

CENTRIFUGAL COMPRESSOR OPERATION AND MAINTENANCE

by

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ABSTRACT

The operation and maintenance of centrifugal compressors varies considerably depending upon the size, speed, horsepower, and type of driver.

Typical centrifugal compressors vary from high speed, low flow applications to very large flow, low speed machines. Each has its own operating and maintenance problems.

Centrifugal compressors are normally driven by electric motors and speed increasing gears for the low flow, low horsepower applications. The middle size ranges are usually motor-gear, steam turbine or gas turbine direct connected combinations. The very large flow, large horsepower machines are practically all steam turbine driven with a few being driven by gas turbine where natural gas fuel is available.

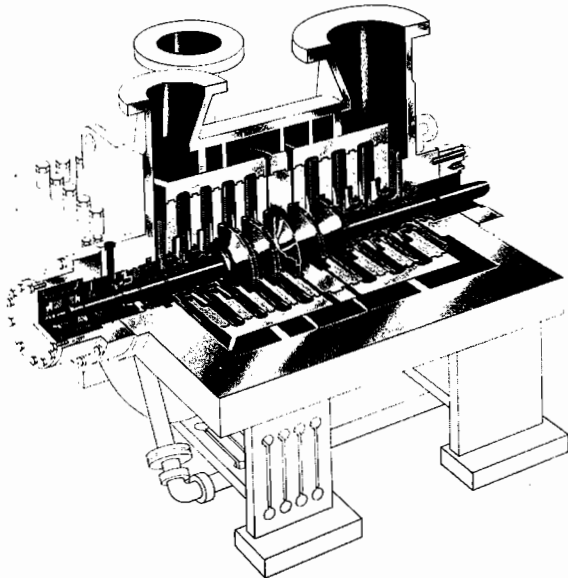


Figure 1. Multi-stage Fabricated Case Compressor.

Centrifugal compressors can be designed and built with operation and ease of maintenance practices in mind. Specific items such as off-design operation, bearings and seals, casing design and mounting, and lubrication systems are discussed.

INTRODUCTION

The process industry has experienced tremendous growth during the past two decades. The growth of single line and continuous process plants and the increasing use of automation have demonstrated the importance of component reliability. Machinery users are now demanding dependable performance, simplicity of operation and ease of maintenance. The suppliers of centrifugal compressors have been forced to review their designs and sometimes to design new equipment to satisfy these users' demands. This paper discusses the ways in which a centrifugal compressor can be built and used to satisfy a customer's operating and maintenance requirements.

TYPICAL MULTISTAGE COMPRESSOR DESIGN

A typical multistage centrifugal compressor, designed to meet a particular customer's needs, is shown in Figure 1. This machine consists of 9 impellers in

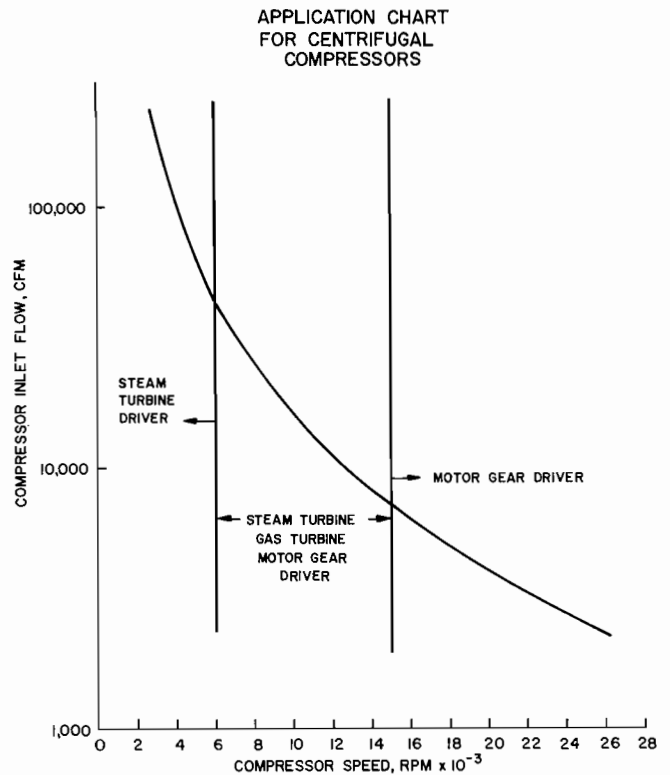


Figure 2. Application Chart for Centrifugal Compressors.

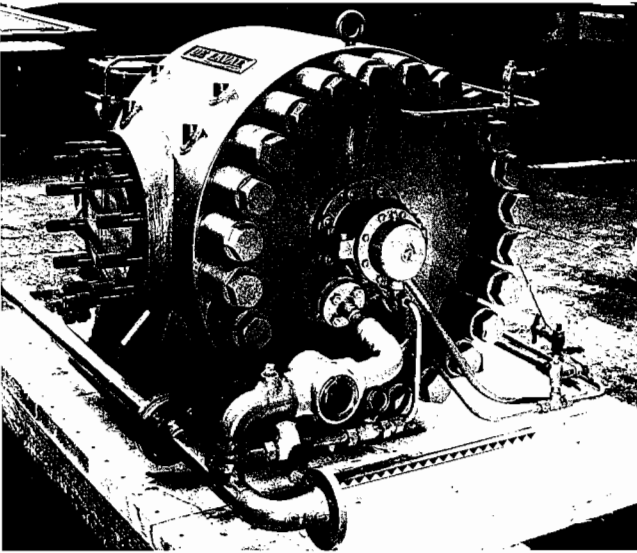


Figure 3. Four-Stage Barrel Compressor.

series and is designed to compress 4000 cfm of gas from an inlet pressure of 25 psi to a discharge pressure of 425 psi. Each impeller imparts velocity (kinetic) energy

to the gas being compressed. This velocity energy is converted into increased pressure in the diffuser passage. The cross-over passage and the return guide vanes lead the gas to the next impeller where the compression is continued. The volume of the gas stream is reduced as it is compressed and each stage is designed to accept a successively smaller flow.

RANGE OF APPLICATION

Figure 2 is a curve which shows the limits of applications for centrifugal compressors in terms of flow and speed. The speed is limited by the stresses in the impellers. The small flow, high speed compressors have the same working stress levels as the large flow, low speed machines. The compressor applications in the low flow range are almost entirely driven by motors and speed increasing gears. The compressors in the mid-range of flows are driven by motor-gears, steam turbines, and some gas turbines. The large, high flow compressors are practically all driven by steam turbines.

The size and operating speed of a centrifugal compressor have a direct effect on the operation and the maintaining of the compressor.

A small machine such as the one shown in Figure 3, besides having high rotating speeds, is usually high pressure as well. Shaft alignment is more critical since the

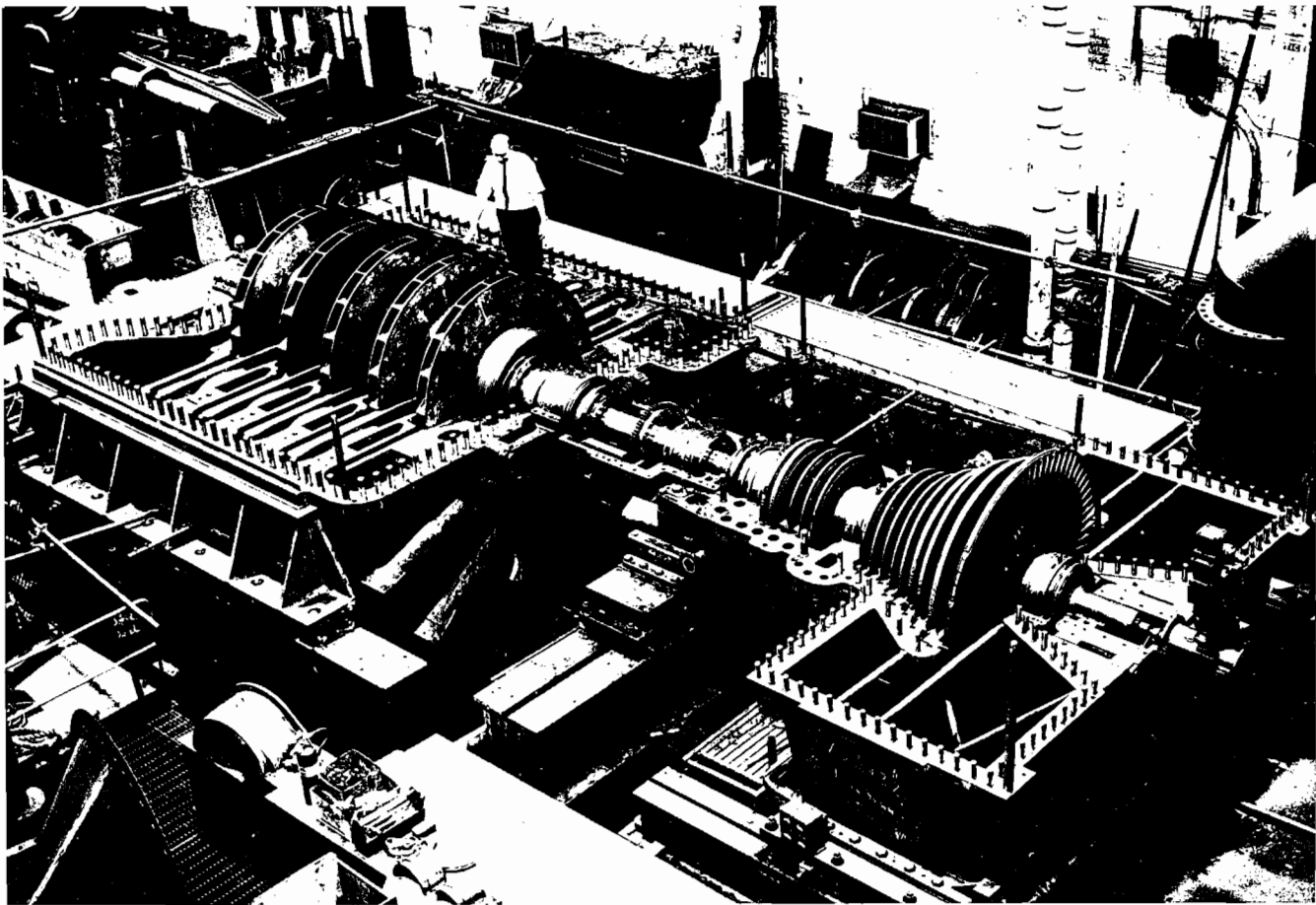


Figure 4. Turbine Driven Blast Furnace Blower.

shaft ends are small and external pipe forces must receive proper attention to ensure good operation of the equipment. Maintenance is easier due to the light weight of the compressor components, but the clearances of the internal seals and bearings must be watched more closely due to their small physical size.

A large compressor, such as the one shown in Figure 4, is more difficult to work on due to its size and weight. The clearances between rotating and stationary parts is less critical but the shaft alignment is more difficult to obtain, although more liberal tolerances are acceptable, because the components are not easy to move and special lifting facilities are required. The foundation for these large machines is also of special concern because unless the supports are designed, constructed, and maintained properly the machinery may never achieve trouble-free operation.

ROTOR DYNAMICS

Modern process compressors are built in accordance with the API specification 617. One important item defined by this specification is the natural frequencies of the rotor. These natural frequencies must not occur in the variable speed range of the compressor. The dynamics of a rotor can be studied with the help of the computer and the effect on the rotor of operative unbalance due to build-up misalignment can be evaluated. These rotor unbalances will load the bearings. Com-

puter analysis allows the engineer to predict these bearing loadings and to design a dependable maintenance free machine.

Figure 5 shows the calculated and measured rotor response curves for an eight-stage compressor rotor. The measured values were obtained first during the mechanical test of the compressor. Although the vibration level was less than 0.7 mils and the bearing forces were below the design dynamic load limit, the steepness of the vibration curve near the maximum operating speed was understandably cause for concern. The rotor was modified and retested.

These results are shown in Figure 6. The shop test shows quite low vibration amplitudes and low bearing loading. The compressor, once in actual service, will become unbalanced due to build-up on the rotor. The difference between the calculated and measured response curves shows that this compressor will be tolerant to considerable rotor deposits before it will have to be cleaned. Lightly loaded journal bearings, such as those used in compressors, can be unstable at high speeds, and a number of solutions to this problem have been used. The tilting pad bearing, like the one shown in Figure 7, is widely used in compressors. Each shoe tilts independently to maintain its load carrying hydrodynamic pressure wedge. Extensive service in many types of compressors have proven the dependability of this bearing.

ROTOR RESPONSE CURVES FOR EIGHT STAGE ROTOR

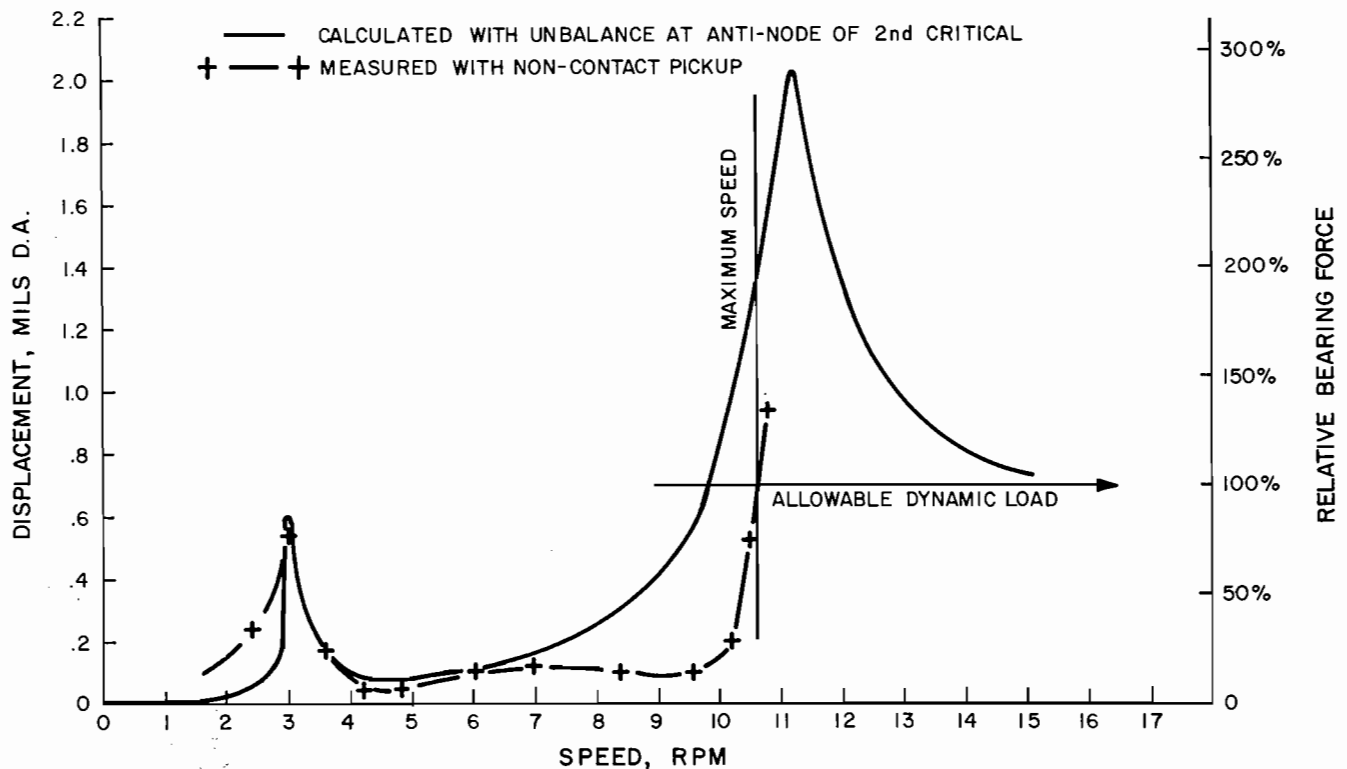


Figure 5. Eight-Stage Rotor Response Curves.

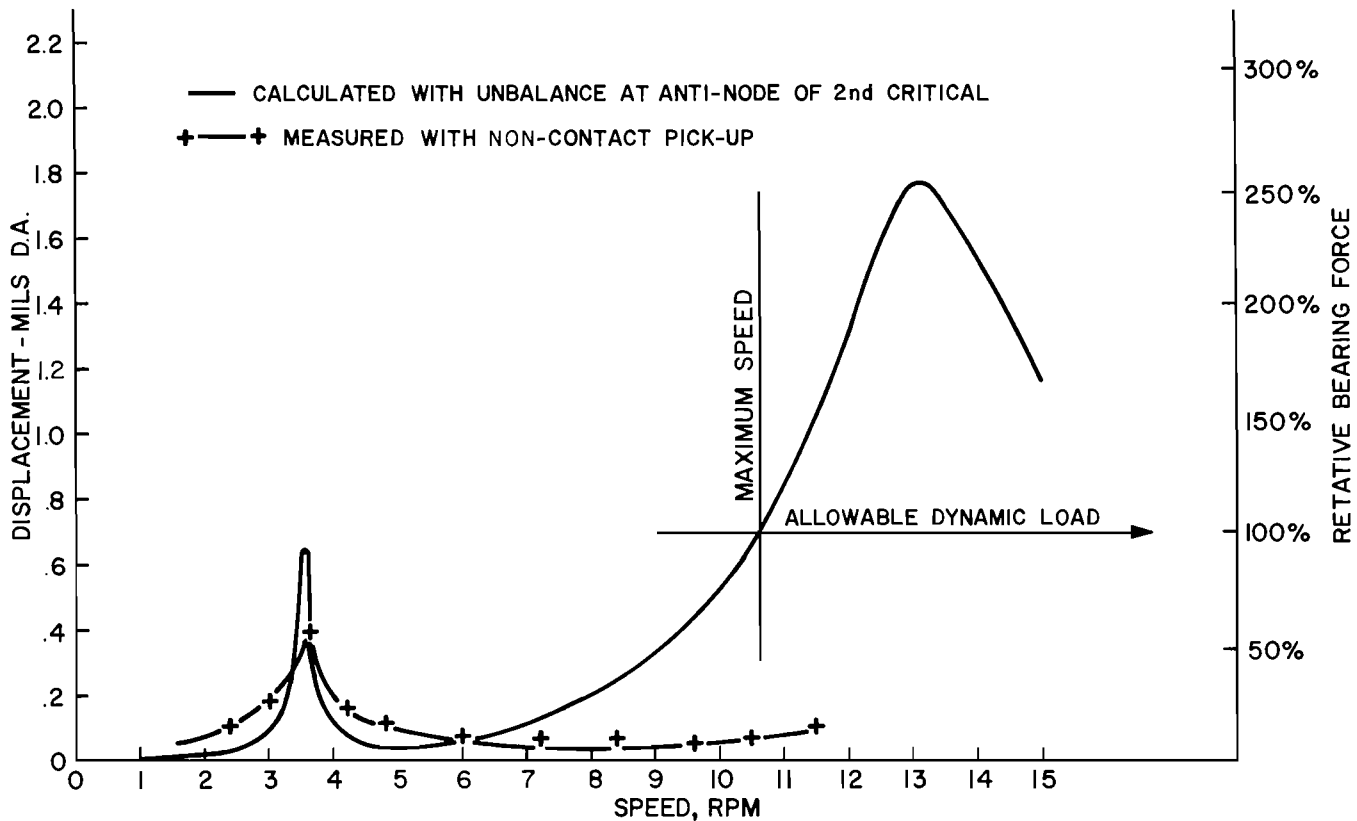


Figure 6. Rotor Response Curves for Modified Eight Stage Rotor.

OFF-DESIGN OPERATION

Most compressor users take trouble-free mechanical operation for granted, but they are concerned with compressor performance. Figure 8 shows a typical compressor performance curve. Uncomplicated and trouble-free operation can be expected in the stable performance region to the right of the surge line. Surging, or unstable operation, can occur in any centrifugal compressor when the inlet flow is reduced to approximately 60% of the design inlet flow or lower. Compressors that produce large pressure ratios, ratio of inlet pressure to discharge pressure tend to have more violent surges. When the compressor is operated repeatedly or for prolonged periods of time in surge the pressure forces can damage the internals of the machine. For those applications where frequent surge operation can be expected the compressor internals should be made of steel, instead of the more common cast iron material. Figure 9 shows the details of a five-stage compressor which has wheels, diffusers and return vanes all made of steel plate material designed to withstand surging. When the compressor is operated in surge continuously it absorbs approximately 40% of the rated horsepower, however the flow thru-put is greatly reduced and under some conditions stops completely. The power required to drive the compressor in surge is therefore largely converted to heat. This causes excessive temperature build-up inside the compressor and in a matter of seconds, if the conditions

are severe enough, can melt the soft labyrinth seals which control the internal leakage. The compressor performance suffers once the seals are damaged and the machine must be opened and the seals replaced to restore it to the original condition. Excessive temperatures in a compressor having a balance drum labyrinth seal made of a soft material with a low melting temperature can melt the seal. This will upset the rotor thrust balance and overload the thrust bearing. When the thrust bearing fails the rotor will shift axially and the impellers will rub against the stationary parts causing further damage. To help avoid these operating problems the compressor can be provided with a high temperature balance drum seal made from compressed metal fibers that will withstand several times the normal operating discharge temperatures.

Another safety feature that can be employed is a high temperature switch located in the balance drum leakage pipe, as shown in Figure 10. It is wrong to locate this switch in the discharge pipe. In this location the switch does not protect the compressor since there is not sufficient discharge flow to carry the heat to the switch when the compressor is operated in surge. The balance drum leakage pipe is the correct location for this protective device. There is leakage flow in this pipe even when the compressor is being operated completely shut-off.

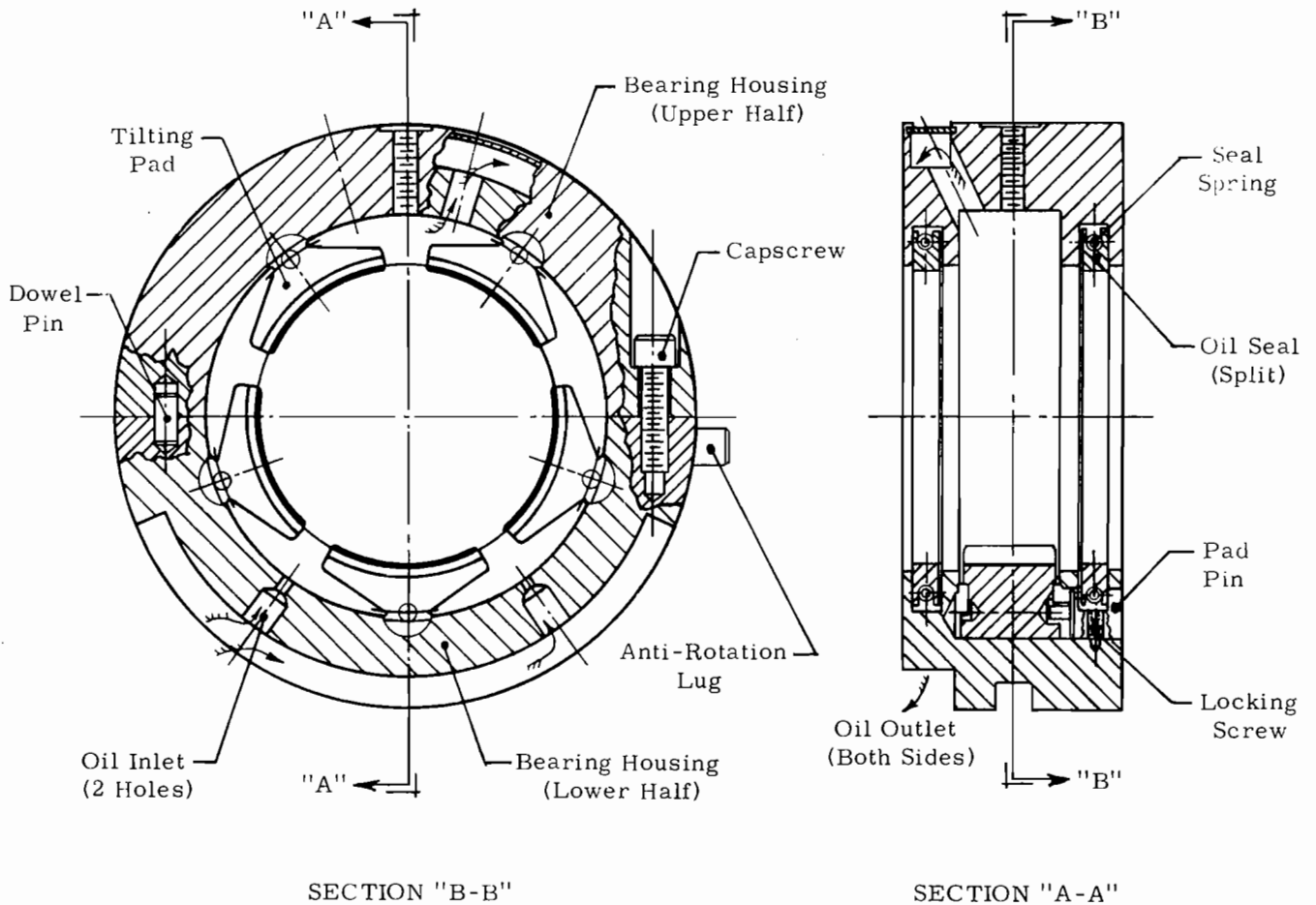


Figure 7. Tilting Pad Bearing.

NORMAL MAINTENANCE ITEMS

It is reasonable to expect trouble-free operation for periods up to two or three years between internal inspections for heavy duty commercial machinery; however some applications where the gas stream being compressed is extremely dirty and internal washing cannot be used it may be necessary to shut down in order to clean the compressor internals to restore full flow capacity.

The internal seals that prevent leakage around the impellers are normally of the labyrinth type, as shown in Figure 11A. They consist of a series of circumferential knife points that are positioned closely to the rotating impeller. In order for the compressor to maintain the design performance these knife points must not be damaged by rubbing, erosion, corrosion, or plugged-up with foreign matter. These seals are normal wearing parts and spares should be maintained in anticipation that they will need to be replaced after an extended operating period.

Labyrinth seals are used in compressors with various design features that will extend their life and make

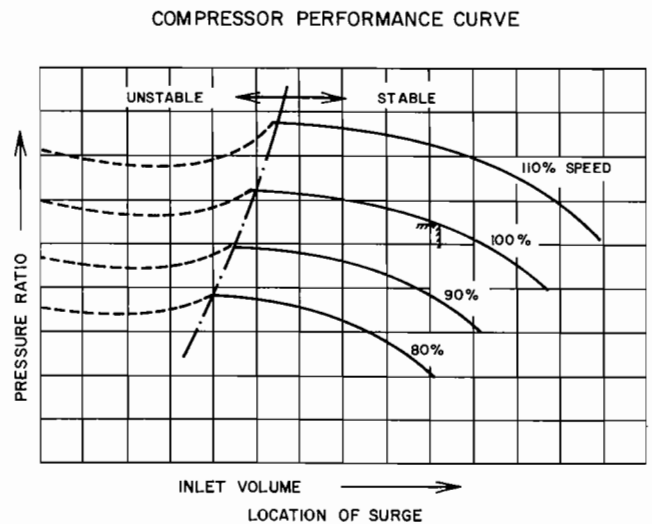


Figure 8. Compressor Performance Curve.

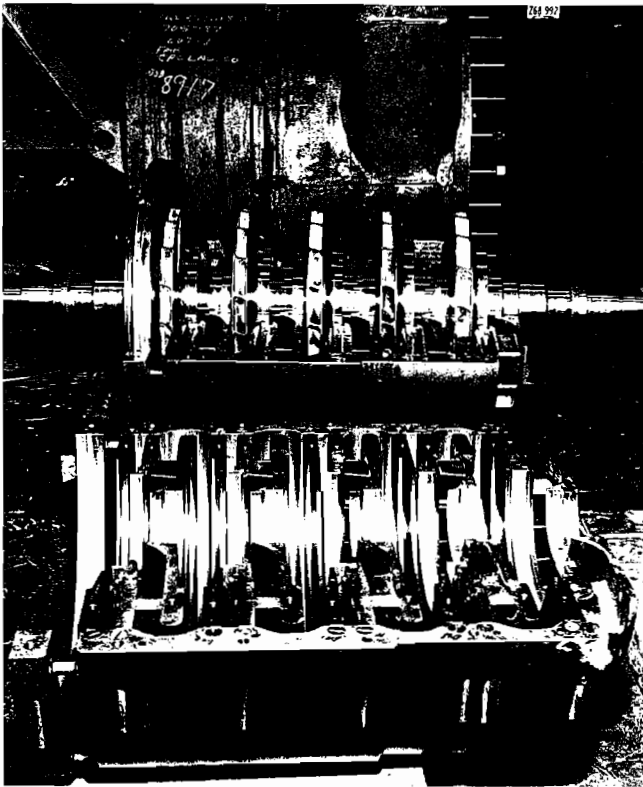


Figure 9. Steel Diaphragms Installed in the Compressor Inner Barrel.

them less acceptable to damage. One popular version is to machine the knives in the rotating part and to position a stationary sleeve of a soft material around the points to obtain a seal, as shown in Figure 11C and 11D. The clearances between the rotating and stationary parts of the seal can be reduced because it is intended that the rotating points cut grooves in the adjacent material under normal operation. This type seal allows the compressor to operate with higher efficiency due to reduced leakage, longer life because the thin knife points are made of steel and resist erosion. Also in this design considerable radial rotor motion can be tolerated without altering the effectiveness of the seal. The stationary part of these seals have been manufactured in babbitt-lined steel, aluminum, non-metallic compounds, and compressed steel fiber materials. Each has its own advantages when considering the particular operating environment of the compressor.

In many compressors labyrinths are also used for the shaft seals. Where the small leakage allowed by labyrinth shaft seals can not be tolerated, either the oil film or the mechanical contact shaft seals can be used.

Compressor shaft seals of the oil barrier type present potential maintenance and operational problems. A recent study of a large cross-section of compressor users showed that approximately 1/3 of all compressor failures and loss of production was due to malfunctioning oil film type seals. Regardless of the cause, whether it was design, operation, or maintenance it points out a par-

ticular component of the compressor that commands respect.

Oil film seals consist of basically two stationary bushings which surround the rotating shaft with a few thousands of an inch clearance, as shown in Figure 12. Seal oil is introduced between the bushings and leaks in both directions along the shaft. "O" rings in the seal housing prevent leakage around the outside of the bushings. The seal oil is maintained at some pressure higher than the gas pressure inside the compressor. The differential pressure across the inner bushing is usually only a few pounds per square inch to limit the amount of inward oil leakage. This leakage is collected in a leakage chamber that is separated from the gas stream by a labyrinth seal and is drained away through a drain trap. If the gas being compressed contaminates the seal leakage, the leakage is discarded. The outer bushing takes the total pressure drop from seal oil pressure to the atmospheric drain.

Oil film seals can cause many different types of operating and maintenance problems. The most common is excessive inward leakage due to increased clearances between the bushing and the shaft. When these clearances become large enough the leakage chamber becomes flooded and oil spills over through the labyrinth and enters the compressor. This malfunction can cause many operating problems ranging from the nuisance of having to continuously add oil to the reservoir to contaminating the main process gas. The bushing clearances can increase in time due to corrosion or erosion of the inner surfaces of the bushing, wiping due to radial shaft vibration, or dirt in the seal oil.

Another shaft seal, which is not as common in process compressors, is the mechanical contact type. This seal has the advantages of being able to maintain low inner leakage ratio with higher oil-to-gas differential pressures and therefore makes the seal oil pressure control system simpler. The seal has a spring loaded carbon face that runs against a radial face of the seal collar. The complete shaft seal either consists of two mechanical contact seals in a back to back arrangement or a combination of the mechanical contact seal on the gas side and a bushing seal on the atmospheric side, as shown in Figure 13. These mechanical contact seals have been used for many years to seal against pressure up to 1200 psi in natural gas service. They presently are being tested for pressures as high as 2500 psi differential across a single sealing face. The mechanical contact seal has an added advantage when applied to high pressure applications in that the radial sealing faces have a minimum effect on the rotor dynamics, whereas oil film bushing seals can lose their free floating feature and can upset the stability of the rotor when operating at high speeds.

High speed compression machinery must be properly balanced, especially when it is designed to operate between the 1st and 2nd lateral critical speeds. The correct method is to first dynamically balance each impeller on an arbor, and to check balance the rotor balance as each impeller is installed on the shaft. The run-out of the shaft must be watched closely as each of the impellers are shrunk on the shaft. Abnormal change in

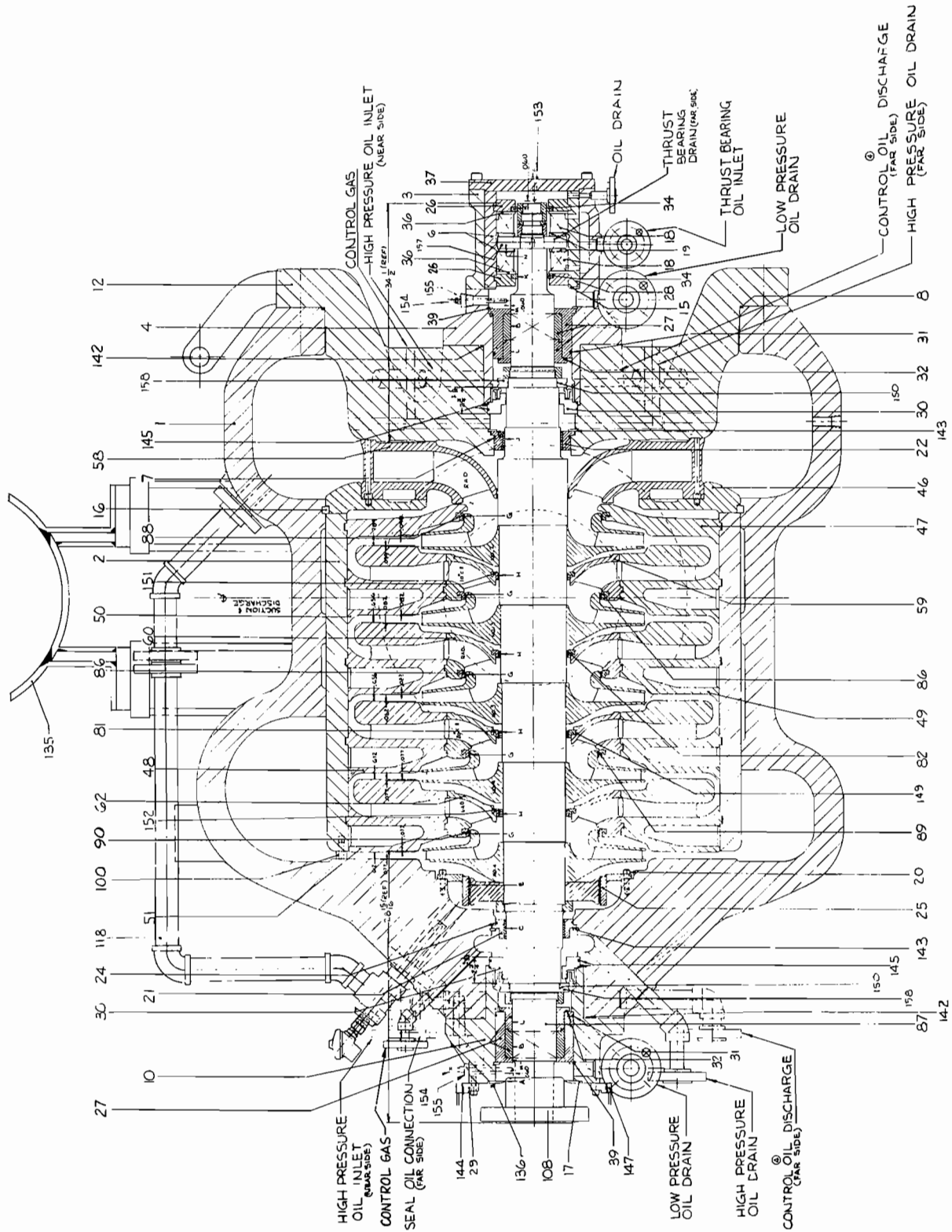
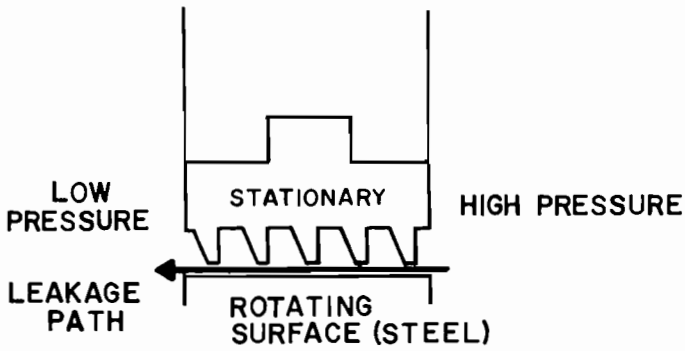
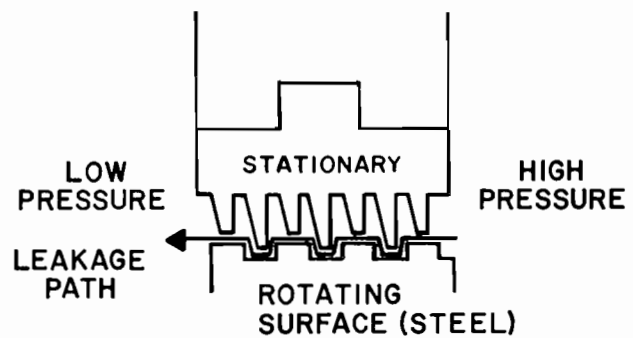


Figure 10. Temperature Switch Mounted in Balance Piping.



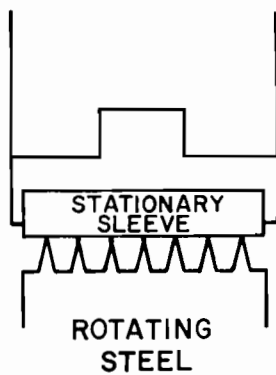
SIMPLEST DESIGN
LABYRINTH MATERIALS
ALUMINUM, BRONZE,
BABBIT, OR STEEL

A



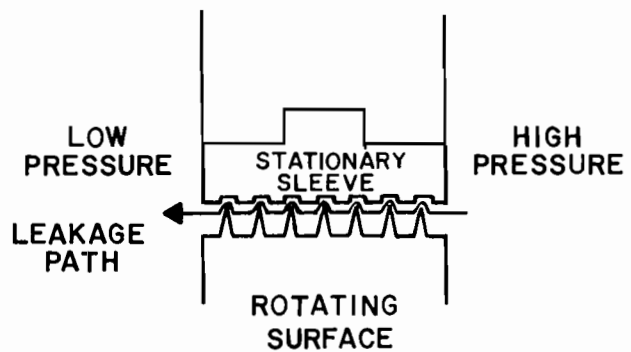
MORE DIFFICULT TO
MANUFACTURE BUT PRODUCES
A TIGHTER SEAL
SAME MATERIAL AS IIA

B



ROTATING LABYRINTH TYPE
BEFORE OPERATION
SLEEVE MATERIAL
BABBIT, ALUMINUM,
NON-METALIC OR OTHER
SOFT MATERIAL

C



ROTATING LABYRINTH AFTER
OPERATION. RADIAL AND AXIAL
MOVEMENT OF ROTOR CUTS
GROOVES IN SLEEVE MATERIAL
TO SIMULATE STAGGERED TYPE
SHOWN IN FIG. IIB

D

Figure 11. Compressor Labyrinth Seals.

shaft runout indicates that the impeller is not square on the shaft and must be adjusted before checking the balance. When the balance check, after installing an impeller on the shaft, indicates that an unbalance exists, the correction is made to the last impeller that was mounted. This procedure if strictly followed can produce a well balanced rotor even when the balancing is done in a low speed balance machine. Figure 14 shows a typical low speed balance machine. Some users rou-

tinely check the balance of complete rotors and make corrections on the first and last impellers. This practice is wrong for flexible rotors that have more than three impellers. The only sure way to balance a completely assembled high speed rotor is to use a high speed balance machine. The rotor unbalance must be checked throughout the operating speed range and any corrections that are made to the rotor must be made at the plane of unbalance. Most balance shops in this country

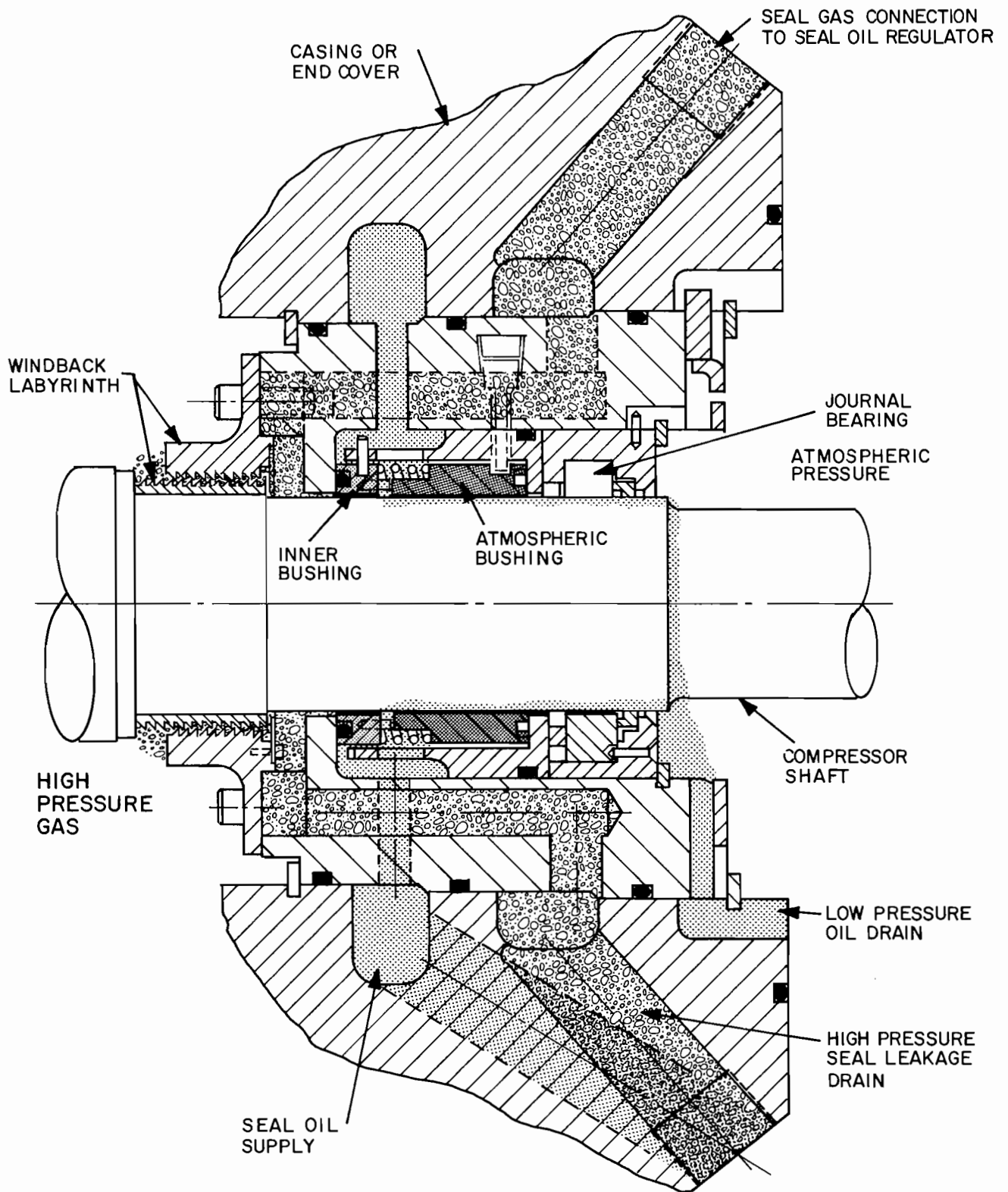


Figure 12. Bushing (Oil Film) Seal and Bearing Assembly.

MECHANICAL CONTACT SEAL WITH ATMOS BUSHING DESIGNED FOR SEPARATE LUBE & SEAL OIL SYSTEMS INCLUDING BUFFER GAS INJECTION

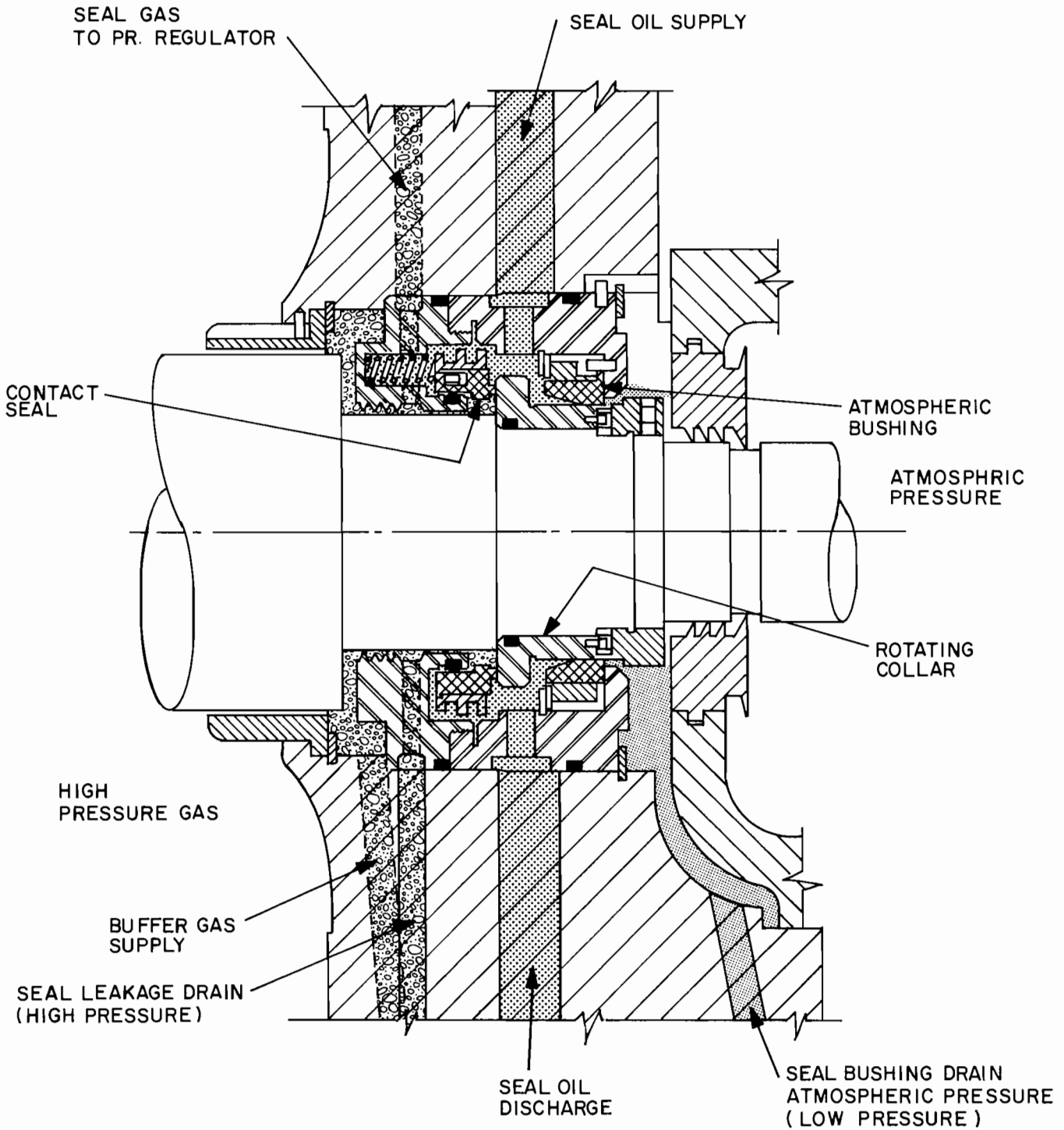


Figure 13. Mechanical Contact Seal.

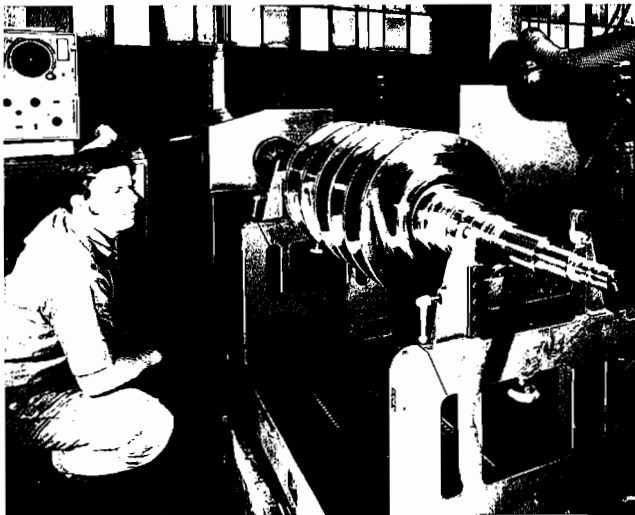


Figure 14. Low Speed Balancing Machine.

have balancing equipment that operates below 1500 rpm. The photograph in Figure 15 shows a balance machine that will balance a 1000 lb. rotor at speeds up to 7000 rpm. The machine will handle most high speed flexible compressor rotors.

Experience has shown that high speed compressor rotors can be successfully balanced in a low speed balancing machine when the progressive impeller by impeller method is followed. The compressor is normally subjected to a mechanical test in the shop before shipping. During these tests the vibration amplitudes and frequencies are measured. When the compressor vibrates excessively the rotor is rebalanced in the high speed balancing machine, however this is normally not required. When a spare rotor is purchased or repaired after the compressor has been installed the rotor should be given a high speed balance check in order to prevent delays later in the users plant.

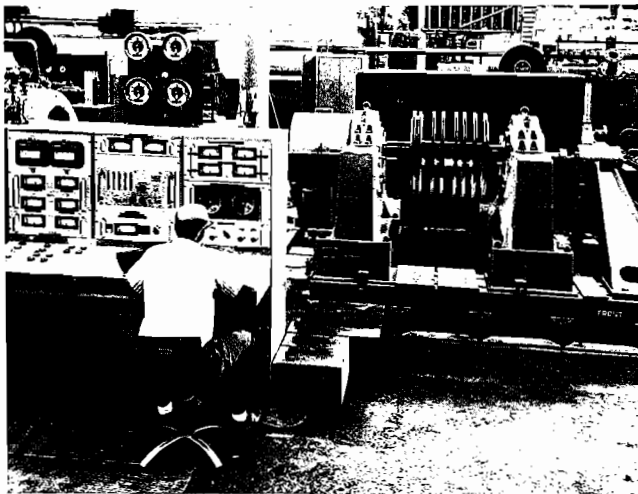


Figure 15. High Speed Balancing Machine.

The alignment between adjacent shafts ends of all rotating equipment in the compressor train is very important to the operation and maintenance of the equipment. All compressors, gears, motors, turbines, etc. have some tolerance for misalignment; however except for rotor balance, misalignment is the most frequent cause of running problems. Excessive misalignment can force the rotating equipment to vibrate and shorten the life of the bearings and gear type couplings. When misaligned shafts are rotated the gear teeth in the coupling must slide back and forth on each other. This causes bending moments and forces to be imposed upon the shaft ends. The shaft end will fail if the conditions are severe enough or the coupling hub will come loose on the shaft.

Most couplings are mounted on a tapered shaft end. It is common that a key and keyway be provided to transmit the torque through the connection. The coupling hub is pushed upon the shaft taper by a nut which also prevents it from coming off the taper during operation. This type of coupling hub mounting is satisfactory for low speed-low horsepower applications but can be a limiting factor for modern high speed machinery. A much better method is to shrink the coupling hub upon the shaft taper using hydraulic pressure and jacking tools, as shown in Figure 16. This type of mounting allows for at least twice as much torque to be transmitted through the same size shafts with the same stress levels. This is possible since the key is no longer needed and the keyway stress concentration is eliminated. The misalignment forces from the coupling teeth will not loosen the coupling hub on the shaft because it is fitted on the shaft taper with considerable interference. To mount a coupling hydraulically, the shaft end is drilled to provide an oil passage for pumping high pressure oil between the shaft taper and the coupling hub bore. A hydraulic hand pump is connected to the shaft end through an adapter. The hub is stretched with hydraulic pressure and the jack screws (Figure 16A), are used to push the coupling hub up on the taper a specified distance. The jack screws hold the hub in the correct position while the oil pressure is released. After the oil has drained the mounting plate is removed, a nut is mounted on the shaft end (Figure 16B) to protect the threads. To remove the coupling hub the procedure is reversed and the hub is "popped-off" against the tools with hydraulic pressure. This type of coupling mounting provides for an easily mounted hub that will transmit as much torque as the shaft material will allow.

Good shaft alignment must be maintained under dynamic conditions if trouble-free operation is to be achieved. To accomplish this the thermal growth of the individual elements must be taken into account when aligning compressor equipment in the cold conditions. For instance, the diameter and length of a compressor case will increase due to the heat of compression.

Normally the casing is supported at the horizontal centerline to allow the case to grow in diameter without changing the position of the shaft. The mounting feet on one end of the case are bolted and doweled to the foundation, as shown in Figure 17. The opposite end of the case must be allowed to move axially as the casing expands. This has been accomplished in the traditional

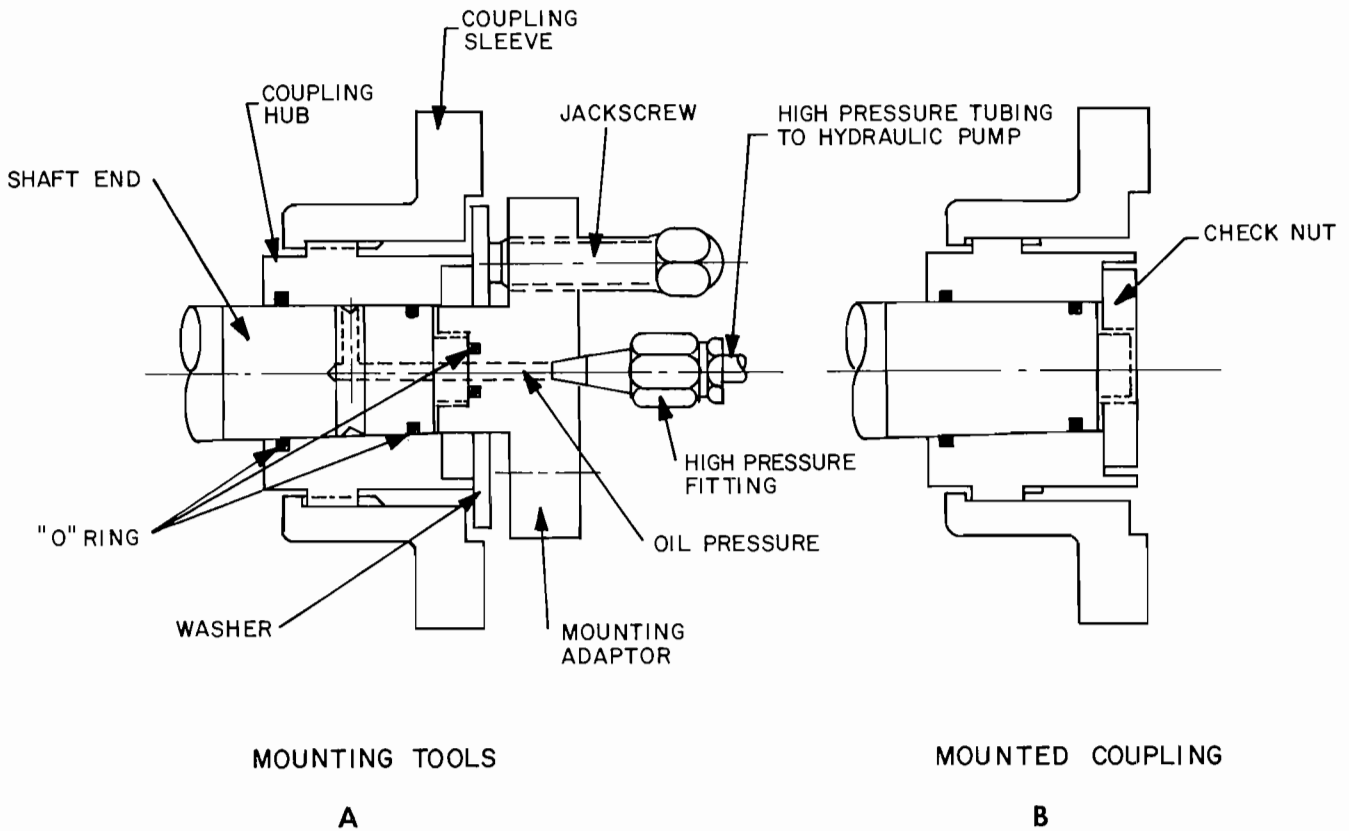


Figure 16. Hydraulically Mounted Coupling and Tools.

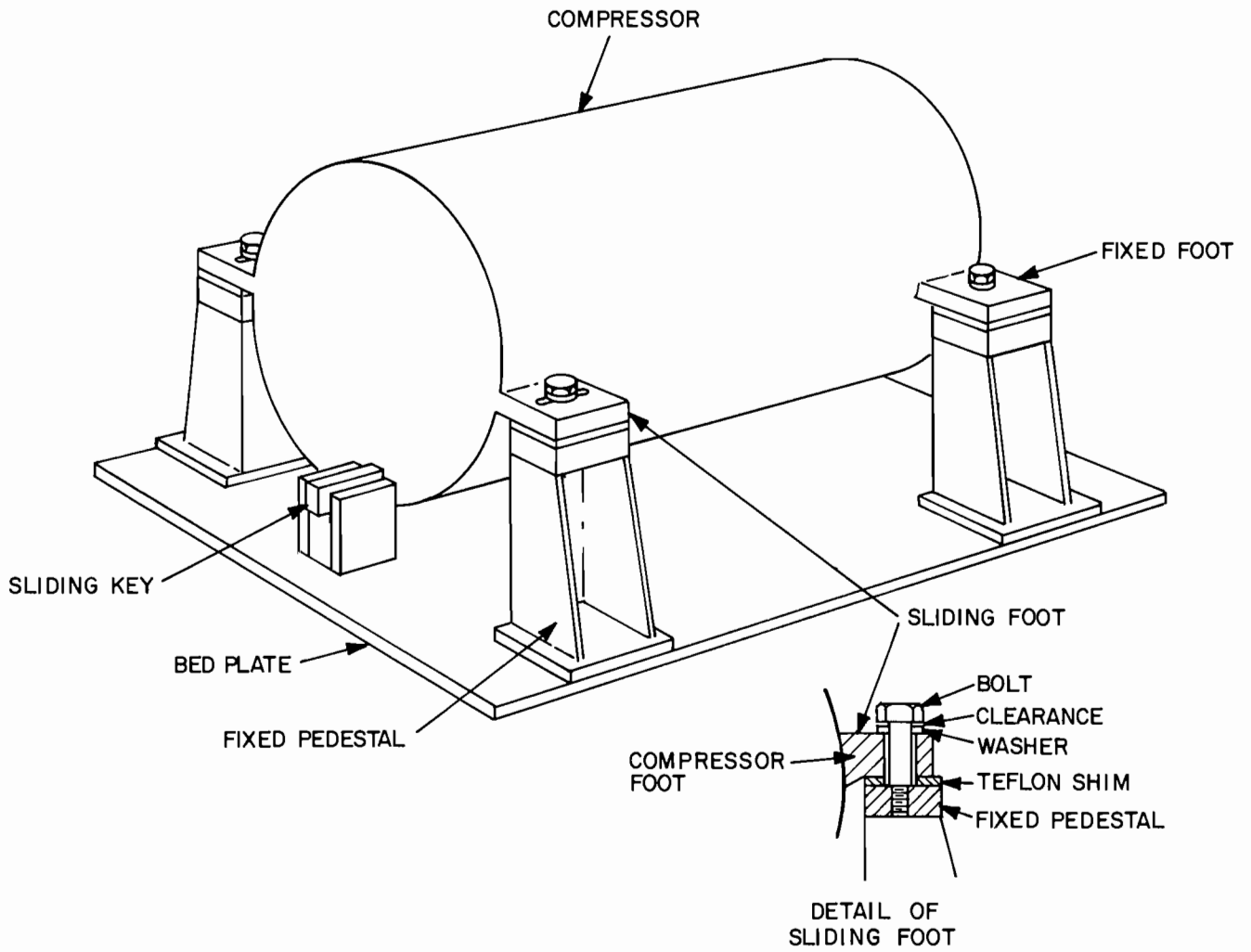


Figure 17. Compressor Case with Sliding Mounts.

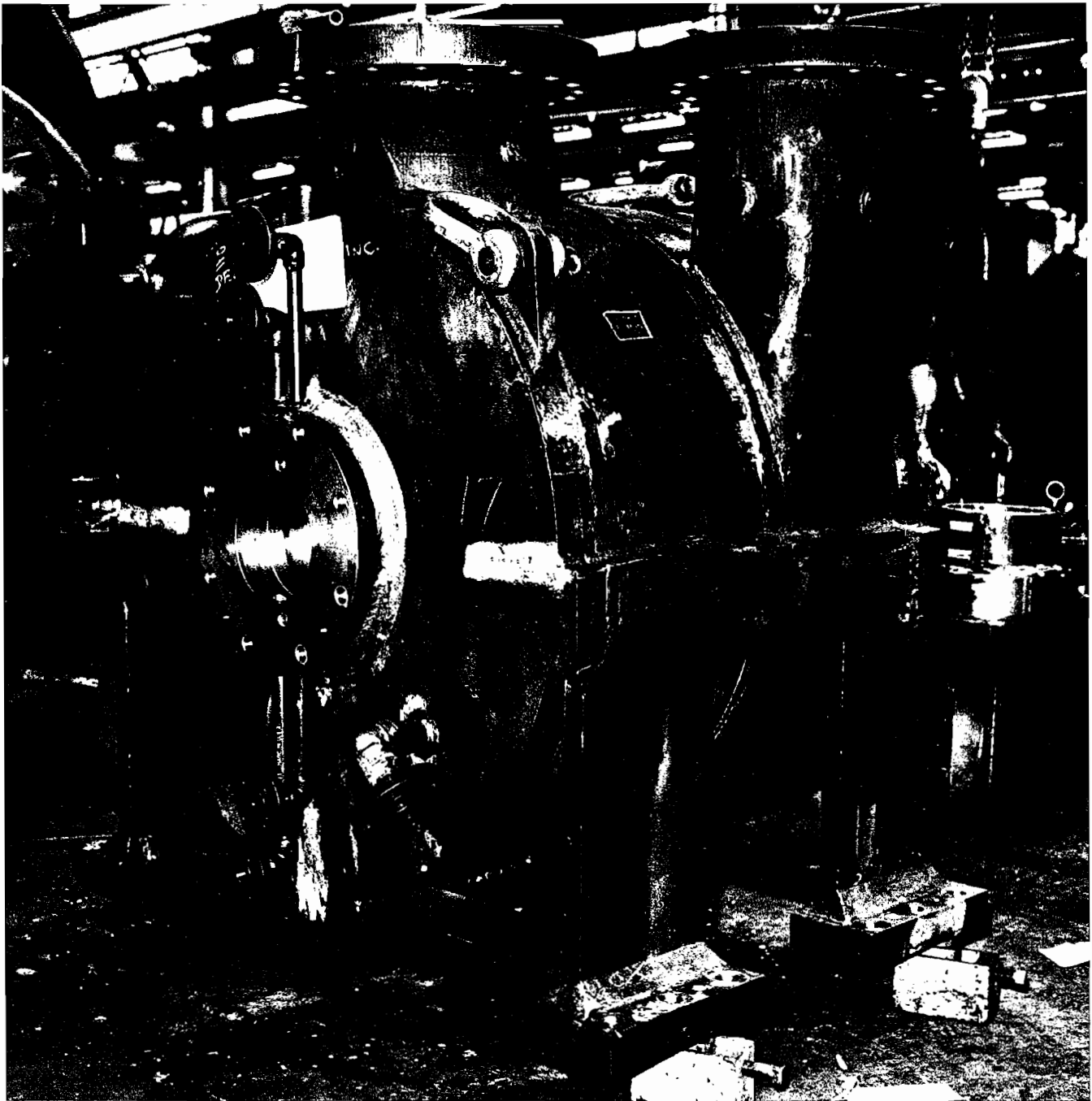


Figure 18. Compressor Case with Centerline Support.

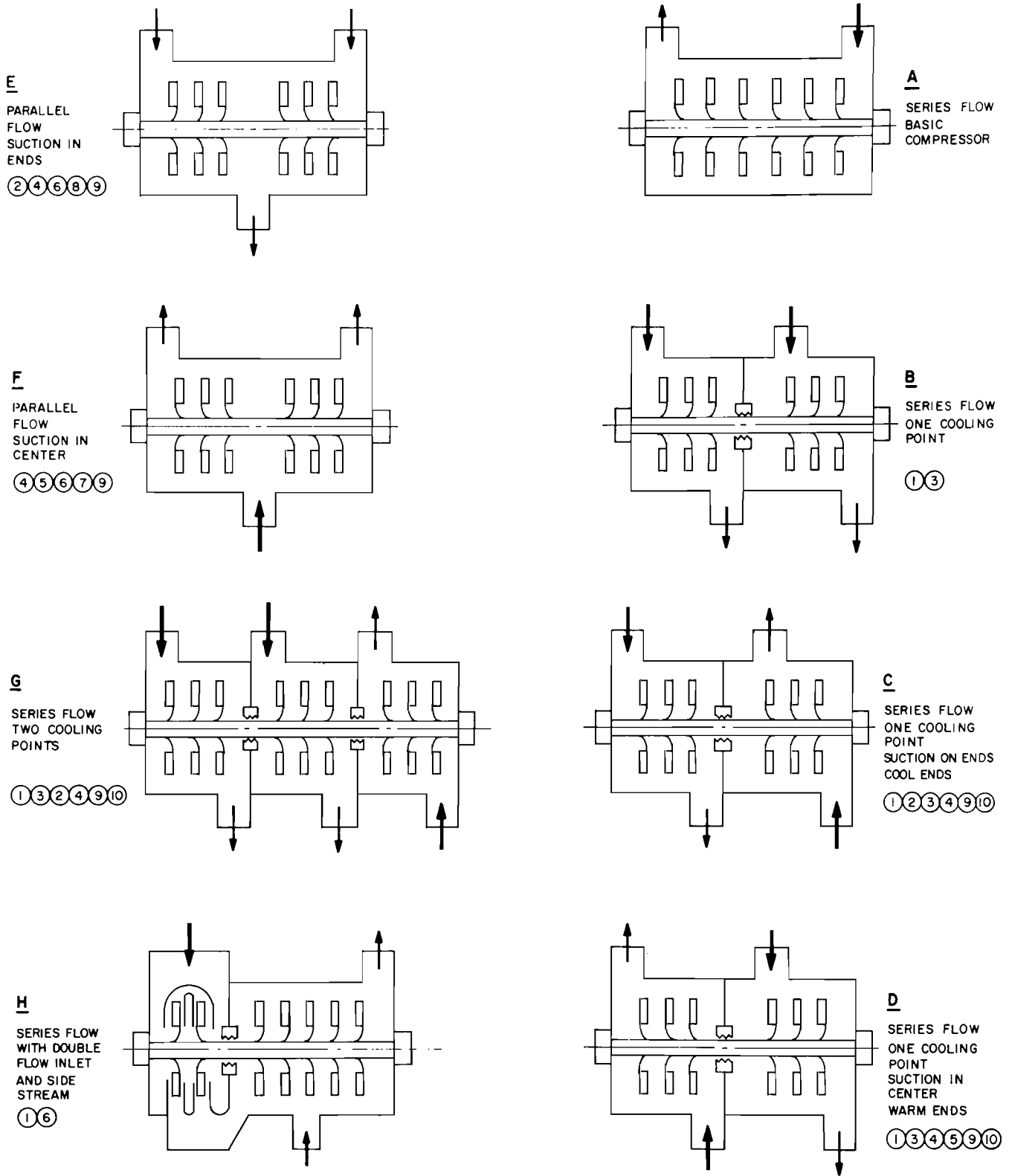


Figure 19. Eight Compressor Case Configurations.

compressor mount with holddown shoulder bolts that allow the case to slide axially upon lubricated shims but limit the vertical movement of the case. With this arrangement a vertical key and keyway are located between the case and foundation on the vertical centerline to prevent transverse movement of the case. This mounting requires a special foundation to support the vertical key and also regular lubrication of the shims.

The compressor case mounting shown in Figure 18 allows for thermal growth of the case in all directions, provides centerline support for the case, has four mounting pads that are all bolted and doweled to the foundation, and requires no maintenance. The ruggedness of this support will allow for considerable external forces to be exerted upon the casing without changing the shaft alignment.

INTERNAL CONFIGURATION

The process market places very demanding and ever changing requirements upon the selection and arrangement of the compressor internals and external casing nozzle configuration. Centrifugal compressor selections should be made with the ease of operation and maintenance aspects in mind as well as the compressing requirements. Figure 19 shows the most typical arrangements of the compressor internals and outer casing in the form of simplified diagrams.

The operation and maintenance advantages are listed for each as follows:

1. Single compressor body instead of two or more results in a simpler system.
2. Hot discharge at center of case to reduce lubrication and oil seal problems.

3. Reduced power required to compress gas results in a smaller driver.

4. Back to back impellers reduce natural rotor thrust and allows for more internal seal wear before overloading thrust bearing. Increase time between overhauls.

5. Cold inlet at center of case to reduce lubrication and oil seal problems.

6. Smaller compressor and higher speed to do the same compression job. Reduced foundation and maintenance problems.

7. Single inlet to better suit external piping arrangement.

8. Single discharge to better suit external piping arrangement.

9. No external balance piston leakage pipe. Compressor can tolerate increased balance seal wear without upsetting thrust balancing system and overloading thrust bearing.

10. Hot or cold sections of case are adjacent to reduces thermal gradients and distortion of the case. Makes alignment easier to achieve.

CONCLUSION

This paper has been presented to show that centrifugal compressors can be designed, applied, and selected with the customers operation and maintenance requirements in mind, without making any compromises in the actual compressor performance. All of the items covered in this paper would make suitable subjects for separate, full length, presentations, however the time and space available does not make this treatment possible.