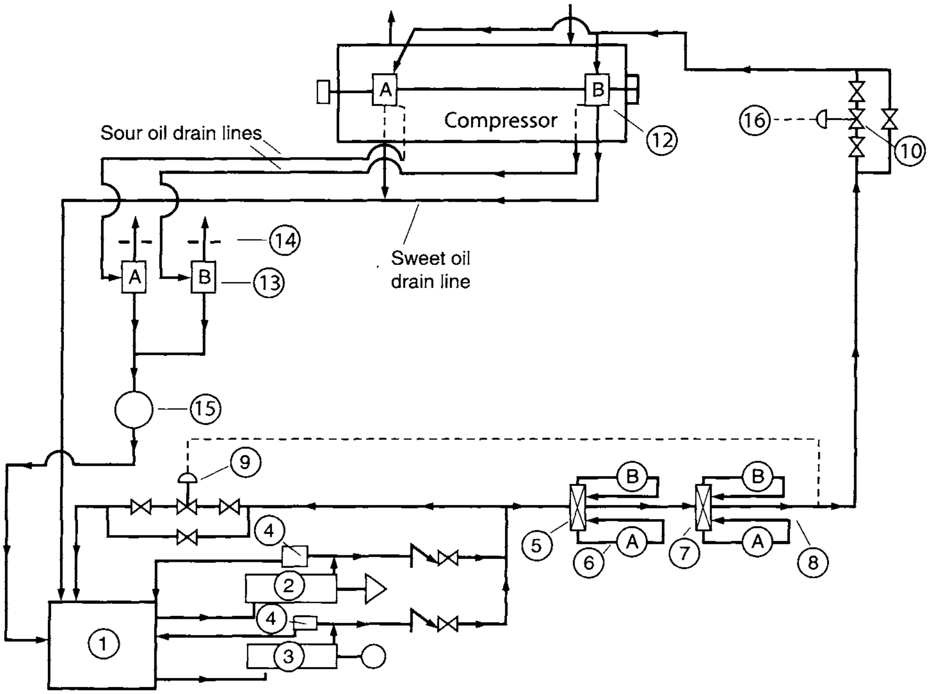
Let us examine the previous example case for this system and observe the differences.

1. *Gas Side Seal Flow = 0 GPM*
2. *Atmospheric Side Seal Flow*
   Reference Pressure = 0 PSIG
   Seal Flow = 5 GPM
   Reference Pressure = 200 PSIG
   Seal Flow = 12 GPM
3. *Flow Through Flow (Orifice)*
   Minimum = 0.5 GPM
   Maximum = 3 GPM

As can be seen, this system is more susceptible to high temperature atmospheric bushing conditions at low suction pressures and must be observed during such operation to assure integrity of the atmospheric bushing. In this system, the control valve will sense supply oil pressure to the seal chamber and control a constant set differential, approximately 35 psid, between the reference gas pressure and the supply pressure. If continued low pressure operation is anticipated with such a system, consideration should be given to a means of changing the minimum flow and maximum flow orifice for various operation points. Externally piped bypass orifices could be arranged such that a bypass line with a large orifice for minimum suction pressure conditions could be installed and opened during this operation. It is important to note, however, that the entire supply system must be designed for this flow condition and control valve must be sized properly to assure proper flow at this condition. In addition, the low pressure bypass line must be completely closed during normal high pressure operation.

Example 3: Bushing gas side seal – bushing atmospheric side seal with no flow through provision

Figure 7.87 shows this type of seal system. In this type of system, the differential control valve becomes a level control valve sensing differential from the level in an overhead tank and is positioned upstream of the unit. Both bushings can be easily conceived as equivalent orifices. The gas side bushing flow will remain constant



1. Oil reservoir
2. Main screw pump - turbine driven
3. Aux screw pump - motor driven
4. Relief valves
5. Transfer valve
6. Oil coolers (A & B)
7. Transfer valve
8. Oil filters (A & B)
9. Back pressure control valve
10. Differential pressure control valve
11. Overhead seal oil tank (typically 15 ft above compressor centerline)
12. ISO sleeve seals (also called liquid bushing seals)
13. Seal oil drainers (A & B)
14. Drainer orifice vents (A & B)
15. Degassing tank
16. Reference gas line

Figure 7.96 API 614 Lube-seal oil system for ISO-carbon seals (Courtesy of Elliott Company) regardless of differential. The atmospheric bushing flow will vary according to seal chamber to atmospheric pressure differential.

Therefore, the atmospheric bushing must be designed to pass a minimum flow at minimum pressure conditions that will remove frictional heat and thus prevent overheating and damage to the seal. Since a gas side bushing seal is utilized, a minimum differential across this orifice must be continuously maintained.

Utilizing the concept of head, the control of differential pressure across the inner seal is maintained by a column of liquid.

As an example, if the required gas side seal differential of oil to gas is 5 psid, by the liquid head equation:

$$\text{Head} = \frac{2.311 \times 5 \text{ psid}}{.85} = 13.6 \text{ ft.}$$

Therefore maintaining a liquid level of 13.6 ft. above the seal while referencing process gas pressure will assure a continuous 5 psid gas side bushing differential. In this configuration, the control valve which senses its signal from the level transmitter, will be sized to continuously supply the required flow to maintain a constant level in the overhead tack. As an example, consider the following system changes from start up to normal operation.

Seal System Flow

Item	Start-up condition	Normal operation
Compressor suction pressure	0 PSIG	200 PSIG
Overhead tank reference pressure	0 PSIG	200 PSIG
Gas side seal bushing flow	0 GPM	0 GPM
Atmospheric side seal bushing flow	5 GPM	12 GPM
Seal pump flow	20 GPM	20 GPM
Bypass valve flow	15 GPM	8 GPM

In this example a change from the start-up to operating condition will increase gas reference pressure on the liquid level in the overhead tank and would tend to push the level downward. Any movement of the level in the tank will result in an increasing signal to the level control valve to open, thus increasing the pressure (assuming a positive displacement pump) to the overhead tank and reestablishing the pre-set level.

In the above example, at 200 psi reference pressure, the bypass valve would close considerably. To increase the pressure supply of the seal oil from 5 psi to 205 psi, the difference of bypass flow through the valve (7 gpm) is equal to the increased flow through the atmospheric bushing at this higher differential pressure condition. Utilizing the concept of equivalent orifices, it can be seen that the additional differential pressure across the atmospheric bushing orifice is compensated for by reducing the effective orifice area of the bypass control valve. This is accomplished by sensing the level in the head tank and maintaining it at a constant value by opening the seal oil supply valve.

As in the case of the orificed through flow example above, this configuration is susceptible to high atmospheric bushing temperatures at low suction pressures and must be monitored during this condition. Repeated high temperatures during low suction pressure conditions should give consideration to re-sizing of atmospheric bushing clearances during the next available turnaround. The original equipment manufacturer should be consulted to assure correct bushing sizing and supply system capability.

Example 4: Gas side bushing seal – atmospheric side bushing seal with through flow design

Refer to Figure 7.97. The only difference between this system and the previous system is that a through flow option is added to allow sufficient flow through the system during changing pressure conditions. The bypass valve in the previous system is replaced in this system by a level control valve referenced from a head tank level transmitter, and is installed downstream of the seal chamber. This system functions in exactly the same way as the system in Example 1. The only difference being that a level control valve in this example replaces the differential control valve in the previous example. Both valves have the same function, that is, to control the differential in the seal chamber between the seal oil supply and the referenced gas pressure. A level control valve is utilized in this example, however, since a bushing seal requires a significantly lower differential between the seal oil supply pressure and the gas reference pressure.

Consider the following example. Assume that a differential control valve would be used as opposed to a level control valve for the system in Figure 7.97. For the start-up case, the differential control valve would have to maintain a differential of 5 psi over the reference gas. When the reference gas pressure were 0 psi, the oil upstream pressure to the valve would be approximately 5 psi. For the operating case, maintaining the same 5 psi differential, the upstream pressure across the valve would be approximately 205 lbs instead of 5 psi. Consequently, the valve position would change significantly, but still would have to control the differential accurately to maintain 5 psi. Reduction of this pressure in any amount below 5 psi could result in instantaneous bushing failure. However, if a level control valve were installed, the accuracy of the valve would be measured in inches of oil instead of psi. Any level control

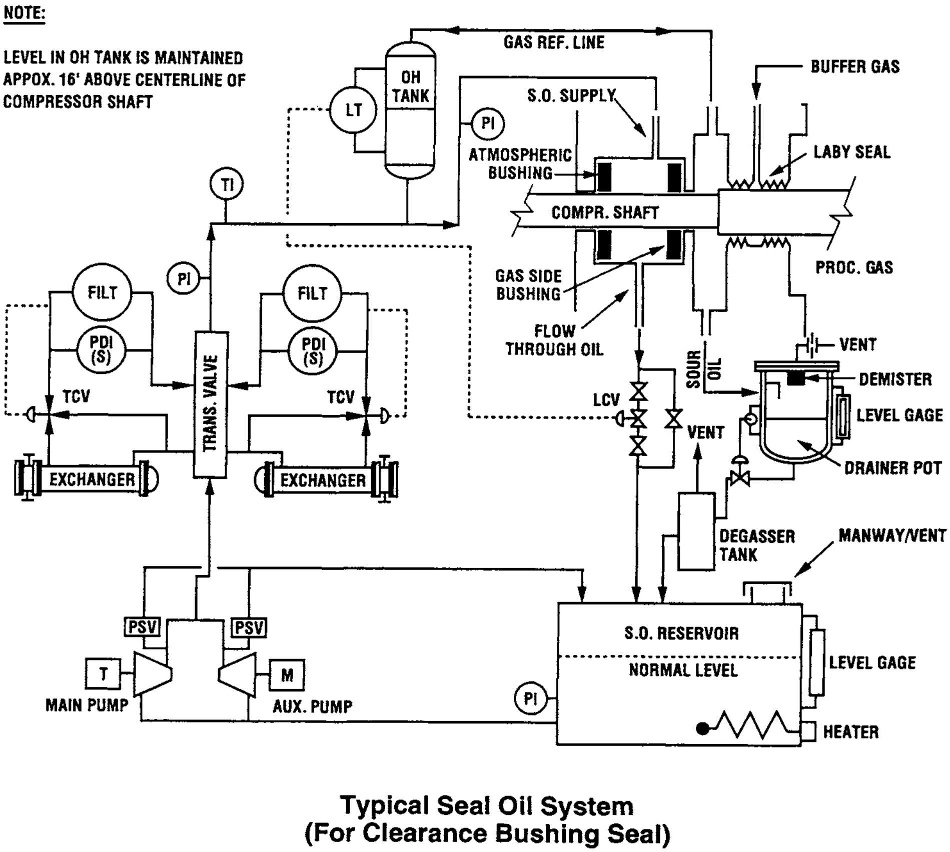


Figure 7.97 Typical seal oil system (Courtesy of M.E. Crane Consultant)

system could control the level within two inches, which would be only a .06 psid variation in pressure differential!

This example shows that the accurate means of controlling differential pressure for systems requiring control of small differential values, is to use level instead of differential control. This system would be designed such that the combination of the atmospheric flow and the through flow through the seal would be equal to the flow from the pump.

Example 5: Trapped bushing gas side seal: atmospheric side bushing seal with flowthrough design

This system would follow exactly the same design as the system described in Example 4. The only difference would be in the amount of flow registered in the seal oil drainer. A trapped bushing system is designed to minimize seal oil drainer pot leakage. Typical values can be less than five gallons/day.

Seal supply system summary

All of the above examples have dealt with a system incorporating one seal assembly. It must be understood that most systems utilize two or more seal system assemblies. Typical multi-stage compressors contain two seal assemblies per compressor body and many applications contain upwards of three compressor bodies in series, or six seal assemblies. Usually each compressor body is maintained at the suction pressure to that body, therefore three discreet seal pressure levels would be required and three differential pressure systems would be utilized. The concepts discussed in this section follow through regardless of the amount of seals in the system. Sometimes, the entire train, that is, all the seals referenced to the same pressure. In this case, one differential seal system could be used across all seals.

In conclusion, remembering the concept of an orifice will help in understanding the operation of these systems. Remember, the gas side bushing is essentially zero flow, the atmospheric side bushing flow varies with changing differential across the seal and any seal chamber through flow will change either as a result of differential across a fixed orifice or the repositioning of the control valve.

Seal liquid leakage system

This seal system sub-system's function is to collect all of the leakage from the gas side seal and return it to the seal reservoir at specified seal fluid conditions. Depending upon the gas condition in the case, this objective may or may not be possible. If the gas being compressed has a tendency to change the specification of the seal oil to off specification conditions, one of two possibilities remain:

- *Introduce a clean buffer gas* between the seal to assure proper oil conditions
- *Dispose of the seal oil leakage*

In most cases, the first alternative is utilized. Once the seal oil is in the drainer, a combination of oil and gas are present. A vent may be installed in the drainer pot to remove some of the gas, or a degassing tank can be incorporated.

This concludes the overview section of seal oil systems. As can be seen from the above discussion, it is evident that the design of a seal oil system follows closely to that of a lube system. The major difference is that the downstream reference pressure of the components (seal) varies, whereas in the case of a lube system it does not. In addition, the collection of the expensive seal oil is required in most cases and a downstream collector, or drainer system, must be utilized. Other than these two exceptions, the design of the seal system is very similar to that of a lube system and the same concepts apply in both cases.

Seal system component design and preventive maintenance

(Compressor liquid seal component knowledge)

- Introduction
- High pressure systems
- Differential pressure control valves
- Level control valves
- Control valve location in the system
- Seal system component design and preventive maintenance

Introduction

In this chapter we will deal with liquid seal supply system component designs of three major areas:

- The pumping system
- The differential control valves
- The level control valves

In the next chapter, we will deal with the downstream side of the seal system, the drainers, demisters, and degassing tanks.

There is a great similarity between the seal system and the lube system. As far as the components are concerned, they are similar. The reservoir, pumps, coolers, filters and control valves are sized in exactly the same way as those of a lube system, the only major areas of differences between a lube and seal supply systems are:

- The arrangement of components in high pressure systems
- Differential control

High pressure systems

High pressure seal systems are defined in general as those systems where the seal pressure is in excess of 1,000 lbs per square inch. This category of sealing systems experiences unique problems. System cooling requirements, high pressure pump start-up, and rapid deterioration of system components in the presence of debris. Refer to

Figure 7.98. This is a typical component arrangement for a combined lube and seal system.

System cooling requirements

In this arrangement, excess (bypass) oil is recirculated uncooled back to the reservoir. The temperature rise of the oil through the pump is proportional to pump differential pressure. In high pressure systems, a significant pump temperature rise occurs. If the combined effect of temperature rise and bypass flow are such that the heat input into the reservoir exceeds the heat loss in the reservoir, a reduction of oil viscosity will occur. High pressure positive displacement rotary pumps are sensitive to lower oil viscosities. At a given pressure, there is a minimum specified operating viscosity. For high pressure systems, careful attention must be drawn to location of the seal oil cooler. In many cases, a bypass cooler or a cooler at the suction of the pump, is recommended to preclude the possibility of pump damage.

Consideration should also be given to cooler leaks in high pressure systems. A small leak can result in a large quantity of oil being taken from the system in a short period of time. Coolers should be considered to be installed in the lowest possible pressure section of the system. In combined lube and seal systems, a recommendation is to install the cooler in the low pressure section of the system and utilize a booster pump for seal oil supply.

High pressure start

In high pressure systems, consideration must be given to the sudden starting of a seal pump against a high system pressure. Pumps can be quickly damaged under this condition. If a pilot operated relief valve is utilized, a time delay can be incorporated to hold the valve open upon simultaneously starting of the standby pump. This will alleviate any high pressure sudden shock problems with positive displacement pumps.

Component failure

The failure rate of components in high pressure systems is greater than other types of auxiliary systems. Pumps are particularly susceptible to failure. High pressure seal pumps can fail from high pressure hydraulic shocks induced during start up. As a result, a significant amount of debris is introduced into the system and the corresponding high pump discharge pressure will cause the relief valve to lift. This action will start the second pump. Since the second pump is usually connected to the same system, this pump will also experience high discharge pressure and recirculate all of its flow. This action will quickly result in high suction temperature in both pumps and dual pump failure, resulting in critical

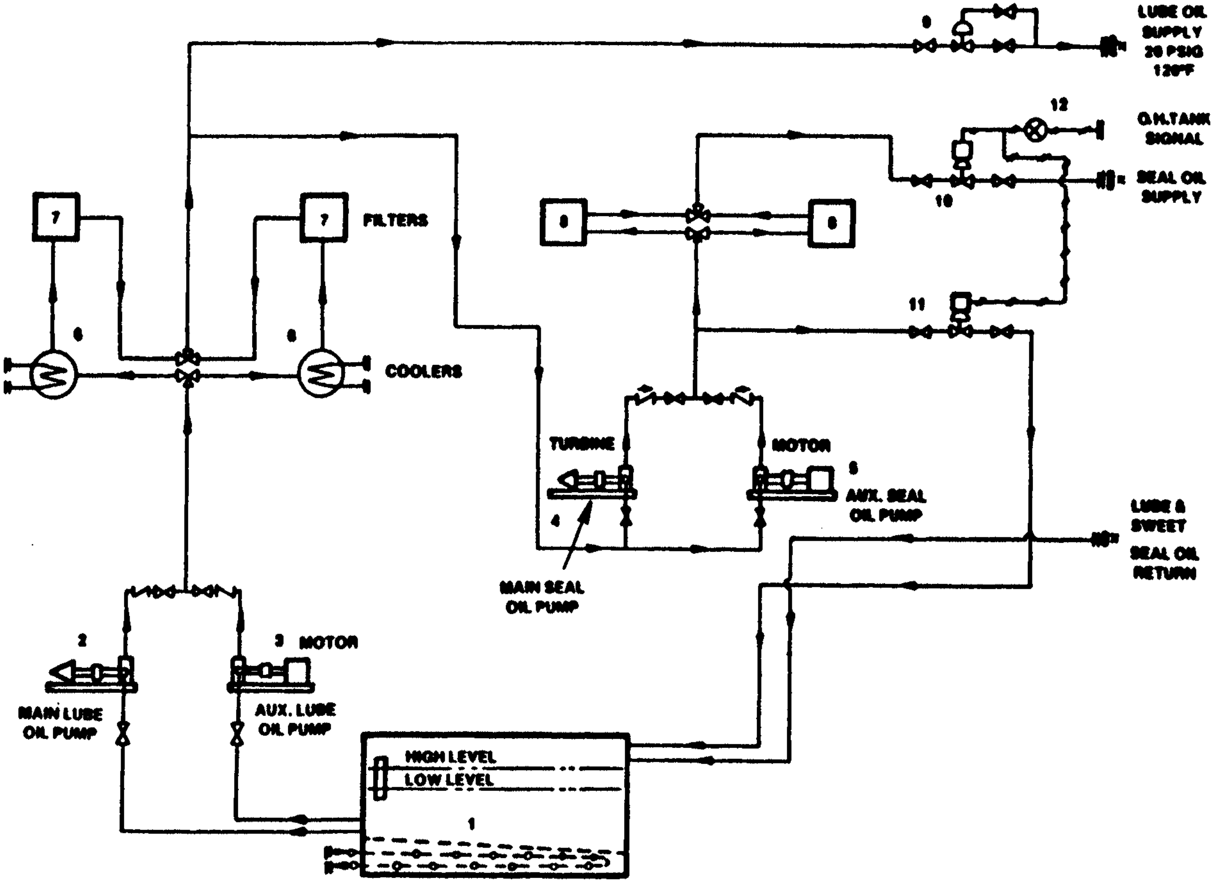


Figure 7.98 High pressure, combined lube and seal system sealing pressurs to 3000 psig (211 Kg/CM2) (Courtesy of Dresser Rand)

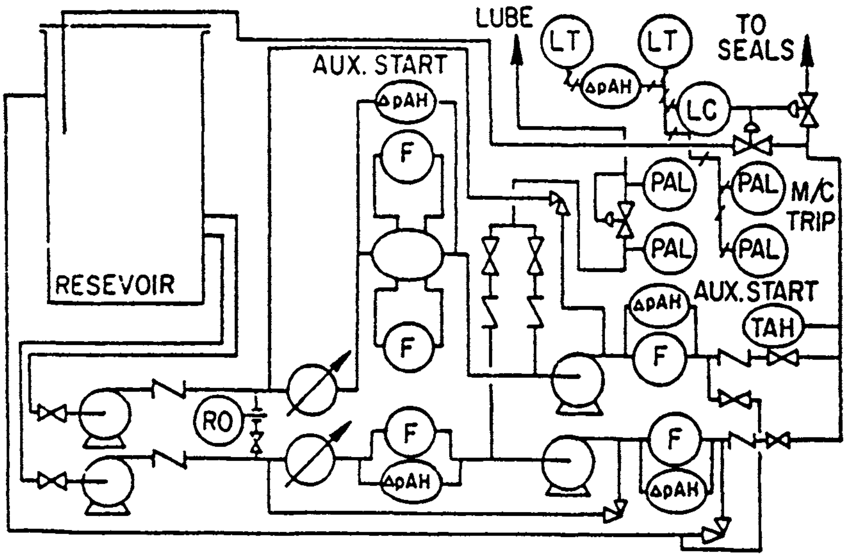


Figure 7.99 High pressure seal system schematic

equipment shutdown. Consideration should be given to separate pump loops as shown in Figure 7.99 for high pressure systems to preclude the possibility of dual pump failure.

Differential pressure control valves

Function

A differential pressure control valve is required in a sealing system since the referenced pressure or the gas side seal downstream pressure, does not remain constant. In order to supply a constant flow across the seal (which is an equivalent orifice) the pressure drop must remain constant. This is facilitated by using a differential pressure control valve, either direct operated as shown in Figure 7.100 or controller operated as shown in Figure 7.101.

The set point differential pressure is determined by the seal design. The required oil to gas differential pressure is that amount necessary to introduce sufficient flow to remove frictional heat from the seal. Refer to Figure 7.102 which shows a differential bypass control valve installed in the seal oil system. Let's examine valve operation during start-up and normal operating conditions.

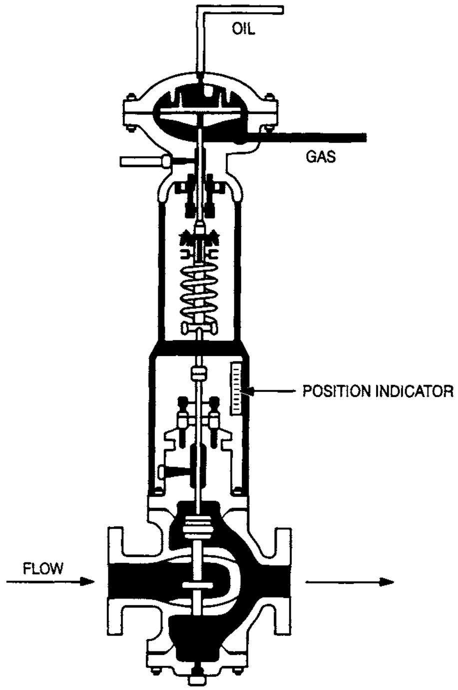


Figure 7.100 Direct acting differential pressure control valve

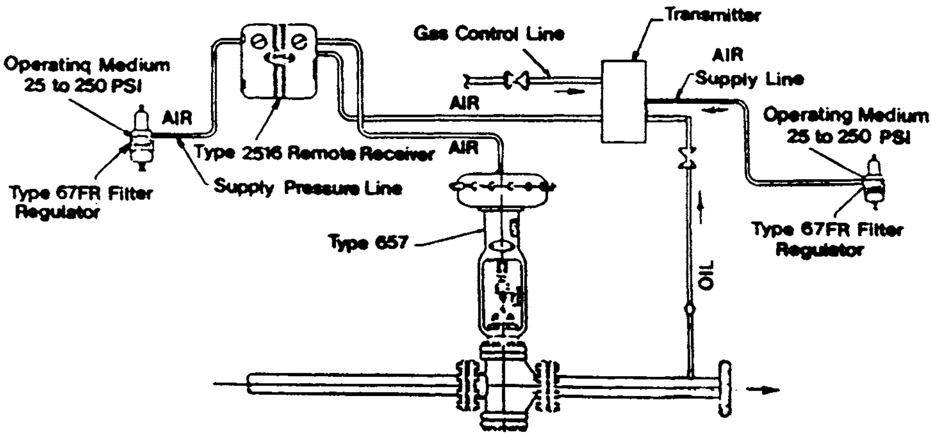


Figure 7.101 Controller operated pressure control valve

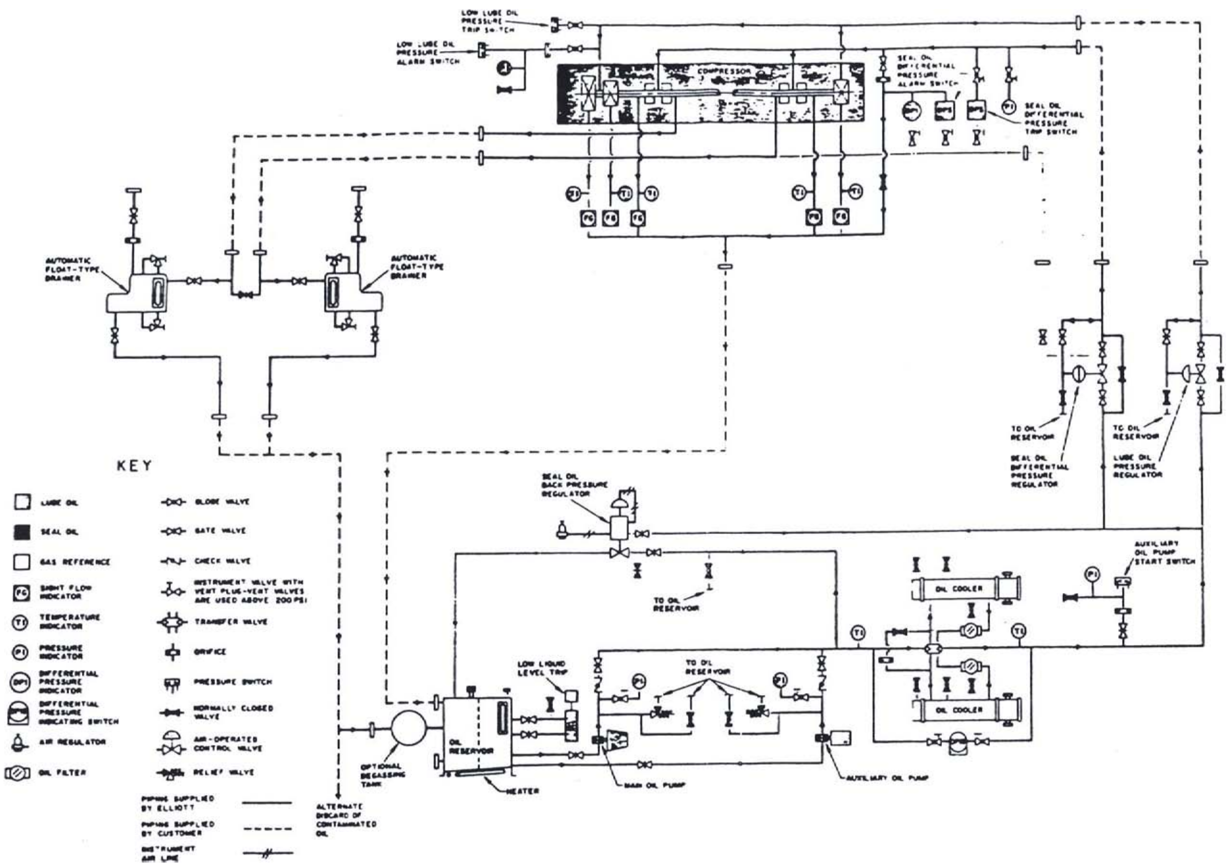


Figure 7.102 API G14 lube-seal oil system for ISO-carbon seals (Courtesy of Elliott Co.)

Start-up

In this example, the compressor case reference pressure during start-up is 100 psi. Therefore, the control valve will regulate pressure downstream of the filter and cooler to be equal to 150 psi. (Fifteen additional psi were added to the specified seal differential of 35 psi to account for the pressure drop between the exit of the seal system and the entrance to the seal chamber at the unit.) The concept to consider in this instance is that of an equivalent vessel (the discharge pipe of the seal oil supply system). The control valve must regulate the supply flow to the equivalent vessel such that the supply flow is equal to the flow demanded by the seal system to maintain a constant supply pressure. In the case of increasing demand flow and constant reference pressure, the pressure in the equivalent vessel would fall. The supply flow would be increased by a decreasing seal oil supply signal thus closing the bypass valve and providing additional supply flow. It can be seen that the functioning of the bypass control valve with a constant seal gas reference pressure, is identical to that of a lube system bypass valve.

Normal operation

For this example, normal operation requires a suction pressure of 500 lbs per square inch. As the suction pressure is increased, the differential pressure across the atmospheric seal, increases while the differential pressure across the gas side seal remains constant. In creasing differential pressure across the atmospheric seal will increase seal oil demand, thus resulting in reduced referenced oil pressure to the seal oil differential valve. This reduced pressure will act to close the valve and provide additional required seal oil to the seal system. If sufficient seal oil flow is not available when the bypass valve is completely closed, the referenced oil pressure will fall until the auxiliary pump starts. It is important to note that differential pressure switches must be used since the reference or seal downstream pressure can change during operation. If the seal oil differential continues to fall, the critical equipment will be tripped on low pressure. Typical set points for differential pressure control systems are to start the auxiliary pump to start at 35 psid and to trip the unit at 25 psid.

It is important to note that action to start the auxiliary seal oil pump and to trip the unit is based on differential seal oil to seal gas pressure. A sudden increase of seal gas reference pressure or a sudden decrease of seal oil pressure, will result in auxiliary pump start up and/or critical equipment trips. Care must be taken to assure the system is designed to prevent sudden oil pressure or reference gas pressure spikes that will lead to spurious trips. Seal systems tend to be softer than lube oil systems since the referenced pressure is a compressible fluid. That is, the requirement to start the auxiliary seal oil pump without tripping the unit tends to be easier and does not require accumulators.

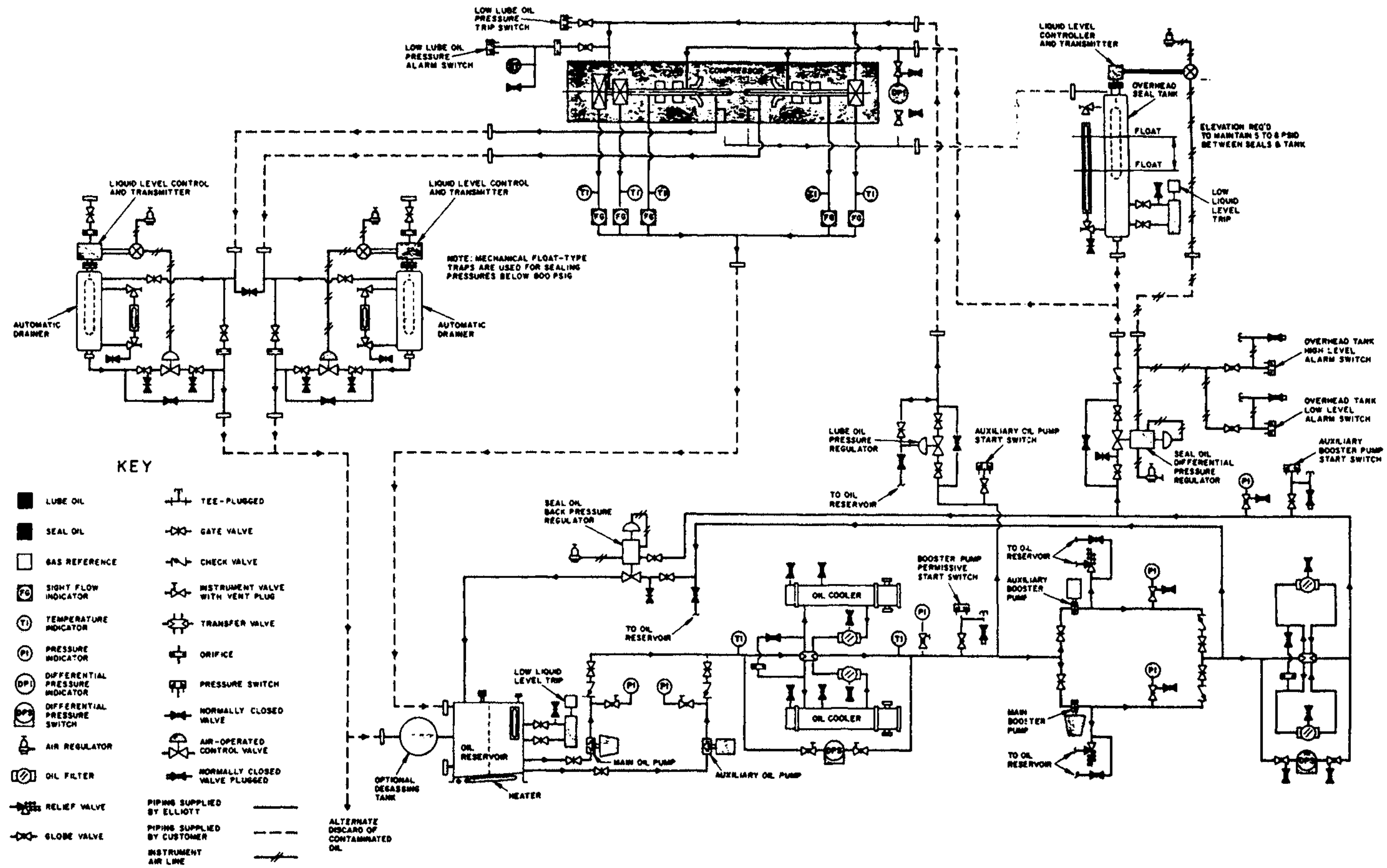


Figure 7.103 API G14 lube-seal oil system for ISO-sleeve seals (Courtesy of Elliott Co.)

Level control valves

Function

Level control valves utilized in seal systems are designed to regulate seal oil flow to or from the seal in the same manner as s differential pressure control valve. Level control valves are used where extreme accuracy in seal oil to seal gas differential pressure is required. They are most commonly employed in bushing seal applications since bushing seal differential pressures are regulated to a minimum value in order to minimize internal seal oil leakage. Refer to Figure 7.103 which incorporates a level control valve used in the system incorporating bushing seals. Again, let's examine the start-up and normal operating case for this seal system configuration.

Start-up

Reference pressure for start-up is 100 psi. This reference pressure is established in the overhead seal oil tank and exerts a pressure of 100 psi on the liquid. Refer again to the concept of an equivalent vessel which is the level tank in this case. As the referenced pressure increases, the differential pressure across the atmospheric bushing of the seal will increase (the differential pressure against the gas side bushing will remain constant). This action will increase the atmospheric bushing seal flow which will reduce the level in the head tank. A reduction of head tank level sends a signal to the level control valve which will open the valve, thereby increasing flow to the supply side of the equivalent vessel unit the supply flow equals the seal demand flow and the equilibrium level is established.

As the suction pressure increases to 500 psi, the atmospheric bushing seal flow will increase proportionally, thereby causing a decease in head level. Again, a signal will be sent to the level control valve to open thus increasing supply flow to the equivalent vessel and re-establishing head level at the same point as before. In the event of referenced pressure decrease, the opposite situation will occur and a signal will be sent to the control valve to close, thereby decreasing supply flow. Remember, that liquid head is proportional to pressure. In this example, unequal demand and supply flows to the equivalent vessel cause changes in vessel liquid level which are pressure changes as previously discussed. Therefore, in this example the function of the level control valve is to act as a variable orifice to change system supply flow for different demand flow requirements to maintain a constant pressure (head) in the level tank.

Control valve location in the system

As seen in the preceding section, the location of the level control valves and differential pressure control valves can vary from design to design. Either valve can be used in any of the following locations:

- Bypass
- Supply
- Return

Bypass control

Depending on the type of seal system and the type of seal design used, the control valves will be positioned accordingly. If the seal design does not incorporate through flow, a differential bypass or bypass level control valve will be incorporated. The function of the valve in this position will be to bypass pump discharge flow and thus reduce seal supply flow. Note that this type of valve arrangement will be used with positive displacement type pumps only. If centrifugal pump were utilized, the valve would be incorporated in the pump discharge and would regulate differential pressure downstream of the valve.

Supply control

If the seal arrangement incorporates flow through with a back pressure orifice, differential pressure supply control or supply level control is incorporated. If the seal is a busing type, level control would be utilized. If the seal is a contact type, differential pressure control would be used. The function of this system is to control supply flow to the seal system in accordance with seal system demand.

As an example, if seal reference pressure increases, atmospheric seal flow will increase and seal system demand will increase. This action will open the supply control valve, thus allowing supply flow to equal demand flow and achieving equilibrium supply system differential pressure.

Return control

If the seal configuration incorporates a flow through system without a fixed orifice, the control valve will be located in the discharge to compensate for oil to gas differential pressure changes.

As an example if a flow through bushing seal combination were used and the seal reference gas reference pressure increased, the atmospheric seal flow would increase, thereby increasing seal demand. In this configuration, seal supply flow is fixed, therefore, increased seal demand requires decreased flow through demand. This action would immediately result in a lower seal oil supply pressure and would close the control valve resulting in a reduced flow through quantity to compensate for the increased atmospheric seal requirement. Refer to Figure 7.104 for a schematic of this system.

Regardless of the location of the control valves in a seal system, the instrumentation is identical, that is, starting auxiliary pumps on decreasing level or differential pressure and tripping the critical equipment on decreasing level or pressure.

This concludes the discussion of the supply side component sizing of a seal system. In the next chapter, we will direct our attention to the return side of the seal system.

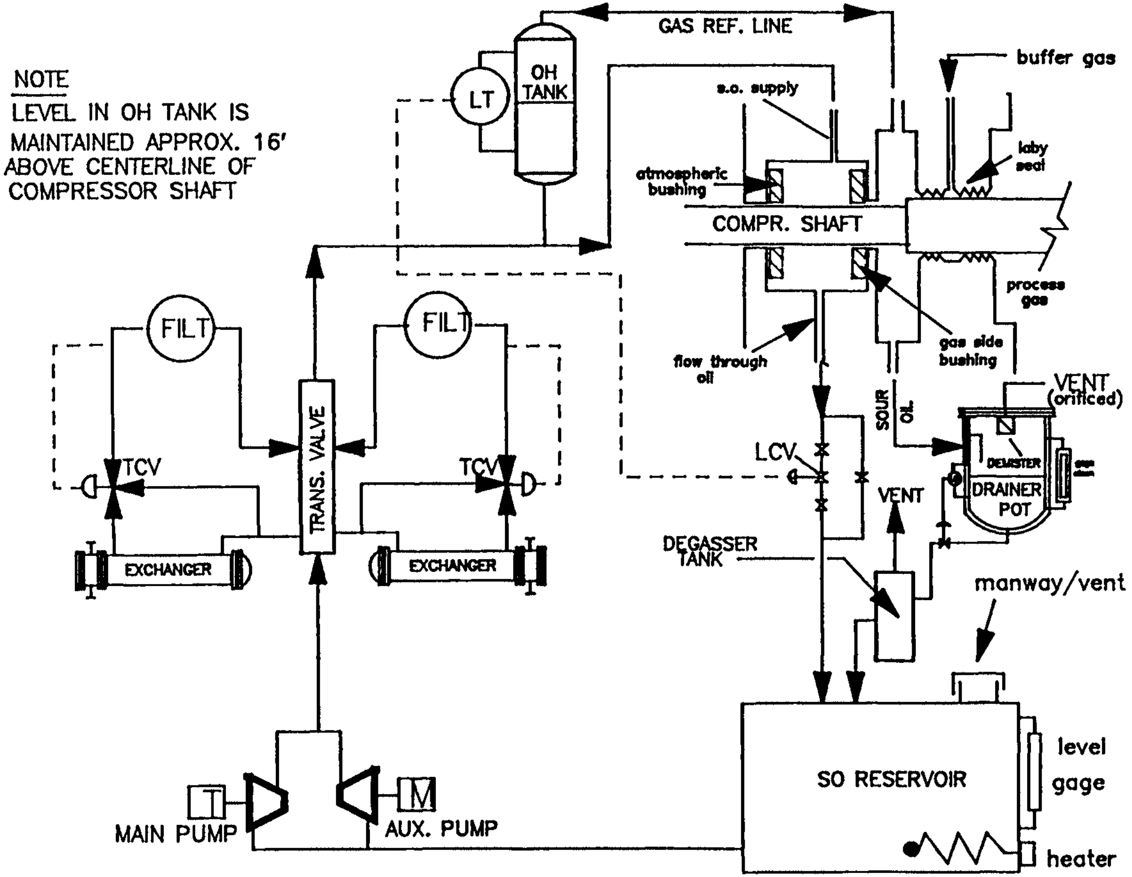


Figure 7.104 Typical seal oil system (Courtesy of M.E. Crane Consultant)

Seal system – the contaminated seal oil drain system

(Compressor liquid seal component knowledge)

- Introduction
- Basic system configuration
- Oil drain system component design
- System reliability considerations

Introduction

The return side of a seal oil system consists of three distinct return lines.

- The atmospheric drain back to the seal oil reservoir.
- The pressurized drain back to the seal oil control valves.
- The contaminated seal oil drain.

All liquid seals are designed such that a small amount of oil leakage (10–20 gallons per day per seal) enter the drain between the inboard gas dies seal and the compressor internals. This drain is commonly known as the contaminated oil drain. To assure that gas does not leak into the seal, a small amount of liquid is designed to continuously flow through the seal. Depending on the nature of the process gas, the oil may or may not be recirculated to the seal oil tank. The function of the contaminated oil drain system is therefore to collect all contaminated oil and direct this oil to the desired location. This may be directly to the seal oil reservoir, to a degassing tank or other oil reclamation device, or to a contaminated oil drain. We will now examine basic contaminated oil drain system configurations and discuss the function of each component.

Basic system configuration

Refer to Figure 7.105 which shows a basic contaminated oil drain system. The system consists of:

- The contaminated oil drain line to a seal oil drainer for each seal in the unit.
- The drainer.

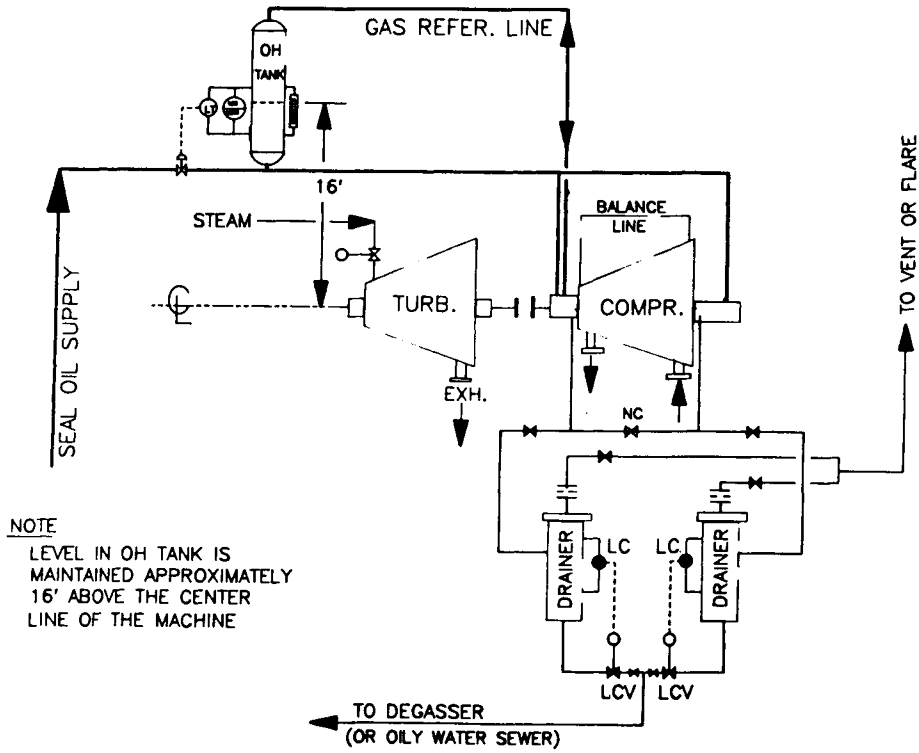


Figure 7.105 Schematic – seal oil arrangement (Courtesy of M.E. Crane Consultant)

- The vent connection or reference connection from the top of the drainer.
- The drainer level control device (internal or external) which automatically drains upon reaching a preset level.
- The degassing tank (optional).
- The oil reclamation device (optional).

The following system design alternatives are available depending upon the operating conditions of the compressor and the process gas.

Sweet hydrocarbon or inert gas service

For sweet or inert gas service, the seal oil drain can be returned directly to the reservoir provided the drainers are sized for adequate residence time and the seal oil leakage is reasonable (less than one gallon per hour per seal). A sweet hydrocarbon gas is defined as a gas that does not contain hydrogen sulfide (H_2S). The vent line on top of the drainer can be routed to a lower pressure source, atmosphere or back to the compressor suction. If routed back to the compressor suction, a demister should be installed to prevent oil from entering the compressor case. The sizing of the orifice in the vent line of each drainer is critical in that it assures that all contaminated oil flow will enter the drainer. Too low a velocity will allow contaminated oil to enter the compressor. Too high a velocity could cause oil to enter the compressor via the vent or reference line.

High suction pressure

In high suction pressure applications, the system configuration changes somewhat as shown in Figure 7.106.

In this application, an external level control valve is recommended on each contaminated seal oil drainer. In addition, venting drainers to atmosphere or to flare will result in exceptionally high pressure drop across the orifice. In this application, the vent line is usually routed back to the compressor suction. Again, an adequately sized demister must be installed to prevent oil ingestion in the compressor.

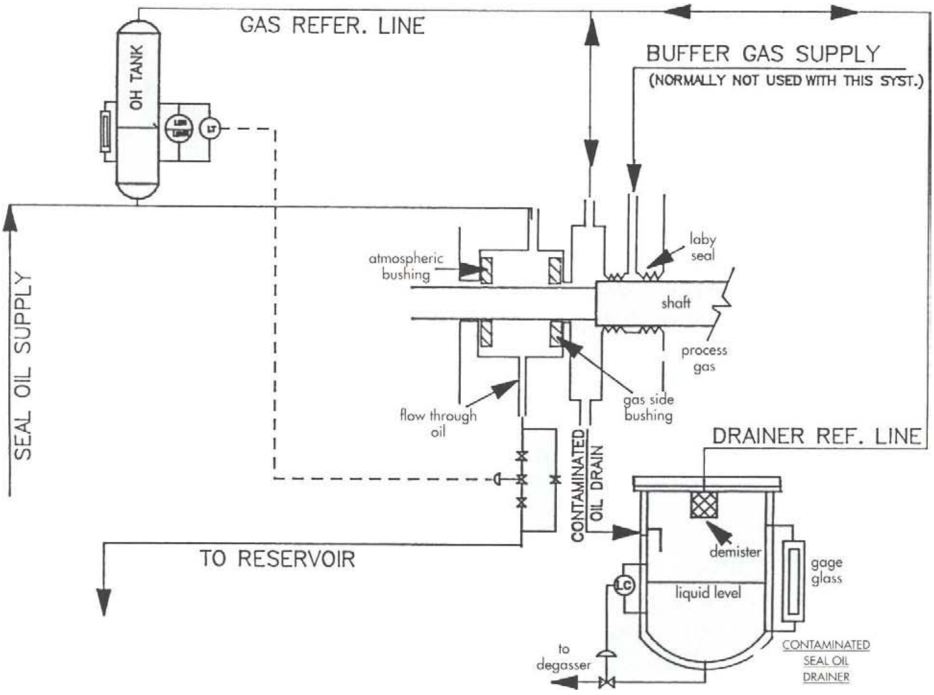


Figure 7.106 Contaminated oil drain system (with referenced drainer) (Courtesy of M.E. Crane Consultant)

Vacuum service

Compressors that act with suction pressure below atmospheric pressure or use a suction throttle valve, can have a reference pressure less than atmospheric. In this case, the contaminated seal oil drainer vent line must be referenced back to the compressor suction, in order to assure proper operation and not allow air to enter the demister through the vent line in the reverse direction. For this application, a buffer gas system must be installed that will be designed to maintain the drainer pressure above atmospheric pressure at all times in order to allow the drainer to be drained.

Sour process gas

Where the process gas can contaminate the seal oil leakage (as in the case of H_2S gas, etc.), the contaminated seal oil drain line is usually routed to the plant contaminated oil system and not back to a degassing tank or the reservoir.

Oil drain system component design

The function, sizing criteria and operating specifics of each major component of the contaminated oil drain system will now be discussed.

The contaminated seal oil drainer

The function of the contaminated seal oil drainer as shown in Figure 7.107 is to contain all of the oil leakage from a specific seal. Normally, drainers are automatic, that is, they drain oil when a specific pre-set level in the drainer is reached. The typical sour oil leakage per day varies from less than five gallons to an excess of 20 gallons per day. Normal vessel capacity of liquid is approximately 1/4– 1/2 gallon. Consequently, drainers must either be manually drained or automatically drained between 40–80 times a per day. The inlet connection to the contaminated seal oil drainer is from the gas side seal oil drain and contains an oil-gas mixture. The gas is either the process gas or a clean buffer gas as will be discussed in following sections. The top of the drainer contains a vent connection which may be routed to a reference connection in the compressor case or directly to vent. Regardless of the configurations, the drainer contains a mixture of free gas above the liquid and entrained gas in the liquid. Drainers are essentially of two designs.

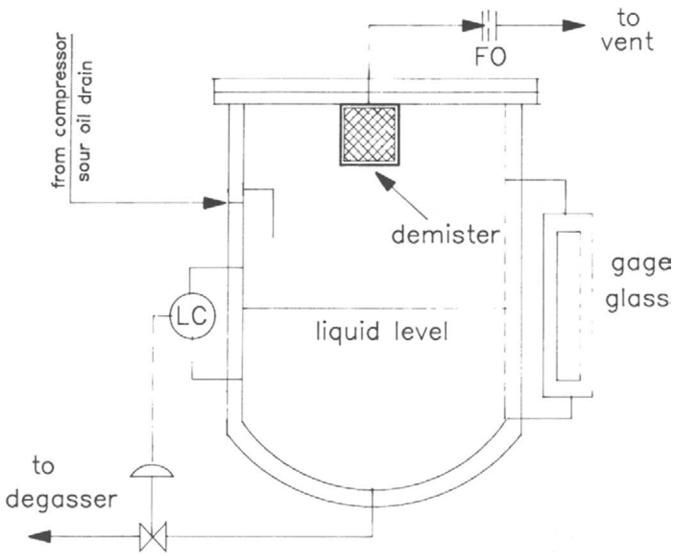


Figure 7.107 Contaminated seal oil drainer (Courtesy of M.E. Crane Consultant)

Internal valve

This design incorporates a ball float valve shown in Figure 7.108 which is internal to the drainer and opens at a prescribed level. In order to control the rate of drainage from the drain pot with the valve open, an orifice is sized. Note that this orifice is sized to allow controlled drainage time under normal operating conditions. Conceptually, this orifice represents another atmospheric bushing in the seal oil system. It is similar to the atmospheric seal in that the differential across the orifice will vary with reference pressure in the compressor. Many times care is not given to adequately sizing this orifice. As a result, during start-up conditions, with low compressor case suction pressure, there is insufficient pressure drop across this orifice to adequately drain the vessel. The vessel will not drain until there is sufficient pressure drop. That is, there must be a column of liquid high enough (head) to drain through this orifice. Considering the installation location of seal oil drainers, many times the height available from the center line of the compressor down to the drainer is insufficient at low or zero pressure conditions to force a drain. Therefore, under these conditions, all contaminated seal oil will drain into the compressor case. As can be seen from Figure 7.109, drainer installations should be provided with bypass valves around the drainer. It may be necessary to open bypass valves during low pressure operation to allow drainage back to the reservoir. *Caution must be exercised during this operation if compressor gas is toxic or flammable since a full stream of gas will be exiting continuously back to the appropriate vessel.*

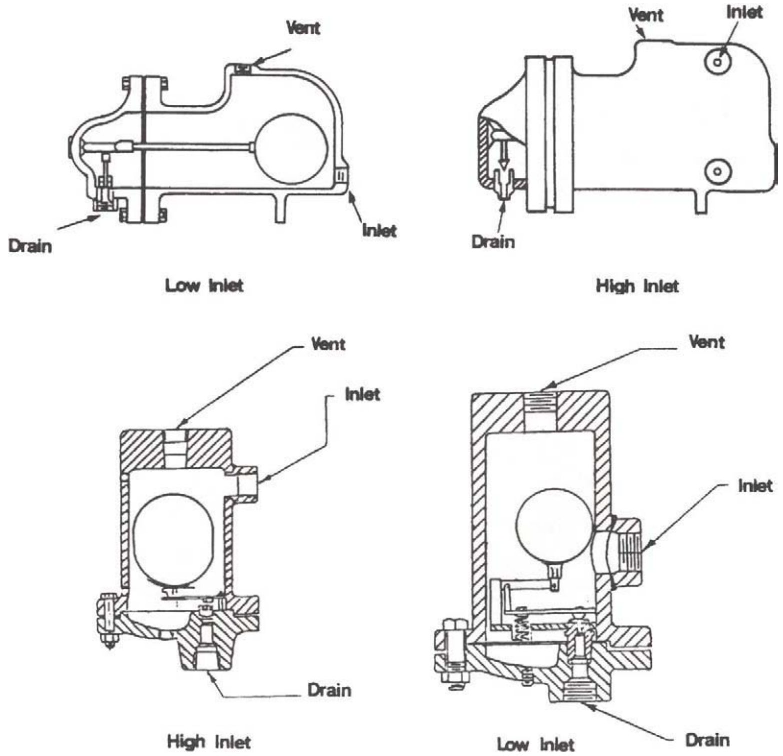


Figure 7.108 Ball float valve. Above: WKM drainers; below: Armstrong drainers (Courtesy of Elliott Co.)

Caution: Prior to opening any system connections or sampling fluids, obtain a site work permit to assure the area and conditions are safe for work or entry.

A properly sized orifice for low pressure conditions must be installed in this line to minimize loss of gas.

Contaminated seal oil drainer with external valves

This application is usually utilized in higher pressure cases such a re-injection compressors where suction pressure and seal oil drainer pressure run between 1,000 and 1,500 lbs per square inch. The external valve is controlled by a level control transmitter which sends a signal to open the valve when the level reaches a specified amount and quickly closes the valve when the level falls below a specified set point. Again, care must be given in sizing the valve to assure that sufficient valve area is available under low suction pressure conditions to allow drainage back to the appropriate vessel. In addition to the valve,

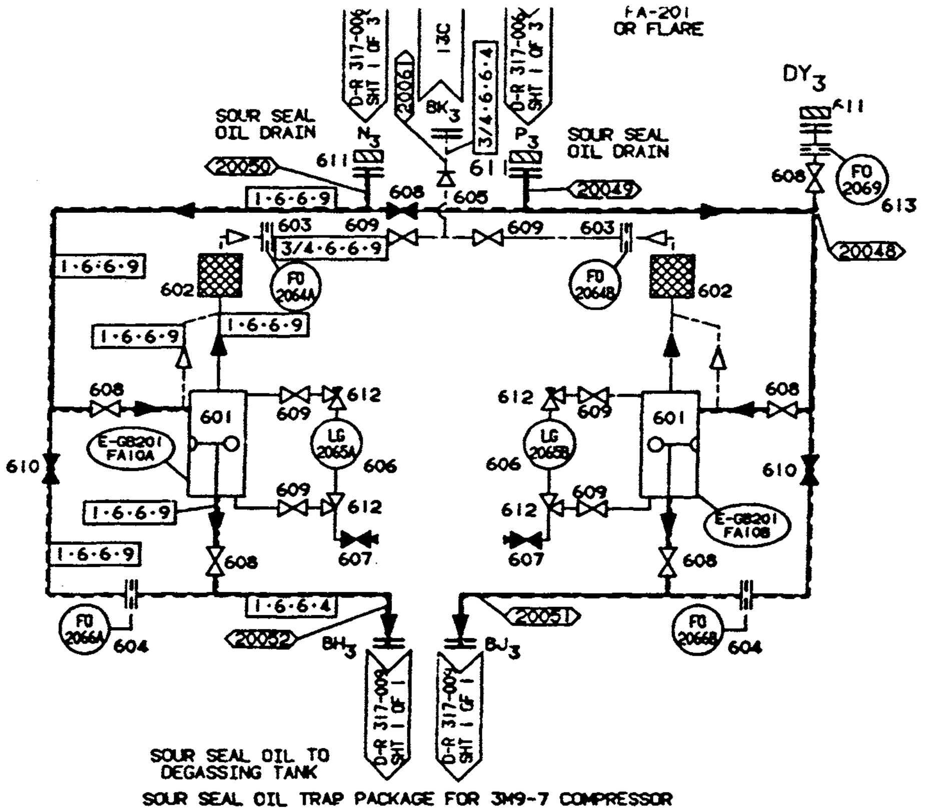


Figure 7.109 Sour seal oil trap arrangement (Courtesy of Dresser-Rand)

downstream piping must be adequately sized and designed to allow contaminated seal oil to exit the drainer.

Drainer reliability considerations – As can be seen, the reliable operation of the drainer valve, internal or external, is essential to safety and reliability. The valve must open and close tightly upon signal to assure a minimum amount of processed gas exits the drainer. Many applications process a gas that tends to be sticky or has a high amount of carbon that will cause internal valves to bind, thus keeping the valve open at all times.

Attention must be paid to the gage glass on the drainer. Gage glasses must be kept clean so that level can be observed. If level is not present, the drainer exit should be checked to assure that gas is not exiting the drainer. If this is the case, the drainer should be isolated and inspected. *Caution – any action involving opening valves and drainers requires an area safety permit.*

Systems should be designed such that drainers can be isolated during operation and one drainer can temporarily service two seals so that maintenance can be performed on a drainer while the unit is in operation. If a continued problem is experienced with clogging of drainer orifices or hanging up of internal valves, consideration should be given to injecting a clean buffer gas which will assure satisfactory operation of the drainer. Note that external valves are also subject to this malfunction in that debris can enter the valve causing it to remain open.

Another reliability consideration is the drainer bypass line. Many times this line is inadvertently left open, either after start-up or opened by operators during operation. This valve should be closed during operation. Otherwise a continuous stream of gas exits and proceeds downstream.

If the drainer gage glass is continually full, this could indicate that the drain valve is not properly opening and contaminated oil could be forced into the compressor under this condition. Many processes prohibit the entrance of oil into the compressor and this action could cause shutdown of the unit. Again, the drainer should be isolated and inspected. Failure to properly monitor and maintain drainers leads to many unscheduled shutdowns of critical equipment.

One instance of false level gauge indication is shown in Figure 7.110. In this case, the top reference line of the level gage is referenced to the drainer vent line. In this application, the vent line was referenced to the suction vessel and the velocity through the orifice was great enough to create a vacuum on the gas in between the top reference line take off and the fluid in the level gauge. The result was that the liquid level was actually forced to the top of the glass (by greater pressure in the drainer), creating a false indication of a full oil drainer. The solution was to install a larger (four inch diameter) section at the drainer vent connection to minimize the velocity in this area. The high velocity steam acted as an eductor creating a lower pressure on the top of the liquid in the level gauge.

As an exercise, let's calculate the reduction in gas pressure on top of the oil in this application that would allow the oil level to rise six inches (the height of the level glass). Solving for pressure,

$$P = \frac{HD \times S.G.}{2.311}$$

$$= \frac{.5(ft) \times .85}{2.311}$$

$$= 0.184 \text{ psi}$$

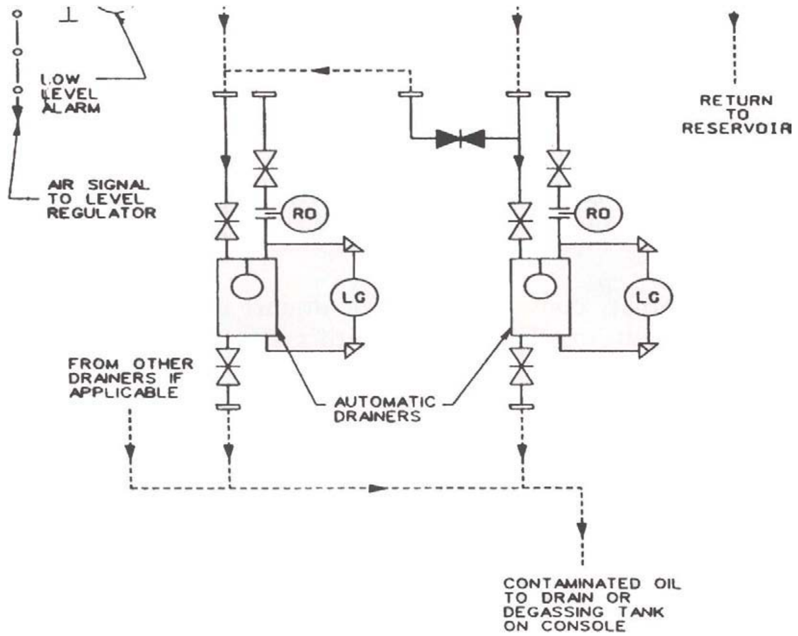


Figure 7.110 False level gauge indication (Courtesy of Elliott Co.)

Therefore, if the high velocity gas stream can reduce the pressure in the trapped volume between the gas stream and the top of the liquid level by .184 psi, the level will rise to the top of the gage glass and give the illusion of a full drainer. To see if this problem exists, on line, briefly close the inlet valve to the drainer. If the level suddenly decreases, this would indicate this type of problem. *Caution: immediately open drainer inlet line to avoid seal oil entering the compressor.*

Demisters

When the vent line from the top of the drainers is referenced back to any section of a compressor, a demister must be installed to minimize the amount of oil entering the compressor. The functions of demisters, therefore, are to eliminate oil migration into the compressor via the vent line. The demisters must be properly sized to assure their efficiency. Care must be given to calculating the velocity through the demister which is dependent on the vent orifice. Maximum differential pressure conditions across the orifice must be considered. Frequently, mesh demisters are designed to be integral with the contaminated seal oil drainer as shown in Figure 7.107. In this case, any oil mist exiting the top of the drainer will be condensed and will fall back into the drainer. It must be understood that demisters are not 100% efficient. If the process cannot tolerate any seal oil, the vent line should be routed directly to flare or to atmosphere, depending upon the gas composition. *Caution – any toxic or flammable gas must be routed to a safe location.*

Degassing tanks

As previously mentioned, the oil that enters the contaminated seal oil drainer is accompanied by free gas which exits the drainer through the vent, and entrained gas in the oil. The oil must be properly degassed prior to entrance back into the seal oil reservoir. The function of a degassing tank, therefore, is to degas the contaminated seal oil so that all oil exiting the degasser is within the original oil specification. A typical degassing tank is shown in Figure 7.111. The tank contains baffles, a heating device, and an overflow drain with a properly sized vent in order to degas all seal oil entering this vessel. Experience has shown that a degasser sized for 72 hours residence time, based on the total estimated sour oil leakage, usually is sufficient to achieve the design objective.

As an example, if a compressor containing two seals each has a maximum specified leakage of 20 gallons per seal per day, the degassing tank should be sized as follows:

capacity = total leakage per day × 3 days = 120 gallons.

A cursory inspection of any refinery or chemical plant will show that most degassing tanks are undersized. As a result, seal oil sampling

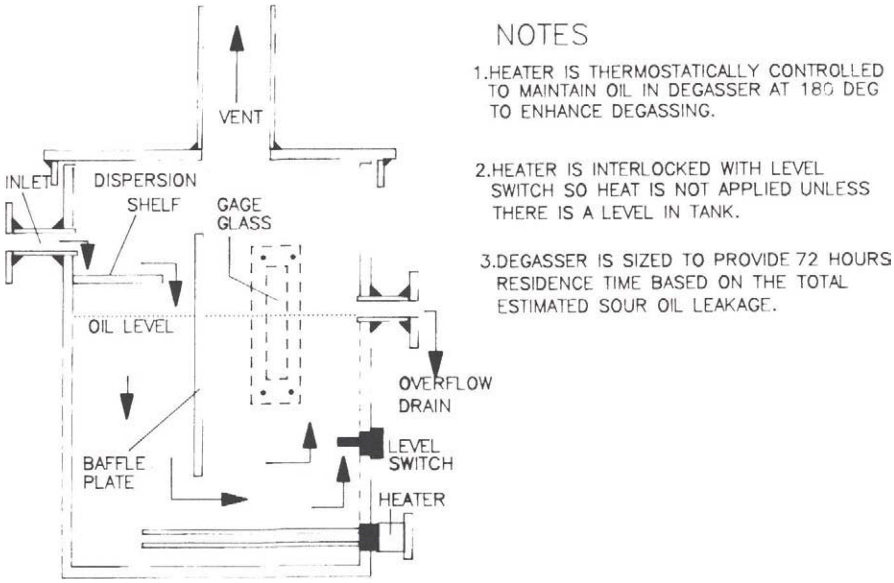


Figure 7.111 Typical degasser arrangement (Courtesy of M.E. Crane Consultant)

usually shows a deterioration of oil viscosity and flash point. As mentioned in the previous chapter, flash point is the temperature at which the oil will sustain combustion. Light gasses, (hydrogen, and hydrogen mixtures), significantly reduce oil flash points. Experience has shown that this value can approach the operating temperature of the system! It is strongly recommended that the following action should be taken when seal oil reservoir samples indicate a low flash point.

- Temporarily isolate seal oil drainer return, collect all seal oil and vacuum degas the seal oil. Provide make up fresh seal as required.
- Adequately size a degassing tank and install at earliest opportunity.
- Consider the installation of an oil reclamation device.

Figure 7.112 is a table of mineral oils used for seal oil service.

Characteristic	Light	Medium
Gravity, api	31.7	30.6
Pour, °F (°C)	20 (-7)	20 (-7)
Flash, °F (°C)	395 (201)	400 (204)
Viscosity		
Sus at 100°F	150/165	215/240
Sus at 210°F	44	48.8
Csi at 40°C	28.8/32.0	41.4/46.0
Csi at 100°C	5.2	6.5
VI, min	95	95
Iso viscosity grade	32	46
Color, astm. max	1.5	2.0
Neutralization number, max.	0.20	0.25
Rust test (A&B) (astm D665-IP135)	Pass	Pass
Demulsibility (astm 1401) 3 ml max.		
at 130°F (54°C) 1/2 hr. at 180°F (82°C) 1 hr	Pass	Pass

Figure 7.112 Typical oil flash points for seal oils

Please note the values of the oil flash points and remember that many operating seal oil systems contain oil flash points that are on the order of 120°F. This is particularly dangerous in the case of combined lube and seal oil systems where the oil in the reservoir will actually enter the bearing system.

Oil reclamation units

In cases where the degassing tanks have proven not to be effective, or are inadequately sized, the use of an oil reclamation unit should be considered. All oil from drainers should be collected or directly piped to an oil reclamation unit. Considering the cost of typical mineral oil (approximately $25 per gallon), a standard compressor with two seals can use a $1,000 of oil per day if oil cannot be returned to the reservoir.

Figure 7.113 shows a typical oil reclamation unit which has capability to degas all oil entering the unit. In large installations, this unit may be justified for direct installation downstream of the drainers. For smaller systems, the purchase of one unit should be considered for the site. Gas entrained liquids can then be collected and transported to the unit for reclamation at specified intervals.

System reliability considerations

The contaminated oil drain system is one of the most neglected sub-systems in a compressor seal oil system. It is out of the way, its function is not fully understood, and it is not usually monitored with accuracy. Following are a few suggestions concerning drainer system reliability:

1. Care should always be given to the inspection of site glasses.
2. Oil samples to determine adequate degassing should be taken periodically.
3. Leakage rates from the drainer should be regularly measured to determine the condition of the seals.

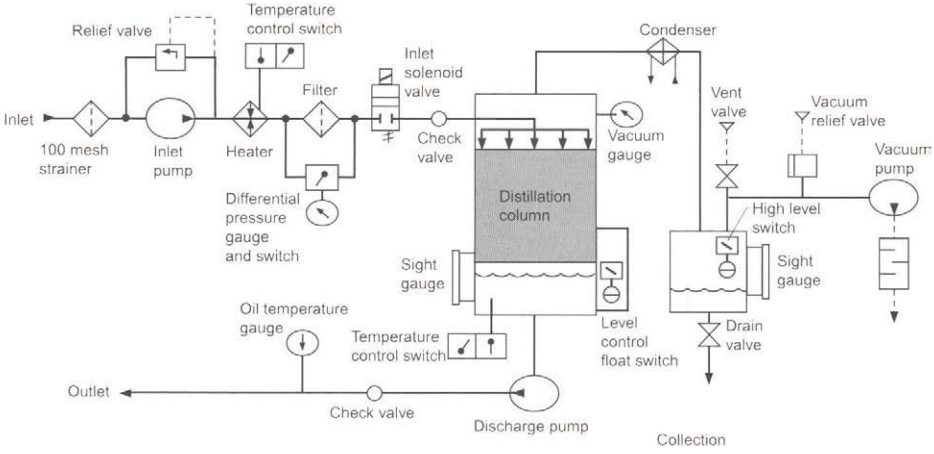


Figure 7.113 Oil reclamation unit schematic (Courtesy of Petroneics, Inc.)

4. The atmospheric side seal drain should incorporate site glasses as a means of monitoring flow visually to determine if flow quantities have significantly changed.

5. It must be remembered that control valves can be used as rough flow meters when pressure across the valve and valve travel are known. A control valve can give an indication of change of flow rates in the system and will indicate any change in seal oil flows. It must be remembered that the atmospheric side seal oil flows will vary with changing with reference pressure. However, in most installations once units are on-line, reference pressure is relatively constant. Therefore, significant changes in seal oil control valves position will indicate deterioration of seals.

This concludes our discussion of the contaminated oil drainage system. We will further discuss this system when the subject of buffer gas systems is studied in subsequent chapters.

Dry gas seal systems
(Compressor dry gas seal component knowledge)

- Introduction
- Dry gas seal design
- Gas seal system types
- Summary

Introduction

Thus far, we have been studying in detail all of the components required for an auxiliary system. Many times, the reliability of critical equipment is dependent on the reliability of each component in every auxiliary system connected with the critical equipment unit. How do we maximize critical equipment reliability? The easiest way is to eliminate the auxiliary systems. Imagine the opportunity to eliminate all of the components; pumps, filters, reservoirs, etc. and thereby increase reliability and hopefully, the safety of the equipment. The gas seal as used in compressor applications affords the opportunity to achieve these objectives. However, the gas seal is still part of a system and the entire gas seal system must be properly specified, designed, maintained and operated to achieve the objectives of optimum safety and reliability of the critical equipment.

In this section, the principles of gas seal design will be discussed and applied to various gas seal system types.

System function

The function of a gas seal system is naturally the same as a liquid seal system. In fact, we have defined such systems as fluid seal systems. To repeat, the function of a fluid seal system, remembering that a fluid can be a liquid or a gas, is to continuously supply clean fluid to each specified seal interface point at the required differential pressure, temperature, and flow rate. Therefore, one would expect the design of a gas seal and a liquid seal to be very similar, which, in fact, they are. Then why are their systems so different?

Comparison of a liquid and gas sealing system

Figure 7.114 shows a liquid sealing system as previously discussed in this section.

Compare this system to Figure 7.115 which shows a gas seal system, if

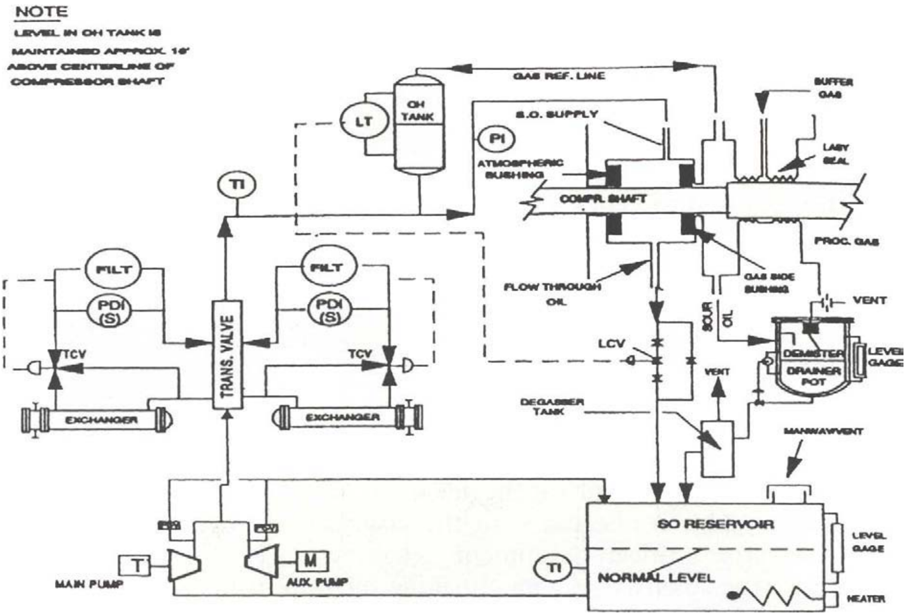


Figure 7.114 Typical seal oil system (for clearance bushing seal)

the same compressor were retrofitted for a gas seal. WOW!! What a difference. Why are there such a small amount of components for the gas seal system? As an aid, refer to Figure 7.116 which shows a typical pump liquid flush system as specified by the American Petroleum Institute. This system incorporates a mechanical seal and utilizes pump discharge liquid as a flush for the seal. Refer now to Figure 7.115 and observe the similarities. It should be evident that a gas seal system is simplified in compressor applications over a liquid seal system merely because the gas seal utilizes the process fluid. This is exactly the same case for a pump. By using the process fluid, and not a liquid, one can eliminate the need to separate liquid from a gas, thereby totally eliminating the need for a liquid supply system and the need for a contaminated liquid (sour oil), drain system.

Referring back to Figure 7.114, therefore, we can see that the following major components are eliminated:

1. The seal oil reservoir
2. The pumping units
3. The exchangers
4. The temperature control valves
5. The overhead tank

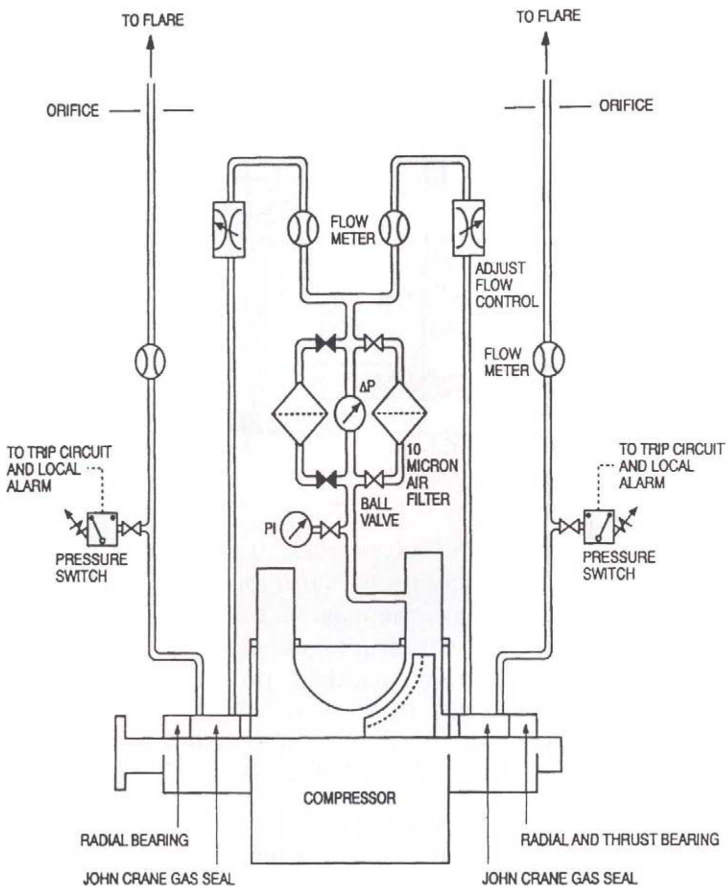


Figure 7.115 Typical gas seal system (Courtesy of John Crane Co.)

6. The drain pot
7. The degassing tank
8. All control valves
9. A significant amount of instrumentation

Referring back to the function definition of the gas seal system, all requirements are met. 'Continuously supplying fluid' is met by utilizing the discharge pressure of the compressor. The requirements for 'specified differential pressure, temperature and flow rate' are met by the design of the seal itself which can accommodate high differential pressures, high temperatures, and is sized to maintain a flow rate that will remove frictional heat necessary to maintain seal reliability. The only requirement not met is that of supplying a clean fluid, and this can be seen in Figure 7.115. This requirement is met by using a dual filter.

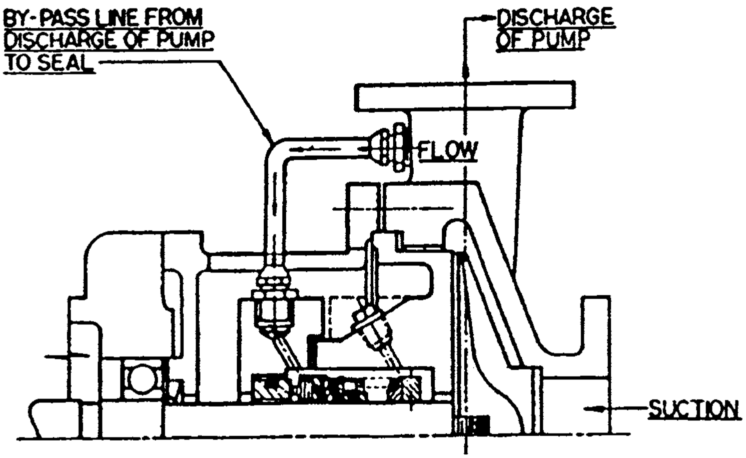


Figure 7.116 Liquid seal flush (Courtesy of John Crane Co.)

When one considers all the advantages, the next question to ask is, okay, what are the disadvantages? Naturally, there are disadvantages. However, proper design of the gas seal system can minimize and eliminate many of the disadvantages. Do not forget that the requirements for any system mandate proper specification, design, manufacture, operation and maintenance. One can never eliminate these requirements in any critical equipment system.

Considerations for system design

As mentioned above, there are disadvantages to a gas seal system which are not insurmountable but must be considered in the design of such a system. These considerations are as follows:

Sensitivity to dirt – since clearances between seal faces are usually less than 0.0005 inch and seal design is essential to proper operation, the fluid passing between the faces must be clean (5–10 microns maximum particle size).

Lift-off speed – as will be explained below, a minimum speed is required for operation. Care must be taken in variable speed operation to assure that operation is always above this speed.

Positive prevention of toxic gas leaks to atmosphere – since all seals leak, the system must be designed to preclude the possibility of toxic of flammable gas leaks out of the system. This will be discussed in detail below.

Possible oil ingestion from the lube system – a suitable separation seal must be provided to eliminate the possibility of oil ingestion from the bearings. Whenever a gas seal system is utilized, the design of the critical equipment by definition incorporates a separate lube oil and seal system. Consideration must be given during the design or retrofit phases to the separation between the liquid (lube) and gas seal system.

If all of the above considerations are incorporated in the design of a gas seal system, its reliability has the potential to exceed that of a liquid seal system and the operating costs can be reduced.

Before moving to the next section, however, one must consider that relative reliability between gas and liquid seal systems are a function of proper specification, design, etc. as mentioned previously. A properly designed liquid seal system that is operated and maintained as detailed in this book can achieve reliabilities of a gas seal system. Also, when one considers operating costs of the two systems, various factors must be considered. While the loss of costly seal oil is positively eliminated, with a gas seal system (assuming oil ingestion from the lube system does not occur) the loss of process gas, while minimal, can be expensive. It is argued that the loss of process gas from a liquid seal system through drainer vents and degassing tank vents, is also significant. While this may be true in many cases, a properly specified, designed and operated liquid seal system can minimize process gas leakage such that it is equal or even less than that of a gas seal.

There is no question that gas seal systems contain far fewer components and are easier to maintain than liquid seal systems. These systems will be used extensively in the years ahead. The intention of this discussion is to point out that existing liquid seal systems that cannot be justified for retrofit or cannot be retrofitted easily, can be modified to minimize outward gas leakage and optimize safety and reliability.

Dry gas seal design

Principles of operation

The intention of this sub-section is to present a brief detail of the principles of operation of a dry gas seal in a conceptual form. The reader is directed to any of the good literature available on this subject for a detailed review of gas seal design.

Refer to Figure 7.117. Figure 7.117 shows a mechanical seal utilized for pump applications, while Figure 7.118 shows a mechanical seal utilized for compressor application. The seal designs appear to be almost identical. Close attention to Figure 7.118, however, will show reliefs of the rotating face of the seal. Considering that both seals operate on a fluid may give some hint as to why the designs are very similar. The objective of seal designs is to positively minimize leakage while removing frictional heat to obtain reliable continuous operation

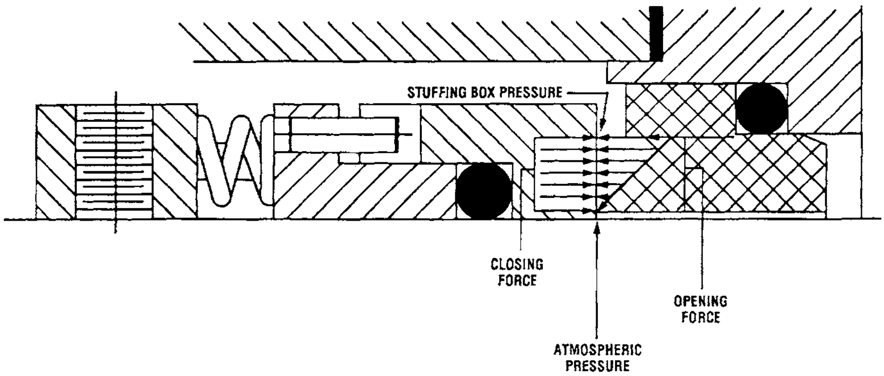


Figure 7.117 Typical pump single mechanical seal

of the seal. In a liquid application, the heat is removed by the fluid which passes between the rotating and stationary faces and changes from a liquid to gaseous state (heat of vaporization). This is precisely why all seals are said to leak and explains the recent movement in the industry to sealless pumps in toxic or flammable service. If the fluid between the rotating faces now becomes a gas, its capacity to absorb frictional heat is significantly less than that of a liquid. Therefore an 'equivalent orifice' must continuously exist between the faces to reduce friction and allow a sufficient amount of fluid to pass and thus take away the heat. The problem obviously is how to obtain this 'equivalent orifice'. There are many different designs of gas seals. However, regardless of the design, the dynamic action of the rotating face must create a dynamic force that will overcome the static forces acting on the seal to create an opening and hence 'equivalent orifice'.

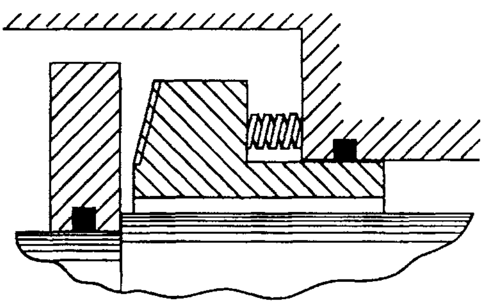
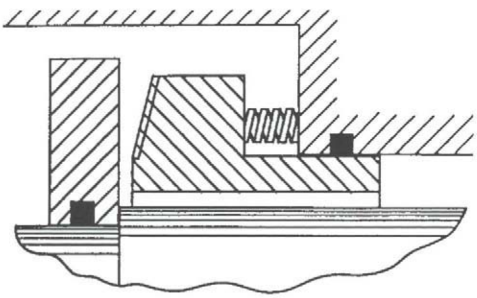


Figure 7.118 Typical design for curved face – spiral groove non-contact seal; curvature may alternately be on rotor



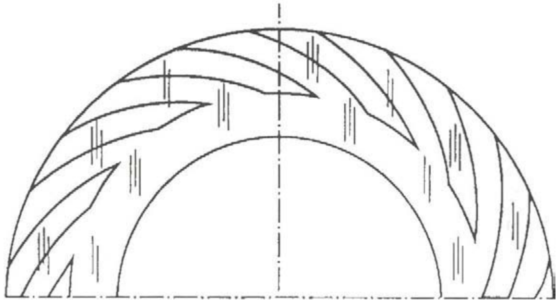


Figure 7.119 Dry gas seal. Top: typical design for curved face – spiral groove non-contact seal; curvature may alternately be on rotor; Bottom: Typical spiral groove pattern on face of seal typical non-contact gas seal (Courtesy of John Crane Co.)

Refer to Figure 7.119 which shows a typical dry seal. Notice the spiral grooves in this picture, they are typically machined at a depth of 100–400 micro inches. When rotating, these vanes create a high head low flow impeller that pumps gas into the area between the stationary and the rotating face, thereby increasing the pressure between the faces. When this pressure is greater than the static pressure holding the faces together, the faces will separate thus forming an equivalent orifice. In this specific seal design, the annulus below the vanes forms a tight face such that under static (stationary) conditions, zero leakage can be obtained if the seal is properly pressure balanced. Refer to Figure 7.120 for a force diagram that shows how this operation occurs.

Ranges of operation

Essentially, gas seals can be designed to operate at speeds and pressure differentials equal to or greater than those of liquid seals. Present state-of-the-art limits seal face differentials to approximately 1,000–1,500 psi and rubbing speeds to approximately 400 feet per second. Temperatures of operation can reach as high as 1,000°F. Where seal face differential exceeds these values, seals can be used in series (tandem) to meet specifications provided sufficient axial space is available in the seal housing.

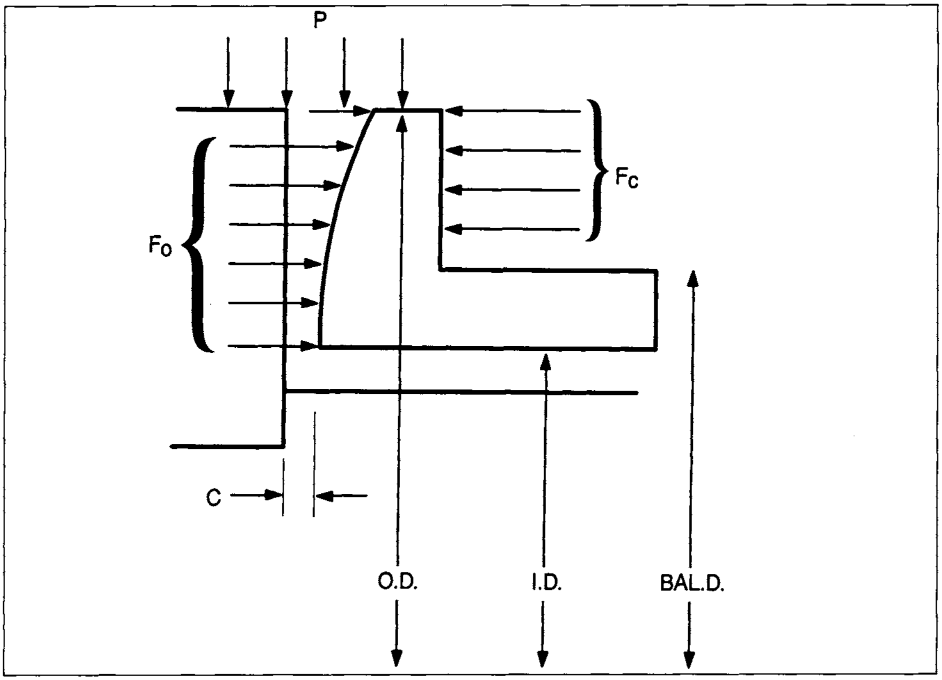


Figure 7.120 Hydrostatic force balance on seal stator ($F_C = F_O$) (Courtesy of John Crane Co.)

Leakage rates

Since the gas seal when operating forms an equivalent orifice, whose differential is equal to the supply pressure minus the seal reference pressure, there will always be a certain amount of leakage. Refer to Figure 7.121 for leakage graphs.

It can be stated in general that for most compressor applications with suction pressures on the order of 500 psi and below, leakage can be maintained on the order of one standard cubic foot per minute per seal. For a high pressure application (1,000–2,000 psi), differential leakage values can be as high as five standard cubic feet per minute per seal. As in any seal design, the total leakage is equal to the leakage across the seal faces and any leakage across secondary seals (O-rings, etc.). There have been reported incidence of explosive O-ring failure on rapid decompression of systems incorporating gas seals, thus resulting in excessive leakage. Consideration must be given to the system in order to tailor system decompression times in order to meet the requirements of the secondary seals. As previously mentioned, all gas seals will leak, but not until the face 'lifts off'. This speed known, oddly enough, as 'lift off speed' is usually less than 500 rpm. Caution must be exercised in variable speed applications to assure the system prevents the operation of the variable speed driver below this minimum lift off value.

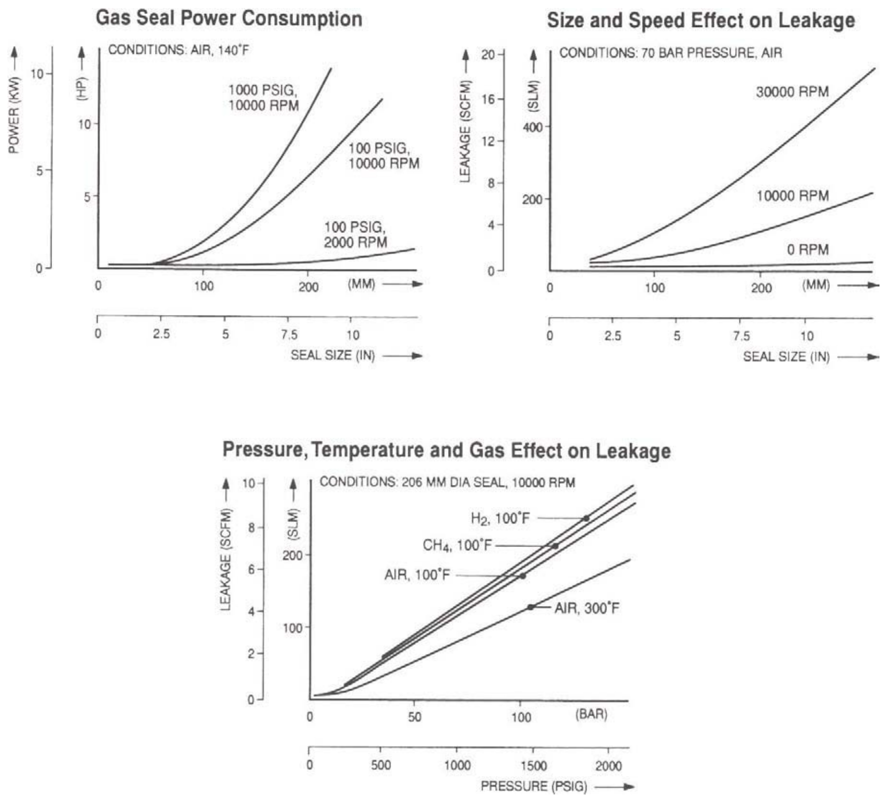


Figure 7.121 Dry gas seal leakage rates (Courtesy of John Crane Co.)

One recommendation concerning instrumentation is to provide one or two thermocouples in the stationary face of each seal to measure seal face temperature. This information is very valuable in determining lift off speed and condition of the grooves in the rotating seal face. Any clogging of these grooves will result in a higher face temperature and will be a good indication of requirement for seal maintenance.

Gas seal system types

As mentioned in this section, in order to assure the safety and reliability of gas seals, the system must be properly specified and designed. Listed below are but a few typical gas seal systems.

Low/medium pressure applications – air or inert gas

Figure 7.115 shows such a system. This system is identical to that of a liquid pump flush system incorporating relatively clean fluid that meets the requirements of the seal in terms of temperature and pressure. This system takes the motive fluid from the discharge of the compressor through dual filters (ten microns or less) incorporating a differential pressure gage and proportions equal flow through flow meters to each seal on the compressor. As previously discussed, compressors are usually pressure balanced such that the pressure on each end is approximately equal to the suction pressure of the compressor. The clean gas then enters the seal chamber and has two main paths:

A Through the internal labyrinth back to the compressor.

B Across the seal face and back to either the suction of the compressor or to vent.

Since the gas in this application is inert, it can be vented directly to the atmosphere or can be put back to the compressor suction. This would be the case also for a flammable gas. It must be noted, however, that this port is next to the journal bearing. Therefore a means of positively preventing entry of lube oil into this port must be provided in order to prevent the loss of lube oil or prevent the ingestion of lube oil into the compressor if this line is referenced back to the compressor suction. A suitable design must be incorporated for this bushing. Typically called a disaster bushing, it serves a dual purpose of isolating the lube system from the seal system and providing a means to minimize leakage of process fluid into the lube system in the event of a gas seal failure. In this system, a pressure switch upstream of an orifice in a flare line is used as an alarm and a shutdown to monitor flow. This switch uses the concept of an equivalent vessel in that increased seal leakage will increase the rate of supply versus demand flow in the equivalent vessel (pipe) and result in a higher pressure. When a high flow is reached, the orifice and pressure switch setting are thus sized and selected to alarm and shut down the unit if necessary. As in any system, close attention to changes in operating parameters is required. Flow meters must be properly sized and maintained clean such that relative changes in the flows can be detected in order to adequately plan for seal maintenance.

High pressure applications – air or inert gas

In this application, for pressures in excess of 1,000 psi, a tandem seal arrangement or series seal arrangement is usually used. Since failure of the inner seal would cause significant upset of the seal system, and large amounts of gas escaping to the atmosphere, a backup seal is employed. Refer to Figure 7.122.

The arrangement is essentially the same as low/medium pressure applications except that a backup seal is used in place of the disaster bushing. Most designs still incorporate a disaster bushing between the backup seal and the bearing cavity. Attention in this design must be

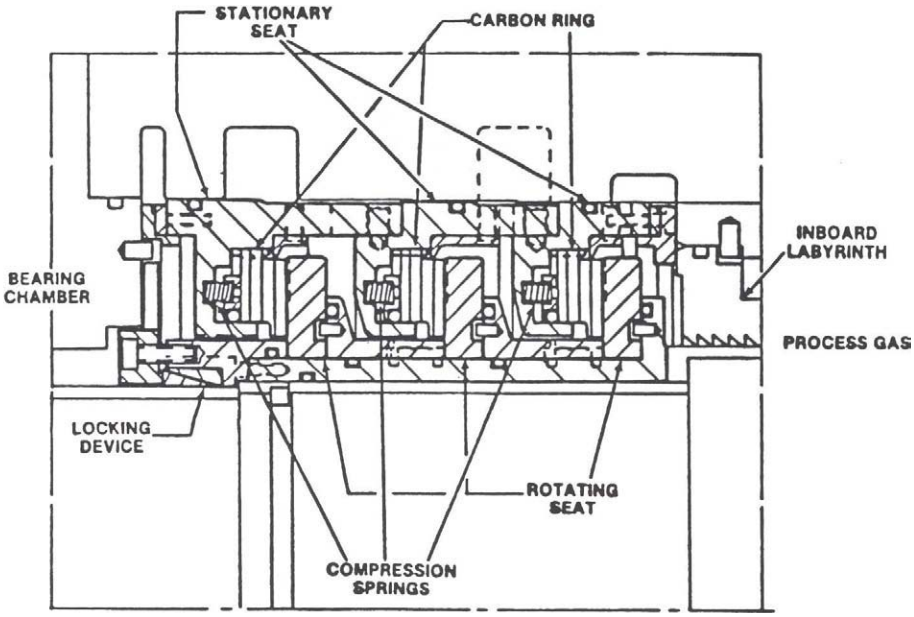


Figure 7.122 Dry gas seal: tandem dry gas seal arrangement (Courtesy of Dresser-Rand Corp.)

given to control of the inter-stage pressure between the primary and backup seal. Experience has shown that low differentials across the backup seal can significantly decrease its life. As in the case of liquid seals, a minimum pressure in the cavity between the seals of 25–30 psi is usually specified. This is achieved by properly sizing the orifice in the vent or reference line back to the suction to assure this pressure is maintained. All instrumentation and filtration are identical to that of the previous system.

Low/medium pressure application toxic or flammable gas

Please refer to Figure 7.123.

This system incorporates a double mechanical seal. In order to eliminate the possibility of flammable or toxic process gas escaping to the atmosphere, a buffer gas system utilizing inert gas is constantly injected between the primary and secondary seal. The buffer gas is regulated such that the pressure at the primary seal interface is greater than the seal process gas pressure. In order to achieve this objective, a differential control valve is installed on the inlet to the buffer gas port which senses pressure inside the cavity between the seals and reference pressure on the compressor side of the primary seal. A pressure differential of three to five psi is maintained to assure flow of buffer gas

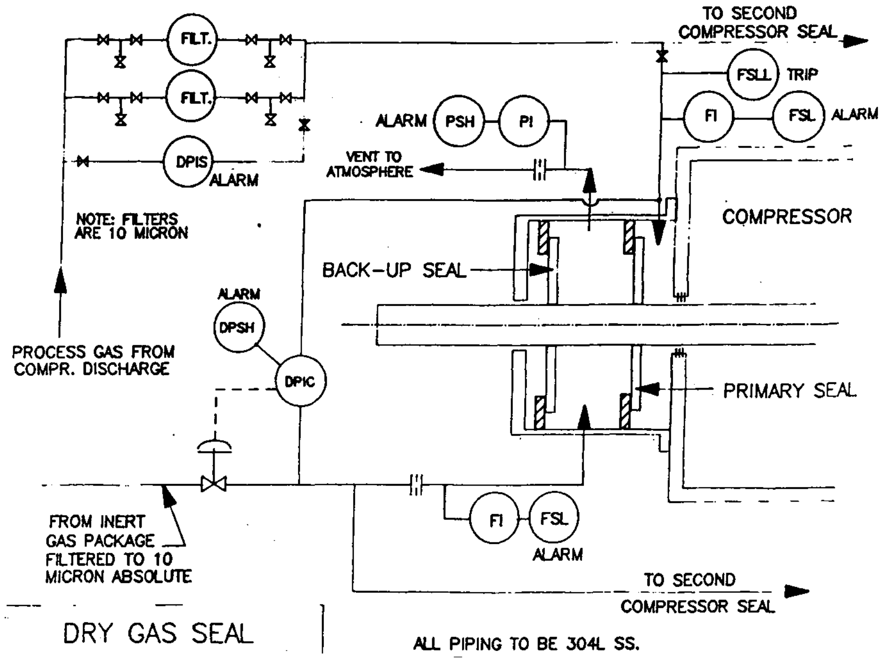


Figure 7.123 Double gas seal configuration for flammable or toxic process gas service (Courtesy of M.E. Crane Consultant)

to the unit. The backup seal then seals between the pressure between the two seals, which is now inert gas, and the atmosphere thus creating a safe seal situation at all times.

In this design, however, absence of proper buffer gas flow (differential between seal faces) will be cause for alarm on decreasing differential and unit shutdown if the condition does not correct itself. The reliability of this system is dependent on the buffer gas system. This means that the buffer gas system must continuously supply the buffer fluid at the correct pressure, temperature, flow rate, etc. A buffer gas compressor shutdown or failure will necessitate shutdown of the critical equipment. The buffer gas compressor could range from a large nitrogen compressor down to a small packaged unit furnished with a buffer gas inert package. Inert gas packages that generate nitrogen directly from atmospheric air are available to utilize as a buffer gas source for this design where a large nitrogen system is not available.

A variation of this design would have to be considered if the process did not allow leakage of inert gas into the process system. This would be the case in any process recycle application such as a hydrogen reformer, etc. In this case, a double seal would be utilized as the primary seal and would seal between suction and discharge pressure. The discharge flow would be filtered as in the case of example one in this section. Leaks from the primary seal into the port between the seals would then be washed with inert gas. The backup seal would be designed such that buffer gas would be admitted to the inner cavity via a tight labyrinth and leakage would be controlled by an orifice to vent or flare. The design would be such that the ratio of inner to process gas flow would be approximately ten to one. The backup seal leakage of inert gas to atmosphere would be controlled tightly by the seal gas face. In this design, flow meters would be typically used to measure maximum flow from the inert gas system. Excess flow would initiate an alarm and final shutdown of the unit.

There are different schools of thought concerning the use of buffer gasses. In pipeline operations, frequently the gas between the primary and backup seal may be sweet fuel gas. In this case, caution must be paid to failure of the backup seal. Failure of the backup seal will admit sweet fuel gas directly to the atmosphere and the lube oil system.

Summary

Since there are significant advantages to the use of dry gas seals, many units are being retrofitted in the field which incorporates this system. In many cases, significant payouts can be realized.

If a unit is to be retrofitted, it is strongly recommended that the design of the gas seal be thoroughly audited to assure safety and reliability. As mentioned in this section, retrofitting from a liquid to a gas seal system renders the unit a separate system type unit, that is, a separate lube and gas seal system. Naturally, loss of lube oil into the seal system will result in significant costs and could result in seal damage or failure by accumulating debris between the seal rotating and the stationary faces. The adequate design of the separation barriers between the lube and seal face must be thoroughly examined and audited to assure reliable and safe operation of this system. Many unscheduled field shutdowns and safety situations have resulted from the improper design of the lube system, seal system separation labyrinth. In addition to the above considerations, a critical speed analysis, rotor response and stability analysis (if applicable) should always be conducted when retrofitting from liquid to dry gas seals.

Auxiliary system types and functions
(Auxiliary systems knowledge)

- Introduction
- Types of auxiliary systems
- Summary of system functions, similarities and differences

Introduction

In this chapter, we will direct our attention to the types of auxiliary systems that will be covered. The major types of systems to be covered will be overviewed and the functions, similarities and differences of each system type will be summarized. Note that only the functional design as detailed on the system schematic will be discussed. The system arrangement design as detailed on the system outline or model will be discussed in a later chapter.

Types of auxiliary systems

The following is a brief summary of the types of auxiliary systems that will be covered. It must be understood that there are many variations of these systems both in terms of functional design and arrangement of components. However, the basic function of a specific type of system is the same regardless of the variations. As we proceed through the course, many of the system variations will be presented. Readers are encouraged to introduce system variations with which they are familiar. All of the system types presented in this section will be covered in detail later in this book.

Lubrication systems

There are about as many different types of lubrication systems as there are lubricants. In this course we will confine our attention to the pressurized types that should always be used with critical equipment. Two types of pressurized lube oil systems using positive displacement pumps are shown in Figures 7.124 and 7.125. Figure 7.124 depicts a system with a shaft driven lube oil pump while Figure 7.125 shows both pumps driven by sources other than the shaft. Figure 7.126 details a lubrication system using dynamic (centrifugal) pumps.

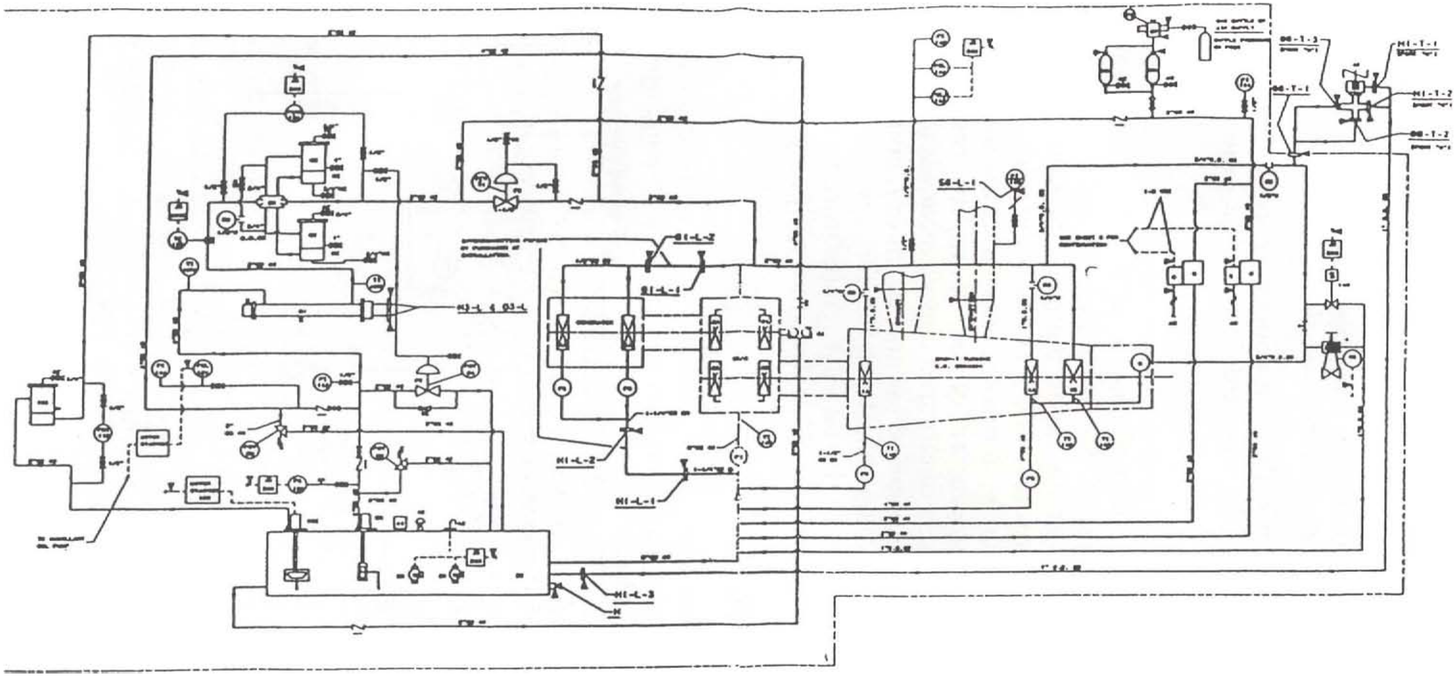


Figure 7.124 Lube oil system main pump shaft driven (Courtesy of Elliott Co.)

Regardless of the design variations or component type, the basic function of any lubrication system in critical equipment service is as follows:

To *continuously supply clean lubricating fluid* to *each specified point* at the required *pressure, temperature* and *flow rate.*

As an exercise, identify the component or components in Figures 7.124, 7.125 and 7.126 that are required to satisfy each italicized phrase in the above definition.

Before we proceed, one comment concerning the types of pumps used in auxiliary systems. The performance of positive displacement pumps (screw, gear, rotary, piston, diaphragm, etc.) is not significantly affected by the characteristics of the liquid pumped. However, dynamic pump (centrifugal) performance varies significantly with different liquid characteristics. As a result, dynamic pumps are used where the pumped liquid characteristics can easily be controlled. Since the liquid characteristics (viscosity) of oil varies considerably with temperature and the oil reservoir temperature is not accurately controlled, positive displacement pumps are usually used for systems containing oil. However, in cases where the oil inlet temperature to the pump can be held relatively constant (±25°F), a dynamic type pump can be used.

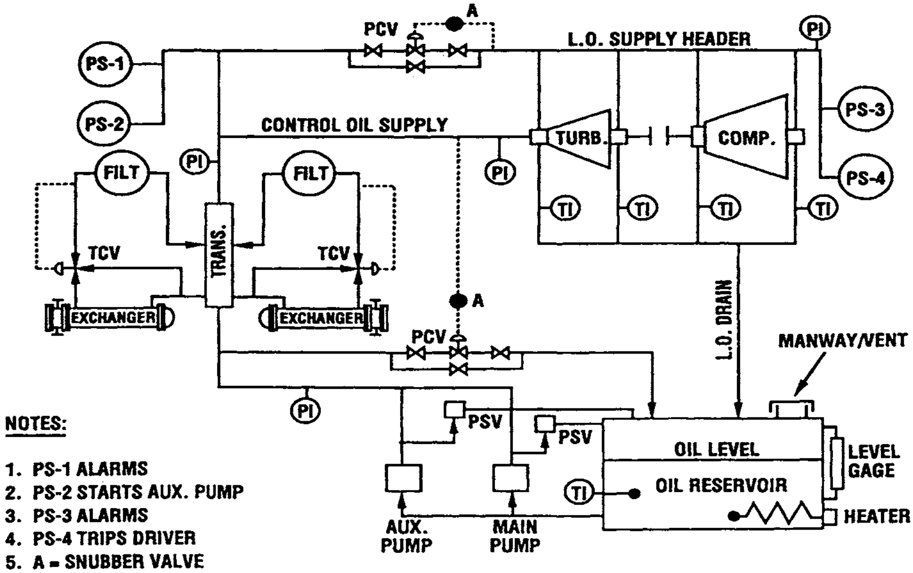


Figure 7.125 Typical lube oil supply system

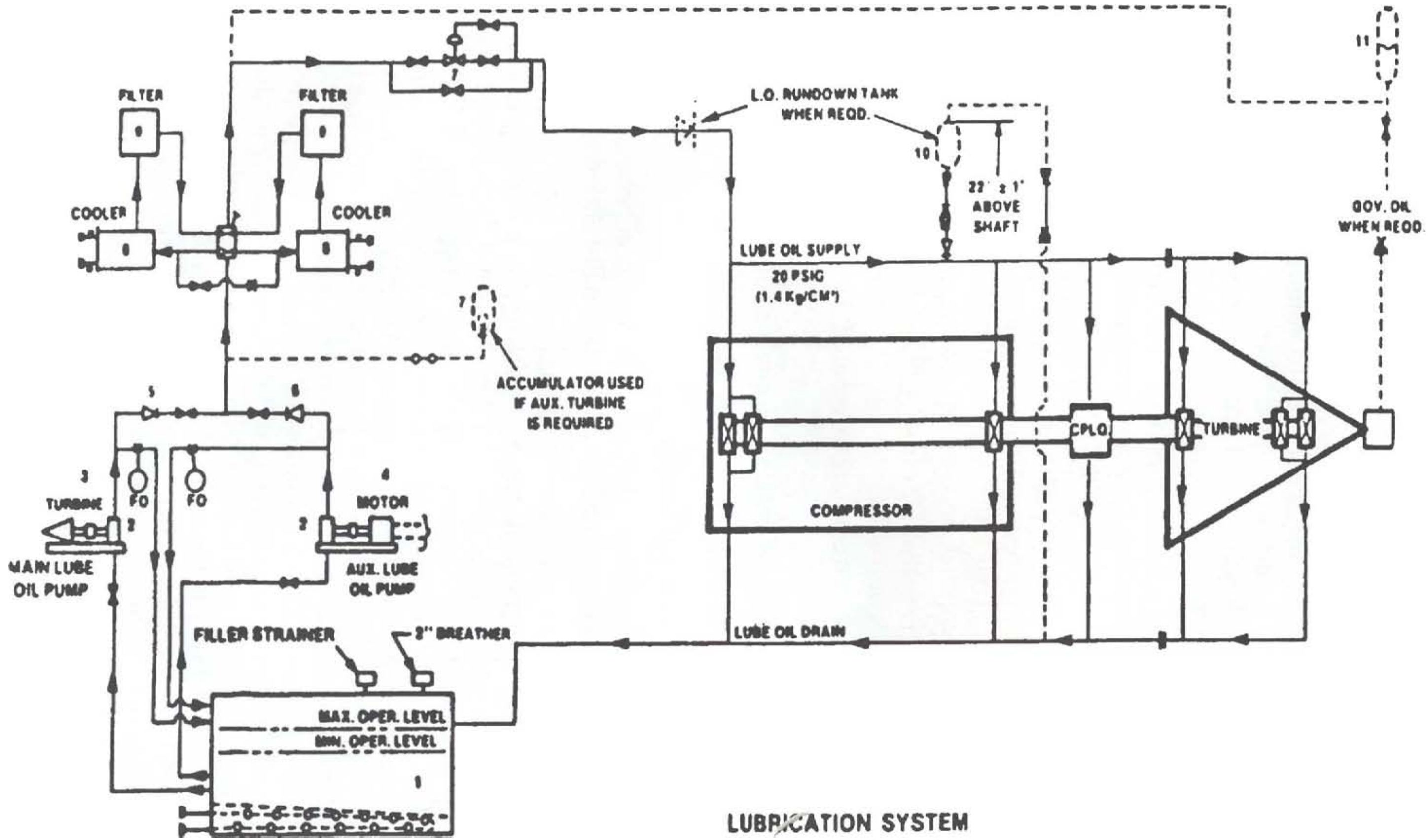


Figure 7.126 Lubrication system (Courtesy of Dresser-Rand)

Fluid sealing systems

Again, there are numerous types of fluid seal systems. In fact, there are more variations than lubrication systems since the types of seals utilized, sealing fluids and sealing pressures vary widely. Notice that this section is entitled *fluid* seal systems as opposed to liquid seal systems. A fluid by definition can be a liquid or a vapor. There are both liquid and vapor (gas) seal systems. Regardless of the type, the function of a critical equipment seal system is:

> To *continuously supply clean fluid* to *each specified seal interface point* at the required *differential pressure, temperature* and *flow rate.*

Naturally, the function of the seal is to contain the process fluid. Since the seal interface is between the rotating and stationary components of the seal, friction and corresponding energy loss (BTU's/#, horsepower) result. The seal fluid (oil, water, vapor) must be introduced into the seal at the correct differential pressure above the process fluid pressure in order to contain the process fluid and to remove the heat of seal interface friction.

Three types of seal systems are presented in this section for turbo-compressor applications:

- Figure 7.127 – Seal oil system – contact seal
- Figure 7.128 – Seal oil system – bushing seal
- Figure 7.129 – Gas seal system

As an exercise, compare the seal system components with the lubrication system components and state the major differences.

Figure 7.130 shows a combined lubrication and sealing system that would be used with a liquid seal. Since the function of a lubrication and a seal system are similar they can be combined in certain instances where required flow rates, pressures and *process gas characteristics* permit. Extreme care must be exercised when designing a combined system containing entrained seal gas in the sealing oil (process gas or inert-buffer gas). Since the system is combined, the seal oil leakage, if it is returned to the reservoir, must be effectively 'degassed' to assure the bearing lubricating film is incompressible (gas free). Failure to do this can result in catastrophic bearing and equipment failure. In addition, a saturated lube oil will have a significantly lower 'flashpoint'. The flashpoint is defined as the temperature at which a lube oil will support combustion. There have been cases where returned seal oil leakage flashpoints have been recorded as low as 120°F! (Typical oil flashpoints are 400–500°F.) Seal oil leakage degassing is a primary factor in seal oil system safety and reliability.

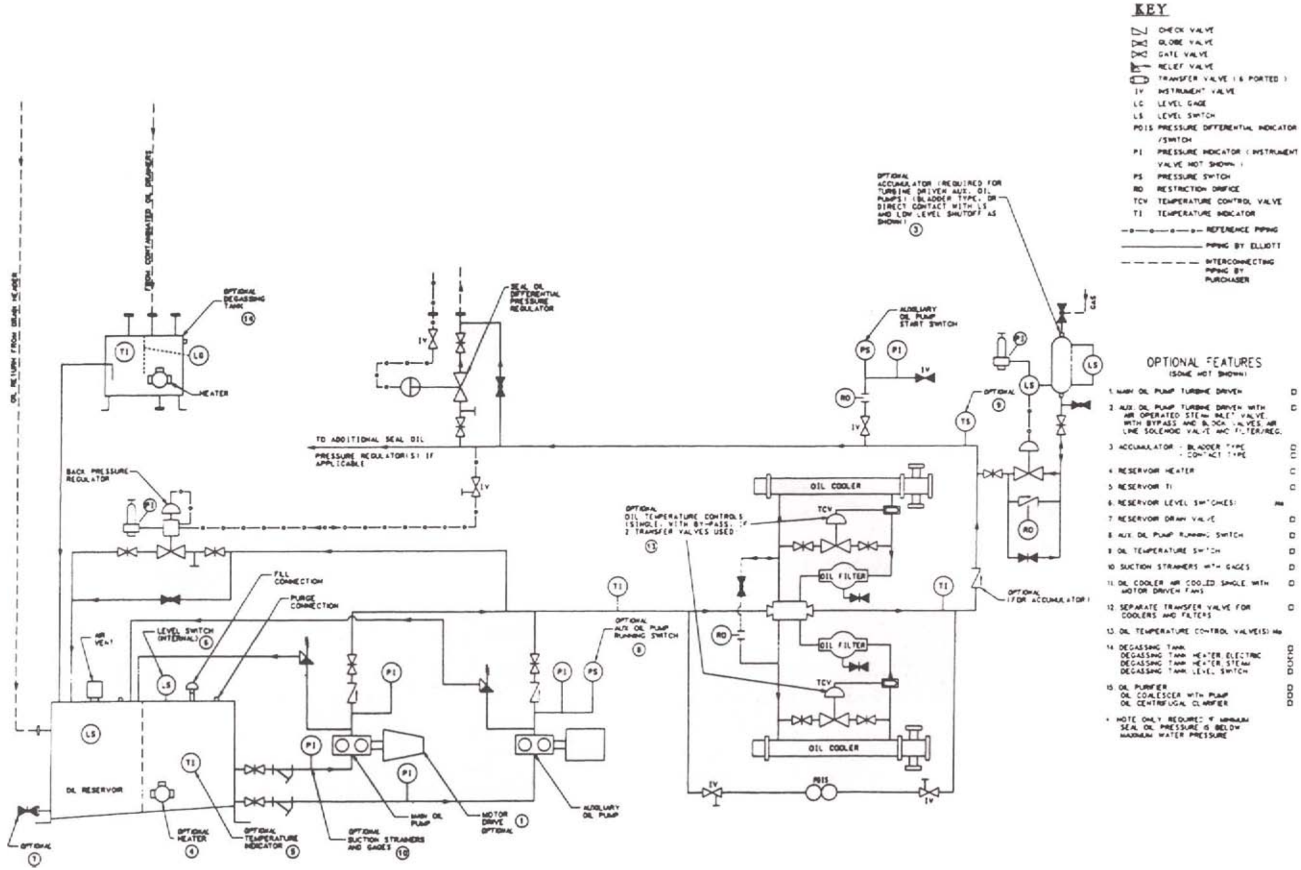


Figure 7.127 Elliott seal oil console schematic for compressors with ISO-carbon seals (Courtesy of Elliott Co)

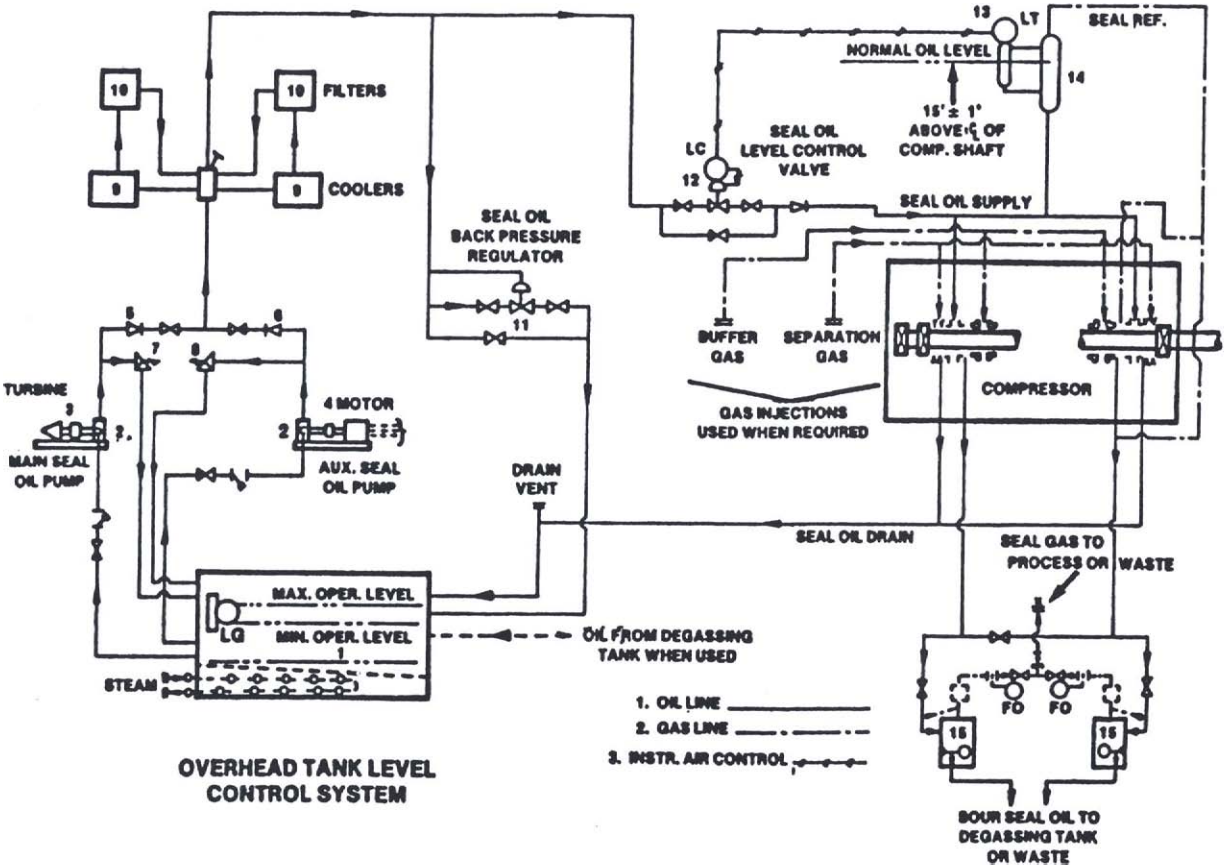


Figure 7.128 Bushing seals (Courtesy of Dresser-Rand)

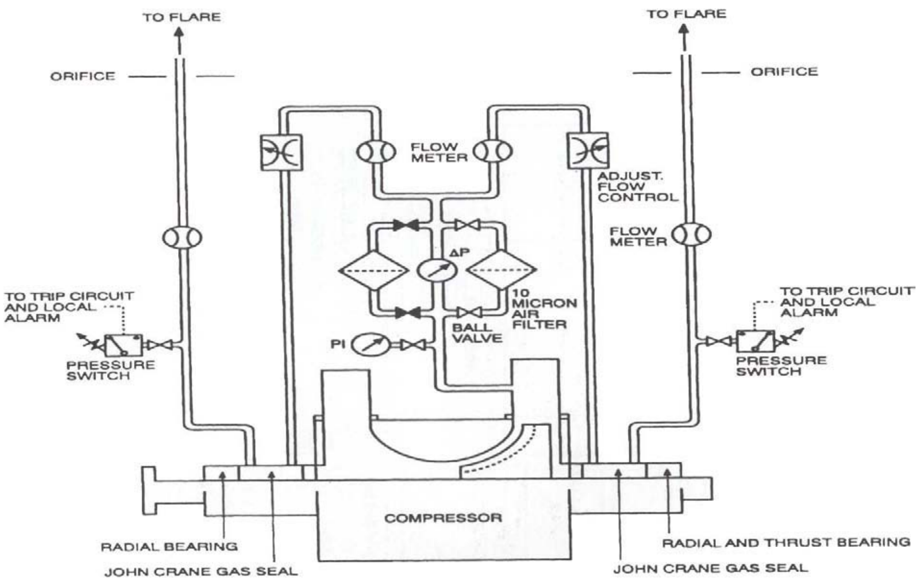


Figure 7.129 Typical gas seal system (Courtesy of John Crane Co)

Buffer fluid systems

This large category of auxiliary systems covers buffer gas systems for turbo-compressors, steam seal systems, eductor systems and pump flush systems.

In turbo-compressor applications, a buffer fluid (gas) can be used for any of the following purposes:

A. Prevent the process gas from coming in contact with the sealing fluid

B. Prevent the sealing fluid from entering the process stream

C. Control the seal area environment (pressure and temperature)

Alternatives B and C above can use the process gas if it is available at the required pressures and temperatures and is compatible with the sealing fluid. Figure 7.131 is an example of a buffer gas system designed for alternative A.

Figure 7.132 shows a system designed for alternatives A and B.

Figure 7.133 shows a steam seal system whose purpose is to direct leak off steam to the specified locations thus preventing steam (moisture) leaking from the shaft seals and entering the lubrication system. This arrangement reduces the process stream to an acceptable pressure level and also uses a subatmospheric pressure source to educt air to prevent external seal steam leakage.

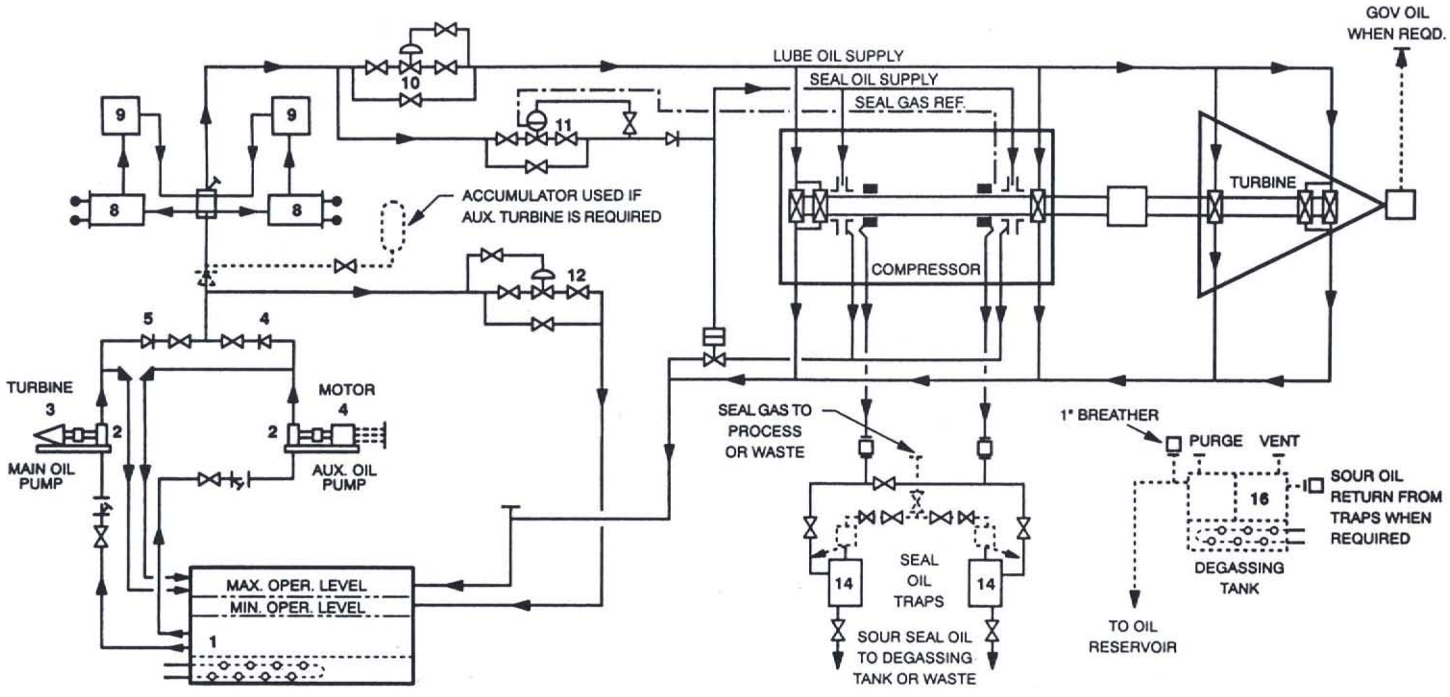


Figure 7.130 Combined lube and seal system (Courtesy of Dresser-Rand)

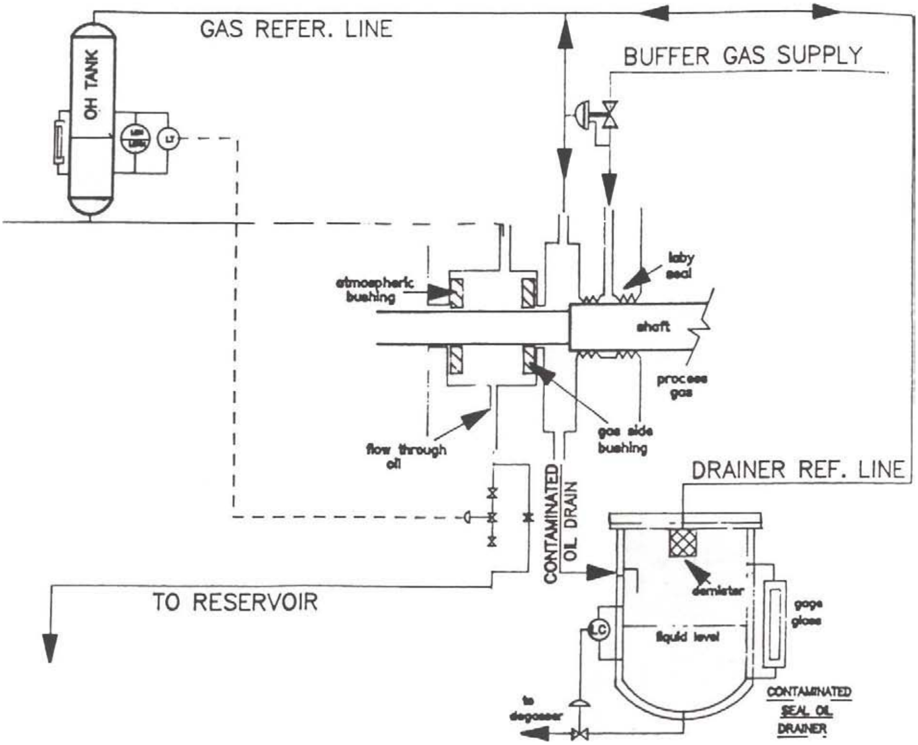


Figure 7.131 Typical buffer gas system (Courtesy of M.E. Crane Consultant)

The final type of buffer fluid systems covered in this chapter involve pump sealing systems. When the pumped liquid conditions do not meet the requirements of the pump seal (pressure, temperature, cleanliness, liquid characteristics, etc.) a buffer system (commonly called a flush system) must be utilized. The buffer liquid can be the pumped liquid modified to meet the seal requirements (filtered or cooled), an externally supplied liquid compatible with the pumped liquid or a barrier liquid used in a tandem or double seal arrangement. Typical examples of these systems are shown in Figures 7.134, 7.135 and 7.136.

As in lubrication and sealing systems, the basic function of any critical equipment buffer fluid system is as follows:

> To *continuously supply clean buffer fluid* to *each specified point* at the required differential *pressure*, *temperature* and *flow rate*.

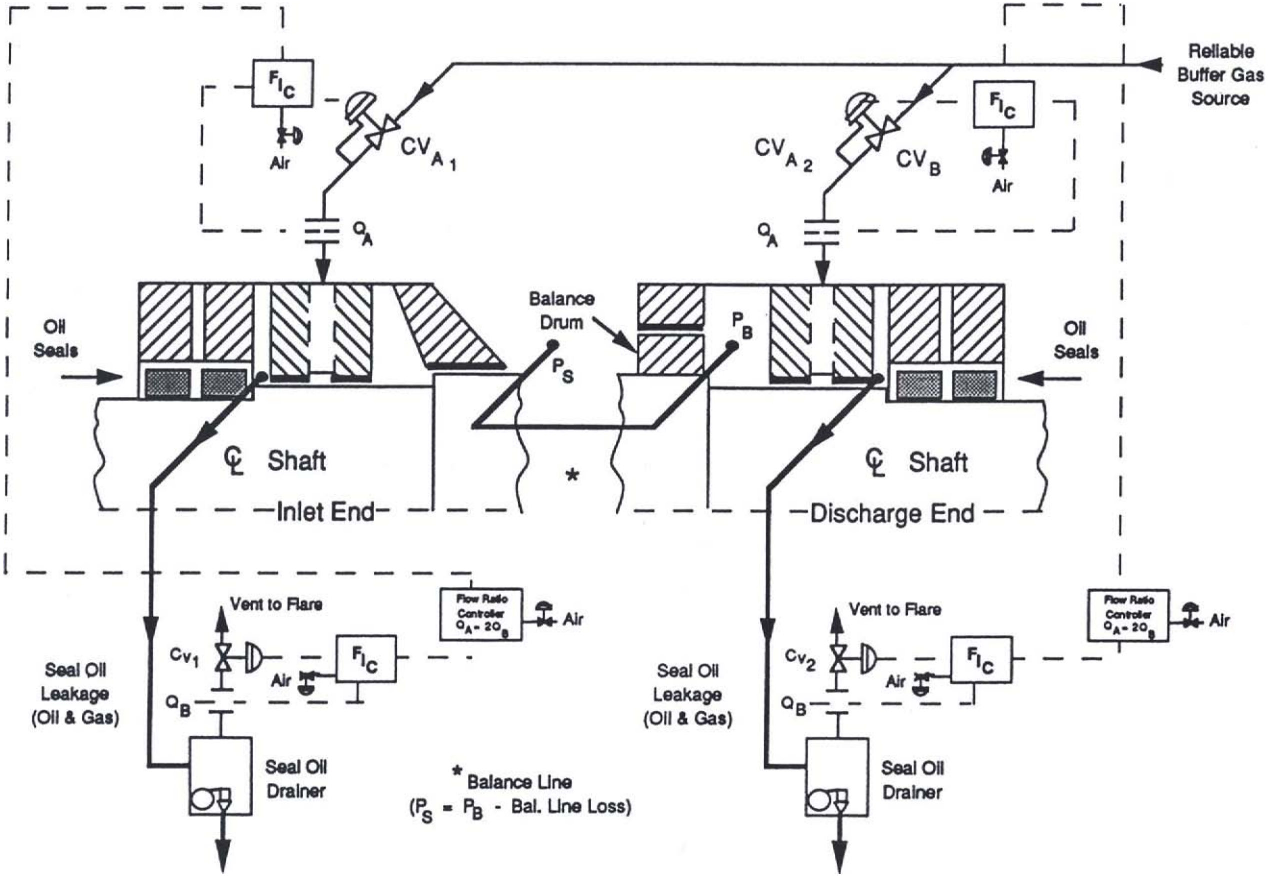


Figure 7.132 Flow ratio buffer gas control system

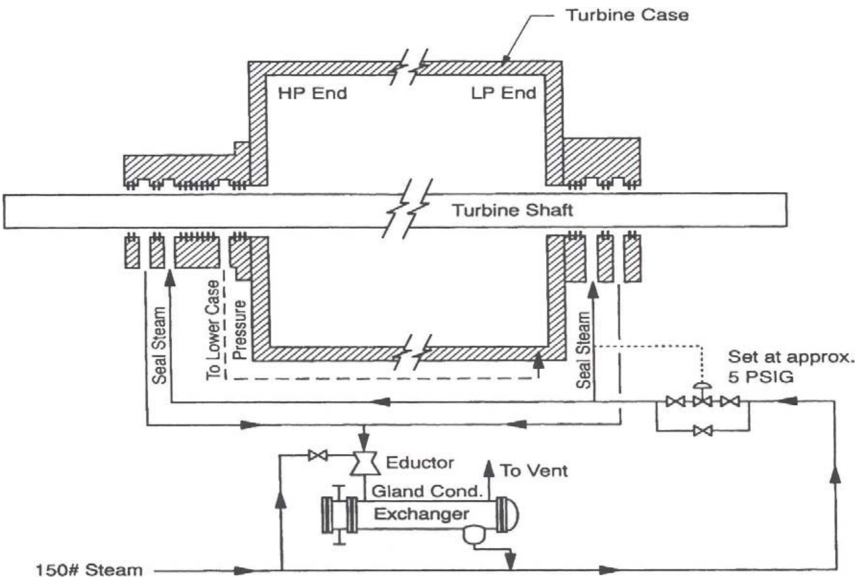


Figure 7.133 Steam turbine gland seal system (Courtesy of M.E. Crane Consultant)

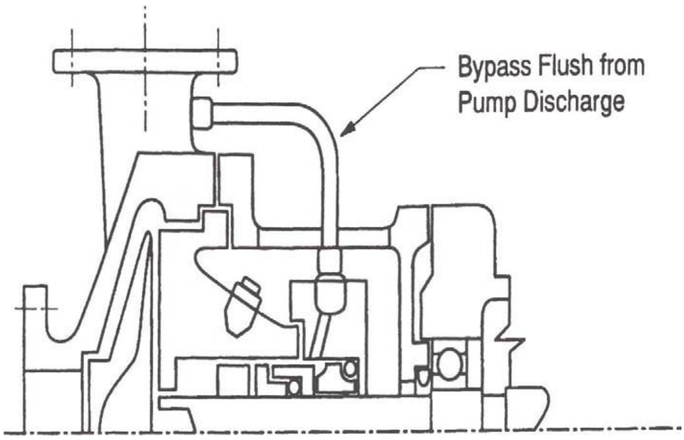


Figure 7.134 Single inside seal with bypass flush from pump discharge (Courtesy of Durametallic Corporation)

Plant utility systems

The purpose of including this topic is to bring attention to the fact that each item of critical equipment and all of its auxiliaries are part of the plant utility system which includes:

- The plant and instrument air system
- The plant steam system

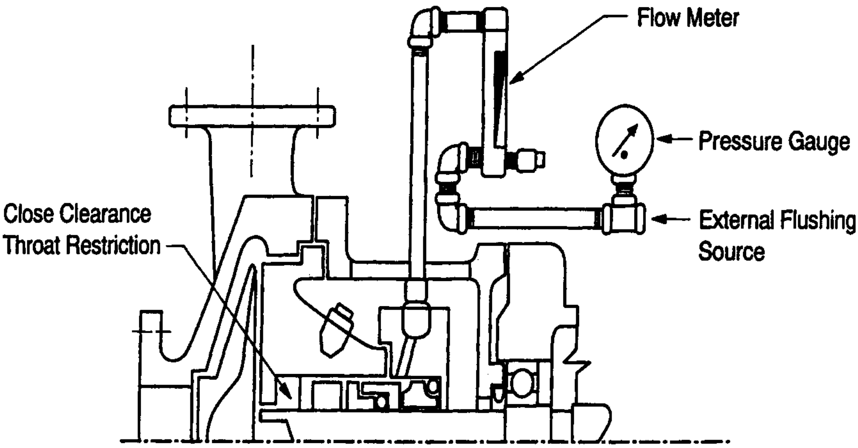


Figure 7.135 Single inside seal with flush from external source (Courtesy of Durametallic Corporation)

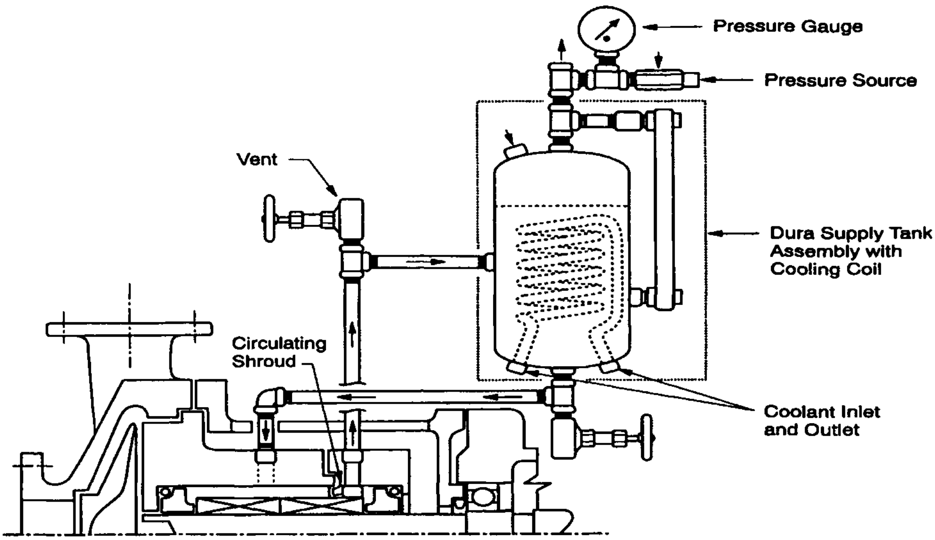


Figure 7.136 Double inside seal with induced circulation through supply tank with cooling coil (Courtesy of Durametallic Corporation)

- The plant cooling water system

To name just a few. They are also auxiliary systems and have the same function:

> To *continuously supply clean utility fluid* to *each specified point* at the *required pressure, temperature and flow rate.*

Summary of system functions, similarities and differences

Having divided critical equipment auxiliary systems into four groups, and having defined the function of each group, let's attempt to create a universal auxiliary system function definition. Reviewing the stated function definition of each group and using only exactly similar phrases we obtain the following definition:

To continuously supply clean fluid to each specified point at the required pressure, temperature and flow rate.

The only major functional difference is that lubrication and plant utility systems require *pressure control* whereas seal and buffer systems require *differential pressure* control. Why is this so? Sealing and buffer systems require differential pressure control because the fluids they are acting on can vary in pressure but lubrication and utility systems ultimately act against atmospheric pressure which is essentially constant.

As one last section exercise let's examine the universal auxiliary system definition above and list the system components common to all auxiliary system groups.

It is hoped that the information contained in this section will enable the reader to better understand the function of each major group of auxiliary system types and how similar all the major auxiliary system groups are in terms of function. Once this fact is understood, the understanding of each specific type of auxiliary system becomes clearer.

Answers for each component and system function knowledge base section in this chapter

1. The rotor and the process system

Function exercise

Component: *Pump or compressor rotor*

I. 'What' - The function of Rotor is to *produce the specified value of fluid head and efficiency without impeller, shaft or any other installed component failure while maintaining acceptable values of machine vibration, axial displacement and bearing temperature.*

II. 'How' - The function of Rotor is accomplished by *proper impeller design tip speed and fluid velocity relative to the impeller and proper stationary passage design (volute, diffuser, stage seal and crossover). It is assumed that the all clearances are to specification, the rotor is balanced and the impeller and all passages are free of fouling. (Debris).*

Figure 7.137 Function exercise - Component: Pump or compressor rotor

Component failure causes

Component: *Pump or compressor rotor*

List all possible failure causes. (Note: examine each of the failure classifications – in preparing this list).

Reference
Number

1. Low value of head required by process causing choke or cavitation
2. High value of head required by process causing surge, recirculation or vaporization
3. Fluid density change causing low or high head required by the process
4. Liquid vapor pressure change causing cavitation
5. Foreign object damage from process system (solids, liquids or gas)
6. Improper impeller/seal radial clearances causing items 1 or 2
7. Improper axial clearances causing impeller/stationary passage failure
8. Operating errors – closing suction valve, discharge valve etc.
9. Improper impeller aerodynamic design (head & efficiency produced) causing choke or cavitation or driver overload
10. Improper impeller aerodynamic design (head & efficiency produced) causing surge, recirculation or vaporization
11. Improper impeller mechanical design (stress, bore, seal clearances) causing impeller and/or shaft failure
12. Improper material
13. Improper manufacturing techniques
14. Natural frequencies of rotor and/or impellers excited by operating conditions (critical speeds)
15. Wear out of components due to excessive liquid entrainment

Figure 7.138 Component failure causes – Component: Pump or compressor rotor

Failure cause monitoring parameters

Component: *Pump or compressor rotor*

List all parameters (instruments) to be monitored or used for each failure cause listed in Figure 7.138. (Note: use corresponding failure cause reference number (same reference number as in Figure 7.138).

Reference
Number

1. Process pressures, temperatures, flows and fluid composition
2. Process pressures, temperatures, flows and fluid composition
3. Process pressures, temperatures, and fluid composition
4. Fluid sample, suction pressure and temperature
5. Suction pressure, temperature, flow, fluid conditions and inspection of suction system including suction screen
6. Vibration analysis
7. Vibration analysis, axial displacement and thrust pad temperature
8. Review of process system trend values (machine suction and discharge pressures and flows)
9. Process pressures, temperatures, flows and fluid composition for performance analysis
10. Process pressures, temperatures, flows and fluid composition for performance analysis
11. Vibration analysis
12. Vibration analysis
13. Vibration analysis and process pressures, temperatures, flows and fluid composition for performance analysis
14. Vibration analysis with machine pressures, temperatures and fluid composition
15. Fluid composition, process conditions and vibration analysis

Figure 7.139 Failure cause monitoring parameters - Component: Pump or compressor rotor

2. Anti Friction Journal Bearings

Function Exercise

Component: *Anti-friction journal bearing*

I. 'What' – The function of an Anti-Friction Journal bearing is to support the Rotor in the radial direction.

II. 'How' – The function of the Anti-Friction Journal bearing is accomplished by having sufficient contact area to support the maximum anticipated radial loads, properly installed and being supplied with clean, cool lubricating oil of the proper viscosity to provide the required lubrication for the frictional heat load

Figure 7.140 Function exercise – Component: Anti-Friction Journal Bearing

Component Failure Causes

Component: *Anti-friction journal bearing*

List all possible failure causes. (Note: examine each of the failure classifications in preparing this list).

Reference
Number

1. Insufficient bearing area
2. More load than anticipated
3. Incorrect installation
4. Unclean oil
5. Hot oil
6. Incorrect oil viscosity used or incorrect oil temperature
7. Incorrect oil mass flow rate

Figure 7.141 Component failure causes – Component: Anti-Friction Journal Bearing

Failure cause monitoring parameters

Component: *Anti-friction journal bearing*

List all parameters (instruments) to be monitored or used for each failure cause listed in Figure 7.141. (Note: use corresponding failure cause reference number (same reference number as in Figure 7.141).

Reference
Number

1. Vibration analysis, bearing housing temperature, oil supply and drain temperature
2. Process conditions (flow, fluid composition (SG), suction and discharge pressure, suction temperature), vibration analysis of pipe supports and foundation
3. Vibration analysis, bearing housing temperature, oil supply and drain temperature
4. Oil particle analysis
5. Oil supply temperature
6. Oil analysis and oil temperature
7. Check oil ring movement with strobe light

Figure 7.142 Failure cause monitoring parameters – Component: Anti-friction journal bearing

3. Thrust Bearings

Note: Please refer to the Hydrodynamic Thrust Bearing Example in this chapter, see pages 102–106.

4. Shaft End Seals

Function Exercise

Component: *Pump mechanical seal*

I. 'What' – The function of a Pump mechanical seal is to minimize fluid leakage to the atmosphere

II 'How' the function of a *pump mechanical seal* is accomplished by having the *proper seal pressure – velocity and fluid condition (Vapor pressure, Temperature, Pressure) and seal flush rate*

Figure 7.143 Function Exercise – Component: Pump mechanical seal

Component failure causes

Component: *Pump mechanical seal*

List all possible failure causes. (Note: examine each of the failure classifications in preparing this list).

Reference
Number

1. Improper fluid condition (vapor pressure)
2. Improper pressure in the stuffing box
3. Improper temperature in the stuffing box
4. Plugging of flush line orifice and/or strainer
5. Excessive throat bushing clearance
6. Improper seal assembly
7. Not installing flush line orifice
8. Improper venting of the pump
9. Not chilling down or warming up the pump properly
10. Not allowing tandem seal pots to vent
11. Improper seal balance design
12. Improper seal materials
13. Seal component wearout ('O' rings and secondary seals)

Figure 7.144 Component failure causes – Component: Pump mechanical seal

Failure cause monitoring parameters

Component: *Pump mechanical seal*

List all parameters (instruments) to be monitored or used for each failure cause listed in Figure 7.144. (Note: use corresponding failure cause reference number (same reference number as in Figure 7.144).

Reference
Number

1. Fluid sample
2. Measure pressure in the stuffing box
3. Measure temperature in the stuffing box
4. Measure pressure in the stuffing box
5. Measure pressure in the stuffing box
6. Measure pressure and temperature in the stuffing box
7. Measure pressure in the stuffing box
8. Measure temperature in the stuffing box
9. Measure temperature in the stuffing box
10. Measure pressure in the seal pot
11. Measure pressure and temperature in the stuffing box
12. Measure pressure and temperature in the stuffing box
13. Measure pressure and temperature in the stuffing box

Figure 7.145 Failure cause monitoring parameters – Component: Pump mechanical seal

5. Pressurized lube oil system

Function Exercise

System: *Pressurized lube oil system*

I. 'What' the function of *A pressurized lube oil system* is to *continuously supply cool, clean oil to the components at the required pressure, temperature and flowrate*

II. 'How' the function of *A pressurized lube oil system* is accomplished by *an oil reservoir with interconnecting pipe to main and auxiliary pumps, relief valves, control valves, transfer valves, coolers, filters, pressure transmitters, accumulator and instrumentation to monitor, auto start the pumps and shut down the unit in the event of a malfunction*

Figure 7.146 Function Exercise – System: Pressurized Lube Oil System

Component failure causes

System: *Pressurized oil system*

List all possible failure causes. (Note: examine each of the failure classifications in preparing this list).

Reference
Number

1. Incorrect oil viscosity
2. Incorrect pressure and temperature settings
3. Incorrect cooling water conditions
4. Incorrect component flows
5. Pump suction strainer blockage
6. Pump malfunction
7. Pump driver malfunction
8. Pump coupling failure
9. Relief valve malfunction
10. Control valve instability
11. Control valve excessive friction

12. Accumulator malfunction
13. Transfer valve malfunction
14. Cooler leaks
15. Filter plugging
16. Transfer to a cooler or filter that is not full, vented and heated (in cold climates)
17. Failure to change install filters correctly

Figure 7.147 Component failure causes – System: pressurized oil system

Failure cause monitoring parameters

System: *Pressurized lube oil systems*

List all parameters (instruments) to be monitored or used for each failure cause listed in Figure 7.147. (Note: use corresponding failure cause reference number (same reference number as in Figure 7.147).

Reference
Number

1. Lube oil sample
2. Check of all pressure and temperature settings
3. Monitor cooling water temperatures
4. Install temporary ultrasonic flow meter to confirm component flows and use control valve position, differential pressure, SG and valve C_v to measure flow where applicable
5. Monitor pump suction strainer differential pressure
6. Monitor pump(s) vibration, seal leakage and bearing temp.
7. Monitor pump driver(s) vibration, seal leakage, bearing temp. and control system (if applicable)
8. Inspect coupling visually and with strobe light during operation if pump and/or driver vibration increases
9. Check relief valves for leaks and proper setting
10. Check control valve controller settings & vent actuator

11. Check control valve actuator diaphragm and packing friction
12. Check accumulator precharge pressure and bladder condition
13. Check transfer valve internals for blockage or excessive friction
14. Check oil sample for water or monitor reservoir level
15. Monitor filter differential pressure
16. Confirm coolers and filters are full, vented and heated (in cold climates)
17. Check oil sample for particle count downstream of filter

Figure 7.148 Failure cause monitoring parameters – System: pressurized lube oil systems