Bruce Andrews

HANDS ON PUMP & MECHANICAL SEAL SCHOOL

IPS

INDUSTRIAL PIPING SYSTEMS

OCTOBER 17 & 18, 2000

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Four Points
Sheraton
Toronita St.
York, PA

COURSE INSTRUCTOR
WILLIAM H. LENTZ III

Index

A. PUMPS

B. PACK PUMPS AND VALVES

C. MECHANICAL SEAL SCHOOL

Pumps

READING PUMP CURVES

In order for a pump company to sell you a pump, they must supply you with curves of all the pumps they sell. Because the pump manufacturer or distributor must give out curves to stay in business, they are easily available. It is a good idea to keep a copy of the pump curve with the pump. Laminating a copy and hanging it on the pump will go a long way in getting everyone familiar with how the pump is supposed to run efficiently without cavitation.

Family of curves. In selecting a pump, it is useful to have the family of curves that the pump company supplies. But once you have the pump, they can be a little confusing. They show the pump operating with a range of impeller diameters. But your impeller only has one diameter. Once you know what an impeller you are using, highlight that one and then learn to ignore all other lines.

TDH in feet. The pump companies use the term. Total Developed Head to describe the height you are pumping to or the pressure difference across the pump converted to height. Along the bottom of the Gusher curve on page 2 it shows you how much will come out of the pump depending on the height you are pumping to. So for example, an 8 inch impeller starts at 71 feet. That is how much head it develops. If you were pumping to 71 feet, you would get out approximately 20 gpm. If you were pumping to 60 feet, the pump would put out approximately 400 gpm, etc.

Height versus pressure

A 100 foot column of water creates a pressure at the bottom of 43.29 psi. It does not make any difference how wide the column. Pressure depends on the height of the water above it. If a fluid weighs 50% more than an equal volume water it has a specific gravity of 1.5. It would produce a pressure 50% more than water or about 65 psi.

43.29 psi x 2.31 = 100 Total Developed Head

2.31 ft of water = 1 PSI

For water you divide the height in feet by 2.31 to find the pressure.

You multiply the pressure x 2.31 to find the height in feet.

On the Gusher 3 x 4 x 8 curve the PSI and feet are both shown. Most curves only show TDH in feet, so it is necessary to use the 2.31 factor to calculate feet to PSI or PSI to feet.

Pumps

Estimating what a centrifugal pump will do.

The tip speed depends on the RPM and the diameter of the impeller. The tip speed determines how high the pump will pump. If you know how high it will pump you can estimate pressure. By coincidence, you can approximate the height by simply taking the diameter of your impeller at its widest point and multiply that number by itself. In theory this will be 90% of the head the pump will develop. In practice you can use the number for approximating height. The pumps themselves are not scientifically accurate.
For example @ 1750 rpm:

10 inch = 100 ft height
9 inch = 81 ft. height
8 inch = 64 ft. height
7 inch = 49 ft. height
5 inch = 25 ft. height
etc..

On the Gusher 7071M hands on pump curve the highest efficiency point for a 8.25" impeller indicates 63ft. height or TDH.

When you're pumping 63 feet, and the fluid you are pumping is water, you will be pumping approximately 450 gpm.

As I previously stated, once you locate the impeller size on your pump, highlight that impeller size line. Today, we are using a 8" impeller so highlight that line and ignore the other impeller lines.

Efficiency of the pump.

On any of the curves you see the efficiency numbers start at zero and build to a maximum and then begin to drop. There is one area where the efficiency is maximum. In the design of the spiral volute centrifugal pump all forces created in the casing are equal at maximum efficiency and there is no side loading on the shaft or the impeller. As you get off that point on either side of maximum efficiency it creates radial side loading which can lead to a variety of other pump problems. The maximum efficiency point on the Gusher pump curve is 75%

Operating at maximum efficiency is not a question of saving power as it is on most other equipment. The pump should be kept at the maximum efficiency to extend bearing and seal life. As a general rule, the higher the efficiency number at maximum, the more unstable the pump becomes when you get off that point.

Most start up procedures were written for packed pumps. The start up procedure on the next page is up to date for mechanical sealed pumps.

Pumps

B. Pump Prep for Start-Up

After the pump has been installed and coupling alignment completed, the following steps should be implemented for a successful start-up.

1. Check the pump and driver for sufficient and proper lubrication.
2. Check the driver for correct rotation.
3. Open flush and barrier fluid lines, heating or cooling lines and quench vent and drain lines.
4. Pump suction valve should be fully opened. Check for leaks.
5. Vent pump case and seal chamber (open vent at top of pump casing until all air is expelled from casing).
6. If product is hot, use warm-up lines and allow ample time for pump case to heat up. Pump case and rotating assembly could distort from uneven heat transfer.
7. Before starting, rotate pump shaft by hand. It should be free, with no rubbing.
8. Open the discharge valve.
9. Once the pump is running, check mechanical seal for leaks and proper circulation. Check temperature of seal gland and flush lines. They should be no hotter than product temperature.
10. Check pump bearings and driver bearings for temperature, noise and vibration.

    If you are satisfied with the results of start-up and the pump has run for 30 minutes, it is good practice to record the vibration level of pump and driver for future reference.

C. Pump in Operation

1. During operation, a centrifugal pump requires occasional inspection.
2. Check the pump and motor bearings to see if they are overheating. Always touch the motor with the back of the hand so that any shock will push the hand away.
3. The mechanical seal should be checked for leakage, particularly during the first hours of operation. Minor leakage through the seal usually stops after a short time. If leakage continues, the pump should be stopped.

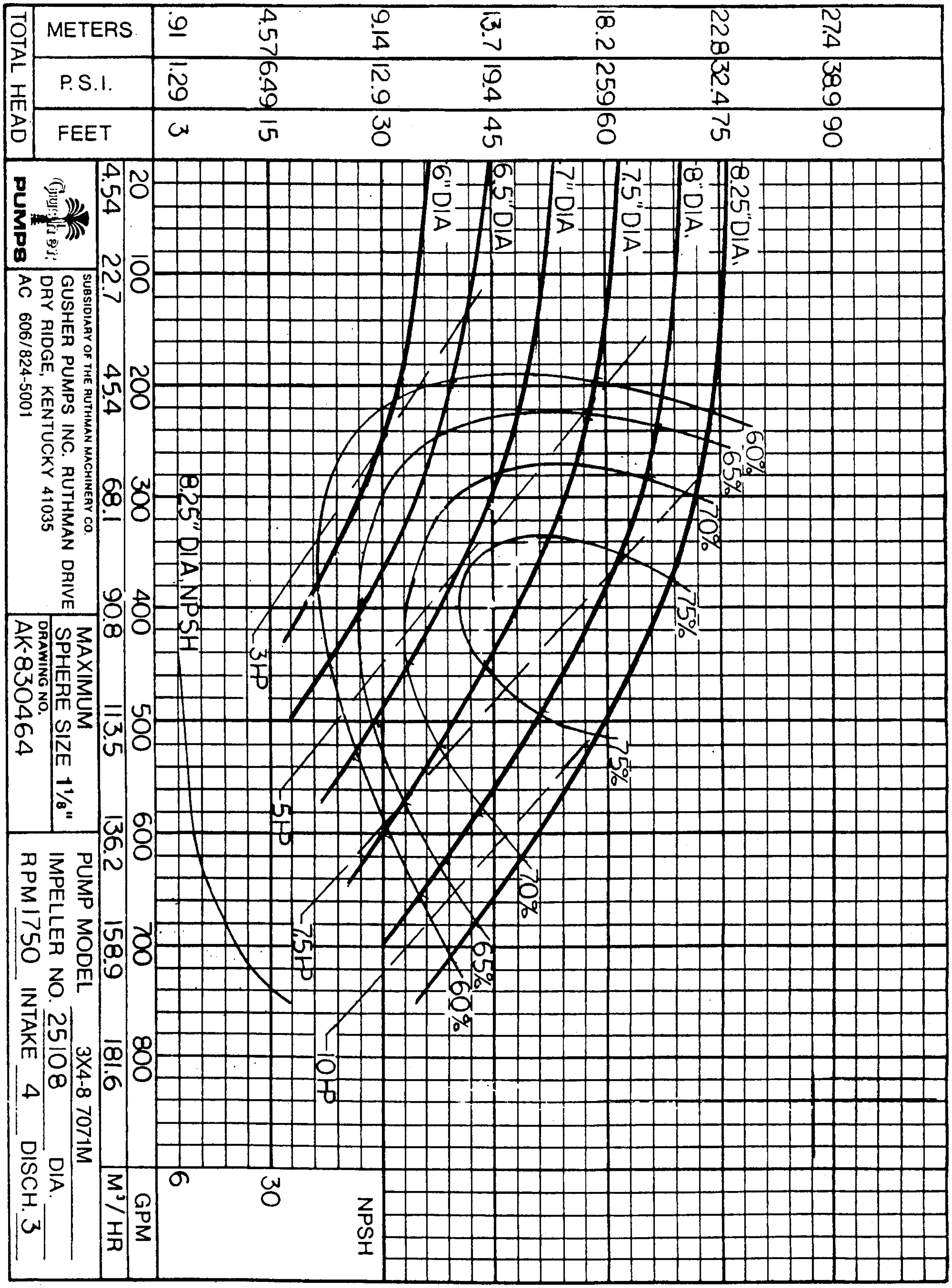
D. Trouble Checklist

1. No liquid discharge from the pump may be caused by:

   a. Pump not primmed.
   b. Speed too low - check voltage.
   c. Suction lift too high. Insufficient NPSH.
   d. Impeller or piping plugged.
   e. Wrong rotation.
   f. Air leak in suction line.
   g. Air pocket in suction line.

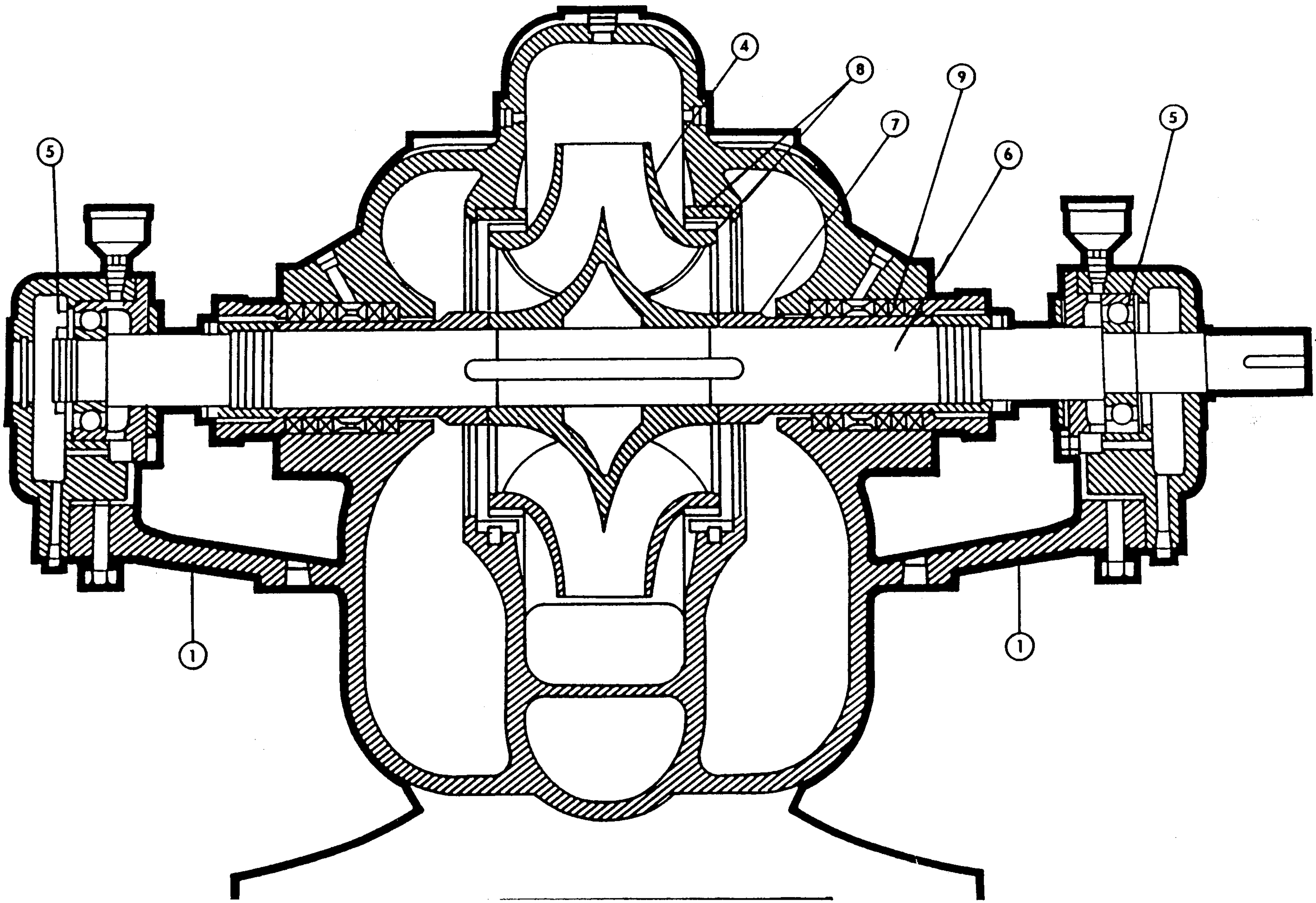
2. Insufficient liquid discharge may be caused by:

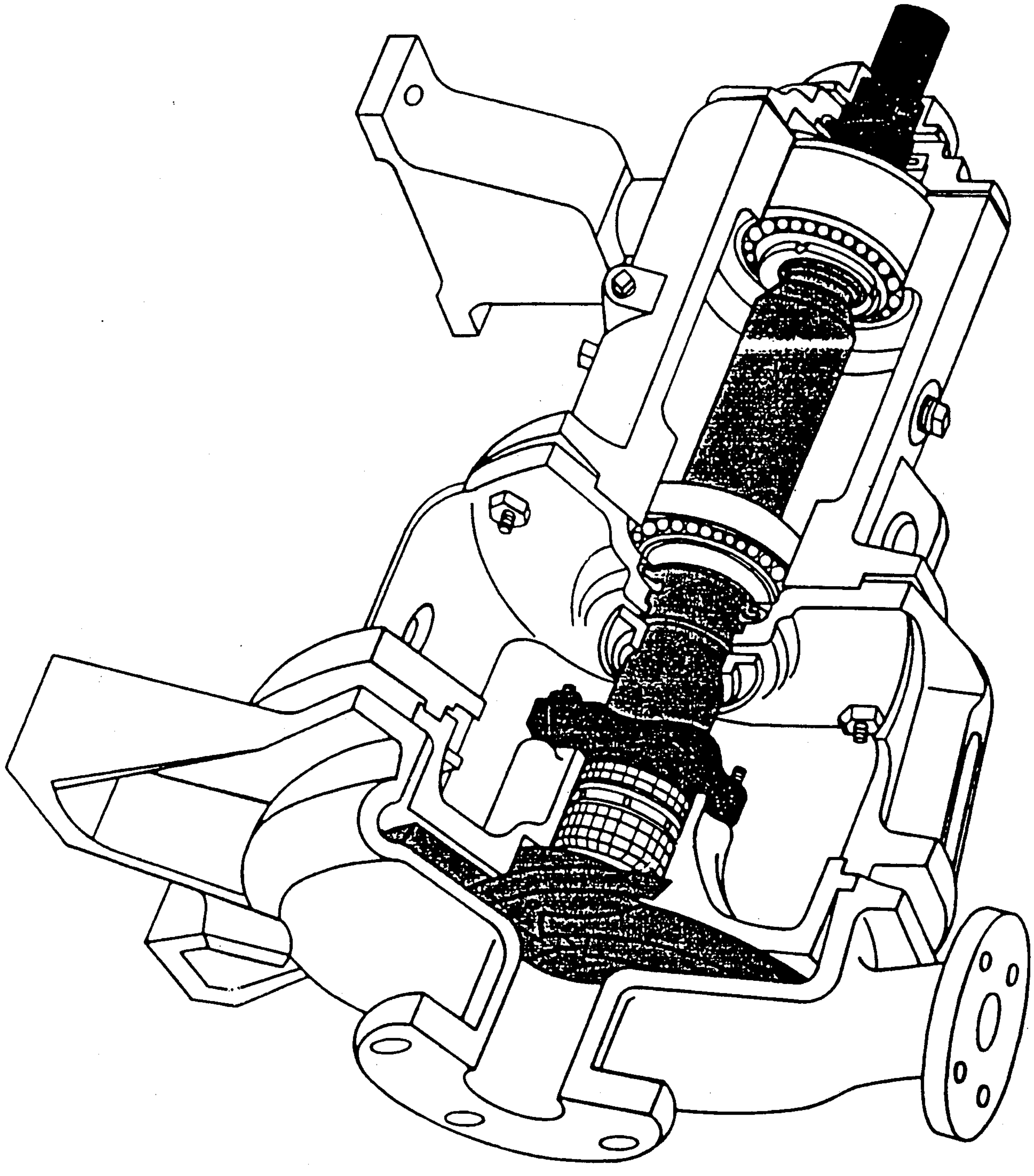
   a. Speed to low.
   b. Discharge head higher than anticipated.
   c. Suction lift too high or insufficient NPSH.
   d. Impeller or piping partially plugged.
   e. Wrong rotation.
   f. Air leaks in suction line.
   g. Air pockets in suction line.
   h. Mechanical Defects
      - Wearing rings worn
      - Impeller damaged.



TOTAL HEAD
METERS
P.S.I.
FEET
.91 4.57 9.14 13.7 18.2 22.8 27.4
1.29 6.49 12.9 19.4 25.9 32.4 38.9
3 15 30 45 60 75 90
6" DIA
6.5" DIA
7" DIA
7.5" DIA
8" DIA.
8.25" DIA.
60%
65%
70%
75%
75%
70%
65%
60%
3HP
5HP
7.5HP
10HP
8.25" DIA. NPSH
NPSH
6
30
GPM
20 100 200 300 400 500 600 700 800
M³/HR
4.54 22.7 45.4 68.1 90.8 113.5 136.2 158.9 181.6
PUMPS
SUBSIDIARY OF THE RUTHMAN MACHINERY CO.
GUSHER PUMPS INC. RUTHMAN DRIVE
DRY RIDGE, KENTUCKY 41035
AC 606/824-5001
MAXIMUM
SPHERE SIZE 1⅛"
DRAWING NO.
AK-830464
PUMP MODEL 3X4-8 7071M
IMPELLER NO. 25108 DIA.
RPM 1750 INTAKE 4 DISCH. 3

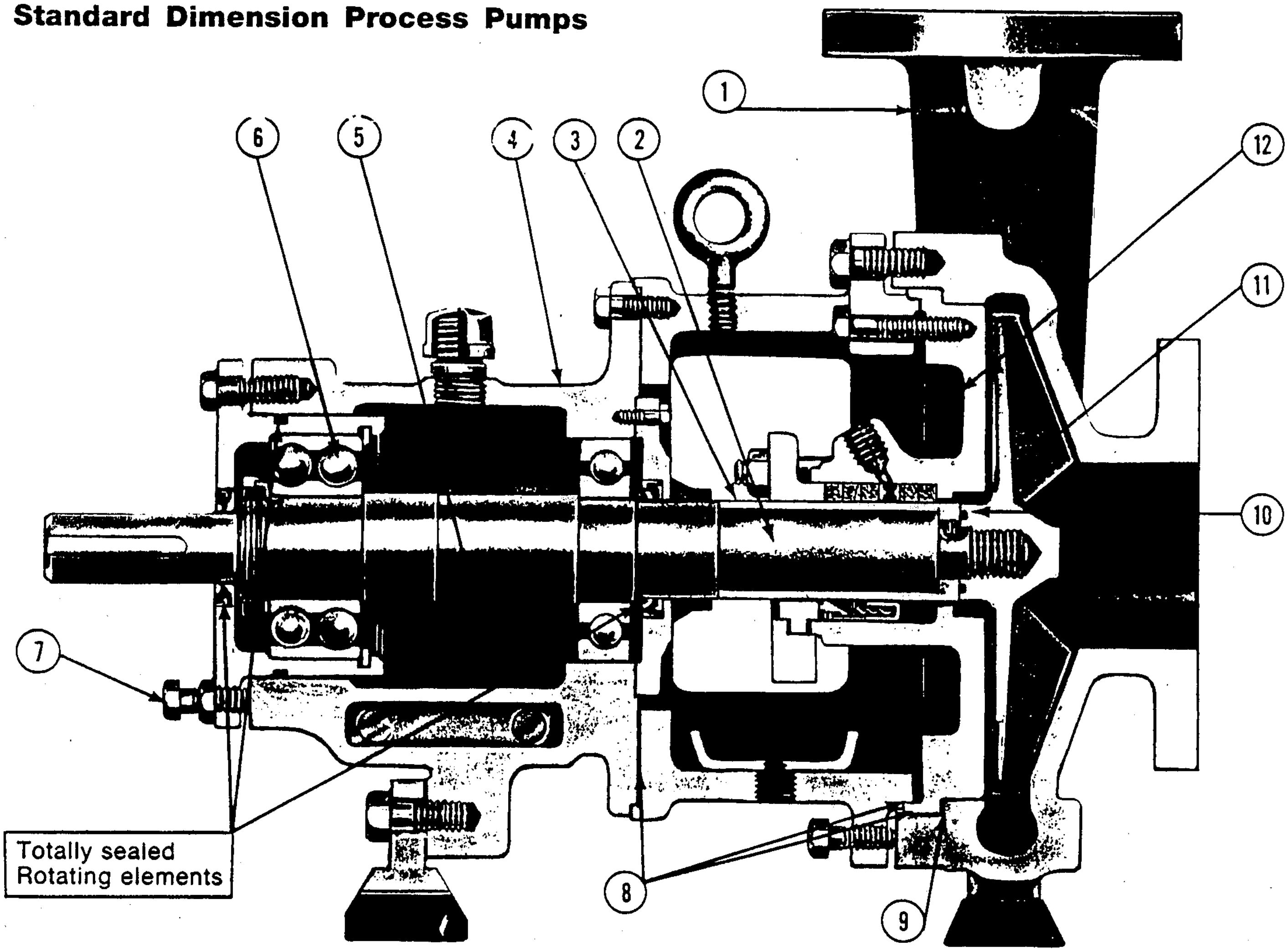
double-volute single-stage pumps





ANSI B-73.1 SPECIFICATIONS

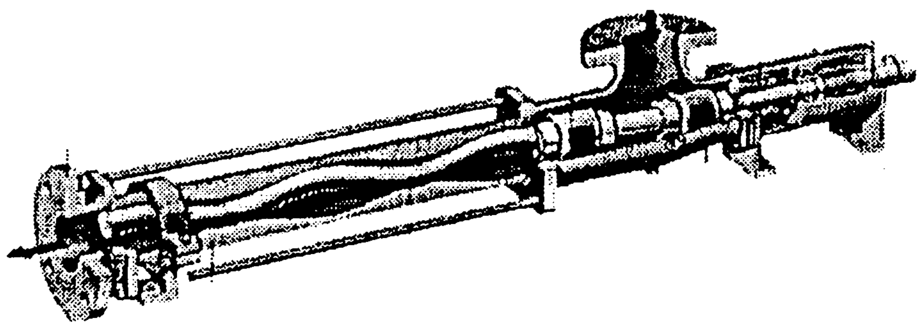
Gusher 7071 Series ANSI Standard Dimension Process Pumps



Offers many features that makes Gusher the best pump for the job:

1. **Heavy Duty Casings** — 300 psi wall thickness increases life under corrosive and erosive conditions. All sizes are self venting with ℄ discharge. Drain plug offered as option.
2. **Heavy Duty Shaft** — Minimum deflection increases life for less maintenance.
3. **Renewable Shaft Sleeve** — Hook type sleeve eases maintenance (easily removed) and is releaved to allow for changes in temperature.
4. **Power Frame** — Heavy duty Power Frame is sealed against contaminents to ensure long bearing life. Water cooled for high temperature applications.
5. **Oil Lubrication** — Oil lubrication is standard. The oil level is maintained by a constant level oiler and power frame is vented. Oil may also be water cooled for high temperature application — all power frames are jacketed as standard.
6. **Heavy Duty Thrust Bearing** — Double row thrust bearing minimizes shaft end play for more effective stuffing box sealing, longer mechanical seal life and withstands greater axial and radial thrust loads.
7. **External Impeller Adjustment** — Like new clearance can be maintained thru simple external adjustment in just minutes to maintain original high efficiencies.
8. **Rabbeted Fits** — Accurately machined rabbets ensure positive alignment, longer seal life, easy replacement of spare rotating element when maintenance is required.
9. **Contained Casing Gasket** — Protects against blow out.
10. **Positive Sealing at Impeller** — Teflon O-Ring protects shaft from liquid being pumped.
11. **Fully Open Impeller** — is designed with back pump-out vane to minimize axial thrust, and designed to handle solids and stringy material.
12. **Jacketed Stuffing Box** — Is offered for high temperature applications to increase seal or packing life in temperatures up to 500°F. (260°C).

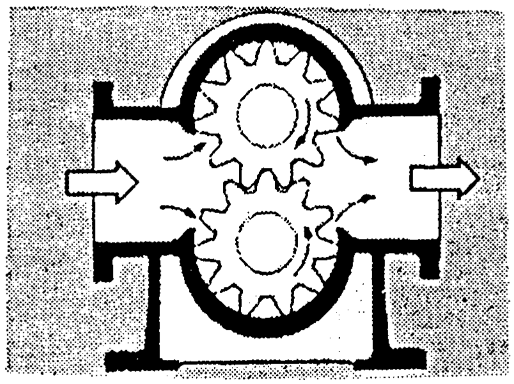
Factors to consider. Abrasive products should not be used in any close fitting metal to metal pumps. The most common PD pumps for abrasives are: The piston pump, the diaphragm pump and the Moineaux.



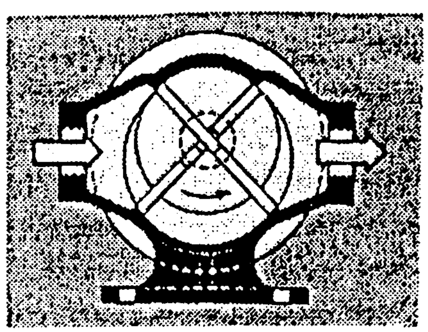
The location of the bearings is an important factor. Ball bearings and roller bearings should always be oil lubricated and so are mounted outside of the pump. Sleeve bearings are used in many PD pumps and are lubricated by the fluid being pumped. This means the fluid should be water, oil or some lubricating fluid. The Viking internal gear pumps is one of the most popular pumps for pumping viscous non lubricating fluids. But because it does have an internal bushing it is subject to wear when the fluid contains abrasives.

POSITIVE DISPLACEMENT PUMPS

Positive displacement pumps have three problems you do not find on a centrifugal. To understand these problems we have to take a look at a few typical pumps.



In the external gear pump shown, the fluid enters from the left, gets caught between the teeth of the gears and moves to the output on the outside of the gear. It can not get back to the input because there is no room between the meshing teeth. The output of this pump is controlled by the **speed** and the **size** of the pump.



In the sliding vane pump shown, the fluid enters from the left, is trapped by the vane and is moved to the right. It can not get back to the suction because there is no room. The output of this pump is controlled by the **size** and **speed** of the pump.

TYPES OF PIPING SYSTEMS

1. SYSTEMS THAT ONLY HAVE HEIGHT. In boiler feed pumps and cooling tower pumps there is often a short run of large pipe to a container with a pressure. There is no pipe loss on the way. The pump only overcomes the pressure and height and this does not vary with the flow. In plotting a system curve it is a simple straight line. The pressure or height may change over time as the tank gets full but it does not change with flow.

2. SYSTEMS THAT ONLY HAVE PIPE LOSS. In a closed loop chilled water system for example there is no static height to overcome. Once the system is full of fluid and all air has been removed the pump has only to overcome the pipe resistance. The closed system pump requires both a suction and a discharge pressure gauge. It is only the difference between the two that is important. This represents the piping resistance. If there is no flow there will be no difference. As the flow increases the pipe loss will increase exponentially.

3. SYSTEMS THAT HAVE BOTH. Most systems are a combination of pipe loss and height. With a simple system that does not contain automatic valves you may be able to get a good approximation of your system curve. You find out the static height through normal means. You then find where your pump is operating on its curve. The difference between the static height and the height pump is pumping will give you an approximate pipe loss number. You can then project out piping loss at all other flow rates. Using any of the appropriate piping charts.

OPERATING MORE THAN ONE PUMP.

PUMPS IN SERIES. When the discharge of one pump feeds the suction of the next the pumps are in series. To determine the combined curve you find the curve for each pump. You then add the heads of each pump at similar capacities. Putting pumps in series has an effect on the thrust of the pump and the sealing
problem and so it is not always a simple decision. As a general rule the pumps should have similar width impellers. The impeller width determines the gallons per minute flow rate. When pumps are in series they will have the same gallon per minute flowing through each stage. The diameter of the impeller determines the pressure or head so these can vary.

PUMPS IN PARALLEL. When the pumps are fed by the same suction line and they discharge to the same line the pumps are parallel. You add the capacities of each pump at similar heads to find the combined curve. As a general rule when pumps are operated in parallel both the diameter and speed of the impellers should be the same.

SYSTEM PROBLEMS VERSUS PUMP PROBLEMS

You have a pump problem if the pump is not putting out as much as the curve says it is supposed to. If the pump is doing what it is supposed to and you are not getting the flow you desire then you probably have a system problem. Often the pump is blamed for system problems. A common example is pumps in parallel feeding a system with pipe resistance. When the first pump is running it develops a certain amount of flow and piping resistance. When the second pump is brought on the piping resistance is increased exponentially while the flow is trying to increase linearly. That is, you get less flow than you would predict. Often a third of fourth pump will not increase the flow at all.

Cavitation

SUCTION SIDE OF THE PUMP

If the fluid level in an open tank is above the suction of the pump the fluid is pushed into the pump by gravity and atmospheric pressure. If the fluid is even with or below the level of the suction of the pump only the air pressure can push the fluid into the pump. At sea level atmospheric pressure equals 14.7 psia. This is equivalent to 33.9 feet of water. So if the fluid level is below your pump, atmospheric pressure can only push it up 33.9 feet. In practice the lift may be considerably less. In order for air pressure to push a fluid into the pump the pressure in the pump must be reduced below 14.7 psia. The pressure a gas exerts on its container depends on how many molecules of gas occupy the space and the temperature of the gas. In many positive displacement pumps air can be pumped as well as fluid. As the air is moved from the inlet to the outlet it lowers the pressure at the inlet. If the suction of the pump is below waterline atmospheric pressure will then push fluid into the pump. The pump is called self priming if it will move air. Most centrifugal pumps can not move air too well. The clearances are too large and they are not going fast enough. Often there is no place for the pump to discharge the air. To get a centrifugal pump to pump out of a well the air must first be removed. This is done by priming the pump. One way to prime a pump is to fill the pump and suction full of water removing all air. The other way is to pump the air out through a float valve. As the air is pumped out fluid is pushed into the pump if the suction is below the waterline. Once the pump is full of fluid and turned on it pushes the fluid out. This leaves a space where the fluid used to be. This is a region of low pressure. Atmospheric pressure will then push fluid to that point. The pump can only move air effectively if it has someplace close to the outlet to vent off the air and if the fluid can not get out before the air. In self priming pumps they sometimes use a large separating chamber on the top of the pump where the fluid keeps going back to the pump until all air is removed. It is important to have a quick exit for the air so it does not get trapped in the elbows and then get pumped right back into the pump. Sometimes a bleed line from the discharge back to the supply tank is necessary.

VAPOR PRESSURE AND BOILING POINT

The molecules of any fluid are in constant motion and trying to escape from the fluid into the atmosphere. The temperature of the fluid determines the pressure of the vapor it produces. This vapor forms a blanket in close contact with the fluid. If the atmospheric pressure is higher than the vapor pressure, the molecules are constantly returned to the fluid and little evaporation takes place. Vapor pressure charts shows the vapor pressure of various fluids at different temperatures. If the pressure on a fluid in a tank drops below the vapor pressure of the fluid, the fluid will boil.. When fluids boil they increase in volume substantially. This boiling forms cavities or bubbles where the fluid used to be. The bubbles absorb the heat of vaporization. If the pressure on the fluid entering the pump drops below its vapor pressure this forms bubble or cavities which reduce the output of the pump. As the bubbles move to the discharge the pressure increases. When the bubbles or cavities implode a high velocity bullet of incompressible fluid hits the impeller and casing. At the same time it is hitting the impeller the fluid is releasing the heat physical damage on some types of impellers and casings. The Higher the specific heat of the fluid the more damage it is likely to cause.

NPSH (required)

The pump curves contains the *Net Positive Suction Head Required.* Th term is called *net* because that means what should be left after the vapor pressure has been subtracted. It is called *positive* because only absolute terms are used. If the pump is cavitating, it is because the head available to the pump is less than the head required. You can determine the NPSH required by looking at the pump curves. You will notice that it increases with flow, so cavitation may be stopped by reducing flow in some cases. This is often done by throttling the discharge. We will see further on, that restricting the flow of the pump can often lead to excessive bearing loads, and bending of the shaft, with consequent seal or packing problems.

NPSH (available)

The head available to a pump should always be more than is required. It consists of: (a) the static height of the fluid level above the centerline of the pump, and (b) the pressure head. Pump companies state that on an open tank, this would be 33.9 feet. On a pressurized tank, if the pressure gauge reads PSIG, it would be that pressure converted to head plus 33.9 feet. If it is PSIA (absolute pressure), it would be multiplied by 2.31 to find the pressure head. But you must subtract the losses in the piping and fitting previously determined. And, you must subtract the vapor pressure of the fluid coming in. It is when you drop below the vpor pressure the pump will cavitate.

Increase the NPSH available by:

1. Increase the pressure head.

   - Pressurize the tank.
   - Increase the pressure in a pressurized tank by external means. (NOTE: In high vapor pressure fluids (refrigerants, liquefied gasses, etc.) the pressure in the tank comes from the vapor pressure of the fluid. By heating the tank, the pressure gauge reading will increase, but this will not be of any help because the vapor pressure has increased, and this figure is subtracted from the pressure head.

2. Increasing the static head.

   - Lower the pump.
   - Raise the level in the tank.
   - Raise the tank.

3. Reduce the piping and fitting loss.

   - Larger Pipe
   - Bell mouth inlets
   - Larger strainers
   - Removing restrictions, etc.

(NOTE: In troubleshooting cavitation that started in a system where there previously was none, this is the usual area to suspect. Common causes include:

- Blocked sumps
- Clogged strainers
- Buildup within the lines
- Clogged foot values.)

4. Reduce the vapor pressure of the fluid

Cavitation

Vapor pressure of a fluid depends on the fluid and its temperature. Lowering the temperature lowers the vapor pressure. *(Note: Pumps that are allowed to recirculate from discharge back to suction often have this type of problem. Recirculation builds up heat, which increases the vapor pressure.)*

AIR OF VAPOR BLOCKAGE

Cavitation which does not seem to be caused by any of the four NPSH elements is often the most difficult to find. Air is either being sucked into your system or it is being dropped in or vapors are being trapped in some section of the system and acting like a restriction.

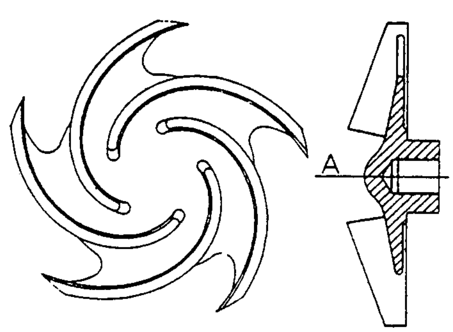
Air in the System

Air in the suction side of your pump will cause a noise, which can be annoying but does not carry with it, the same type of destruction caused by cavitation. When an air bubble enters the pump, it will expand based on the law of gasses (if the pressure, twice the volume). This is nowhere near the type expansions encountered in boiling. The air occupies space and therefore reduces the output of your pump and its efficiency. Sometimes air accompanies classic cavitation, i.e., a cracking noise accompanied by severe pump vibration and internal damage. This happens when the air or vapor bubble gets trapped in the eye of the pump. Because air is lighter, it has a tendency to stay at the eye, while the fluid gets pushed to the outside. This air restricts the input to the pump, causing a lowering of the NPSH available. If the pump is taking suction from a tank near its own level or below it, the air could cause the pump to lose its prime and stop pumping altogether.

Air tends to go to the lowest pressure through the easiest path. The lowest pressure is at the eye of your impeller and the easiest path is along the shaft from the outside of the pump.

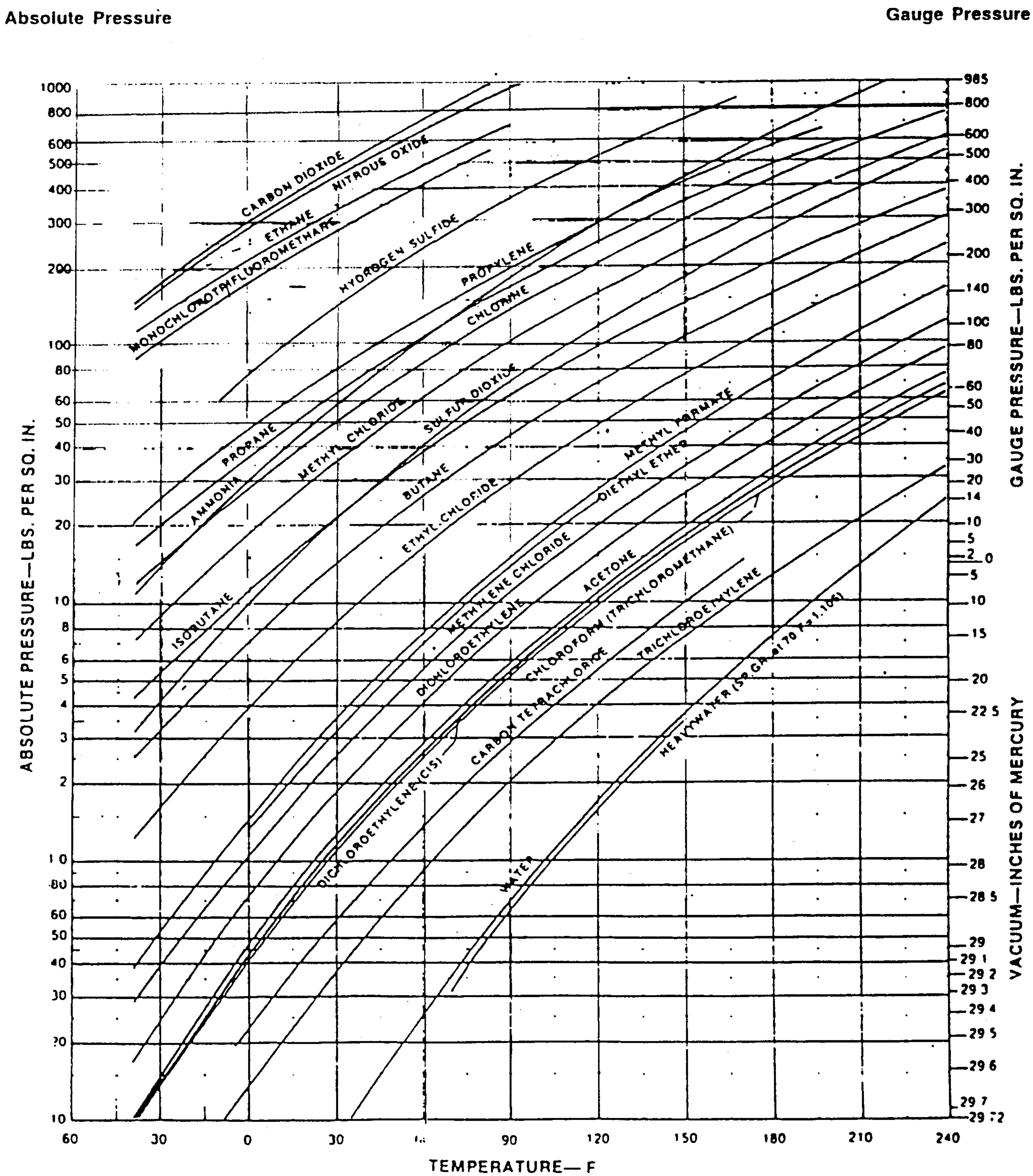
1. Check the shaft. In a packed pump there should be visible leakage. If not spray some shaving soap at the gland area and see if it gets sucked in. Or you can stop the pump and pack some grease around the leak path and see if the problem changes temporarily. Packed pumps should have a spacer or lantern ring as the middle ring of packing to solve this problem. You bring in a fluid or grease until it leaks out or the problem stops.

Mechanically sealed pumps should not have any visible leakage. They can be checked with shaving soap or grease. You should not put a water hose on the packing or seal to check for leakage. Though this will block the leakage temporarily it can break the mechanical seal and contaminate your bearings. If your seal is allowing air to enter it should be pressurized with a line from the discharge. When you replace the seal ensure that it can hold a vacuum the next time. The rubber bellows water pump seal properly installed will hold a vacuum.



SOURCES OF AIR

VAPOR PRESSURE OF VARIOUS FLUIDS



Cavitation

FOR SUCTION SYSTEMS UNDER VACUUM

2. **Check the suction valve packing.** It can be checked by placing grease, or water, or shaving soap around the valve stem. If leaking, repack the valve. Cut the rings correctly and put a plastic ring between any of the two braids.

3. **Check the gasketed joints.** Joints at flanges, valves or strainers can be checked by wrapping them with Saran wrap or some safe similar pliable material. Listen for a change in the noise.

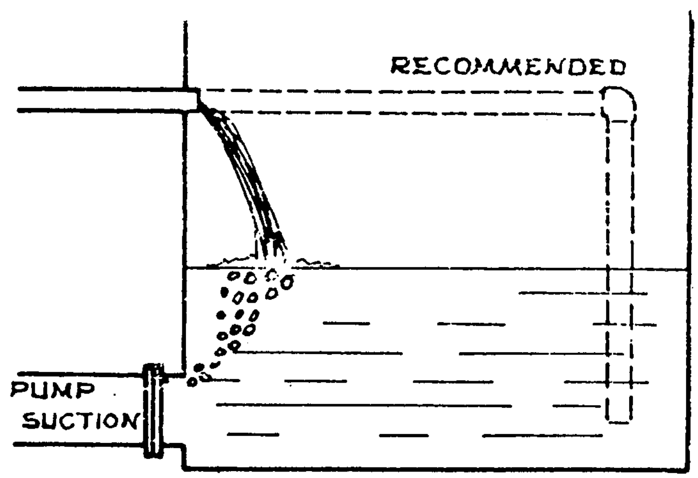
4. **Check the suction piping.** The air can be coming in anywhere between the fluid level and the pump if the tank is below or even with the pump. The lines can be checked with saran wrap or tape. If possible you would pressurize the suction pipe with a dye in the fluid to see where it leaks out. This requires a foot valve as well as a place to bring in the pressure. The suction pipe must be visible.

- Go to the following procedure if you have not found the problem.

CHECK THE SUPPLY TANK

- **Look for a waterfall.** Water plunging into the tank is the most common cause of cavitation. This is found by raising floats or raising the level of the fluid. It is solved by diverting the fluid, moving the suction away from the waterfall, by putting baffles between the supply and suction etc.

- **Look for a Vortex or whirlpool.** Whirlpools can introduce large volumes of air into your system. They are broken by interfering withthem. Placing 4x4 lumber on the surface of a pond is a common technique. Baffles that look like ice cube trays, flow straighteners and anything that interferes with the surface energy patterns have all been effective.



SUMMARY

- **Find out what is good. Whatever is left is bad.**

1. Pressure head
2. Height
3. Pipe Loss
4. Vapor Pressure.

- **If you have ruled out all 4 items look for air:**

1. Along the shaft
2. On the Suction Valve
3. On the suction pipe
4. Look for a waterfall into your supply tank
5. Look for a vortex in the tank

- **Suspect a vortex or trapped air in your system.**

CURING THE PROBLEM:

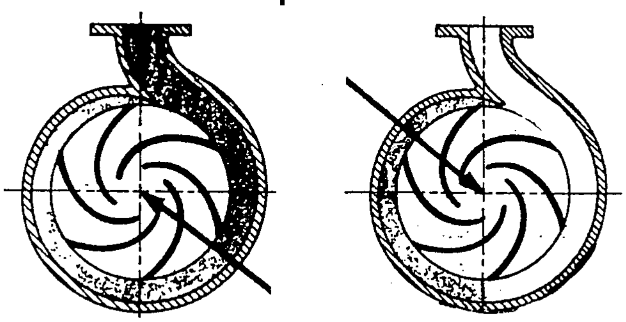
1. Increase the pressure head
2. increase the static height
3. Reduce the piping loss
4. Reduce the vapor pressure (cooling)

Cold Water pump cavitation can usually be blamed on a blocked suction or air. If you have a suction pressure gauge it will tell you if the suction is blocked.

Troubleshooting

SHAFT DEFLECTION

With the spiral shaped single volute type pump, forces act on the shaft when the pump is not operating at its most effective point. To understand the following explanation, you should look at the view of the impeller below:



The fluid has a high velocity at the impeller's tip. At the fluid moves towards the casting, the velocity is reduced as it is converted to pressure against the walls of the casting. The casting is gradually expanding in areas as it goes towards the output, so the pressure is being developed across a gradually expanding area. The force
acts on the rotating impeller and shaft, and puts a significant load on the bearings of he pump. If the design is poor, or the clearances are too tight, this will lead to rubbing of the pump parts. The rubbing will be in only one section of the stationary part of the pump, while the rotating part may be rubbed all around. Quite often the rubbing leads to high heat which can cause a variety of problems, including corrosion, seal failure, coupling failures, and excessive power consumption. The bending of the shaft and the loads on it will shorten the life of your seals, because it causes them to move. It can severely shorten the life o packing, and it determines the life of the bearing. The centrifugal pump is a very simple device.

TROUBLESHOOTING RUBBED PARTS

CAUSE	SYMPTOM ON ROTATING UNIT	SYMPTOM ON STATIONARY UNIT
Shaft Deflection	All Around	One Section
Pipe Strain	All Around	One Section
Shaft/Pump Not Concentric	All Around	One Section
Bent or Bowed Shaft	One Section	All Around
Impeller Unbalanced	One Section	All Around
Wrong Clearances/	All Around	All Around
Bearing Failure	All Around	All Around

PUMP TROUBLESHOOTING

In most cases, it is no more than a spinning disc of cast iron in a casing. So there are not too many things to look for when you take a pump apart. But you should be sure to look. Many of the pump problems will show up on the mechanical seal. Because this is one of the most likely areas of pump failure, the method used to troubleshoot seals is described in detail in this book.

Look For:

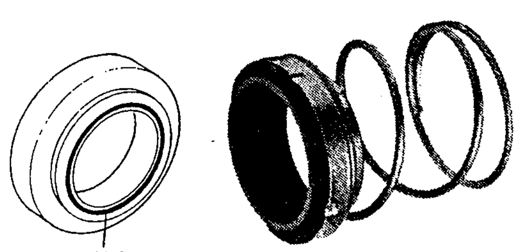
- Parts which have been rubbed
- localized erosion or polishing
- localized corrosion
- signs of heat

Look At

- the wear rings
- the bottom of the stuffing box
- the volute, or casting
- the impeller
- the shaft (especially where it goes through the bottom of he stuffing box, the packing, and the seal gland

Look At:
THE WEAR TRACK OF SEAL

Width of the Wear Track



RECIRCULATION AND PUMP EFFICIENCY

The pumped fluid goes from the low pressure section of the eye to the high pressure of the discharge. If there is a path from the discharge to the suction, a percentage of the pumped fluid travels back, and this will reduce the efficiency of the pump. If the impeller has an open space, it should be close to the casting to restrict this recirculation path. The distance between the impeller face and the casting to restrict this recirculation path. The distance between the impeller face and the casting may vary from .008" to .035" or more, depending on the design of the pump. The amount of running clearance should usually be less than .015", but the cold adjustment clearance may need to be more. The pump manufacturer may have had to allow for differential expansion of the metals in the pump, and he would allow for deflection of the shaft. Clearances are usually adjusted from the bearing end by sliding the bearings until the impeller bottoms, and then pulling it back the designed amount. If you are not able to bottom the impeller with the bearing adjustments available, you may need to check your actual clearance by ensuring the distance from the impeller to the casting wall. In some cases of odd-shaped impellers, modeling clay is placed on the vanes and that is compressed to the wall. When the pump is again disassembled, the compression of the clay can be measured. It is also important to make sure that the impeller is perfectly square with the centerline of the shaft.

SUCTION AND DISCHARGE RECIRCULATION

On some pump designs, where the eye of the impeller is large compared to the impeller diameter, a localized vortexing, or recirculation, of the inlet fluid occurs. This vortexing can drop the pressure to the point where the fluid cavitates and causes damage at the eye of the pump. Though the symptoms are similar to cavitation, the localized erosion occurs at the suction eye of the pump and it is usually on the pressure side of the vanes. (Whereas normal cavitation will often damage the impeller somewhere between the eye and the edge of the impeller.) Discharge recirculation occurs on some pumps, also. This is a localized vortexing at the tips of the impeller blades causing cavitation-like damage. Often this is associated with high efficiency pumps operating outside their most efficient area.

Both suction and discharge recirculation are signs of fundamental design problems. Though the causes are known, the cure is not so easy. It involves getting the pump operating at its most efficient point or changing to a different impeller design.

SUMMARY

In order to troubleshoot a pump problem, you need to know if you have a pump problem or a system problem. A pump problem is when the pump is failing to meet its design curve. There are very few things that can cause that to happen. In a centrifugal pump, it will most likely be excessive clearance on the wear rings or impeller adjustment, a completely eroded impeller or a physical restriction in the pump.

Bearings

BEARING LIFE

Oil comes halfway up to the middle of the bottom ball in an oil lubricated pump ball bearing. As the ball turns the roughness picks up a him of oil which keeps the parts of the bearing separated. The oil film is two to three times as thick as the average peak to peak roughness of the metal parts. Because the parts can not come into contact the pump bearing is not subject to wear. Pump bearings are subject to repeated stress. How many times a metal can be stressed depends on the amount of stress and the type. In extensive testing of ball bearings over the years they have come up with methods for projecting out the life of bearings .

If, for example, one thousand similar bearings were tested under 2000 lbs. of load at 3600 rpm and 10% failed after 100 hours that would be called the B-10 life. If a new set of bearings were tested at 1000 lbs. of load and10% failed after 400 hours that would be the B-10 life. If the load was cut to 500 lbs and a new set of bearings was tested 10% would fail after about 1600 hours. This means if we were plotting out the life versus load we could see that if the load were low enough, the life would be infinite. When pump companies say they have a bearing life of 40 years that means for the amount of load and the rpm, tests have shown that 10% would fail in that time span.

In practice original pump bearing failure from fatigue is rarely a problem. Most pump bearings fail because they have worn out or there has been a lubrication failure.

OIL CONTAMINATION

Water in a pure mineral oil at levels as low as .2 % can increase bearing failures substantially. This is confusing in that sometimes water is used as a bearing lubricant .

WATER LUBRICATED BEARINGS

Water is a thin film lubricant. The film thickness runs about 11 millionths of a inch which is less than the roughness of the metal. So parts are in contact and they wear out. Water will not support the pressure that oil will so it should not be used around ball or roller bearings. Because ball bearings only have a small area in contact with the load they are extremely high pressure devices. Water lubricated bearings are bushings. Though the term journal bearing applies to any bearing holding a journal(shaft) the term generally refers to sleeve bearings or bushings. The common water lubricated bushing materials are Bronze, Rubber, and Graphite. Bronze bearings must be kept lubricated and have high expansion in relation to most shaft and casing materials. This means excessive clearance must be allowed for startup expansions. Once the equipment is at operating temperatures this clearance leads to a problem called shaft whirl. The larger pumps that have bronze journal bearings are more likely to have seal and packing problems because of this type of radial run out. The rubber bearings are becoming much more popular where water is the lubricant as in many well pumps. The bearings have the low friction of wet rubber and freedom from the type of differential growth of the bronze bearings.

Bearings

Surprisingly, the simple routine of changing the oil at regular intervals of two thousand hours could stop many of the reported problems.

Grease lubrication

Grease is a soap base that acts a sponge for oil. When it is heated the oil is released. Usually pumps use the same lithium based #2 grease you use on your automobile. Grease can not be changed easily in most equipment and so is more prone to contamination than oil.

Grease must be used in three cases:

1. On an oil lubricated pump bearing oil is only half way up on the lowest ball. The bearings are made of steel and they rust rapidly. So, you grease equipment that is not running frequently to keep it from rusting.

2. On vertical equipment there is no way to keep oil halfway up on the ball so you grease vertical equipment.

3. On mobile equipment there is no level and so you must use grease.

Regreasing your equipment:

On a pump it is a good idea to remove the old grease before you put new grease in. If there is a drain on the bottom and on the opposite side from the grease fitting the routine is easy. You remove the grease fitting. With the equipment running and the drain removed you keep adding SAE 30 oil .This will wash out the old soap base. When clean oil is coming out the drain you are now ready to regrease. You stop the equipment, put the grease fitting back on and pump grease in until it comes out the drain. You then start the equipment and let it run for at least 20 minutes. This gives it time for the grease to expand out and for the grease to get thrown out of the way. You then close the drain and you are off and running for another 500 to 1000 hours.

GREASING FREQUENCY

SERVICE	GREASE EACH
Normal, 8 hour day operation. Room free of dust and damaging atmosphere.	6 Months
Severe, 24 hour day operation. Room with moderate dust and/or damaging atmosphere, or outdoor service	1 Month
Light, approximately 10 hour week. Room relatively free of dust and damaging atmosphere.	1 Year

They cannot run dry. The graphite journal bearings are self lubricating and free from expansion problems. They can take dry running to some extent. The disadvantage is higher initial cost.

SOURCES OF WATER CONTAMINATION

Pump bearing housings are vented to atmosphere. In a high humidity environment (such as the Gulf Coast Region of the U.S.) just the drop in temperature at night when the equipment is turned off is enough to put water in the oil. Additives are normally in the oil to prevent breakdown but their life and effectiveness are limited. One of the most common sources of water contamination is from a leaking packing or mechanical seal. In many food processing plants they hose down the equipment periodically for cleaning purposes. Of course any equipment outside is subject to the weather.

PREVENTING CONTAMINATION

Lip seals made out of the common rubber materials have a limited life and should be avoided. The rubber-on-metal line contact generates high localized heat which cooks out the rubber. It may also cause damage where the rubber has picked up abrasives and grinds the shaft. This makes a replacement lip seal unsuitable.

Some specialty lip seal suppliers can provide different materials which last a great deal longer.

The in-place flinger found on most pumps should be maintained. If the source of failure is from a leaking packing or seal then an extra flinger should be made and put between the packing gland and the bearing housing. This extra flinger should be as big as will fit and as close to the packing or seal gland as possible. The basic idea is to keep the leakage away from the other flinger.

Today, Labyrinth seals easily replace lip seals. Labyrinth seals have been around for many years. However, early ones were made of bronze with o-rings to drive the heavy rotary, they were also hard to retro fit were the lip seals area.

Recently a company named JM Clipper invented a TFE and graphite labyrinth seal that snaps in where the lip seal was. So just pop out that National or Chicago lip seal, copy the part number and it can be crossed to a labyrinth seal. Labyrinth seals are many times the cost of lip seals. However, they last longer due to no contact between the rotor and stator and the TFE/graphite construction works up to 500°F. *****

Oil misting systems have also proven highly effective in many refineries. They redesign the bearing housing so an oil mist is pumped through the bearings and then collected as it leaves. By keeping a slight positive pressure on the bearing housing it stops contamination and in many reports stops bearing failures. The oil is drained from the sump for the most effective misting but that involves an element of risk. The commercial systems used in refineries use back up systems of air or nitrogen to maintain a pressure no matter what happens.

Monitoring the condition of the oil is another method used in larger equipment that may contain gallons of oil.

LUBRICATION PROBLEMS

The first place a lubrication failure will show up is on the retainer. If you have had retainer failures, bearing wear and signs of excessive heat you have probably had a lubrication failure.

Bearing Tolerances

ABEC tolerances for pump bearings are described as class 1, class 3, class5, etc. This describes how closely the dimensions match the ideal. For all practical purposes the class 3 tolerance will be within .0002 of design and is as close as you will need for a pump. Higher precision bearings are much more costly and do not give you any advantage. Classes of fit describe the internal clearance and are important where heat or expansions are a problem. Most industrial pump bearings use a class 3 fit or loose fit bearing. The term loose and very loose should not be confused with play in the bearing. They are relative terms used by bearing manufactures and will have no effect on your seals. Industrial bearings are loose fit. If you are using stainless steel shafts and steel bearing, you may have a problem with expansions. If you do it will show up as banding on the balls and the possibility of a lubrication failure from excessive heat. This would be the sign to go to a looser fit. The manufacturers use a slightly smaller ball.

Brinnelling

A person named Brinnel invented a machine for testing the hardness of metals. In the device they press a round ball into a piece of metal under a known load. This leaves a dent in the metal. The bigger the dent, the softer the metal. The dent size or brinnel number is used for describing hardness. A more common method is to use a diamond point and a Rockwell machine for testing hardness. Someone picked up the terminology used in this field to describe what happens when you press a bearing the wrong way. If you push or bang on the other race you will cause brinnelling. That is, you will be pushing a round ball into a piece of metal under load. This will cause dents in each ball location. You can also get brinnelling by dropping a piece of equipment. False brinnelling is a term used to describe the dent made at the ball location when a direct force is not applied. In a vibrating situation, if the oil from the ball gets squeezed out the metal on metal contact can cause damage. This is prevented by using high pressure additives and rotating your equipment at frequent intervals. If you must move heavy equipment it is a good idea to tie the shafts down so the bearings do not bounce and cause brinnelling.

Heating a Bearing

The hole in a 2 inch bearing heated to 180 degrees F on an 80 degree day will expand .0012. This is the internal clearance on a 2 inch loose fit bearing. This means the bearing should slide on the shaft easily. You use 180 degrees F because you do not need any more. If you need a higher temperature there is probably something wrong with the fit or a dimension. At 190 degrees rubber seals begin to cook. At 200 degrees, the grease may begin to smoke. At 250 degrees many steels start to lose their strength. The bearing can be heated in oven at 180 degrees with no risk. If you use any other method you should have some way of monitoring the temperature.

Freezing a Shaft

CO_2 comes out of a fire extinguisher at -140 degrees F. This is a good way to freeze a shaft. If you know the dimensions are correct it is not a bad idea to freeze the shaft and heat the bearing. This gives you more time. If the bearing gets stuck halfway down the shaft there is nothing wrong with continuing with a press to get it all the way. The practical problem with pressing bearings onto a shaft is the tendency to keep pressing even if they don't fit.

Troubleshooting Pump Bearings

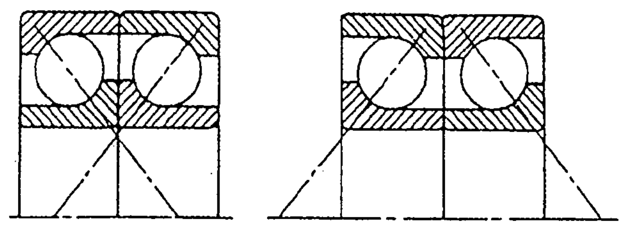
A wide variety of information is available from bearing distributors concerning exotic bearings failures. The most common pump bearing failures however are from contamination of the oil and from problems of fit

For two direction / thrust loads, use paired bearings

The back-to-back arrangement is recommended for many pump applications since the clearance of the pair is controlled by clamping the inner rings, and no clamping of the outer rings is necessary.

The API 610 Standard specifies that the pump thrust loads shall be supported by two 40°, single row angular contact ball bearings, arranged back-to-back. The need for bearing axial clearance or preload is to be based on the requirement of the application.

The face-to-face arrangement is used when <u>misalignment is unavoidable,</u> such as in double suction pumps with slender shafts and housings bolted on the pump frame. The main advantage with the face-to-face arrangement is less sensitivity to misalignment. For proper function, the outer rings of bearings arranged face-to-face must be securely clamped in the housing. The axial clamp force for bearings arranged face-to-face must be greater than the axial load supported by the bearings but less than the limiting clamp load. The axial clamp force can be applied to the outer rings by the clamping of the housing cover.



FACE-TO-FACE BACK-TO-BACK

Alignment

Why You Are Aligning

Though most alignment information is supplied by the coupling manufacturers, you must keep in mind that they are more concerned with their coupling than they are with your equipment. Many couplings are flexible and can work within high limits of misalignment without damage to the coupling. In some cases, the alignment instructions can give you the impression that misalignment within the coupling limits is perfectly acceptable. But, if the pump and motor centerlines are at an angle to each other, the motor will be transmitting radial and axial forces to the pump shaft. These forces may cause bending or flexing of the shaft. This will add load cycles to the bearings and shorten their life. They will also cause the shaft to move axially and radially within the bearings clearance and that motion will be felt at the mechanical seal. Any motion of the shaft will cause a serious seal problem. Mechanical seals have to maintain a very thin film of lubricating fluid between a rotating face and a stationary face. If the shaft is moving under the seal, then the faces may separate. Face separation leads to seal leakage which will shorten the life of the seal. Face separation also allows abrasive particles to get between the faces, which can lead to considerable damage. How this leakage shortens the life of the seal is described in detail in the seal section of the manual.

Alignment Preparations

The base plate should be clean and level so that any horizontal motions will not affect the vertical position. You must also ensure that all four feet of the motor and pump are resting evenly on the base plate or shim pads. If they are not, you have soft foot. If the motor was a table, it would be rocking if all four feet did not hit evenly. But because it is bolted securely to the base plate, it will be distorted. This distortion can lead to problems of alignment within the pump or motor.

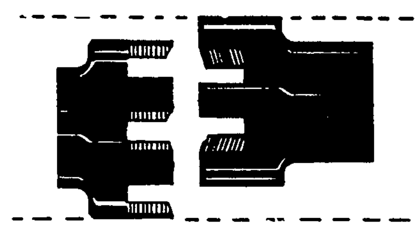
Checking for Soft Foot

When the motor or pump is being unhooked, you should place a dial indicator on the baseplate with the probe resting on the foot you are loosening. If the foot moves more than .002, then you have a soft foot and this must be shimmed independently of the others until a tightening or loosening does not make any different in the dial indicator readings.

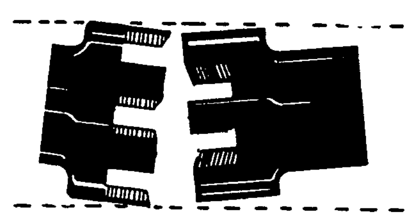
A flexible coupling is not designed to compensate for excessive misalignment. Its purpose is to permit slight movement of pump or driver shafts or any driven equipment while transmitting power. Excess misalignment can cause short coupling life due to sliding and working action of coupling connectors. Modern machinery operates at ever increasing speeds; even minor alignment errors lead to high vibration on bearing loads resulting in machinery damage and production downtime.

Solving Misalignment Problems

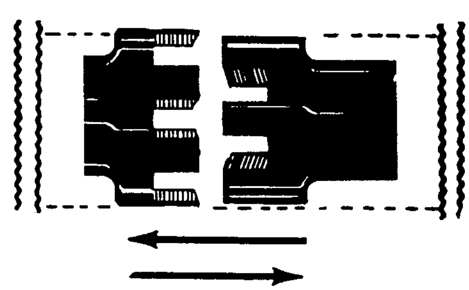
1. Radial/Parallel



2. Angular



3. Axial



INSTALLATIONS OF PUMP AND VALVE PACKINGS

Packing The Pump Correctly

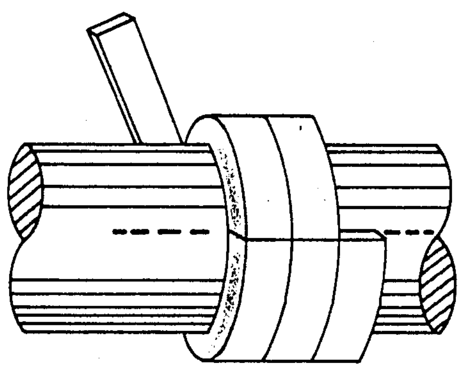
The importance of packing the pump correctly cannot be overemphasized. Many packing failures are due to incorrect installation of the packing. The following steps have been devised to ensure effective installation of packings on pumps:

1) **REMOVE ALL THE OLD PACKING FROM THE STUFFING BOX.** Clean box and shaft thoroughly and examine shaft or sleeve for wear and scoring. Replace shaft or sleeve if wear is excessive.

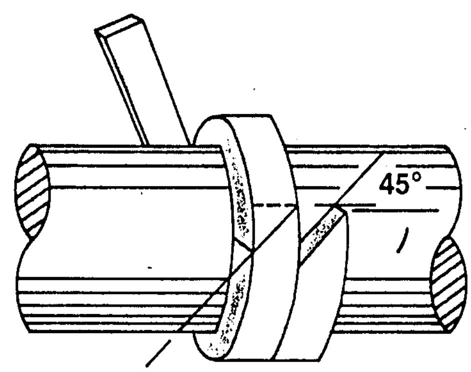
2) **USE THE CORRECT CROSS-SECTION OF PACKING OR DIE-FORMED RINGS.** To determine the correct packing size, measure the diameter of the shaft (inside the stuffing box area if possible) and then measure the diameter of the stuffing box (to give the OD of the ring). Subtract the ID measurement from the OD measurement and divide by two. The result is the required size.

3) **WHEN USING COIL OR SPIRAL PACKING, ALWAYS CUT THE PACKING INTO SEPARATE RINGS.** Never wind a coil of packing into a stuffing box. Rings can be cut with butt (square), skive (or diagonal) joints, depending on the method used for cutting. The following illustration shows these methods of preparing bulk packing. The best way to cut packing rings is to cut them on a mandrel with the same diameter as the shaft in the stuffing box area. If there is no shaft wear, rings can be cut on the shaft outside the stuffing box.

   Hold the packing tightly on the mandrel, but do not stretch. Cut the ring and insert it into the stuffing box, making certain it fits the packing space properly. Each additional ring can be cut in the same manner, or the first ring can be used as a master from which the balance of the rings are cut.



BUTT JOINT



SKIVE JOINT

If the butt cut rings are cut on a flat surface, be certain that the side of the master rings, and not the OD or ID surface, is laid on the rings to be cut. This is necessary so that the end of the rings can be reproduced.

When cutting diagonal joints, use a miter board so that each successive ring can be cut at the correct angle.

It is necessary that the rings be cut to the correct size. Otherwise, service life is reduced. This is where die-cut rings are of great advantage, as they give you the exact size ring for the ID of the shaft and the OD of the stuffing box. There is no waste due to incorrectly cut rings.

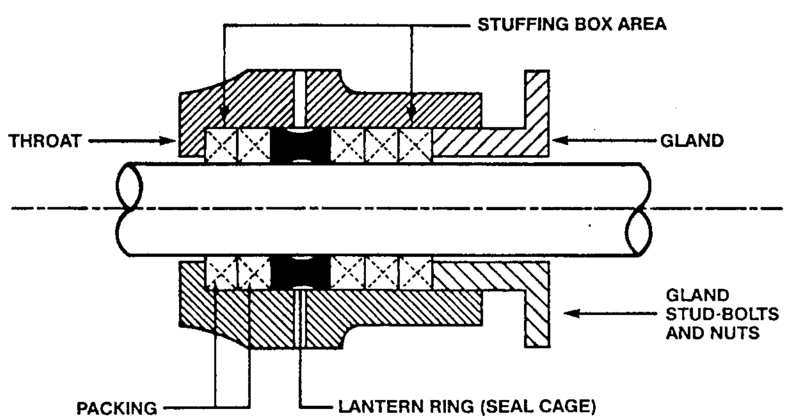
4) **INSTALL ONE RING AT A TIME.** Make sure it is clean and has not picked up any dirt in handling.

Seat rings firmly (except PTFE filament and graphite yarn packings, which should be snugged up very gently, then tightened gradually after the pump is operating). Joints of successive rings should be staggered and kept at least 90 degrees apart. Each individual ring should be firmly seated with a tamping tool, or suitable split bushing fitted to the stuffing box bore. When enough rings have been individually seated so that the nose of the gland will reach them, individual tamping should be supplemented by the gland.

5) **AFTER THE LAST RING IS INSTALLED,** take up gland bolts finger tight or very slightly snugged up. Do not jam the packing into place by excessive gland loading. Start pump and take up gland bolts until leakage is decreased to a tolerable minimum. Make sure gland bolts are taken up evenly. **STOPPING LEAKAGE ENTIRELY AT THIS POINT WILL CAUSE THE PACKING TO BURN, HARDEN, AND DAMAGE EQUIPMENT.**

6) **ALLOW PACKING TO LEAK FREELY STARTING UP A NEWLY PACKED PUMP.** Excessive leakage during the first hour of operation will result in a better packing job over a longer period of time. Take up gradually on the gland as the packing seats, until leakage is reduced to a tolerable level, preferably 8-10 drops per minute, per inch of shaft diameter. Some packing can run virtually leak free. Contact your packing manufacturer for specific recommendations.

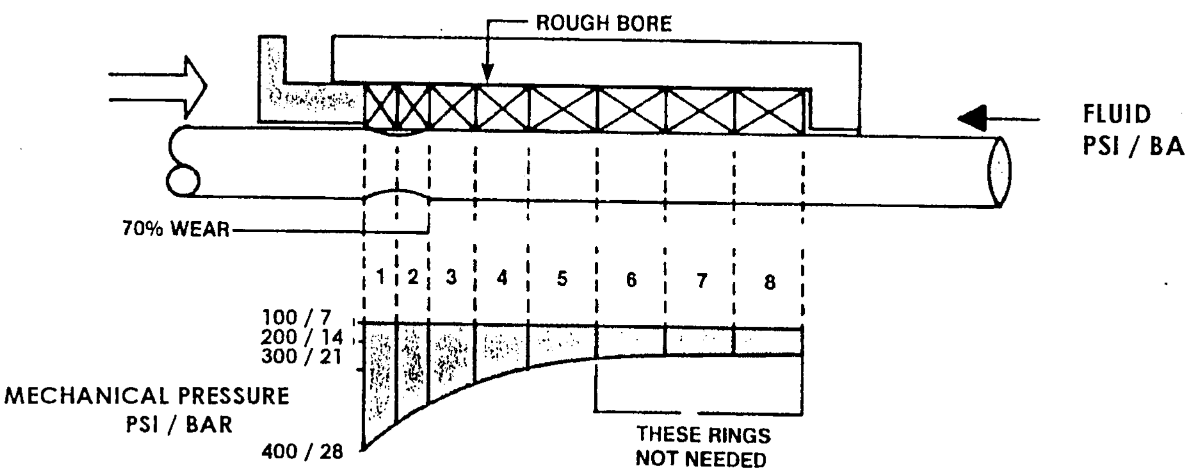
7) **WHEN SPECIFIED BY THE PUMP MANUFACTURER, PROVIDE MEANS OF LUBRICATING THE SHAFT AND PACKING THROUGH THE LANTERN RING BY SUPPLYING WATER, OIL, GREASE, OR LIQUID HANDLED IN THE PUMP.** Fittings for this purpose are standard on many pumps. Flush pressure should be minimum 15 psi (1 bar) above stuffing box pressure.



8) **IF THE STUFFING BOX HAS A LANTERN RING (SEE FIGURE 5),** make sure that the lantern ring is installed properly so it will remain under the inlet as gland pressure is applied.

9) **REPLACE PACKING WHEN LEAKAGE CANNOT BE CONTROLLED BY FURTHER TAKE-UP ON THE GLAND. DO NOT ADD MORE PACKING RINGS.**

10) **ON BOTH CENTRIFUGAL AND RECIPROCATING PUMPS,** about 70% of wear is on the outer two packings nearest the gland. However, each additional ring does throttle some fluid pressure. On most pumps, there must be enough rings so if one fails, another does the sealing, and the pump need not be shut down.



The mechanical pressure curve above shows eight packing rings. The first five rings do the majority of the sealing. The bottom three do little sealing, but are needed to fill the available space. The advantage of using fewer rings is less shaft or sleeve wear. Also, the stuffing box design is simpler and takes less material. But, wear isn't the only problem. With high temperatures, high pressures, corrosive chemicals, or abrasive particles in the fluid, more rings may be the only solution for some services. In such cases, the bottom ring controlling the fluid may have the most wear from these severe service conditions.

<u>CAUTION</u>: ALL PACKINGS <u>MUST</u> BE INSTALLED IN ACCORDANCE WITH MANUFACTURER'S INSTRUCTIONS.

TROUBLESHOOTING PACKING FAILURES

Packings may fail for a variety of reasons. Besides improper installation, packing failures are often due to worn or faulty equipment, shaft misalignment, uneven take-up on the gland bolts, and other causes.

If you are having trouble, carefully remove and examine the old packing set. DO NOT THROW THE SET AWAY, because it often gives clues as to the condition of the equipment and may be the means of solving the problem. The following clues and possible causes were found by examining sets of packing which failed in service:

CLUE 1: Excessive reduction in cross-section of packing directly beneath the rod, shaft, or plunger.

CLUE 2: Excessive reduction in the thickness of the packing directly over or on either side of the rod or shaft.
POSSIBLE CAUSE: Rod or plunger out of alignment, and in the case of the rod or shaft, the bearings may be badly worn, causing whipping of the shaft.

CLUE 3: A whole ring or part of a ring is missing from set.
POSSIBLE CAUSE: Bottom of stuffing box badly worn, with packing being extruded into the system.

CLUE 4: Wear on the outside of one or more rings.
POSSIBLE CAUSE: Rings rotating with shaft or loose in the box. Packing too small.

CLUE 5: Axial bulge in one or more rings.
POSSIBLE CAUSE: Adjacent rings cut too short or too long, depending on the style of material used, causing packing under pressure to be deformed.

CLUE 6: Packings show tendency to extrude between rod or shaft and the gland follower.
POSSIBLE CAUSE: Excessive gland bolt pressure and/or too much clearance between rod or shaft and the gland follower.

CLUE 7: Rings next to gland follower badly damaged, with bottom rings in fair condition.
POSSIBLE CAUSE: Improper installation of packings and excessive gland bolt pressure used.

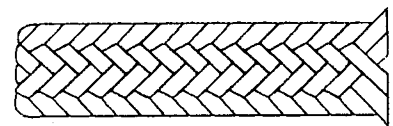
CLUE 8: Wearing surface of rings dried and charred with rest of packings in good condition.
POSSIBLE CAUSE: High temperatures and lack of adequate lubrication.

CLUE 9: Innermost ring deteriorated.
POSSIBLE CAUSE: Packing incompatible with fluid handled.

Pure White Teflon, Slow Speed

Made of Pure White Teflon and Lubricated with Teflon Suspensoids

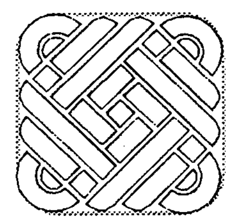
Most all Chemicals and Speeds Only to 900 FPM



Centerlock® braided from pure PTFE filament and lubricated with PTFE suspensoid.

This construction creates a dense firm packing with excellent chemical resistance and is impervious to vapor and gases.

Conforms with Mil-P-24396, Type B.



Centerlock®

Construction
Centerlock®
Base Material
Pure PTFE
Shape
Square
Lubricant
PTFE suspensoid
Temperature
-400°F/-240°C to 550°F/288°C
pH
0-14
FPM
900

Applications
Low speed centrifugal, rotary pumps, valves □ Air, water, sea water, brines, superheated steam, sludges, slurries, waste, petroleum oils, petroleum fuels, solvents, chemicals, liquified gas, strong chemicals and harsh gases.
Advantages
Non-asbestos. Provides longer packing life than asbestos. PTFE suspensoid provides added lubricity.
Maximum Size
2"

Feet Per Pound

1/8"	3/16"	1/4"	5/16"	3/8"	7/16"	1/2"	9/16"	5/8"	11/16"	3/4"	7/8"	1"
76.66	3.33	18.00	11.89	8.88	6.15	4.76	3.84	3.12	2.59	2.19	1.65	1.27

Feet Per Minute =

RPM of Pump x Shaft Diameter in inches x 3.14 ÷ 12

Example:

1. **1750 RPM x 2" Shaft Dia. X 3.14 ÷ 12 = 916 FPM**
2. **3600 RPM x 2 Shaft Dia. X 3.14 ÷ 12 = 1884 FPM**

Genuine GFO

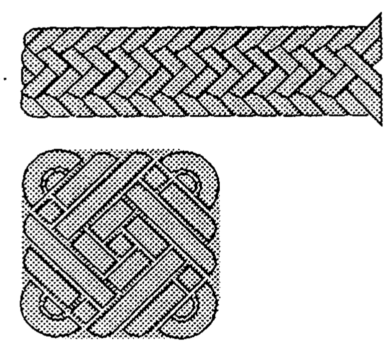
Best General Service Packing

Made of Gore's Soft Teflon Sleeves Stuffed with Pure Graphite Particles and Coated with a Special Inert Lubricant

No Substitution, Many Chinese Imitations are on the Market Today Make Sure it's Genuine GFO.

Most All Chemicals and Speeds up to 4400 FPM

Centerlock® braided from PTFE fiber containing a high percentage embodiment of submicron size Graphite particles and thoroughly impregnated with an inert lubricant.



Centerlock®

Our braided construction creates an exceptional, high speed packing far outperforming and outlasting similarly construction packing.

Construction
Centerlock®
Base Material
GFO® Graphite/PTFE yarn
Shape
Square
Lubricant
High temperature inert
Temperature
-400°F/-240°C to 550°F/288°C
pH
0-14
FPM
4400

Applications
Best general service high speed pump packing □ Air, water, sea water, brines, superheated steam, sludges, slurries, waste, petroleum oils, petroleum fuels, solvents, chemicals, liquified gas.
Advantages
Comparison testing has proven this style far out performs all other material in a wide variety of applications and industries. Field testing reports available upon request.
Maximum Size
2"

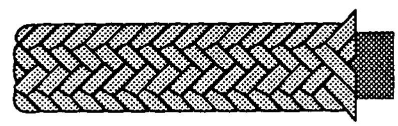
Feet Per Pound

1/8"	3/16"	1/4"	5/16"	3/8"	7/16"	1/2"	9/16"	5/8"	11/16"	3/4"	7/8"	1"
105.00	45.25	23.58	15.72	10.13	7.86	6.00	4.73	3.80	3.22	2.70	1.99	1.52

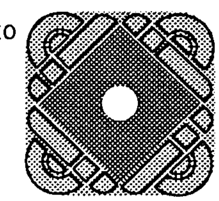
GFO® is a registered trademark of W.L. Gore & Associates

GFO Rubber Core Packing

GFO Packing is Soft by Nature so Adding a Rubber Core Evenly Spreads Wear Over the Packing Set, Not Just the First Two Rings. Therefore, Leakage can be Controlled with Lighter Gland Loads.



The resilient core evenly spreads wear over the packing set when gland follower pressure is applied. This packing was especially designed for pumps where there is an axial or elliptical shaft run out. Made with a core of Silicone, ZZR765 spec grade. The core offers a combination of the highest chemical resistance and temperature ability. The outside braided yarn is a combination of graphite, fluorocarbon and oil to promote the maximum sealing and wear resistance. This also makes an excellent door gasket.



Centerlock®

Construction
Centerlock®
Base Material
GFO® Graphite/PTFE Yarn
With 7165 Silicone Core
Lubricant
High Temperature Inert
Temperature
-104°F to 500°F/260°C
pH
1-14

FPM
2000
Applications
Agitators mixers, worn pumps with an axial or elliptical run out handling water, waste water, brines, oils also static door or lid gaskets.
Advantages
The silicone core allows the packing to continually seal on pumps that are less than perfect mechanical conditions.

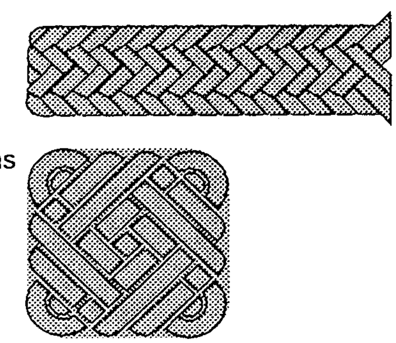
Feet Per Minute

5/16"	3/8"	7/16"	1/2"	9/16"	5/8"	3/4"	7/8"	1"
16.28	11.67	9.10	7.14	5.54	4.30	2.97	2.26	1.672

Synthetic Yarn Impregnated with Teflon and Graphite

Best Low Cost Black Synthetic
Best Abrasive Resistance

Many Chemicals and Speeds to 3200 FPM



Centerlock®

Synthetic Yarn Impregnated with Teflon and Graphite
Centerlock® braided from synthetic yarn thoroughly impregnated with a unique combination of PTFE and Graphite.

This innovative packing offers excellent capability in a wide spectrum of applications including high speeds and aggressive media, yet is sufficiently economic to standardize for general purpose use.

Construction
Centerlock®
Base Material
Synthetic yarn
Shape
Square
Lubricant
PTFE suspensoid, Graphite
Temperature
550°F/288°C
pH
1-13

FPM
3200 maximum
Applications
Best general service, high speed abrasive resistant pump packing □ Dredges, air, water, sea water, brines, superheated steam, sludges, slurries, waste, petroleum oils, petroleum fuels, solvents, chemicals.
Advantages
Excellent for pumps with high speed shafts to extend packing and shaft life. Excellent abrasion resistance.
Maximum Size
2"

Feet Per Pound

1/8"	3/16"	1/4"	5/16"	3/8"	7/16"	1/2"	9/16"	5/8"	11/16"	3/4"	7/8"	1"
109.70	48.78	27.32	17.45	12.97	9.10	6.99	5.50	4.44	3.81	3.27	2.14	1.77

Pure Flexible Graphite

Braid Over Core

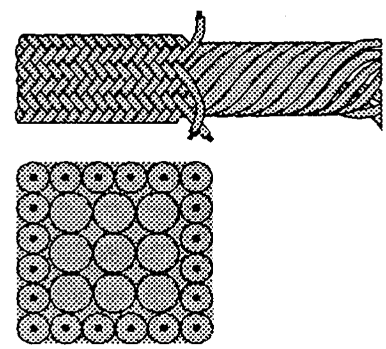
Made of Pure Flexible Graphite
It is Compatible with most Chemicals and Temperatures
and can be <u>Run Dry</u>

No Substitution
Competitors Conventionally Braid Flexible Graphite,
but our Higher Density Braid Over Core Construction
Provides the Proper Compression to Run Dry

Pure Flexible Graphite
Braided from flexible pure graphite.

This is a high density, pure graphite braid; not a man made carbon/graphite fiber.

This packing has excellent high and low temperature operating ability. It can be ran dry in pumps and withstand dramatic pressure and temperatures in valves.



Braid Over Core

Construction
Braid Over Core
Base Material
Flexible Pure Graphite
Shape
Square
Temperature
Up to 800°F totally oxidizing atmosphere
5400°F in non-oxidizing atmosphere
pH
0-14
Pressure
Valve 3500 PSI
Rotating Pump 426 PSI
Reciprocating Rod 3500

FPM
Rotating pump: To 3000
Reciprocating pump: To 390
Application
Best general service low leak pump and valve operation □ Air, water, sea water, brines, super heated steam, sludges, slurries, waste, petroleum oils, petroleum fuels, solvents, chemicals, agitators, dryers, mixers, blowers.
Advantages
The heat dispersion properties and compressibility of graphite makes this the most capable packing on the market today.
Maximum Size
2"

Feet Per Pound

1/8"	3/16"	1/4"	5/16"	3/8"	7/16"	1/2"	9/16"	5/8"	3/4"	7/8"	1"
109.37	62.49	29.82	17.15	14.18	10.02	7.60	6.20	5.19	3.55	2.61	2.00

Special Installation Instructions Next Page

Installation Procedures for Pure Flexible Graphite For Pumps Only

3. For maximum protection against suspended solids, compress the first ring into the bottom of the stuffing box using a split bushing or stacked lantern.

4. Install rings (for best performance compress each ring individually).

5. Tighten the rings until they resist compression (at the ideal density the rings will then resist further compression). A maximum of from 6 to 8 rings is recommended. If a spacer is required, you may place a metallic lantern ring between the gland and the last ring.

6. Loosen gland nuts and re-tighten, finger tight, plug the flush connections.

7. If possible, rotate the shaft manually, the packing should not bind the shaft. Manual rotation should free the shaft if the packing has been over-tightened.

8. At initial start-up, increased average demands and smoke may be observed for about 3 to 5 minutes, this is nominal.

9. Zero leakage performance is directly related to the degree or shaft whip or run out, excessive radial movement may prevent zero leakage, however, even under poor conditions the packing should run flush free with minimum leakage.

COMMENTS

- Clean shaft and stuffing box, do not lubricate the packing or shaft.
- At start-up, micro-particles or graphite plate the shaft forming a microfilm of graphite to reduce friction and average.
- The graphite packing transmits heat rapidly, forming a heat sink throughout the stuffing box, eliminating the need for cooling flush for thermal venting, also the dead air space between the shaft and sleeve prevents heat venting through the shaft.
- Eliminating highly oxygenated flush water reduced the risk of fretting corrosion to the sleeve.
- Worn sleeves: smoothly worn sleeves may perform better than new sleeves. The packing holds into the surface geometry to seal against the several surface planes.
- Rings should be angle cut and have 1/8" to 3/16" oversize to allow for good distribution of their physical mass.

Packing Versus Seals

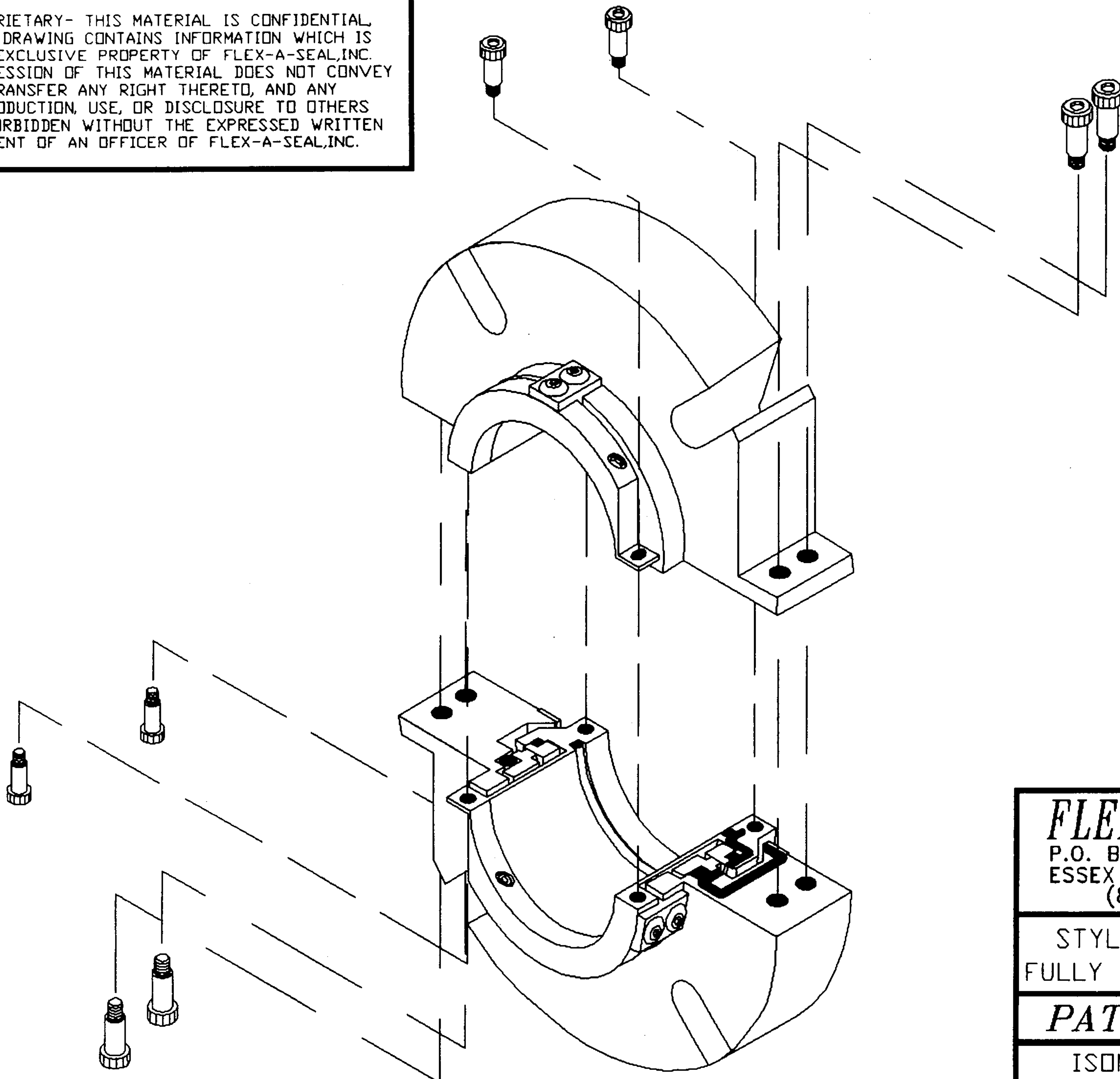
There has been a trend in the pumping business to supply pumps with mechanical seals rather than packing. Often, they are offered at the same price. Each choice has its advantages and disadvantages.

Packing's advantages. Easy to install. Inexpensive compared to seals. Its use may be perfectly satisfactory.

Packing's disadvantages. Usually has to leak and wears out the sleeve or shaft. Leakage may be a hazard. It can be expensive and often cause corrosive damage in the area of the pump. It may cause a sanitation problem. In some cases, the leakage is a direct cause of frequent bearing failures. On lift pumps or vacuum service may draw air into the product stream. Requires routine maintenance and monitoring.

Seal's disadvantages. Requires some special handling. They require initial conversion costs. However, most pumps will accept a split seal with no pump overhaul. Just remove the packing and install the split seal.

Seal's advantages. No routine maintenance. No leakage. Seals are usually the better choice when sealing a vacuum because they will prevent air from entering the fluid. Also, 1) the use of cartridge seals has eliminated set dimensions which make them easier to install; 2) with cartridge seals you don't handle the faces, they are preset and have been tested for leaks at the factory, therefore, no leaky start ups.



FLEX-A-SEAL,INC.
P.O. BOX 184/1 JACKSON ST.
ESSEX JUNCTION, VT 05453
(802) 878-8307

STYLE 85 MULTI-SPRING
FULLY SPLIT CARTRIDGE SEAL

PATENT PENDING

ISOMETRIC EXPLOSION

DRAWN BY:	DATE:	DRAWING NUMBER:
R.B.	4/20/93	SK1123

Mechanical Seals

MATERIALS OF CONSTRUCTION

To make educated decisions in selecting seals, it is necessary to have a working knowledge of the materials of construction for mechanical seals. Sealol provides this information to assist users in choosing a seal that is compatible with the pumped product.

Because seals are exposed to a wide range of applications, careful selection of compatible materials to the pumped fluid is essential. Each material is selected for its unique properties. Corrosion, heat, abrasiveness, viscosity, and lubricity are just a few factors paramount in material selection.

A. Face Materials

(Rotary and StationaryJ

One of the most important steps in seal selection is choosing wear face materials that will perform well in operation. This section discusses available face materials and their characteristics. Factors such as thermal conductivity, density, elastic modulus, hardness, toughness, and chemical compatibility should all be carefully considered.

During operation, faces of the same material can exhibit a fast wear pattern due to frictional contact. If used in the wrong application, the frictional heat build-up that occurs between the two faces of the same materials can cause seal failure.

Selecting two dissimilar materials (one softer than the other), such as carbon-graphite vs. silicon carbide, minimizes potential damage from frictional heat build-up. However, in abrasive service, it is sometimes better to use two hard faces, such as tungsten carbide vs. silicon carbide. Lubrication between two hard faces is especially important to guard against excessive frictional heat.

Face material combinations should be selected on the basis of corrosive resistance; hardness; and PV (pressure-velocity). PV is a measurement representing the seal face contact pressure and the velocity (SFPM) of the rotating seal face.

PV indicates some measure of service severity that relates to a seal's wear life. As PV increases, seal face selection becomes more critical.

PV = Face Unit Load x Velocity

$$\text{Face Unit Load} = \frac{\text{Net Closing Force}}{\text{Face Contact Area}}$$

Velocity = Mean Face Circumference x Speed

Carbon-Graphite

Carbon Graphite is the most commonly used seal face material. The advantages that carbon offer include excellent friction and wear characteristics, low cost and it is easily machined. Its ability to resist corrosive attack in many applications adds to its popularity. However, it is a soft material and is subject to abrasive wear.

Because of its low modulus of elasticity, carbon-graphite faces may bend and distort under pressure. Carbon-graphite is strongest when in compression. Carbon vs. Silicon Carbide has the best friction and wear characteristics of any seal face combination.

Hard Faces

Three hard-face materials are presently being used for seal applications: aluminum oxide (ceramic), tungsten carbide and silicon carbide.

Aluminum Oxide-Ceramic

Aluminum oxide, or ceramic, exhibits a tendency toward thermal-shock, and has a relatively low hardness compared to silicon carbide. It is also brittle. A solid, high-purity ceramic seal face usually contains 99.5% aluminum oxide. It is chemically inert and can be applied to nearly any product. It is normally limited to moderate pressures and speeds. It is most commonly found in less severe applications such as light duty water pumps.

Tungsten Carbide

Tungsten carbide is manufactured by sintering tungsten carbide panicles into a matrix of metal, which acts as a binder. The performance of tungsten carbide greatly depends on the size of the particles, and the distribution and type of binder used.

Tungsten carbide is a heavy, tough, abrasion-resistant material with good thermal conductivity. It exhibits good performance in abrasive service, especially running against silicon carbide. However, in high-speed applications with marginal lubrication, excessive heat can be generated at the interface, causing thermal cracks at the surface known as heat checking. Tungsten carbide may also be subject to corrosive attack on its binder, causing premature failure.

Mechanical Seals

Silicon Carbide

Silicon carbide is a seal face material that exhibits the highest degree of hardness of any seal face material and excellent thermal conductivity. It is a tough, hard, heat resistant, long wearing inert seal face material. There are two types of silicon carbides: reaction bonded and sintered.

In reaction bonded materials, the silicon carbide particles are surrounded by free silicon. The siliconizing process limits the porosity of the material, offering certain advantages in some seal applications. However, because free silicon is more susceptible to corrosion and erosion, applications of reaction bonded materials must be chosen carefully.

A. Sintered silicon carbide is an alpha base polycrystalline material that does not contain free silicon. While it exhibits excellent corrosion resistance, it is brittle and must be handled carefully.

Faces should be selected on the basis of corrosion resistance, hardness and PV. The common high quality face materials available are Carbon, Aluminum Oxide (ceramic), Stellite™, Silicon Carbide and Tungsten Carbide. All are chemically inert where stainless steel is acceptable. If you need to use a special alloy seal, avoid the use of Stellite™ and nickel bound Tungsten Carbide. Aluminum Oxide and Silicon Carbide are very hard with Silicon Carbide being the hardest. PV ratings of common face combinations are shown:

Carbon vs. Cast Iron	50,000
Carbon vs. Aluminum Oxide	500,000
Carbon vs. Stellite	750,000
Carbon vs. Tungsten Carbide	1,000,000
Carbon vs. Silicon Carbide	5,000,000
Tungsten Carbide (TC) vs. TC	250, 000
Silicon Carbide vs. Silicon Carbide	350,000
RB Silicon Carbide vs AS Silicon Carbide	750,000
Silicon Carbide vs. Tungsten Carbide	500,000

B. Secondary Seals
(O-Rings and Gaskets)

The two most critical characteristics of secondary seals are temperature limitations and chemicals compatibility to the pumped product.

If you are selecting a seal with an elastomeric o-ring secondary seal, consult the selection guide from a major o-ring manufacturer. These guides show temperature and chemical limitations on all major rubber compounds produced.

When selecting secondary seals, carefully consider whether or not they will be operating in a dynamic or static manner. Some compounds are not recommended for dynamic applications. Other compounds swell in certain applications, causing seal hang up.

Some common secondary seal materials are Buna-N rubber, Viton™ rubber, EPR (ethylene propylene rubber), neoprene rubber, Kalrez™, Aflas™, Creavey™, Teflon™, Grafoil™ and silicon.

Buna-N

A copolymer of butadiene and acrylonitrile with excellent resistance to petroleum products and wide acceptance in water applications. Buna-N has low resistance to ozone, sunlight and weather and should be stored in a protected area with limited exposure to sunlight or ozone generating electrical equipment. Temperature range is -40°F to +225° (-40°C to +120°C). FDA and USDA grades are available. Most commonly referred to as nitrile.

Creavey

This is a trademark name for a TFE encapsulated elastomeric o-ring made of Viton or Silicon. This product offers the elastomeric properties of the o-ring and the chemical and heat resistance qualities of TFE. It is not recommended for dynamic service. Temperature range is -40°F to +350°F (-40°C to +180°C).

Chemraz

A member of the perfluoroelastomer polymer family, polymer chemists describe the base (raw) perfluoroelastomer as polymers of three or more monomers in which all hydrogen positions have been replaced with fluorine. The outstanding resistance of perfluoroelastomer vulcanizates to heat and most chemicals and solvents is the result of this state of complete fluorination. Temperature range of -20°F to +450°F (-20°C to +232°C).

Ethylene propylene (EPR)

Ethylene propylene (EPR) is a copolymer made from ethylene and propylene monomers. EPR has excellent corrosion resistance to diluted acids and alkalis, ketones, alcohols, water, steam and phosphate ester hydraulic fluids. It is not recommended for use with hydrocarbon based fluids or diester base lubricants. Temperature range is -40°F to +350°F (-40°C to +180°C).

Grafoil

This graphite expanded material is formed into rings to provide a secondary static seal in high temperature applications. Temperature range is -400°F to +1300°F (-240°C to +700°C). In addition, this material displays excellent corrosion resistance.

Kalrez

This perfluoroelastomer is a copolymer of tetrafluoro-ethylene and perfluoromethl vinyl ether. Kalrez has many of the elastomeric properties of Viton and the chemical and heat resistance of TFE. Kalrez has excellent corrosion resistance to solvents, inorganic and organic acids and bases, strong oxidizing agents, halogen compounds, hot mercury, chlorine, fuels and heat transfer fluids. It is not recommended for use with amines and fluorinated solvents. Temperature range is -0°F to +500°F (-18°C to +260°C).

Neoprene

This is a homopolymer of chloroprene and chlorobutadiene. It is used for sealing refrigerants, such as Freon™ and ammonia. Neoprene has a unique resistance to petroleum lubricants. Temperature range is -40°F to +225°F (-40°C to +120°C).

Solid TFE

TFE has made such an outstanding record as a packing and gasket material that it is often considered for o-ring applications. The major handicap to the use of TFE as an o-ring has been its inelasticity in comparison to rubber. This often creates a problem were a TFE o-ring must be stretched and snapped into a groove or where compression set may occur. Virgin TFE is completely unaffected by all common fluids. Temperature range is -100°F to +400°F (-73°C to +200°C).

Aflas™

A TFE based elastomer available to meet a variety of industrial and processing requirements. There are five grades of Aflas™; all grades provide the same exceptional heat range to 500°F in superheated steam, most chemicals, and electrical resistance properties, differing primarily in the physical properties. Temperature range is -100°F to +500°F (-73°C to +260°C).

Viton

Also known as fluorocarbon.
Applications include petroleum oils. diester base lubricants, silicon fluids, halogenated hydocarbons, *water and low temperture steam only to 250F,* and a wide variety of acids. Viton is not acceptable in applications with keytones, anhydrous ammonia, amines, hot hydrofluoric acid,sulfuric acid or other strong caustics.
Temperature range is 0 F to & 400F (-18C to & 200C) Special grades of Viton are available that meet FDA and USDA requirements.

C. Mechanical Seal Metallurgies

The metal parts of a seal are selected for their strength and corrosive resistance and their ability to drive the seal head. If the pump's wetted parts are made of iron, steel, or stainless steel, the body of the seal should be made out of stainless steel. If the pump is a special alloy material, try to use the same alloy for the seal. An outside seal should be considered if the wetted parts are non-metallic, or if they are showing signs of excess corrosion.

Potential materials are Alloy 20 stainless steel, Hastelloy C, Inconel 718 and X-750, Monel and titanium. In addition to Type 316 stainless steel, a heat-treatable stainless steel grade AM350 is a common metal bellows material.

Bellows and springs can be made from stainless steel, but may be subjected to stress corrosion in the presence of chlorine, fluorine, bromine, iodine and some caustic fluids. Hydrogen embrittlement can also affect stainless steel when used in sulfur-based applications. Use Hastelloy C, Inconel, or Monel in these fluids.

Types 304 and 316 Stainless Steel

Grades 304 and 316 stainless steel are the metals commonly used in the production of rotary seal parts, glands and sleeves. While both alloys are similar in composition, 316 stainless steel has slightly higher resistance to chemical attack. It is rarely used for welded metal bellows seals.

Alloy 20 Stainless Steel

Alloy 20 stainless steel is used in applications requiring greater corrosion resistance than either 304 or 316 stainless steel. Sealol uses Alloy 20 for all the metal components in its Type 680 welded metal bellows seal. Alloy 20 is specially designed for sulfuric acid service.

AM350 Stainless Steel

Grade AM350 stainless steel is heat-treatable stainless steel having corrosion resistance similar to Grade 304 stainless steel.

This material is the most commonly used material for welded metal bellows. Properly heat treated, AM350 is a high strength metal available to the bellows manufacturer with three times the strength of either Alloy 20 or Hastelloy.

Hastelloy C

Hastelloy C is a nickel-molydenum material and is one of the most universally used, corrosion-resistant alloys. It has exceptional resistance to a wide variety of chemical process environments.

Inconel

Inconel 600, 718 and X750, nickel-chromium iron alloys are excellent for use in corrosive environments at elevated temperatures.

These heat-treatable alloys retain excellent mechanical properties over a wide temperature range. Inconel 600 is normally used for seal end fittings and Inconel 718 and X750 are used as bellows plate material. Inconel 718 is the highest strength bellows material available and is also more resistant to stress corrosion cracking than Inconel X750.

Monel

Monel is a nickel-copper alloy used for bellows, springs and other seal hardware material when its special corrosion-resistant properties are required. Because of its low iron content, Monel is more resistant to corrosion in some applications than Alloy 20 stainless steel.

Titanium

Commercially pure Titanium CP Grade 2 has been successfully used in bellows applications. It has excellent corrosion resistance combined with a high strength/weight ratio. Applications include ammonium chloride, calcium hypochlorite, chlorine, chlorine dioxide, nitric acid, and potassium chloride.

MECHANICAL SEAL METALLURGIES

Material	Composition % of Each Element by Weight	Ultimate Strength (psi)	Yield Strength (psi)	Modulus of Elasticity
304SS	18-20 Chrome, 8-12 Nickel 64-71 Iron, .08 Carbon, <1 Silicon <2 Manganese <.03 Sulfur <.45 Phosphorus	90,000	37,000	28,000,000
316SS L	16-18 Chrome, 10-14 Nickel, 2-3 Moly 62-71 Iron, .08 Carbon, <1 Silicon <2 Manganese <.03 Sulfur <.045 Phosphorus	85,000	70,000	28,000,000
AM350 Annealed	75 Iron, 16.5 Chrome, 4.3 Nickel 2.75 Moly, .8 Manganese, .25 Silicon .1 Nitrogen, .08 Carbon	145,000	60,000	26,700,000
AM350 Heat Treated	75 Iron, 16.5 Chrome, 4.3 Nickel 2.75 Moly, .8 Manganese, .25 Silicon .1 Nitrogen, .08 Carbon	186,000	160,000	29,000,000
Alloy 20	19-21 Chrome, 32.5-35 Nickel, 2-3 Moly 37-43 Iron, <.07 Carbon, <1 Silicon <2 Manganese 3-4 Copper <.035 Phosphorus <.035 Sulfur, <1 Columbium and Tantalum	93,000	46,000	28,000,000
Hastelloy C™ 276	50-63 Nickel, 14.5-16.5 Chrome 15-17 Moly, 4-7 Iron, <.08 Carbon <1 Silicon, <1 Manganese, 3-4.5 Tungsten <2.5 Cobalt, <.04 Phosphorus <.03 Sulfur, <.035 Vanadium	118,000	51,000	30,000,000
Inconel™ 600	72 Nickel (with Cobalt) 14-17 Chrome 6-10 Iron, <.15 Carbon, <1 Manganese <.015 Sulfur, <.05 Silicon, <.5 Copper	125,000	100,000	30,000,000
Inconel™ 718 Annealed and Age Hardened	50-55 Nickel (with Cobalt) 17-19 Chrome 12-24 Iron, 4.75-5.5 Columbium and Tantalum, .2-.8 Aluminum .65-1.15 Titanium, .2-.8 Aluminum <1 Cobalt, <.3 Copper, <.006 Boron <.08 Carbon, <.35 Manganese <.35 Silicon, <.015 Phosphorus <.015 Sulfur	200,000	175,000	30,000,000
Monel™	63-70 Nickel, 24-31 Copper <.30 Carbon <2.50 Iron, <.50 Silicon, <2 Manganese <.024 Sulfur, Trace of Cobalt	80,000	35,000	26,000,000
Titanium	99.2 Titanium, <.10 Carbon <.2 Nitrogen, <.015 Hydrogen	65,000	55,000	15,000,000

GLANDS

Gland jackets provide a cavity for containing and circulating a heating or cooling fluid near the seal. This is analogous to what the pump jacket does for the liquid deeper in the seal chamber.

Glands perform the functions of closing off the end of the seal chamber and holding the stationary seal face in position.

Sealol glands are designed and precision-manufactured to minimize distortion of the seat face. Glands may have one or a combination of connections.

A. Circulation/Flush Glands

These glands are designed to allow circulation of fluid at the seal faces and used to provide lubrication, heating or cooling, and cleansing action.

B. Vent and Drain Glands

Vent and drain glands with throttle bushings are constructed to enclose the area outside the seal. In the event of seal failure, fluid can escape through the vent fitting or drain out of the drain port. Close clearance between the throttle bushing and the shaft (.025 diametrical) allows only relatively small amounts of fluid to pass up the shaft toward the coupling or bearing. Vent and drain glands also allow the collection of any leakage to be piped from the drain fitting to a scavenge sump or to a drain.

C. Quench Glands

Glands of this type are fitted with vent and drain ports for the introduction and removal of quench fluids. The quench fluid can be contained by a throttle bushing, lip seal, face seal, auxiliary packing, or a quench seal. The method used for a particular application depends upon the nature of the quench fluid, its pressure and temperature and tolerance of leakage. A throttle bushing is normally recommended when steam is the quench fluid.

Quench fluids are introduced to the low pressure side of seals for four different reasons.

a. As a cooling fluid to remove heat from the seal and from the pumped product in the seal chamber.

b. As a heating fluid to add heat to the pumped product in the seal chamber.

c. As a cleansing fluid to remove any accumulation of coking of crystallizing matter that my develop at the seal face's I.D.

d. As a wetting agent to dissolve leakage across the faces.

D. Barrier Systems

Another design of a circulation-style gland is used for barrier systems on double and tandem seal arrangements. This gland normally has two ports arranged 180° apart and labeled "Circulation In" and "Circulation Out."

The outlet connection should be drilled on a tangential basis that relates to the seal's rotation. This will gain maximum effect from centrifugal action in circulating the barrier fluid back to the reservoir.

The optimum arrangement would have the inlet connection on the bottom of the gland and the outlet connection on the top.

E. Gland Piloting

A piloted gland is required in API applications to ensure the seal is centered correctly and to prevent movement of the gland due to improper installation or vibration. The piloted gland may be centered in the seal chamber bore by means of a gland ring or a stationary seat insert. The gland may also be centered by a machined step on the gland plate. This step will match the step in the pump body outside the pump seal chamber.

In most ANSI cartridge seal designs, piloting is accomplished by using centering clips or rings.

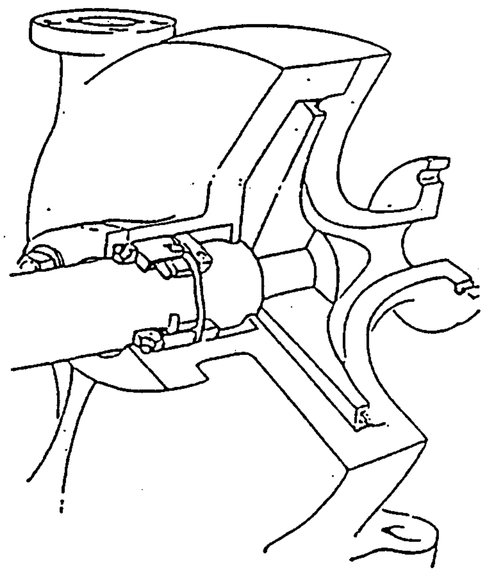
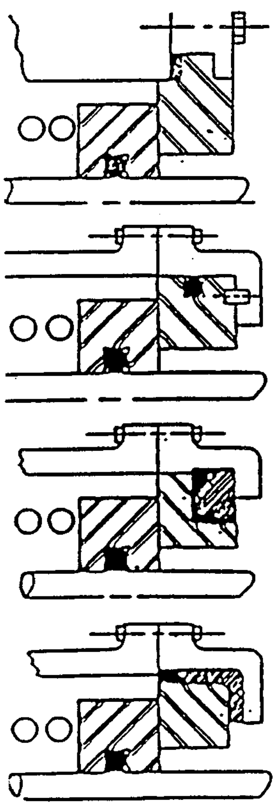


Figure 30



(a) CLAMPED

(b) O RING MOUNT

(c) CUSHION

(d) CUP

Why Seals Fail

Mechanical seals such as the one in *Figure 30* should last until the carbon face wears out. If the seal starts leaking before that happens and the pump must be shut down for repair then the seal has failed. Two fundamental problems underlie many seal failures: 1. Leakage clogs the outside section of the seal, causing hang up. Leakage is caused by face seperation and the next section discusses some of the reasons by face separate.
2. Heat.

Hysteresis

When a stationary ring is not perpendicular or square with the shaft, the sliding elastomer must move back and forth on each revolution. The amount of motion depends on how much misalignment. Axial motion of the seal can also be caused by:

1. A BENT SHAFT.
2. WORN BEARINGS.
3. MISALIGNED COUPLING.
4. PIPE STRAIN.
5. IMPELLER UNBALANCE.
6. SHAFT DEFLECTION.

The seal is alternately pushed away from the face and back toward it. Anything that impedes that motion contributes to face separation. In seal design the major impediment to back and forth motion (in addition to inertia) is the secondary seal. The more drag it has on the shaft the better the changes the faces will stay apart. The delay or lag between the axial motion of the seal back and forth is called hysteresis. In seal usage hysteresis also refers to the measurable drag of the secondary seal on the shaft and can be expressed in ounces or pounds.

Face Separation

The faces of a seal are normally flat to less 11 millionths of an inch and separated by a film up to about 30 millionths of an inch. This leaves space between the faces of less than a micron and so the faces should act as a natural excluder of particles which could cause abrasive damage. When the faces are moved axially on each revolution there is a tendency for them to separate distances measured in thousandths of an inch. From .005 to .030 misalignment is not at all uncommon.

A pump rotating at 3600 rpm moves the seal over 10,000,000 times a day! The separation allows large particles to get between the faces where they can embed in the soft face and cause grinding on the hard face and consequent failure. The leakage out may cause clogging of the sliding secondary seal where it is supposed to be clean. Once the secondary seal sticks to the shaft and is unable to slide to compensate for wear, the seal is bound to fail.

Mechanical Seals

The most important concept in balance is that force can be calculated by multiplying a pressure by an area. Therefore, if the pressure is constant, the force can still be changed by increasing or reducing the area upon which the pressure acts. An example of this can be seen at the local automobile repair shop. The hydraulic lifts in these shops use pressure on the inside diameter of a cylinder to raise the cars. The examples below show how the cylinder diameter can be varied to change the amount of weight the lift can raise.

Car Weight	2500 lbs (1134 Kg)
Hydraulic Cylinder Diameter	8 in. (20 cm)
Pressure	40 psi (2.9 Kg/cm7)

In this example, the car cannot be raised because the total force acting to lift the car is less than the weight of the car as shown below:

Area = 3.14 x 1/2 Diameter2
Lift = Pressure x Area
=40psix3.14x42
(2.9 Kg/cm2 x 3.14 x 102)
= 2010 lbs (911 Kg)

Assuming the pressure remains constant, this car can be lifted if the hydraulic lift cylinder diameter is increased to 9 inches (23 cm).

Lift $= 40$ psi x 3.14 x 4.5^2
(2.9 Kg/cm^2 x 3.14 x 11.5^2
= 2543 lbs (1204 Kg)

This example shows that relatively small pressures can create high forces when large enough areas are involved.

In a mechanical seal, these forces can cause excessive loading of the faces if the seal is not designed properly. This will break down the lubricating film between the faces causing excessive wear and heat generation.

Unbalanced Seal

Figure 9 is a drawing of an unbalanced pusher seal. It is unbalanced because the area upon which pressure is acting to close the faces (Area A) is greater than or equal to the area upon which pressure is acting to open the seal faces (Area B). The use of an unbalanced seal can lead to excessive face loads and heat generation as shown in the example below:

Spring Load	60 lbs (27 Kg)
System Pressure	150 psi (10.5 Kg/cm2)
Rotating Head Area (Area A)	2 sq. in.(12.9 cm2)
Seal Face Contact Area (Area B)	1 sq. in.(6.45 cm2)

The closing force of this seal in pounds can be calculated as follows:

Closing Force = Pressure x Area A + Spring Force
= 150 psi x 2 sq. in. + 60 lbs
(10.5 Kg/cm2 x 12.9 cm2 + 27 Kg)
= 360 lbs (152 Kg)

There is a pressure drop across the seal faces. The O.D. of the seal faces typically sees seal chamber pressure, while the I.D. of the faces typically sees atmospheric pressure. The pressure profile across the seal faces is not constant and varies with product, hydraulic load, and the vapor phase conditions, as well as other factors.

For this example, a linear pressure drop across the seal faces is assumed. Thus, the average pressure between the faces is 75 psi. The Opening Force can therefore be calculated as follows:

Opening Force = Pressure x Area B

= 75 psix 1 sq. in.
(5.25 Kg/cm^2 x 6.45 cm^2)

= 75 lbs (34 Kg)

In this example, the closing force is 285 lbs (128 Kg) I greater than the opening force. This high face load will cause high wear rates and excessive heat generation.

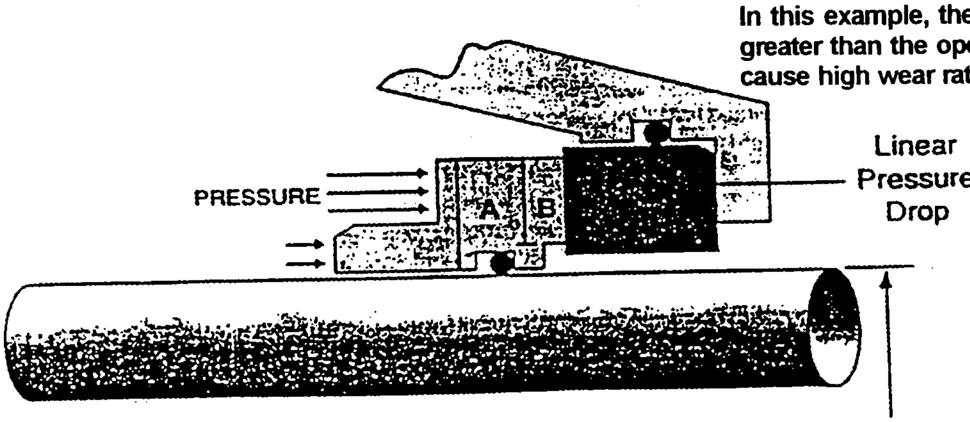


Figure 9

Mechanical Seals

However, the closing force can be decreased by increasing Area B. This Area B must be adjusted in order to achieve an optimum closing force. This is commonly referred to as "balancing" a seal. The balance ratio is the ratio of the closing area (Area A) divided by the opening area (Area B) and is commonly used to simplify the face load calculation.

Balancing the individual seal designs will be discussed later.

Pusher Seal Balance

As explained earlier, balance is used to reduce the Net Closing Force on the seal faces. Remember, this is done by changing the area upon which pressures act.

In order to calculate the balance ratio of a seal, the location of the "effective balance diameter" must be determined. In a pusher seal, it is the diameter of the shaft or sleeve upon which the O-ring rides. Once the effective balance diameter is determined, the balance can be defined as a ratio of seal face contact area above the effective balance diameter to the total seal face contact area.

Since the system pressure and the spring load remain essentially constant, the areas upon which pressure are acting must be changed in order to balance conventional pusher seals *(figure 10)*.

The universally accepted method used to balance a pusher seal is to machine a step in the pump shaft or sleeve to reduce the area of the seal face exposed to the pump system pressure. The use of a balanced

I seal reduces the force acting on the seal faces as shown below. Please note that this is the same example as the one used to depict an unbalanced pusher seal. The only changes are the step in the sleeve which reduces the area of the seal face exposed to pump pressure to .5 sq. in. and the spring load has been reduced to 30 lbs.

Spring Load	30 lbs (13.6 Kg)
System Pressure	150 psi (10.5 Kg/cm^2)
Rotating Head Area (Area A)	.5 sq. in. (3.23 cm^2)
Seal Face Contact Area (Area B)	1 sq. in. (6.45 cm^2)

The closing force of this seal in pounds can be calculated as follows:

Closing Force = Pressure x Area A + Spring Force

= 150 psi x 5 sq. in. + 30 lbs
(10.5 Kg/cm^2 x 3.23 cm^2 + 13.6 Kg)

= 105 lbs (47.5 Kg)

Once again it is assumed that the pressure drop across the seal faces is linear, resulting in an average pressure between the faces of 75 psi. The Opening Force can thus be calculated as follows:

Opening Force = Pressure x Area B

= 75 psix 1 sq.in.
(5.25 Kg/cm^2 x 6.45 cm^2)

= 75 lbs (34 Kg)

In this example, the hydraulic pressure acting to close the seal lace is completely offset by the pressure acting to open the faces. The resultant face load (Opening Force subtracted from the Closing Force) of 30 lbs is the load provided by the spring.

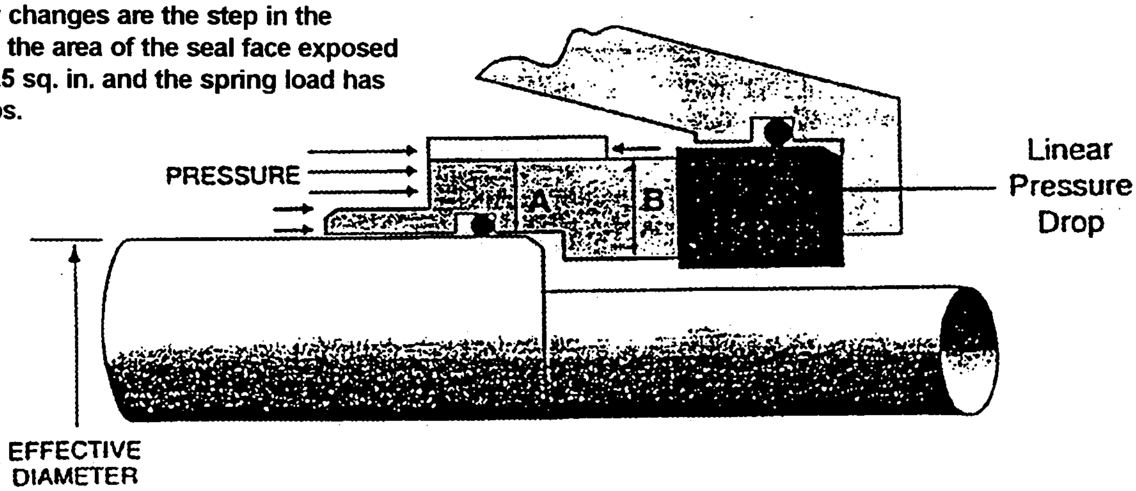


Figure 10

Mechanical Seals

A seal that relies only upon spring pressure to keep the faces closed can be susceptible to fluid system upsets, which could cause the faces to pop open. Seals are therefore typically designed such that Area A is 70% of Area 8 (giving the 70% balance) rather than the 50% given in the example.

$$\text{Balance Ratio} = \frac{\text{Closing Area}}{\text{Opening Area}}$$

BASIC MECHANICAL SEAL DESIGN

As mentioned, mechanical seals are classified as one of two types: Pusher and Non-Pusher. Non-Pusher seals include rubber diaphragm, TFE bellows, and metal bellows seals.

Pusher Seals

The pusher seal is so called because it uses a secondary packing that must be ~pushed" back and forth axially on each shaft revolution to maintain seal face contact and compensate for misalignment. This ~dynamic" secondary seal usually slides on a shaft or sleeve; however, in a few designs, the packing slides on an integral seal component. Pusher seals are available in a wide variety of types and styles, which we will cover in this section.

Advantages

The pusher seal is the most common seal on the market. It is a popular seal in the industry because of its low manufacturing cost and adaptability to special design configurations. Pusher seals are relatively inexpensive because they do not have complicated machine requirements and tight tolerances. They are available in a wide variety of equipment. Made of elastomers or TFE, the secondary seals can only be used in moderate temperature ranges.

Disadvantages

Pusher seals have some major design flaws that can lead to early seal failure. The most common failures are spring pocket clogging, and secondary packing hang-up. The constant back-and-forth action of the dynamic secondary seal often allows din particles to embed themselves in the packing. Instead of having smooth surfaces sliding on each other, the secondary seal packing now acts like sandpaper or a grinding wheel and abrades the shaft or sleeve.

This abrasion is called "shaft fretting". The rouge, surface left by fretting hinders the secondary seal's movement and holds the seal faces open, resulting in premature seal failure.

The pusher seal's closing mechanism is typically a coil spring, with either a single-coil design or a multiple-spring arrangement.

Closing Mechanisms

Single-Spring (Figure 11)

A single-coil spring's advantages include a larger diameter wire, lower spring rate, and less susceptibility to clogging. Disadvantages include the need for a different spring for each shaft size, a longer axial seal length, and a tendency toward uneven face loading. Large inventory requirements to stock the springs in various sizes, loads and materials make it difficult to standardize within a manufacturer.

Multiple-Spring (Figure 12)

Multiple-spring design requires far less operating length than the single-spring design. In addition, the multiple-spring design uses only one or two different springs and the face load can be changed by varying the number of springs used. This design's disadvantages include higher susceptibility to corrosion because of smaller wire diameter, less tolerance to installation errors, and a strong tendency to dog in the spring pockets. Multiple-spring seals are typically short-lived in any application with suspended solids in the fluid.

Mechanical Seals

Secondary Packings

1 O-Flings

• O-Ring Seal *(Single Spring)*

Figure 11 shows an O-ring single-spring seal design. In this seal, a large, single spring drives the seal and keeps the faces pushed together. The spring can absorb some torsional stress from the seal faces without breaking the drive pins, as may happen in other seal designs. Because of its simple rugged nature, the spring seldom clogs.

This seal has a few disadvantages. The single spring's wind-up feature makes these seals uni-directional. If a seal with the wrong directional spring is installed, the seal will not have a positive rotational drive and may fail prematurely. This type of seal also requires additional axial space. Some pumps designed for packing do not have the room to hold a seal with a single spring.

· Built-in Sleeve

Figure 12 is an example of an O-ring seal with an O-ring sliding on a seal component instead of the shaft sleeve. in this design, the multiple springs are not exposed to the pumped product and are not as susceptible to clogging, unless some leakage occurs across the seal faces or past the dynamic O-ring between the barrels.

This seal has a pressed-in carbon rotary face. Temperature range is determined by the interference fit on the face insert, metallurgy of the seal, and secondary packing materials.

This seal demonstrates two ways to use an O-ring. 1) The O-ring on the shaft is a static or non-moying design that can use heavy compression. 2) The O-ring on top is dynamic, rubs on a smooth surface, and requires less compression. The dynamic O-ring between the internal barrel and the shell must not swell and must be free to roll or slide if there is a stationary seal ring or other pump part misalignment. This design is subject to corrosion, O-ring hang-up, and fretting on the barrel.

O-Ring Seal
(Single Spring)

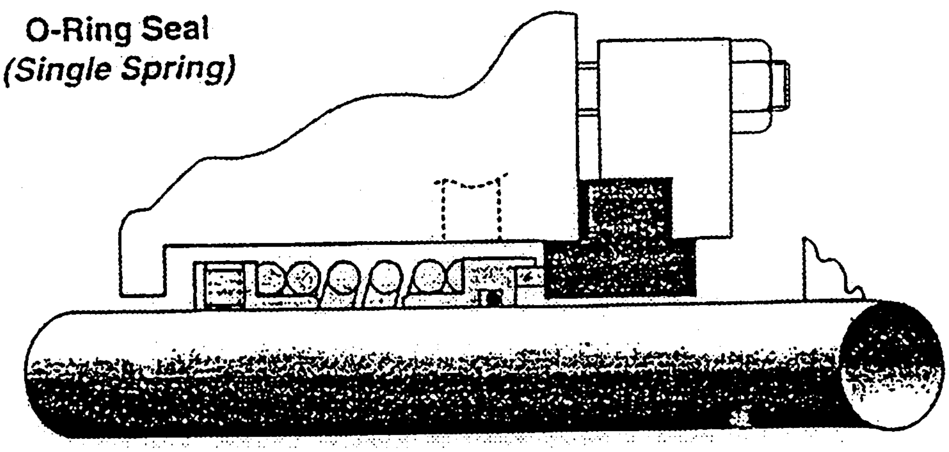


Figure 11

O-Ring Seal
(Built-In-Sleeve)

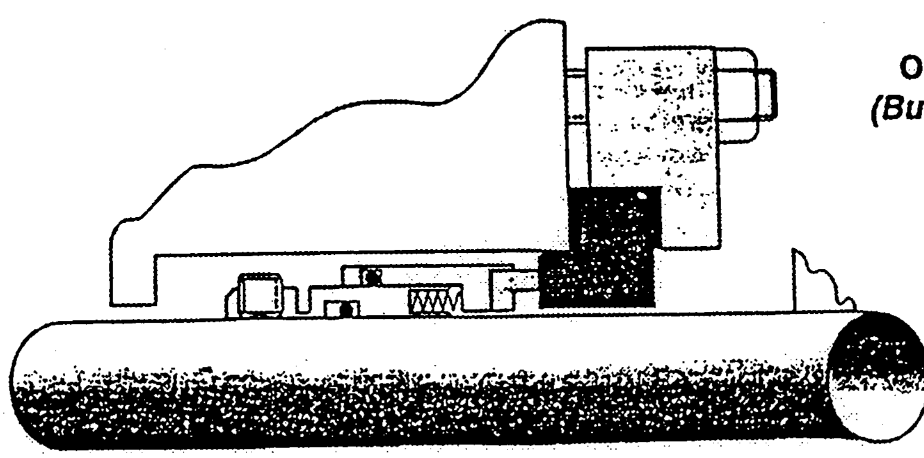


Figure 12

2 TFE Wedges

The most common wedge-shaped secondary seal is a multi-spring seal with a rotating carbon face *(Figure 13)*. The stationary seal seat is usually a T-clamped ceramic or O-ring block ceramic design.

The springs and fluid pressure push the secondary seal against the carbon and the shaft. The seal shell, which retains the springs, carbon, and other parts, is available in different forms. Machined or stamped, the seals of this type can appear quite different. Holes in the shell allow fluid to circulate through the spring area for a cleaning effect.

Because the wedge only takes pressure from one direction, a wedge secondary seal should not be used in applications subjected to vacuum service, unless it is mounted outside the chamber.

Any misalignment in the pump or stationary ring will cause the TFE wedge to move back and forth on each revolution resulting in shaft or sleeve fretting. Because the wedge-shaped seal does not easily follow axial motion of the shaft, any reverse movement may cause face separation and excessive face loading. In addition, the seal may lock onto the shaft causing seal hang-up and subsequent seal damage or failure. Excessive loading on the shaft also contributes to shaft fretting.

3 V-Rings

This multi-spring, rotating seal uses two TFE V-rings for the secondary seal *(Figure 14)*. The rotating face and pusher assembly are separate, so the seal can be assembled on the shaft one piece at a time, which helps prevent damage to the V-rings.

Torque is transmitted from the shaft, through the set screws and drive pins, to the rotating seal head.

The multiple springs keep the V-rings loaded in a shorter axial space than a single spring, which allows this design to fit into pumps with limited space requirements.

In most cases, the rotating face is made of ceramic, stainless steel or tungsten carbide to handle the stress caused by the loaded V-rings. Solid ceramic rotating faces should be avoided because of the possibility of catastrophic fracture from the ceramic being hit by drive pins.

The stationary seal ring is clamped carbon, machined to the dimensions of the seal chamber. A plain carbon face also may break unless it is held in compression by a pressed fit or other arrangement. To contain pump pressure, the face is quite often wide. Standard dimensions are not common, so carbons must be custom made for each pump. Users of this seal often have problems with shaft fretting.

Seals with V-rings cannot take a reverse pressure and should be turned around if sealing a vacuum or if the seal is mounted outside the seal chamber.

4 U-Cups

The U-Cup single-spring design *(Figure 15)* is a dynamic seal that seals with increasing pressure. Torque is transmitted from the shaft sleeve through a pin to the spring cage.

The large single spring keeps the faces together and loads the U-Cup by means of a spreader ring.

The stationary seal ring design is flexible and, unlike a clamped-stationary seat, less likely to cause movement of the elastomer in the seal from misalignment.

Capable of retaining high pressures (more than 400psi/28 bar), the U-cup seal is less likely to clog than multiple-spring designs.

The materials for rotating hard faces are often limited to Stellite™, tungsten carbide, or silicon carbide.

The U-cup seal cannot retain a reverse pressure and should not be used as an inside seal for a vacuum or as an outside seal if a pressure is being sealed.

TFE Wedge Seal

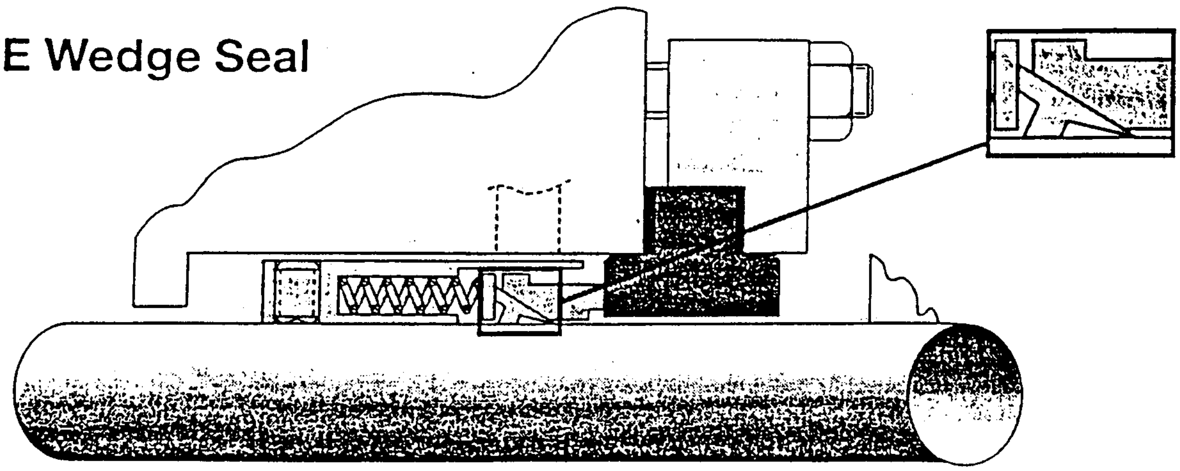


Figure 13

V-Ring Seal

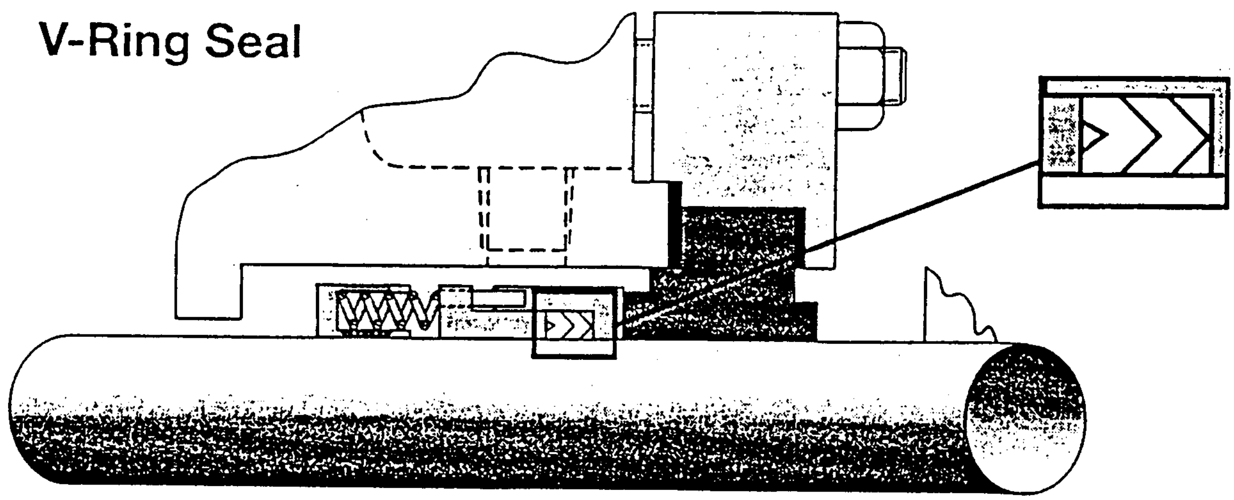


Figure 14

U-Cup Seal

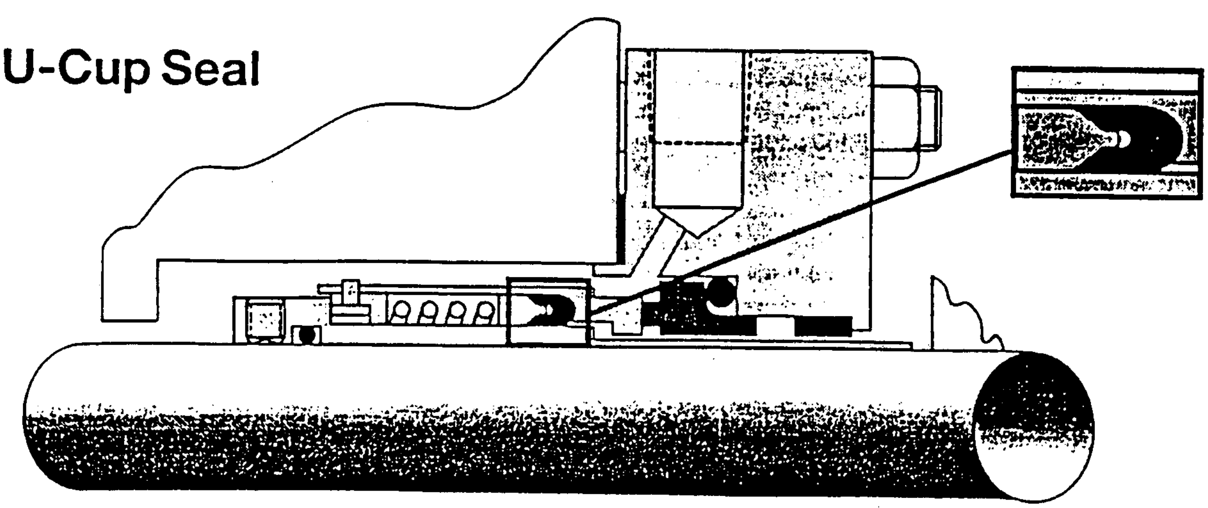


Figure 15

5 BELLOWS SEALS

Bellows seals, sometimes referred to as non-pusher seals, share a common design feature. The secondary shaft seal in all bellows seals is static or non-moving, and does not cause shaft or sleeve fretting, or lead to seal hang-up.

Low Temperature Sealing

The low temperature designation for mechanical seals has traditionally been limited to applications where the product does not exceed 400°F (200°C). This allowed seal manufacturers to use elastomeric or TFE secondary packings.

Metallic components are generally stainless steel or Hastelloy. Rotas faces can be carbon, tungsten carbide or silicon carbide. Stationary seats can be ceramic, cast iron, Ni-resist, tungsten carbide, or silicon carbide.

Rubber Diaphragm Seals

This is the world's most common seal type (Figure 16). It uses the rubber diaphragm as a friction drive mechanism. The rubber diaphragm is mounted directly on the pump shaft or sleeve. When the rubber comes in contact with a liquid, a slight swelling (3% to 5%) will occur. Since the rubber is trapped on the outside diameter by a drive band, the swelling takes place toward the shaft.

Relatively quickly, the rubber will vulcanize itself to the shaft and provide the necessary drive. Axial flexing occurs in the thinner goose neck area of the diaphragm. Since no sliding takes place between the shaft and the rubber diaphragm, no shaft fretting occurs.

In addition to providing the drive mechanism for this seal, the rubber diaphragm functions as the secondary shaft seal. This seal uses a large single spring that keeps the faces together when the pressure is low and positions the seal initially. The spring will not clog easily. However, it may cause more uneven lace loading than multiple spring seals.

Although this is the most popular seal design, it has several maintenance shortcomings.

Installation of this seal is difficult, The requirements are as follows:

1. The shaft diameter must be within a tolerance of -.002 and +.000".

2. The shaft or sleeve finish should be between 40-50 micro inches (RMS).

3. The rubber diaphragm can be lubricated with a petroleum-based lubricant, such as light oil, or soapy water. Only soapy water should be used as a lubricant in EPR (ethylene propylene rubber) applications.

As mentioned earlier, the rubber swells when it comes in contact with the fluid, so it is necessary to work quickly and locate the seal in the correct position before the swelling takes place.

Typically the allowable time from seal lubrication to complete pump assembly is 10 to 20 minutes. Often, users will try to substitute a silicon base lubricant for the oil.

However, this may prevent the rubber from vulcanizing to the shaft and allow the shaft to spin within the seal. Early seal failure will occur because of fretting and heat generated in the rubber diaphragm.

Selection of the materials of construction should be considered on the basis of chemical compatibility, temperature, and fluid pressure. Incorrect selection will drastically affect the seal's performance.

TFE Bellows Seal

The TFE bellows seal (Figure 17) is normally used when there are no readily available metals that can withstand the product's corrosive environment. The bellows core is manufactured from a short tube of TFE and glass-filled TFE with both ends machined f rom the glass-filled portion. The glass is added to the virgin TFE to provide more rigidity and reduce the "Cold Flow" characteristics of TFE.

Cold Flow occurs when TFE is clamped to the shaft, or squeezed as in a gasket, causing the TFE to extrude from under the clamp. The extruding takes place until there is no material left to squeeze. Unfortunately, the added glass only postpones Cold Flow. It does not solve the problem and the seal eventually slips on the shaft. However, slippage can be prevented by adding a clamp ring behind the seal.

Mechanical Seals

Since the seal is externally mounted, the metal components do not come in contact with the pumped product. As with all externally mounted seals, any seal failure results in product leakage that sprays out from the rotating seal. This hazard is compounded by the fact that the TFE bellows seal is normally used in highly corrosive and dangerous fluids.

For safety reasons, a shrouded gland is normally recommended to contain any leakage should a seal failure occur.

TFE Bellows Seals are pressure balanced. Careful consideration should be given to the system pressure, temperature, and rotational speed when selecting an outside TFE Bellows Seal.

Rubber Diaphragm

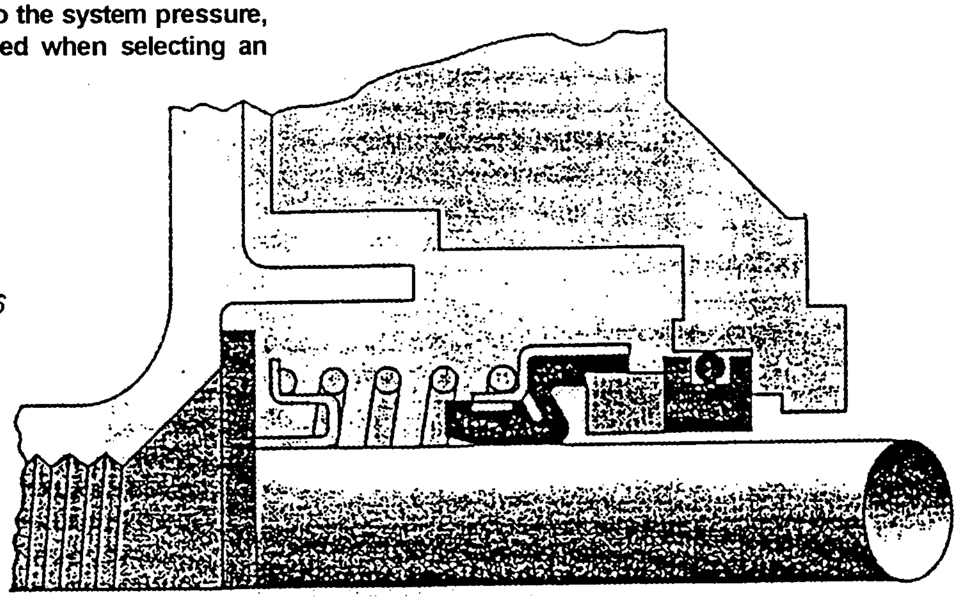


Figure 16

TFE Bellows Seal

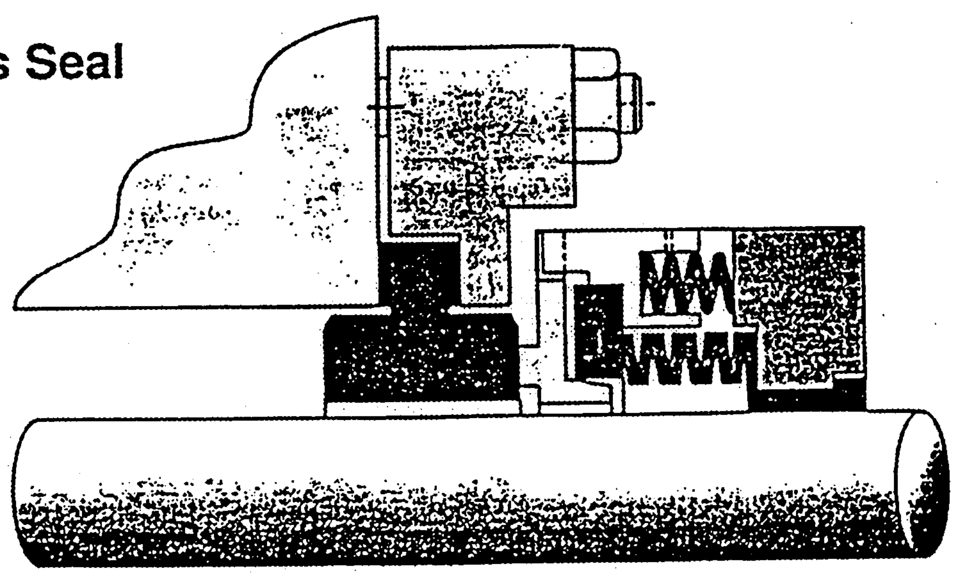
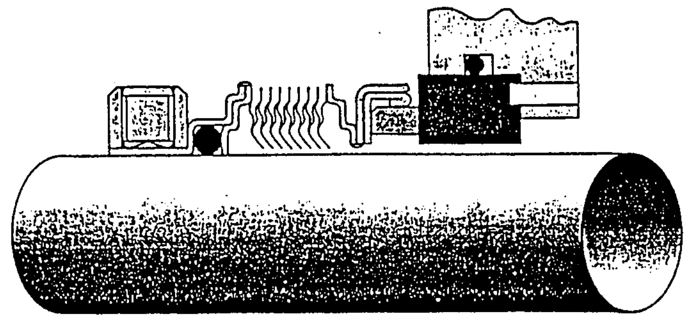


Figure 17

Mechanical Seals

6 Welded Metal Bellows Seals



The welded metal bellows seal consists of a series of stamped plates welded together at the inside and outside diameters.

Although bellows plates can be produced from any weldable material, the most common are AM350, AM350 Heat-Treated, Alloy 20, Hastelloy C™, Inconel™, Monel™, and Titanium.

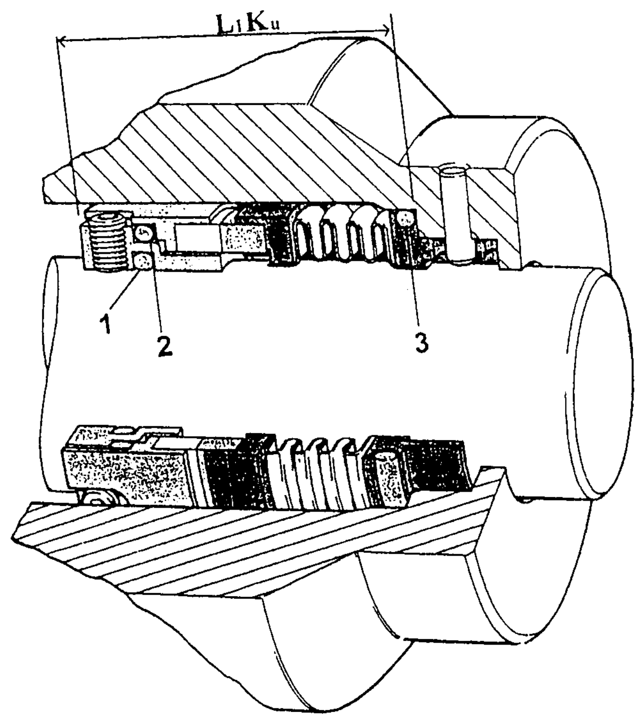
The individual stamped plates are joined at their inside diameters to create a convolution. The convolutions are then joined at their outside diameters to form a bellows core. Plate thickness and number of convolutions vary with each manufacturer. Current designs typically offer plate thickness from .0004" to .010", and cores with 4 to 20 convolutions.

The welding process used is called T.I.G. (Tungsten Inert Gas) Welding and uses a mixture of helium and argon gas. The gas isolates the welding area from the atmosphere and eliminates the carbonizing or oxidizing problems that occur when welding in an open environment.

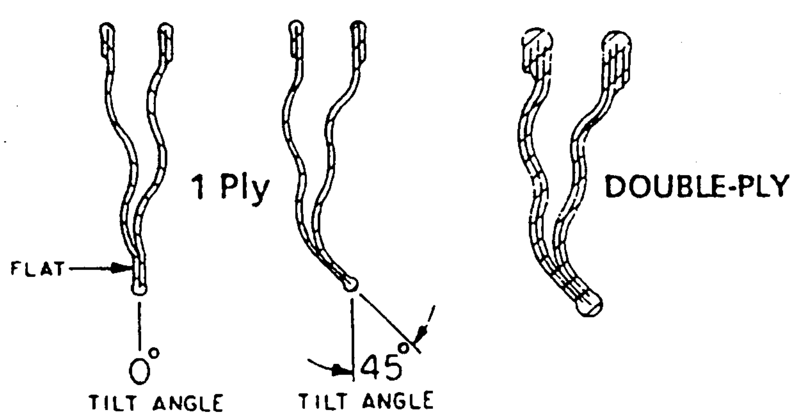
When more convolutions of a given plate thickness are used, the seal has a lower spring rate which allows it to compensate for face wear, pump end play, installation inaccuracy, shaft growth, and mating ring tilt. Thicker plates are easier to stamp and weld, but create high total face loads for the seal assembly and are less forgiving in installation and operation.

To produce a seal head, the bellows core is welded to stamped or machined end fittings. The seal face is held in a machined end fitting by an interference fit. The other end fitting contains the drive collar and the secondary shaft seal.

7 The latest generation of mechanical seal technology, the stationary rolled bellows.



Bellows Plates



Mechanical Seals

SEAL INSTALLATION

A. Seal Installation

If a mechanical seal fails soon after installation, it is usually caused by improper start-up procedures or installation errors. Some common examples of installation errors include:

1. Mistake in assembly. Some examples: installing the seals or V-rings backwards; omitting seal parts; not installing anti-rotating pins in stationary seats; or not tightening or overtightening set screws. A lack of understanding of how seals work may contribute to seal failure at installation.

2. Damage to secondary seals. Cuts and damage to O-rings, V-rings, and gaskets; damage to the shaft or gland surfaces; or improper shaft finishes or dimensions.

3. Damage to faces. Dirt, distortion, scratches or chips caused by handling; or improper clamping.

4. Improper spring tension. Incorrect positioning of the seal. Spring tension not only holds the faces together, but in many seals loads the secondary seal to the shaft. Too much compression or too little can cause early failure.

5. Misalignment. Certain alignments are critical for successful seal operation. The main objective is to ensure there is no axial shaft motion and that both the stationary and rotating faces run perpendicular to the shaft. Misalignment usually shortens seal life rather than causing immediate failure.

   The use of cartridge seals virtually has eliminated most seal failure due to installation errors.

1. Protecting the Seal Faces

New seal faces are lapped flat by the manufacturer to tolerances of one to three Helium Light Bands (one light band is equal to 11.6 millionths of an inch).

Even very stiff materials, such as ceramic, will warp enough to leak if they are overstressed by clamping. It is important to use 1) clamping surfaces that are flat and smooth, or 2) gaskets on both sides of the clamped surface. Four gland bolts should be used if at all possible. Glands clamped with only two bolts have a tendency to become cocked and affect the alignment of the seal faces.

On a horizontal-split case pump, the seal chamber faces of each half of the pump should be machined after they are together to ensure a good sealing surface.

2. General Rules

a. If a seal face has been used, repaired, or relapped, check for flatness before reusing it. It is always better to relap the seal faces to ensure that no wear track exists.
b. Keep the faces covered or protected until they are installed in the pump.
c. Do not lubricate the seal faces.
d. Keep your hands and the work area as clean as possible.
e. When using clamped rings, make sure clamping surfaces are smooth, flat and clean.
f. Use clean cloths if you push on a seal face during installation.
g. To lightly polish or clean a face, use an alcohol-based solvent and a lint-free tissue.
h. Where possible, handle seals on their outside diameters. Faces should be cleaned if contact occurs.
i. Use caution when tightening set screws with an Allen wrench.
j. When using a new shaft sleeve, chamfer the end to ensure no damage tot he secondary seal on installation.

3. Positioning the Seal

In an overhung impeller pump, after it has been overhauled and seal installation is ready to begin, the first step is to re-install the seal chamber (with all gaskets installed) onto the pump frame and bolt into its normal position. The impeller should also be assembled and adjusted to its operating position.

Next color the shaft or sleeve at the face of the seal chamber with machinist blue. Manually turn the shaft and scribe a line at the seal chamber face in the colored section using a straight edge held across the seal chamber face. This line is represented by the letter "B" of *Figures* 42*and* 43 This line gives the point of reference needed to set the seal at its proper operating length.

Then, remove the impeller and seal chamber, exposing the shaft or sleeve and the seal installation reference line "B". Check the I.D. of the seal chamber to ensure the seal will fit.

Mechanical Seals

If the stationary seal ring extends into the seal chamber, as shown in *Figure 42*, measure the distance and label this dimension "X". This dimension is added to the operating length "A" to find the distance "D" from seal chamber line "B" to reference line "C" to the back of the seal set position. All gaskets should be in place for this measurement.

Example: *(Figure 42)*

Seal Operating Length "A"	= 1-1/2"
Seat Pilot Length "X"	= 1/2"
Seal Set Dimension "D" from Seal Chamber "B"	= 2"

Scribe the shaft or sleeve with line "C" 2 inches from the seal chamber line "B". The back of the seal rotary head can be installed and attached to the shaft or sleeve at line "C".

Using the same example as before, if the seal faces meet outside the seal chamber face, as shown in *Figure 43*, the "X" dimension would be subtracted from the operating length "A" to find the distance "D" from the seal chamber line "B" to reference line "C" to the back of seal set position.

Example: *(Figure 43)*

Seal Operating Length "A"	= 1-1/2"
Faces Meet Outside Box "X"	= 1/2"
Seal Set Dimension "D" from Seal Chamber "B"	= 1"

Then scribe the shaft or sleeve with line "C" 1 inch from the seal chamber line "B". The back of the seal rotary head can be installed and attached to the shaft or sleeve at line "C".

4. Summary of Steps

a. Mark the seal chamber face position on the shaft or sleeve.

b. Determine the operating length of the seal. A seal's operating length is always the measurement from where the faces meet to the back of the rotary head.

c. Determine the distance from the seal chamber face to face of the stationary ring.

d. If the stationary face pilots into the seal chamber *(Figure 42)*, add the dimension "X" to the operating length to determine the position of the back of the seal.

e. If the stationary face is recessed in the gland *(Figure 43)*, subtract the dimension "X" from the operating length to determine the position of the back of the seal.

f. Most seal manufacturers give you "D", the seal set dimension from the seal chamber face to the back of the seal, with an installation drawing.

g. Check all components to the assembly drawing to assure that the part numbers match.

5. Protecting the Secondary Seal

If the secondary seal is between the spring and the face, it is usually dynamic. This means that the secondary seal must be able to move to compensate for misalignment, face wear, and axial shaft movement.

A static secondary seal does not have to move once it is installed on the shaft or sleeve. It is usually more heavily loaded on the shaft than a dynamic secondary seal.

The term elastomer refers to rubber-like materials which have resilience. That is, after they are stretched, they return to their original shape. Materials such as Buna-N, EPR, Viton, and Kalrez O-rings are elastomers. TFE and Creavey O-rings, and Grafoil packing are not elastomers and have to be handled more carefully.

6. General Rules for Secondary Seals

a. The shaft of sleeve should be clean, free from set screw marks, sharp obstructions, rust and sharp edges.

b. Threaded sections of the shaft should be covered to prevent damage to the seal.

c. Keyways should be filled with wax or covered.

d. Seals should be pushed, not twisted or hammered, down the shaft.

If your seal has a dynamic elastomer, the shaft or sleeve should be highly polished with the surface finish in the 16 to 32 microinch (RMS) range.

Dynamic secondary packings are normally designed with a very light squeeze, usually no more than .005 inches on a side. O-rings should be stretched around the shaft and sized to its inside diameter, not its outside diameter. Stretching up to five percent will not cause problems. O-rings should never be removed from grooves using a sharp metal object. The scratch can set up a leak path around it.

Mechanical Seals

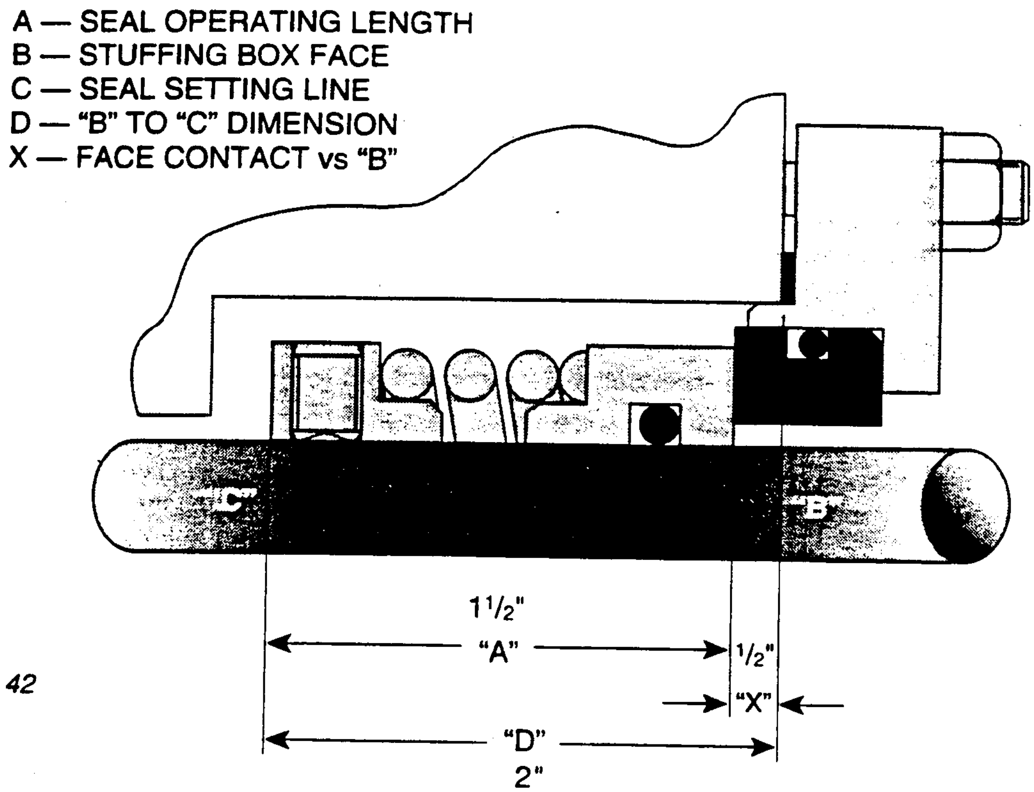


Figure 42

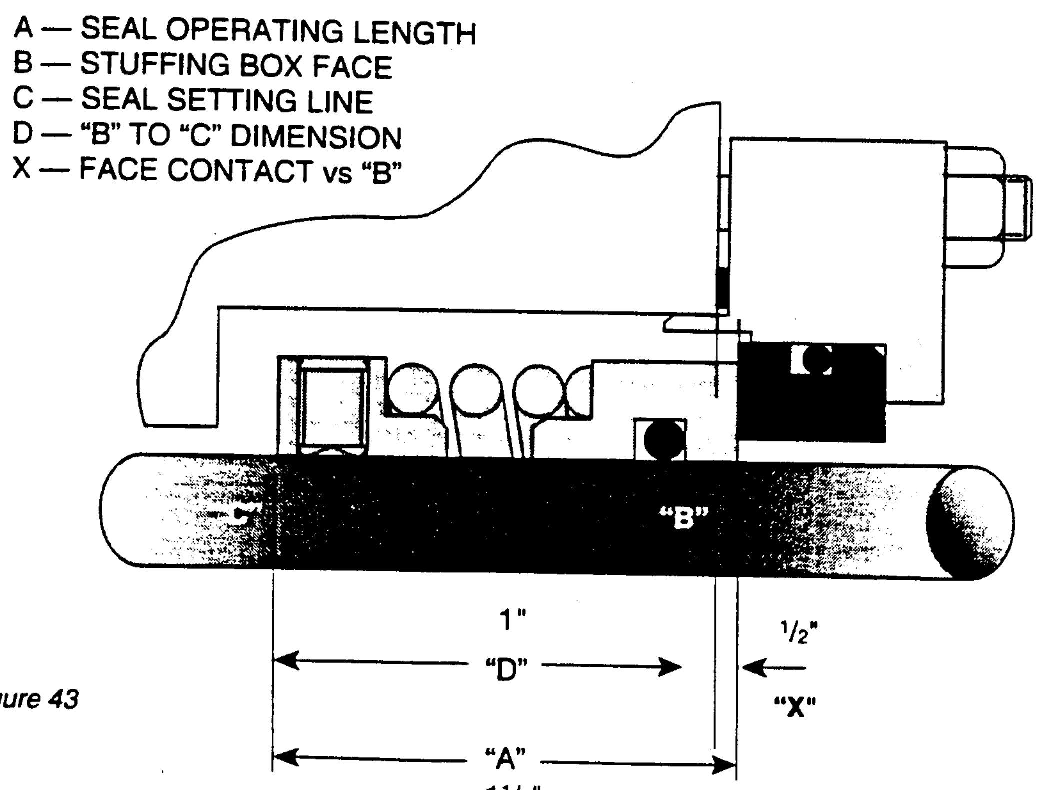


Figure 43

INSTALLATION INSTRUCTIONS – SHAFT/SLEEVE MOUNTED SEALS

These instructions are applicable for back pull-out, overhung centrifugal pumps Other pump tvoes mav require different assemblv procedures.

PREPARATION

1. Disassemble the pump as far as is necessary to remove the old packings or seal, gland and shaft sleeve.
2. Verify that the shaft diameter meets the tolerances and surface finish shown on the assembly drawing.
3. IF—The seal assembly is to be mounted on a shaft sleeve, this sleeve must be gasketed to the shaft and leak tight.
4. To avoid damage to O-rings and packings during assembly, chamfer shaft and sleeve ends and shoulders 10° x .125" /3mm (fig.1), and remove all sharp edges from keyways and grooves.
5. Check the condition of the pump bearings and replace them if necessary. Verify shaft end play does not exceed the pump manufacturer's limit, usually .005" / .1mm, and shaft runout does not exceed .001" per inch/.001mm per mm (TIR) of shaft diameter (figs. 2 & 3).
6. Carefully clean the stuffing box bore and face. Verify that the stuffing box bore diameter complies with the dimension shown on the installation drawing.
7. IF—A shaft sleeve is being used, install it in its operating position, complete with shims and/or gaskets.
8. Reassemble the stuffing box housing, complete with shims and/or gaskets. Verify stuffing box face perpendicularity to the shaft (fig.4), and shaft concentricity to the stuffing box (fig.5) do not exceed .001" per inch/.001mm per mm (TIR) of diameter.
9. With the stuffing box and shaft/sleeve in their correct operating positions, use a straight edge to scribe the position of the stuffing box face onto the shaft/sleeve at A (fig.7). Use machinist's blue to make the scribe easier to see.
10. Again remove the stuffing box housing. From the installation drawing determine the distance from the stuffing box face to the seal set length, and scribe line B (fig.7) onto the shaft/sleeve at this distance.
11. Without disturbing the scribe lines, wipe the shaft/sleeve clean and apply a lubricant which is compatible with the sealed fluid and the gasketing materials. Both Dow 55 and 111 are acceptable.

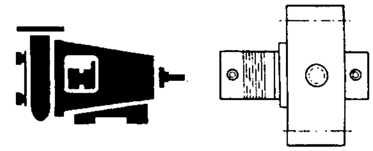
ASSEMBLY

1. Unwrap the mechanical seal components, taking care not to scratch or damage the seal faces.
2. (IF—A stepped stationary seat is supplied, first install the stationary O-ring into its groove in the gland counterbore.) Lightly lubricate the stationary packing and press the assembly into the gland plate counterbore, ensuring that the slot engages the drive pin in the gland, if applicable.
3. Carefully slide the complete gland assembly, including the gland gasket, onto the shaft as far as possible.
4. Slide the rotating assembly onto the shaft/sleeve, being careful not to damage the O-ring or packing. Referring to the assembly drawing, align the appropriate surface of the rotating assembly with scribe line B (fig.8), and tighten the set screws evenly. (Once tightened, set screws should not be re-used. IF—You must loosen the set screws for any reason, replace them before repeating step 4.) Then, IF APPLICABLE—tighten the socket head cap screws evenly to compress the grafoil packing.
5. Being careful not to damage the seal, reassemble the stuffing box housing and install the impeller.
6. Ensure the gland gasket is in place, and slide the gland assembly into position against the face of the stuffing box. Assemble the gland bolts finger tight. Continue tightening alternately until secure. Do not distort the gland or overtighten. Recommended torque is 30 ft/lbs. Verify the gland is concentric with the shaft/sleeve to prevent possible damage due to rubbing.
7. Complete reassembly of the pump, frequently turning the shaft slowly by hand to check for free rotation. The seal faces are dry so turn slowly to avoid damage. IF—The shaft will not turn, seal has been improperly set.
8. Refer to assembly drawing and/or pump manual for piping connections and coupling alignment (fig.6). Proceed as indicated.

OPERATION

1. Follow the pump manufacturer's recommended warm-up and start-up procedures found on page . Whenever possible vent the stuffing box to ensure that the seal area is flooded. DO NOT RUN THE PUMP DRY.
2. Some minor leakage may occur on start-up, but should progressively diminish. Constant leakage indicates an improperly set seal, or damaged O-ring/packing.
3. The gland plate should not be noticeably warmer than the pump casing. Heat (and squealing) indicates that the seal has been improperly set, or is running without lubrication.

INSTALLATION INSTRUCTIONS—CARTRIDGE MOUNTED SEALS



These instructions are applicable for back pull-out, overhung centrifugal pumps. Other pump types may require different assembly procedures.

PREPARATION

1. Disassemble the pump as far as is necessary to remove the old packings or seal, gland and shaft sleeve.
2. Verify the shaft diameter meets the tolerances and surface finish shown on the assembly drawing.
3. IF—The seal assembly is to be mounted on a shaft sleeve, this sleeve must be gasketed to the shaft and leak tight.
4. To avoid damage to O-rings and packings during assembly, chamfer shaft and sleeve ends and shoulders 10° X .125" / 3mm (fig.1), and remove all sharp edges from keyways and grooves.
5. Check the condition of the pump bearings and replace them if necessary. Verify shaft end play does not exceed the pump manufacturer's limit, usually .005"/.1mm, and that shaft runout does not exceed .001" per inch /.001mm per mm (TIR) of shaft diameter (figs 2 & 3).
6. Carefully clean the stuffing box bore and face. Verify that the stuffing box bore diameter complies with the dimension shown on the installation drawing.
7. IF—A shaft sleeve is being used, install it in its operating position, complete with shims and/or gaskets.
8. Reassemble the stuffing box housing, complete with shims and/or gaskets. Verify stuffing box face perpendicularity to the shaft (fig.4), and shaft concentricity to the stuffing box (fig.5) do not exceed .001" per inch / .001mm per mm (TIR) of diameter.
9. Again disassemble the stuffing box housing. Wipe the shaft/sleeve clean and apply a lubricant which is compatible with the sealed fluid and the gasketing materials. Both Dow 55, and 111 are acceptable.

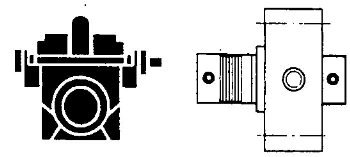
ASSEMBLY

1. Take the complete cartridge from its package. Do not disassemble or alter the unit.
2. Slide the complete cartridge assembly as far as possible onto the shaft sleeve, being careful not to damage the O-ring or packing inside the cartridge sleeve.
3. Reassemble the stuffing box housing and the impeller.
4. With the gland gasket in place, slide the complete assembly into position against the face of the stuffing box. Assemble the gland bolts finger tight. Continue tightening alternately until secure. Do not distort the gland by overtightening. Recommended torque is 30 ft/lbs.
5. IF—The cartridge sleeve packing is grafoil, tighten the socket head cap screws evenly to compress the grafoil packing onto the pump shaft/sleeve. Leave the eccentric washers in place to maintain the axial position of the cartridge sleeve.
6. IF—The assembly drawing calls for holes or countersinks to be drilled under the cartridge sleeve set screws, remove the set screws and mark their location. Unbolt the cartridge gland and loosen the socket head cap screws, remove the impeller, stuffing box housing and cartridge assembly. Drill the shaft/sleeve in the positions marked. Repeat steps 2-5
7. Tighten the cartridge sleeve set screws evenly. (IF—The shaft/sleeve has been drilled, ensure that the set screws align with the appropriate drilled holes.)
8. IF—Clips are present, remove the shipping clips, or rotate the eccentric washers 180° to clear the slot in the cartridge sleeve.
9. Complete reassembly of the pump, frequently turning the shaft slowly by hand to check for free rotation. The seal faces are dry so turn slowly to avoid damage. IF—The shaft will not turn, the seal has been improperly set.
10. Refer to assembly drawing and/or pump manual for piping connections and coupling alignment (fig.6). Proceed as indicated.

OPERATION

1. Follow the pump manufacturer's recommended warm-up and start-up procedures found on page . Whenever possible vent the stuffing box to ensure that the seal area is flooded. DO NOT RUN THE PUMP DRY.
2. Some minor leakage may occur on start-up, but should progressively diminish. Constant leakage indicates an improperly set seal, or damaged O-ring/packing.
3. The gland plate should not be noticeably warmer than the pump casing. Heat (and squealing) indicates that the seal has been improperly set, or is running without lubrication.

INSTALLATION INSTRUCTIONS—CARTRIDGE MOUNTED SEALS



These instructions are applicable for horizontally split, double bearing centrifugal pumps. Other pump types may require different assembly procedures.

PREPARATION

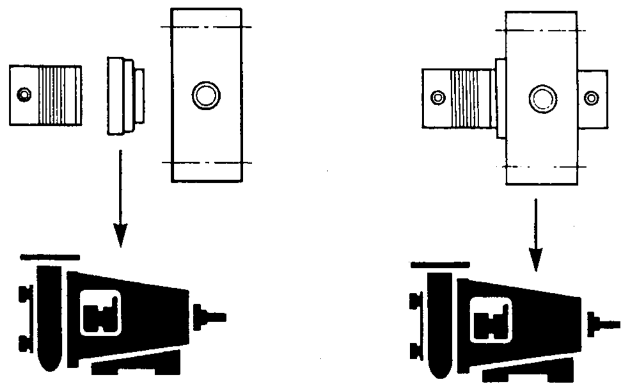
1. Remove the bearings and bearing housings, and disassemble the pump as far as is necessary to remove the old packings or seals, glands and shaft sleeves.
2. Verify shaft/sleeve diameters meet the tolerances and surface finish shown on the assembly drawing.
3. IF—The seal assemblies are to be mounted on shaft sleeves, these sleeves must be gasketed to the shaft and leak tight.
4. To avoid damage to O-rings and packings during assembly, chamfer shaft and sleeve ends and shoulders 10° X .125" / 3mm (fig. 1), and remove all sharp edges from keyways and grooves.
5. Carefully clean the stuffing box bores and faces. Verify stuffing box bore diameters comply with the dimension shown on the installation drawing. IF—Stuffing boxes are split, ensure gaskets extend flush with the stuffing box faces, and halves are in alignment.
6. Reassemble the sleeves, bearing housings and bearings, and adjust the shaft to its actual operating location. Verify shaft end play does not exceed the pump manufacturer's limit, usually .005" /.1mm, and shaft runout does not exceed .001" per inch / .001mm per mm (TIR) of shaft diameter (figs 2 & 3). Verify stuffing box face perpendicularity to the shaft (fig. 4), and shaft concentricity to the stuffing box (fig. 5) do not exceed .001" per inch / .001mm per mm (TIR) of diameter.
7. Again disassemble the stuffing box housing. Wipe the shaft/sleeve clean and apply a lubricant which is compatible with the O-ring and the gasketing materials. Both Dow 55, and 111 are acceptable.

ASSEMBLY

1. Take both complete cartridges from their packages. Do not disassemble or alter the units.
2. Slide the complete cartridge assemblies onto the shaft/sleeves, being careful not to damage the O-rings or packings inside the cartridge sleeves.
3. Reassemble the bearing housings and bearings, and complete all required axial adjustments to the pump rotating assembly.
4. With the gland gasket in place, slide the complete assembly into position against the face of the stuffing box. Assemble the gland bolts finger tight. Continue tightening alternately until secure. Do not distort the gland by overtightening. Recommended torque is 30 ft/lbs.
5. IF—The cartridge sleeve packing is grafoil, not an O-ring, tighten the socket head cap screws evenly to compress the grafoil packing onto the pump shaft/sleeve. Leave the eccentric washers in place to maintain the axial position of the cartridge sleeve.
6. IF—The assembly drawing calls for holes or countersinks to be drilled under the cartridge sleeve set screws, remove the set screws and mark their location. Unbolt the cartridge gland and loosen the socket head cap screws, remove the bearings, bearing housings, and cartridge assemblies. Drill the shaft/sleeves in the positions marked. Repeat steps 2-5.
7. Tighten the cartridge sleeve set screws evenly. (IF—The shaft/sleeve has been drilled, ensure that the set screws align with the appropriate drilled holes.)
8. IF—Shipping clips are present, remove the clips, or rotate the eccentric washers 180° to clear the slot in the cartridge sleeve.
9. Complete reassembly of the pump, frequently turning the shaft slowly by hand to check for free rotation. The seal faces are dry so turn slowly to avoid damage. IF—The shaft will not turn, the seal has been improperly set.
10. Refer to assembly drawing and/or pump manual for piping connections and coupling alignment (fig. 6). Proceed as indicated.

OPERATION

1. Follow the pump manufacturer's recommended warm-up and startup procedures found on page . Whenever possible vent the stuffing box to ensure that the seal area is flooded. DO NOT RUN THE PUMP DRY.
2. Some minor leakage may occur on start-up, but should progressively diminish. Constant leakage indicates an improperly set seal, or damaged O-ring/packing.
3. The gland plate should not be noticeably warmer than the pump casing. Heat (and squealing) indicates that the seal has been improperly set, or is running without lubrication.



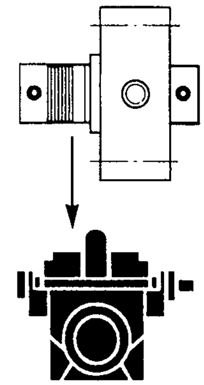


FIG #1

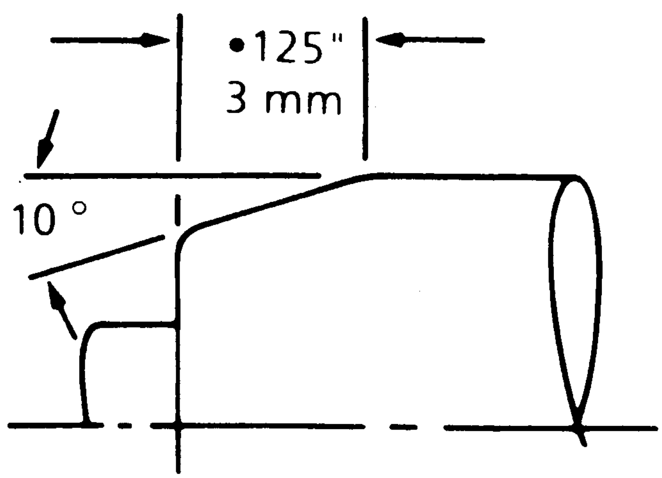


FIG #2

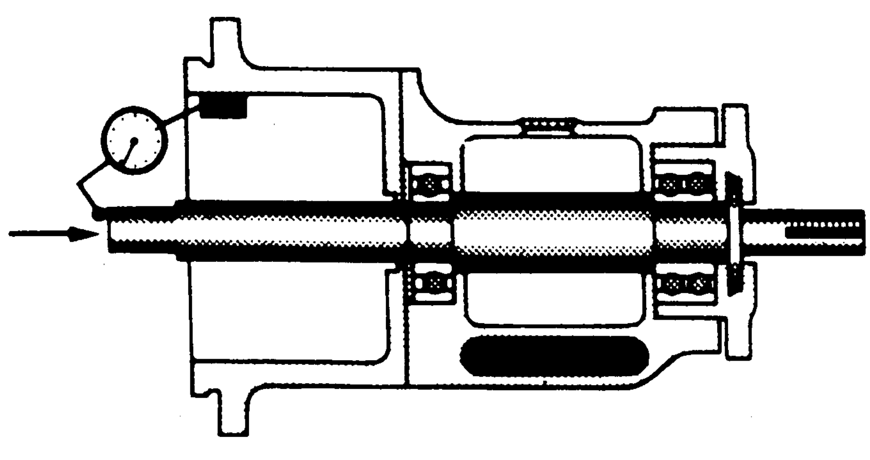


FIG #3

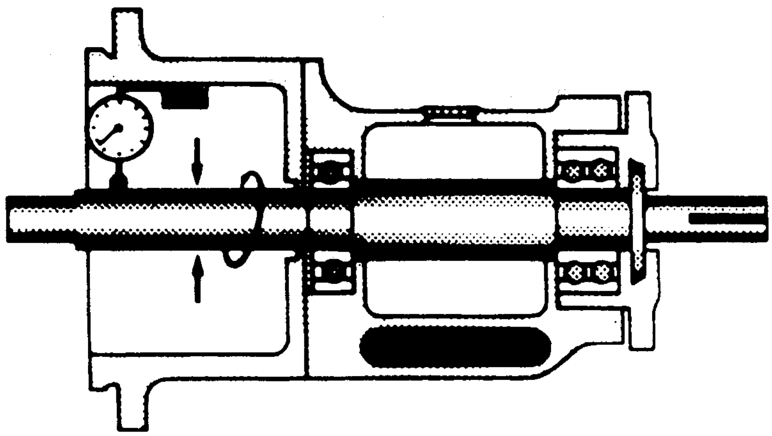


FIG #4

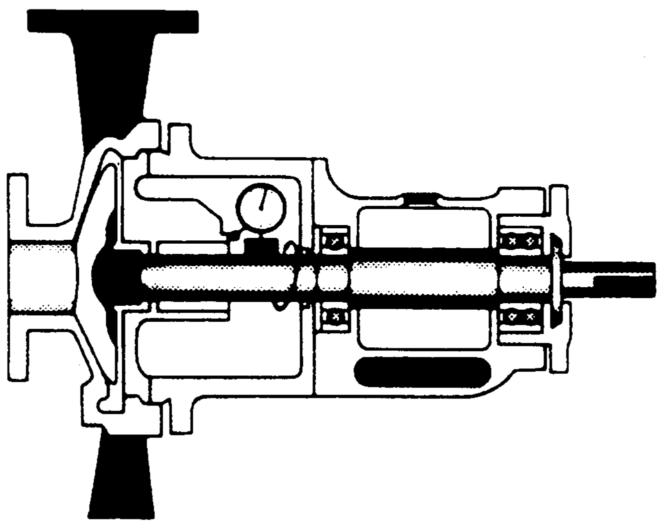


FIG #5

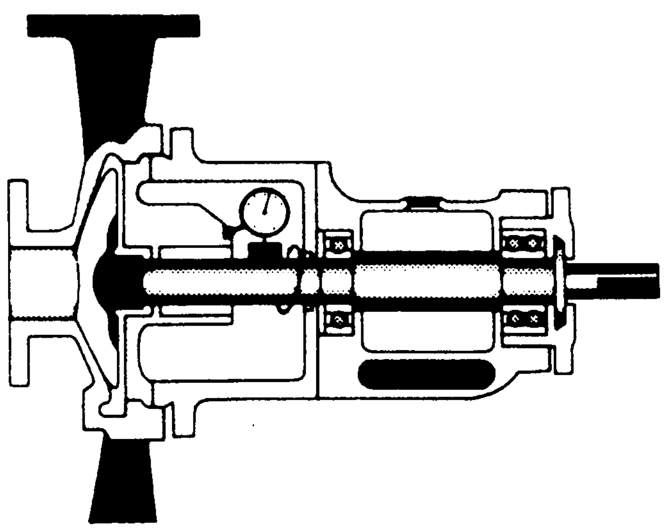


FIG #6

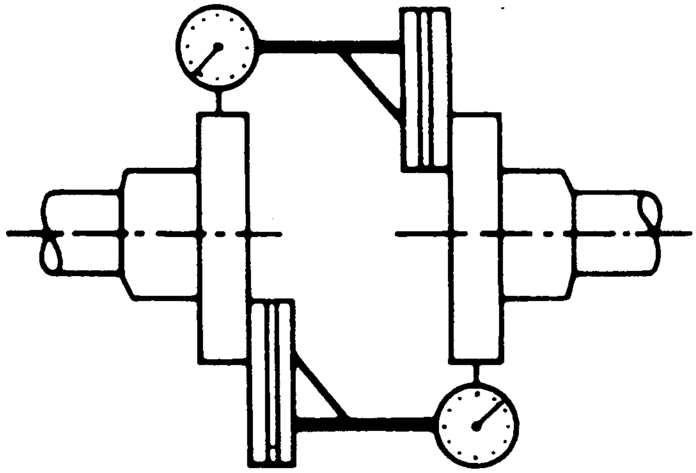


FIG #7

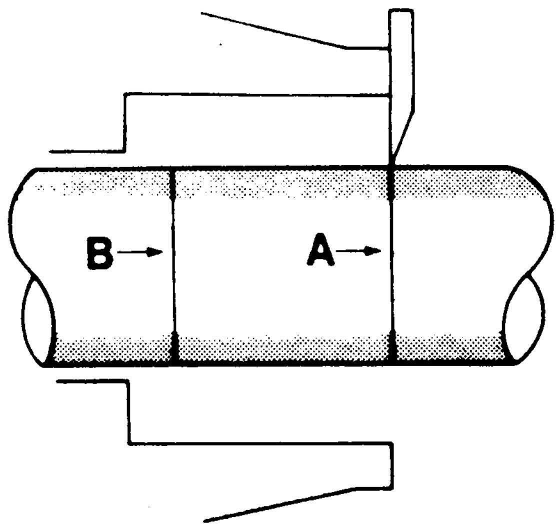
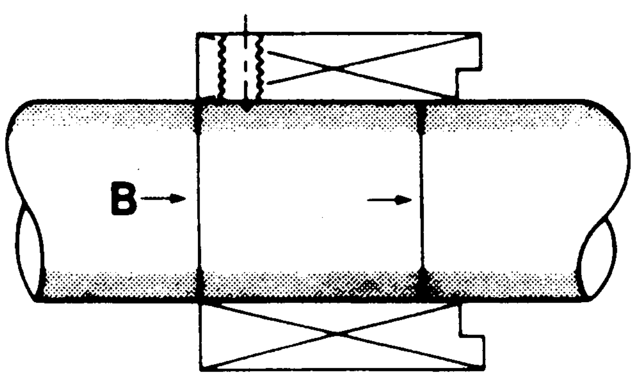


FIG #8



Mechanical Seals

Elastomers are sensitive to chemical attack, therefore use only silicon grease or TFE lubricants for installation. V-rings, wedges, U-cups and other secondary seal shapes made from TFE, should be unloaded during the seal's installation. If TFE is cold, it will shrink and become hard and stiff, but it can be softened in hot water. It is very important that the shaft or sleeve be free from old set screw marks, scratches, sharp corners, keyways and rust. TFE cuts easily and a small leak will often cause the seal to lock onto the shaft.

TFE-covered O-rings should only be used in static applications. They are very sensitive to cutting and crimping. If they must be placed in an O-ring groove, boil them first in water, then lubricate them well before installation. They must not be installed over a sharp edge or twisted onto the shaft or damage may occur.

Pump Conditions

Rapid back-and-forth motion of the seal of only a few thousandths of an inch will shorten seal life. This section describes the alignments and parts of the pump that can cause these small but destructive motions.

The most critical alignment in seal installation is that of the stationary seal ring. Misalignment leads to a back-and-forth motion on each revolution of the shaft.

3600 rpm	x 2 movements = 7,200 per minute
	x 60 minutes = 432,000 per hour
	x 24 hours = 10,368,000 per day

Separation of the faces by .0001" will lead to leakage, abrasive damage, clogging of external springs, hang-up of sliding secondary seals, coking of high temperature fluids, and other problems associated with seal failure.
The seal chamber face should be machined and checked for squareness to the shaft. A dial indicator is clamped onto the shaft with the probe resting on the face of the seal chamber as the shaft is rotated to bring the unit into proper specification.

Once the seal chamber face is square to the shaft, it is important to make sure the stationary seat is also square to the shaft.

It is best to consult your supplier for recommended alignment tolerances and clearances.

Shaft Straightness

The straightness of the shaft can be checked by dial indicating the shaft as you move it by hand. A rule of thumb is that there should be no more than .002 Total Indicator Reading (TIR) per inch of shaft diameter radial runnout on smaller pumps and no more than .001 TIR per inch of shaft diameter on larger pumps. After .005 runnout, most seals will start having problems.

Bearings

The end play should also be checked with a dial indicator. Before placing the indicator pin against the shaft shoulder (impeller end), ensure that the shaft is all the way in, then tap the end of the shaft near the thrust bearing with a soft hammer. This will verify the condition of the thrust bearing and its fit in the housing. Shaft lift can be used to determine the condition of the radial bearing and its fit in the housing. This is checked by placing a dial indicator at the end of the shaft on the impeller end and lifting up.

Couplings

A misaligned coupling can transmit vibration through the bearings to the seal. Rapid bearing wear will result from this condition leading to bearing failure.

Unbalanced Impeller

Severe vibrations caused by an unbalanced impeller can result in face separation, shaft fretting, drive lug wear, and premature seal failure.

Pipe Strain

Pipe strain can cause misalignment between the stationary and rotary seal faces, resulting in seal motion, face separation, leakage and failure.

Shaft Deflection & Cavitation

Shaft deflection due to unbalanced radial thrust and the weight of the impeller will have the same effect on the seal as with a bent shaft.

Cavitation occurs when the available pump head is equal to or less than the system required head. This causes severe vibration in the pump, leading to bearing failure. Furthermore, if the pumped product is flashing at the impeller eye, this will typically cause a flashing problem at the faces.

CONTROLLING THE SEAL ENVIRONMENT

A. Seal Piping Arrangements

1. API Plans/ANSI Plans

The American Petroleum Institute (API) develops standards and specifications for the petroleum industry. The standard for centrifugal pumps used in general refinery service is API 610.

There are 17 Piping Plan arrangements in API 610 to ensure safe seal and pump operations. The following is a description of the most common plans used today.

ANSI uses the same piping plan arrangements as API, except that it adds the number preface 73.

API Plan 2 ***(ANSI Plan 7302)***

This plan calls for a dead-ended seal chamber with no circulation of flush fluid *(Figure 32).* The seal cavity or gland may be jacketed and a throat bushing may be required when specified.

API Plan 2 is used when circulation is not required to prevent flashing or when the jacket is used to control the product temperature in the seal chamber.

The use of big bore and tapered bore seal chamber can eliminate the need for a circulating flush in some applications. Studies have shown, with proper design, these arrangements effectively dissipate heat generated by the seal, thus minimizing the temperature rise at the seal faces when API Plan 2 is used. The major drawback of these designs is that, in most cases, a throat bushing cannot be installed, making it difficult to change the environment in the seal chamber (e.g. raise the pressure or introduce an external flush).
If the environment may need to be changed, it is more practical to use a pump that offers the option of a restrictive throat bushing along with a flush that circulates the fluid to prevent heat build-up.

API Plan 2 is frequently used in conjunction with a water jacket and a close-clearance throat bushing in heat transfer fluids, hot oils near their vapor pressure, and other applications where cooling is advantageous. The bushing is used to minimize mixing of the fluid behind the impeller and the cooled fluid in the seal chamber. By using this arrangement, a 400°F (200°C) product can be cooled by as much as 100°F (38°C), with a three gallon per minute flow of cooling water through the jacket.

In some applications, such as asphalt, the same arrangement is used with a steam jacket to help keep the fluid warm and in a liquid state. When steam is used, a close-clearance throat bushing is usually not required.

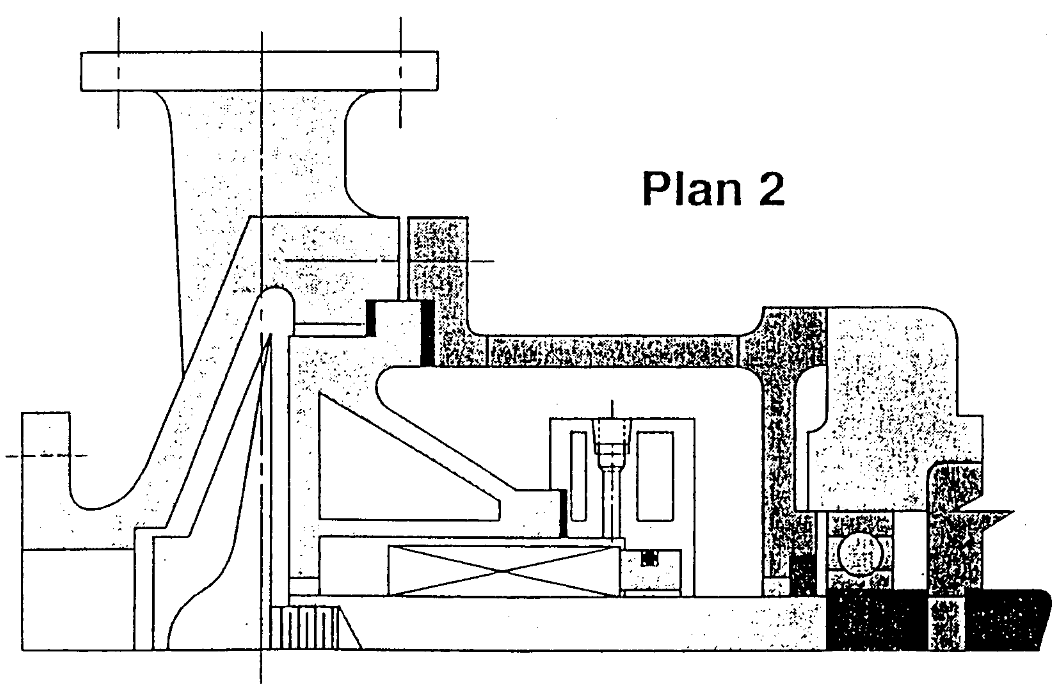


Figure 32

Some of the precautions that must be taken when using a water or steam jacket include:

1. The jacket must be kept clean to ensure an adequate heat transfer.
2. A water jacket should not be used if the product sets up at ambient temperature.
3. The cooling water should not be blocked in when the pump is down and freezing conditions could occur. This could cause the water jacket to crack.

API Plan 11 (ANSI Plan 7311)

This plan requires installation of a recirculation line from the pump case discharge through an orifice to the seal chamber *(Figure 33).*

API Plan 11 is the most common piping plan for single seals and it is also frequently used for the inner seal in multiple seal arrangements. The primary purpose of this plan is to dissipate heat generated by the seal and/or build seal chamber pressure.

API Plan 11 should be used when a single seal is running in a product that is near its vapor pressure and cooling is not practical. Vaporization can be further suppressed by installing a close-clearance throat bushing (less than .012 inches of diametrical clearance) to increase seal chamber pressure. While this is not as efficient as cooling for vapor suppression, it is much less expensive.

It sometimes can be advantageous to take the recirculating flush from downstream of the discharge check valve. When this is done, the flush to the seal will be maintained in case of pump suction loss. This makes the seal more forgiving to pump upsets, particularly if a close-clearance throat bushing is used.

Some of the potential problems that can arise when using this plan include:

1. Not effective when the seal chamber is at or near discharge pressure. When this is the case, API Plan 13 is frequently used.
2. Thermosensitive or viscous products can set up in the flush line, particularly when the pump is down. This can be prevented by heat tracing and insulating the piping to keep the pumped fluid at the proper temperature or by flushing out the piping when the pump is down.
3. When this piping plan is used in a heavy slurry, the flush line can plug. API Plan 32 is usually recommended for these applications.
4. If there is a large pressure differential between the seal chamber and pump discharge, an orifice, or a series of orifices, must be installed to break down the pressure. In order to minimize the chance of plugging, the smallest orifice diameter that should be used is 0.125 inches.
5. Erosion of the seal parts can occur if the flush enters the seal chamber with excessive velocity. This can be prevented by using tangential flush ports and installing the orifice(s) as far from the seal chamber as possible.

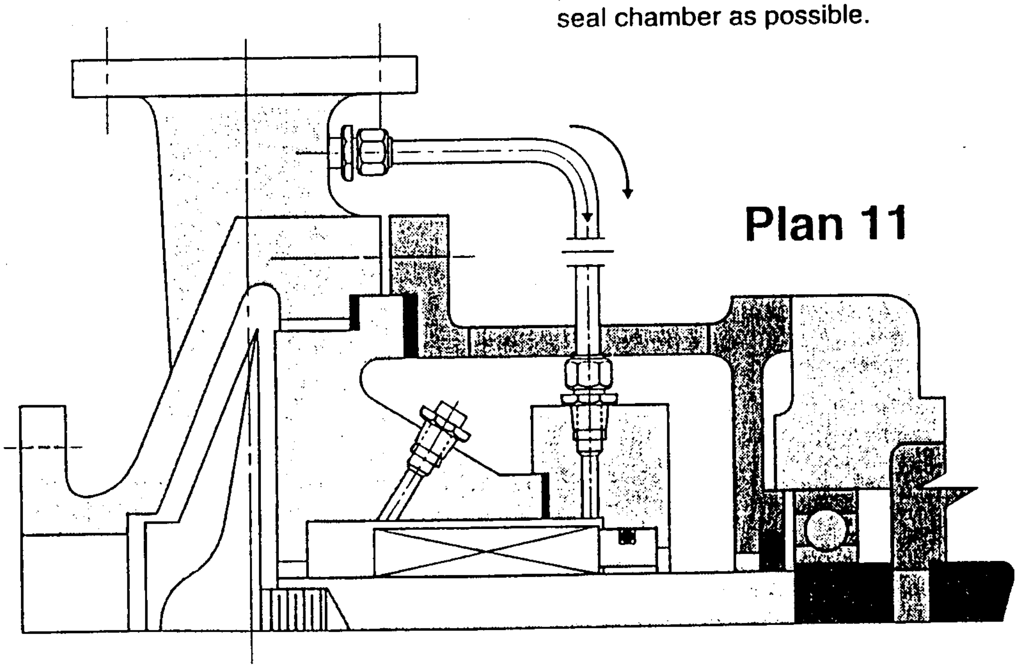


Figure 33

API Plan 13 *(ANSI 7313)*

This plan requires the installation of a recirculation line from the seal chamber, through an orifice, and back to the pump suction *(Figure 34)*.

API Plan 13 is frequently used in vertical pumps to vent vapors from the seal chamber. It is also used in applications where the seal chamber pressure is at or near discharge pressure, or when it is advantageous to lower the seal chamber pressure. When using API Plan 13 on a horizontal pump, the flush connection should be located at the top of the gland to ensure there are no air pockets in the seal chamber. Also, the flush line should be orificed to prevent excessive flow; however, the smallest orifice diameter that should be used is 0.125 inches.

A potential problem with API Plan 13 is that the reduction in seal chamber pressure can make the pumped product more susceptible to flashing. When this is a concern, this plan can be used in conjunction with API Plan 11 to minimize the reduction in pressure.

API Plan 21 *(ANSI Plan 7321)*

API Plan 21 requires the installation of a recirculation line from the pump case discharge, through an orifice and a cooler or heat exchanger, to the seal chamber *(Figure 35)*.

This plan is recommended for applications under 300°F (150°C), where cooling is required to prevent flashing. API Plan 21 has been used to reduce the temperature to where elastomers can be used. A welded metal bellows design with flexible graphite packing eliminates the need for cooling unless flashing is a problem.

This plan also can be used in applications over 300°F (150°C). However, API Plan 21 has been used to reduce the temperature to where elastomers can be used. A welded metal bellows design with flexible graphite packing eliminates the need for cooling unless flashing is a problem.

This plan also can be used in applications over 300°F (150°C). However, API Plan 23 is frequently recommended in these cases to improve cooling efficiency and extended the life of the heat exchanger. High product temperatures on the tube side of the exchanger tend to cause deposits to come out of the solution with the cooling water and build up in the shell. This will eventually result in a significantly reduced heat transfer rate.

Some of the problems that can be encountered when using API Plan 21 are:

1. Even when the product temperature is below 300°F (150°C), the heat exchanger will eventually become fouled. The best way to check for fouling is to compare the current product and cooling water temperatures into and out of the exchanger, to temperatures when the system was initially installed. Because a change in the flow rate of the product or cooling water can also affect these temperatures, it is always best to use seals designed for high temperature applications.
2. This piping plan should not be used for applications where the product sets up at ambient temperatures. These applications typically do not require cooling.
3. API Plan 21 is not effective when the seal chamber is at or near discharge pressure. When cooling is required in these applications, Plan 2 or Plan 23 should be used.
4. If there is a large pressure differential between the seal chamber and pump discharge, an orifice or a series of orifices must be installed to break down this pressure. In order to minimize the chance of plugging, the smallest orifice diameter that should be used is 0.125 inches.
5. Erosion of the seal parts can occur if the flush enters the seal chamber with excessive velocity. This can be prevented by using tangential flush ports and installing the orifices as far from the seal chamber as possible.

API Plan 23 *(ANSI Plan 7323)*

API Plan 23 provides for a circulation pipe attached to the seal chamber with fluid flow generated by a seal pumping ring through a heat exchanger and returned to the seal chamber *(Figure 36)*.

This piping plan is frequently associated with boiler feedwater applications. Because it is essentially a closed-loop system, the same product is continually circulated between the seal and the heat exchanger. This leads to a much more efficient cooling system than API Plan 21. Furthermore, since the product entering the cooler is already at a lower temperature, the life of the heat exchanger will be extended.

Plan 13

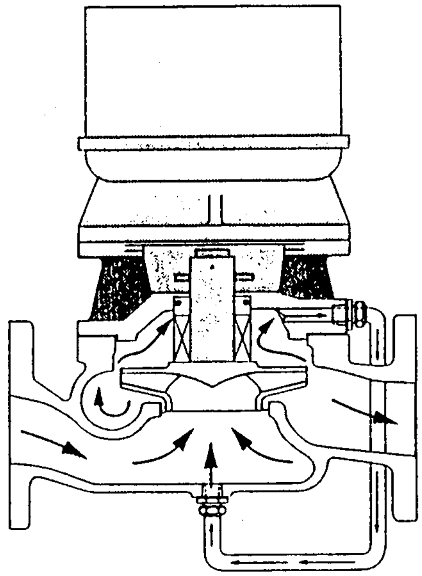


Figure 34

Plan 21

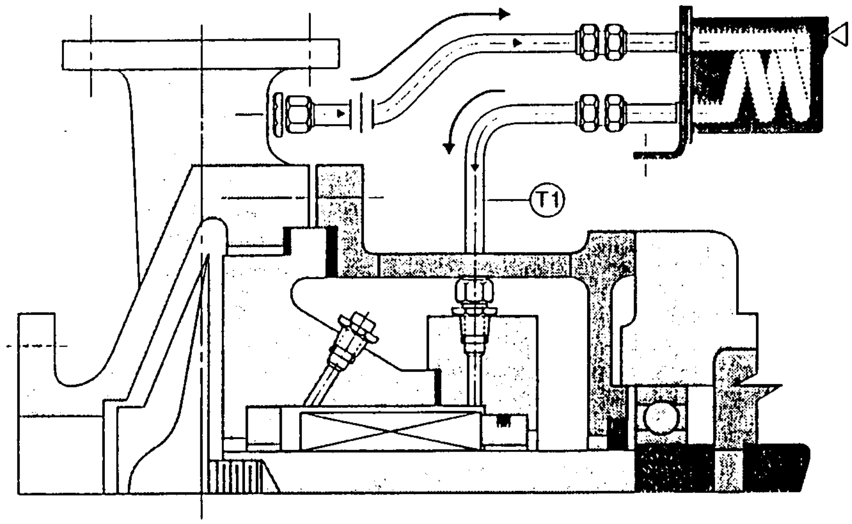


Figure 35

Plan 23

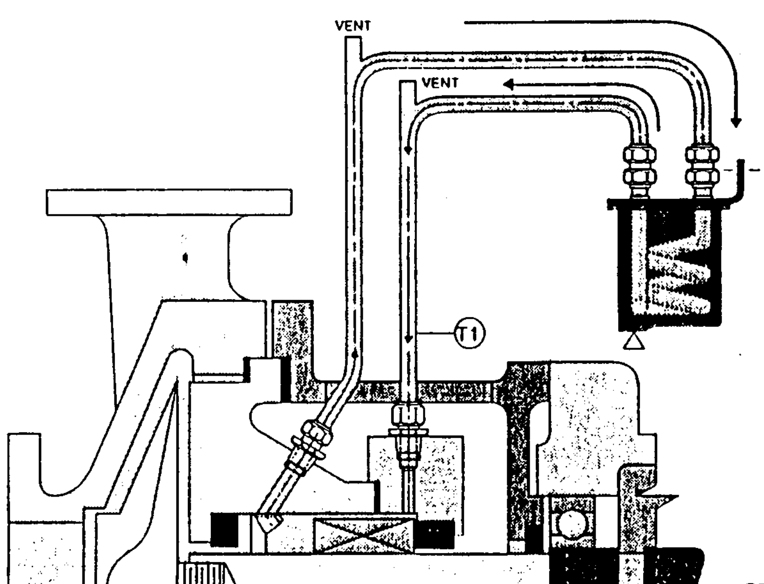


Figure 36

A close-clearance throat bushing is generally recommended when using API Plan 23 to prevent mixing of the hot product behind the impeller and the cool product in the seal chamber. The water jacket is also frequently used to further reduce the seal chamber temperature.

The piping and installation of this system is critical to its success. Below are several rules-of-thumb to keep in mind during installation of an API Plan 23 flush system.

1. The distance from the bottom of the heat exchanger to the centerline of the shaft should be approximately 12 inches.
2. The horizontal distance from the heat exchanger to the seal should be as short as possible, with a maximum of 3 feet.
3. There should be no sharp elbows or bends in the tubing. A gentle sweep, although more difficult for pipe fitters, will yield better results.
4. There must be sufficient shaft surface speed for the pumping ring to be effective.
5. The line from the heat exchanger to the seal and from the seal to the heat exchanger should be 1/2 inch (minimum) tubing. (API 3/4 inch minimum)
6. Provisions should be made for venting the tubing to ensure a fluid-packed system.
7. Horizontal runs of tubing should be sloped slightly upward (2.5 to 5 degrees or a minimum of 1/2" rise per 1 foot of run) from the seal to the cooler to prevent low spots in the line.
8. Any valves in the circulation loop should be of the free-flowing type.
9. The pumping ring should be located in the gland rather than the seal chamber whenever possible so the clearance can be more tightly controlled.
10. Tangential in and out connections in the gland should be used to improve pumping ring efficiency.

Some potential problems that can arise when using this piping plan include:

1. The pumping ring may not provide circulation if the product becomes viscous after it is cooled.
2. All air should be vented from tubing and seal chamber prior to start-up. This will ensure a fluid-packed system and therefore, more efficient circulation.
3. As in API Plan 21, the heat exchanger will eventually become fouled. The best way to check for fouling is to compare the current product and cooling water temperatures into and out of the exchanger, to temperatures when the system was initially installed. Please note that a change in the flow rate of the product or cooling water can also affect these temperatures.

API Plan 32 ***(ANSI Plan 7332)***

API Plan 32 (*Figure 37*) requires the injection of a clean fluid from an external source to the seal. This plan is generally recommended when a single seal is used in a product with poor lubricating properties. A close-clearance throat bushing should be installed to isolate the pumped product from the seal chamber and to minimize the amount of flush fluid required. The flush should be introduced at a pressure of at least 25 psi, but not more than 50 psi over seal chamber pressure.

It is generally economically advantageous to eliminate external flushes. While they can extend seal life, thus reducing seal costs, the cost of product dilution or downgrading typically far exceeds any initial savings.

Plan 32

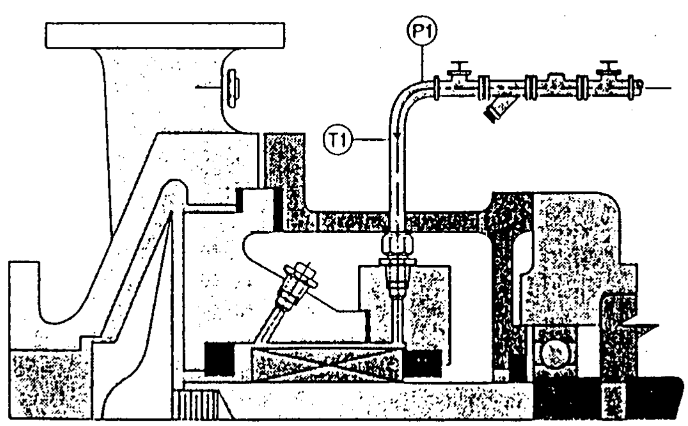


Figure 37

API Plan 52 *(ANSI Plan 7352)*

This plan promotes induced or forced circulation of a barrier fluid in a closed loop arrangement between the seal chamber and an unpressurized external supply tank *(Figure 38).*

API Plan 52 is used in conjunction with multiple seal arrangements to isolate the pumped product from the atmosphere or to extend service life by providing a better environment for the seal. It is commonly used in toxic, carcinogenic and flashing applications as well as fluids that change state when exposed to atmosphere.

The barrier fluid will lubricate the outer seal, while the inner seal is lubricated by the pumped product. To provide a driving force for the barrier fluid to flow across the outer seal, it can be advantageous to maintain a minimum pressure of 5-10 psig on the barrier fluid.

In this plan, any emissions of the pumped product past the inner seal will migrate to the barrier fluid. If the pumped product is immiscible (incapable of being mixed) with the barrier fluid, or has a higher vapor pressure, these emissions can safely be vented to the flare or any other vapor recovery system.

By using a low-leakage inner seal, zero emissions of the pumped product to atmosphere can be achieved. If the pumped product is miscible with the barrier fluid, or has a lower vapor pressure, the barrier fluid must be changed on a regular basis to prevent contamination and to provide essentially zero emissions of the pumped product to the atmosphere.

Some form of forced circulation (either a pumping ring or a circulating pump) should be used whenever possible. Thermal convection is sometimes used, however, barrier fluid flow is easily interrupted. In addition, thermal convection cannot be counted on for heat removal.

Because installation of this system is critical to its success, please keep in mind the following rules-of-thumb when installing an API Plan 52 with a pumping ring.

1. The distance from the bottom of the reservoir to the centerline of the shaft should be approximately 18 inches.
2. The horizontal distance from the reservoir to the seal should be as short as possible, with a maximum of three feet.
3. The size of the reservoir is generally determined by the shaft size. As a general rule, it should be one gallon per inch of shaft diameter, with a two gallon minimum. However, if following API-682, the reservoir should be sized to contain a minimum of five gallons (20 liters) of liquid.
4. The minimum barrier fluid level in the reservoir should be maintained at least one inch above the return line fitting. This ensures a fluid-packing system so the pumping ring will only need to overcome line losses.

Plan 52

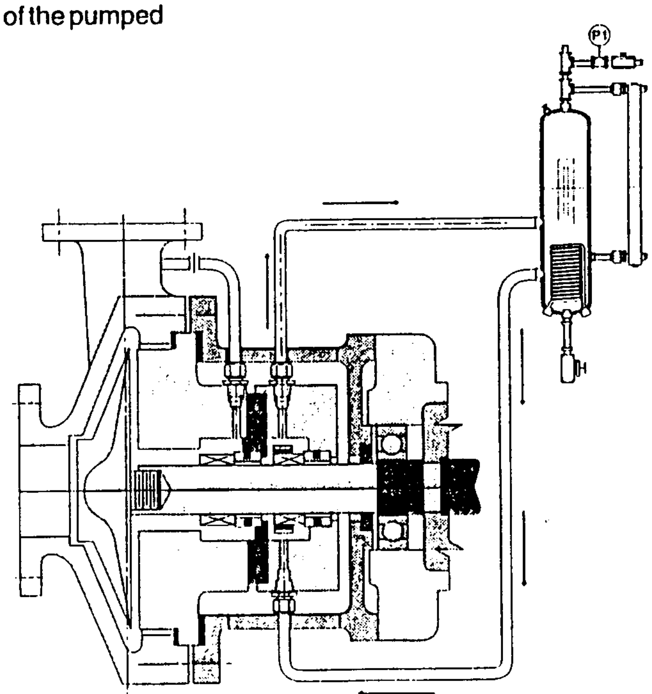


Figure 38

5. There should be no sharp elbows or bends in the tubing. A gentle sweep will yield better results.

6. The line from the reservoir to the seal and from the seal to the reservoir should be 1/2 inch (minimum) tubing (API - 3/4 inch minimum).

7. Horizontal runs of tubing should be sloped slightly upward (2.5 to 5 degrees or a minimum of 1/2" rise per 1 foot of run) from the seal to the cooler to prevent low spots in the line.

8. Any valves in the circulation loop should be of the free-flowing type.

9. The pumping ring should be located in the gland rather than in the seal chamber whenever possible so the clearance can be more tightly controlled.

10. Tangential in and out connections in the gland should be used to improve pumping ring efficiency.

The barrier fluid should be compatible with the pumped product and have good lubricating properties. The properties of some commonly recommended barrier fluids are given in the table below.

Since the seal chamber pressure is higher than the barrier fluid pressure, a failure of the inner seal can be detected by an increase in the barrier fluid level or pressure. If the reservoir is vented, an orifice in the vent line should be installed to ensure detection of excessive leakage.

API Plan 53 *(ANSI Plan 7353)*

This plan is identical in purpose and design to API Plan 52, except that it utilizes a pressurized barrier system *(Figure 39).*

API Plan 53 also is used in conjunction with multiple-seal arrangements to isolate the pumped product from the atmosphere or to extend service life by providing a better environment for the seal. It is used in the same applications as API Plan 52, as well as in products that are very poor lubricants.

When using API Plan 53, the barrier fluid will lubricate both the inner and outer seal faces. The small amount of barrier fluid that passes across the inner seal faces will enter the pumped product. The barrier fluid pressure should be maintained at least 10 percent above seal chamber pressure with a 25 psi minimum and a 150 psi maximum differential pressure. The insert for the inner seal should be rebalanced whenever the barrier fluid pressure is more than 75 psi over the seal chamber pressure. The stationary seal face should be retained when this piping plan is used.

This plan virtually ensures zero emissions to atmosphere. However, the barrier fluid can eventually become contaminated due to mixing at the inner seal faces. This contamination will occur much more slowly than with a non-pressurized system.

Fluid	Min. Temp.	Max. Temp.	Comments
Water	40°F/5°C	180°F/82°F	Use corrosion-resistant materials
Propylene Glycol	-76°F/-60°C	368°F/185°C	Must be mixed with water to reduce viscosity
N-Propyl Alcohol	-147°F/-100°C	157°F/70°C	
Automatic Transmission Fluid	55°F/130°C	200°F/93°C	Contains additives
Kerosene	0°F/-18°C	300°F/150°C	
Diesel	10°F/12°C	300°F/150°C	Contains additives

Since the barrier fluid is at a higher pressure than the seal chamber pressure, an inner seal leak can be detected by a decrease in the barrier fluid levels. As in API Plan 52, thermal convection should not be counted on for circulation of the barrier fluid due to its lack of reliability. Also, cooling coils can be used to prevent flashing of the pumped product or barrier fluid.

Again, the piping and installation of this system is critical to its success. The same rules-of-thumb given for API Plan 52 are applicable here. Also the same barrier fluids listed for API Plan 52 can be used here.

Plan 53

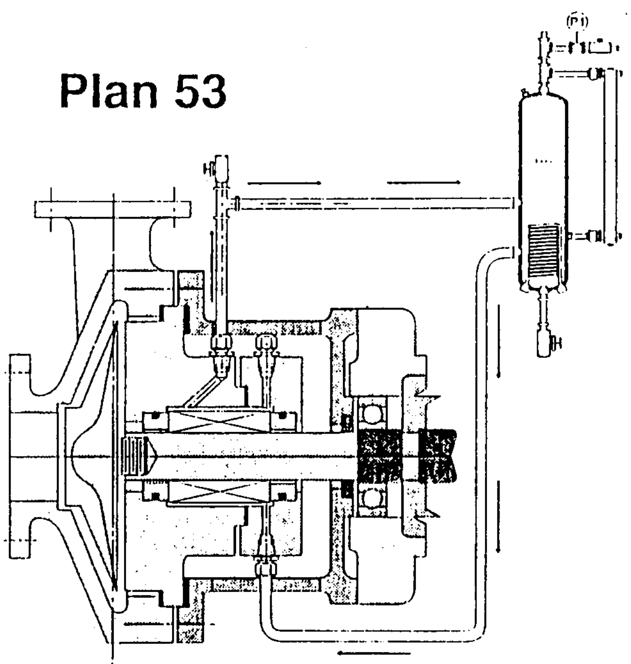


Figure 39

API Plan 62 ***(ANSI Plan 7362)***

API plan 62 *(Figure 40)* provides for an external quench to be delivered through a gland connection from the low pressure side of the seal to the seal's I.D. The quench fluid is used for cooling, heating or cleaning, depending upon the pumped product. The quench fluid is contained using a throttle bushing, lip seal, auxiliary packing or an auxiliary sealing device.

A quench is most frequently used in applications where the pumped product changes state when exposed to the atmosphere. For example, steam is frequently used to prevent coke formation in hydrocarbon applications exceeding 350°F (175°C). Also a steam or water quench can prevent buildup of crystalline deposits in caustic applications, and water is used to cleanse the atmospheric side of a single seal in a variety of other applications. A quench can provide the additional benefit of helping to cool the pumped product to prevent flashing, or to warm the pumped product to keep it from setting up.

Plan 62

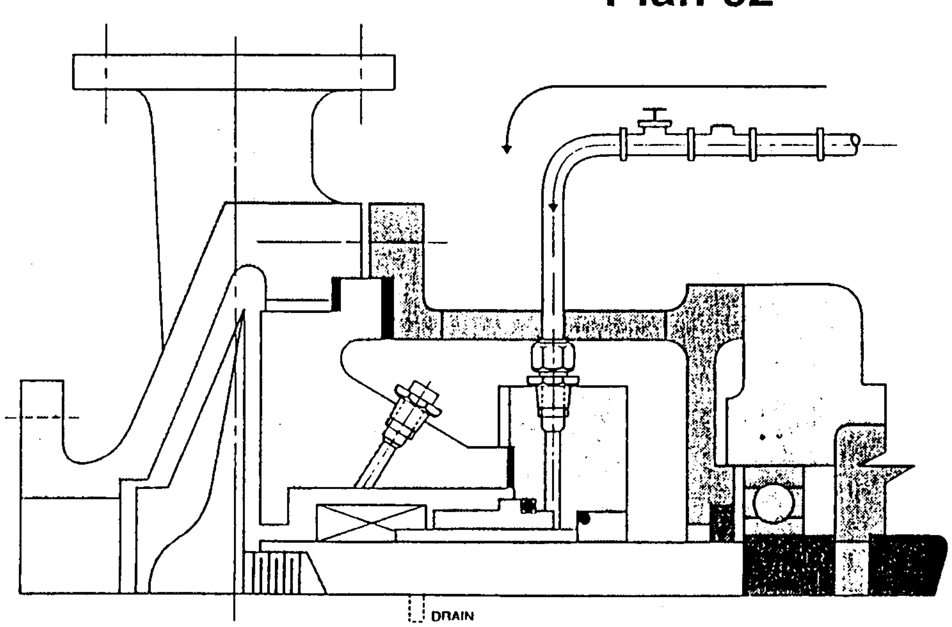


Figure 40

The quench should be regulated so that there is minimal amount of flow to the atmospheric side of the seal. The most common methods of regulation are needle valves and pressure regulators. Needle valves are generally preferred because they are typically more reliable at these pressures (less than 5 psi) than pressure regulators.

Problems that can be encountered when using a steam quench include:

1. Excessive quench flow can damage the bearings. This can be prevented by reducing the quench flow or installing an auxiliary sealing device.

2. A wet steam quench should not be used, because it can enter the gland as a liquid and vaporize near the seal faces, causing the faces to pop open.

3. The quench should be turned on before the pump is started, particularly when steam is being used in a high temperature application. If it is turned on after the pump is started, a slug of water could damage the seal.

4. The needle valve will sometimes plug during operation. If this occurs, precautions must be taken to prevent the seal from being hit with a slug of cold water when unplugging the line. This can be accomplished by installing a block valve and blowdown on the line downstream of the needle valve *(Figure 41)*.

This concludes the summary of frequently used API piping. Following is a brief discussion of other API piping plans.

API Plan 1 ***(ANSI Plan 7301)***

This plan calls for an internal recirculation from pump discharge to the seal and is seldom used. It is essentially the same as the API Plan 11, except there is no external piping.

API Plans 12, 22, 31 and 41 ***(ANSI Plans 7312, 7322, 7331 and 7341)***

API Plans 12 and 31 are variations of API Plan 11; and API Plans 22 and 41 are variations of API Plan 21; but include a strainer or cyclone separator in the piping to improve the seal chamber environment.

API Plan 51 ***(ANSI Plan 7351)***

This plan provides for the use of a dead-ended barrier blanket typically used with an auxiliary sealing device. It can be used for light hydrocarbon or cryogenic products to prevent icing at the seal faces. However, multiple-seal arrangements are generally recommended for cryogenic applications, and methanol blanket is not necessary when sealing light hydrocarbons with a single seal.

API Plan 54 ***(ANSI Plan 7354)***

This plan provides for the circulation of a clean fluid from an external source and is typically used in multiple-seal arrangements. It is essentially the same as the API Plan 53. The only difference is that there is no barrier fluid reservoir for each pump. The barrier fluid comes from an external system that may supply several pumps. The barrier fluids recommended earlier also apply here.

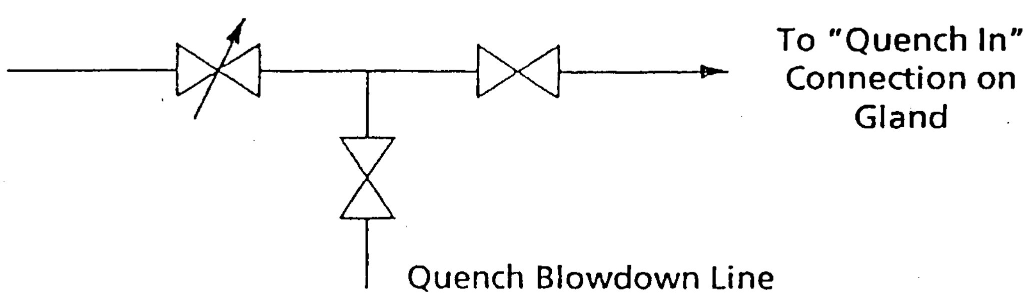


Figure 41

Mechanical Seals

PROPERTIES OF FLUIDS

What are they and what do they do? Mechanical seal applications are divided into 14 major services.

A. Crystallizing fluids

B. Abrasive fluids (slurries)

C. Fluids that set up or harden

D. Fluids that flash

E. Dangerous fluids

F. Hot fluids (over 400°F/200°C)

G. Cryogenic fluids

H. Non-Lubricating products

I. Corrosive fluids

J. High speed (over 5000 sfpm)

K. High vacuum service

L. Water (above and below 180°F/80°C)

M. High pressure

N. Light specific gravity

A. Crystallizing Fluids *(also Hydrocarbon Coking)*

Fluids that crystallize or convert to a solid after passing across the faces, encompass a wide group of products, including salts, caustics and hydrocarbons.

Circulation of the product from pump discharge keeps fresh fluid in the seal chamber at all times. Using a quench fluid from the atmosphere side of the seal keeps the area under the seal faces clean. Quench fluid also excludes air from this area and provides heating or cooling as necessary. Quench fluids should be selected for compatibility with the pumped product and availability at the pump site.

Selection of seal materials, face materials and secondary elastomeric seals depends upon chemical compatibility and temperature capabilities.

B. Abrasive Fluids

Abrasive fluids exist in basically all industries, from slurries in power plants and mining complexes to chemical and petrochemical processing plants which handle wide varieties of abrasive fluids from solvents to hydrocarbons.

The key to successfully sealing abrasive fluids is to keep the abrasive solids in suspension and select seal face materials that will withstand the abrasive attack. Fluid characteristics, such as hazardousness or corrosiveness, need to be considered when selecting seal materials, and determining the use of circulation and quench arrangements for seal protection systems.

In some cases, flush fluids may not be available or cannot be used in a certain process. In these situations, connecting the circulation line from the pump discharge to the seal chamber lantern ring connection or gland circulations connection will remove heat from the seal area and provide a constant flow of fluid in the seal chamber.

Welded metal bellows seals have proven to operate very well in abrasive media without the use of an external flush.

C. Fluids that Solidify or Harden

Thermosensitive and viscous fluids can harden or solidify in the pump seal chamber if heated or cooled beyond certain limits. Examples of these fluids include resins, asphalt, heavy crude oils, molasses and grease.

Pumps equipped with jacketed seal chambers and/or jacketed glands may be used to handle these products. Vents and drain gland connections should be used as needed to provide a steam quench to heat or cool the seal area and provide cleaning.

Hard face seal combinations are normally recommended with a tungsten carbide rotary face and a Sealide™ stationary seat. Drive lugs in the rotary seal head may be necessary in viscous applications. Selection of the seal materials and secondary seals are dependent on chemical compatibility and temperature limitations.

D. Fluids that Flash

Flashing results when the operating temperature/pressure of the fluid causes it to change into a vapor in the seal chamber area. Flashing generally occurs before the fluids passes all the way across the faces.

As the liquid flashes to a vapor, the seal faces pop open, allowing excess product between them. Any abrasive particles subsequently trapped between the faces may cause severe wear to the seal faces, resulting in premature seal failure.

The key to these applications is to keep the product in a liquid state as far across the seal faces as possible.

A rule of thumb for fluids that flash is to maintain 1) the seal chamber temperature 25°F (-4°C) below the flash or boiling point of the liquid and 2) the seal chamber pressure 25 psi (1.7 bar) above the vapor pressure threshold. However, each application should be examined based on its own characteristics for proper seal recommendations.

Temperature in the seal chamber area can be regulated by using the API piping plans discussed earlier. Pressure can be increased by the use of API Plan 11 in conjunction with a close-clearance throat bushing in the bottom of the seal chamber.

E. Dangerous Fluids

This group generally includes toxic, carcinogenic and other fluids determined to be hazardous to human life.

To eliminate any leakage or contamination into the atmosphere, these applications are generally handled with double or tandem seal arrangements with pressurized barrier systems. The barrier fluid must be compatible with the pumped product fluid. The barrier fluid pressure should be 10% higher than the seal chamber pressure with 25 psi minimum differential pressure to ensure no leakage of the pumped product into the atmosphere.

F. Hot Fluids
(over 400°F/200°C)

These fluids are generally present in the petrochemical industry, particularly in oil refineries. They are also present in many other industries. Selection of seal materials and secondary seals is critical in these applications.

Hot temperatures can lead to accelerated corrosion rates, coking, flashing, or hardening of the product in the seal chamber. It is therefore frequently necessary to control the sealed fluid temperature through the use of flushes, gland jackets, pump jackets, and steam quenches.

Please note that when pump jackets are used, the fluid in the seal chamber should not be circulated unless API Plan 23 is used. The seal chamber is normally dead ended.

Many major oil companies classify applications above 350°F to 400°F (175°C to 200°C) as high temperature, with an upper limit of 800°F (425°C). The selection of mechanical seals specifically designed to function in these high temperatures is paramount for peak performance. Desirable design features for metal bellows seals in the high temperature category are:

a. Ability to retain its spring tension through temperature cycles from ambient to 800°F (425°C). This is accomplished with the use of metals such as heat-treated AM350 stainless steel, and Inconel 750 and Inconel 718 in special applications.

b. Ability to retain the seal face insert in the adaptor shell at high temperatures. Many metal bellows seals utilize Alloy 42 with its inherent low thermal expansion to maintain face retention at elevated temperatures. However, this material is not chemically compatible with all high temperature applications.

   Sealol has patented a shell design, made of a corrosion-resistant material, that solves the problem of face retention and maintains face stability through a wide range of pressures and temperatures.

c. Suitable elastomers. With a development of Grafoil, an expanded graphite fiber tape, the high temperature, elastomer problem was eliminated.

G. Cryogenics

These applications involve liquids at extremely low temperatures (< -100°F/- 73°C). These low temperatures are typically required to maintain the fluid in a liquid form. Most cryogenic fluids have poor lubricating properties.

Tandem seals, with a barrier fluid between them, are generally required for cryogenics. Compatibility with the cryogenic fluid, lubricity, and availability at the pump site should be considered when selecting barrier fluids. Material selection is critical in these applications.

Mechanical Seals

H. Non-Lubricating Fluids

For pumped products that have little or no lubricating characteristics, a method to replace that fluid in the seal chamber area must be determined.

This can be accomplished by introducing a flush fluid, with lubricating capability, into the seal chamber area and installing a close clearance throat bushing in the bottom of the seal chamber to restrict the flow of the pumped product. In practice, the flush fluid is sealed rather than the pumped fluid. Compatibility with the pumped product must be considered when selecting a flush fluid.

Tandem and double seal arrangements with pressurized barrier systems have proven to be effective in these applications. The barrier fluid provides the face lubrication for both sets of faces.

I. Corrosive Fluids

Corrosive fluids are present in almost all industries. As temperature increases, corrosion rates are accelerated. Corrosion in excess of .002" per year is detrimental to most mechanical seal designs available today.

The solution to these applications lies in the selection of the correct materials. Select metal components (except bellows or springs) for the seal as follows:

1. If the pump is made of iron, steel or stainless steel, then use stainless steel for the seal body.
2. If the pump is made of another material, use the same material for the seal body if it is available.
3. If the pump is non-metallic, the best choice would be an outside-mounted TFE bellows seal.

   Mixtures of various corrosive fluids create even more problems. Selection of an outside TFE Bellows seal is often the best choice for these applications, provided you are within the pressure and temperature limitations for outside seals.

   Selection of seal face materials and secondary seal elastomers is also very important regarding their chemical compatibility to the corrosive environment.

J. High Speed (over 5000 sfpm)

Sealol has always been the leader in high speed sealing since the early days of the aircraft industry. High-speed sealing with minimum lubrication led to the development of stationary seal heads and rotating seats. These are used today on applications where peripheral speeds exceed 5000 sfpm. Sealol recommends stationary bellows and rotating seats.

The formula for peripheral speed in ft./min. is:

$$\frac{\text{Seal Face O.D. x rpm x 3.14}}{12}$$

Note: Seal Face O.D. is represented in inches.

K. High Vacuum Service

Hydraulically balanced bellows seals perform well in vacuum service as do double and tandem seal arrangements. When a single seal is used, the seal chamber pressure must be increased by injecting a flush, either from discharge or an external source, and installing a close-clearance bushing.

L. Water

There are two categories for sealing water: applications below 180°F (80°C) and applications above 180°F (80°C). Applications below 180°F (80°C) are relatively simple and can be easily sealed using the proper materials. Applications above 180°F (80°C) are typically cooled to prevent flashing of the liquid between the faces, leaving behind any undissolved solids as abrasive crystals.

The following means are recommended to cool the seal chamber if necessary:

1. Water-cooled seal chamber associated with a pump jacket.

2. Attach a pumping ring to the shaft behind the rotary seal head to circulate water out of the seal chamber, through a cooling device, and back through a circulation connection on the gland (API Plan 23/ANSI Plan 7323).

3. For applications less than 300°F (150°C), a circulation line from the pump discharge to a gland connection at the sea faces is adequate.

M. High Pressure

The definition of high pressure in the seal industry begins with 125 psi (8.5 bar) seal chamber pressure, and extends to a normal top limit of 1000 psi (69 bar). However, applications in the 1500 psi (103 bar) range appear in the nuclear, oil field injection, boiler circulation pumps and special mixing equipment industries.

Welded metal bellows seals all have a standard seal chamber pressure rating of 300 psi (20 bar). Above this point Sealol recommends its patented DOUBLE-PLY™ design, which raises the pressure capability to over 1000 psi (69 bar).

N. Light Specific Gravity

The refining industry normally classifies liquids with a specific gravity of below .6 as "Light ends". These products range progressively from C1 hydrocarbons, such as methane, through ethylene, ethane, propylene, propane, isobutane, butane, butadiene, to C5 hydrocarbons of pentane and isopentane.

Light ends require a certain amount of pressure to keep the liquid from vaporizing at any given temperature. This is known as the vapor pressure. The lighter the gravity, the higher the pressure required to keep it liquid.

Tandem seals are usually required for methane, ethylene and ethane applications, while single seals can be used on propylene, propane, butane and hexane. However, environmental and safety considerations often dictate tandem seals for all of these products.

For light ends ,which usually are non-corrosive, Sealol prefers welded metal bellows seals with carbon and Sealide faces and Viton o-rings. For pressures above 300 psi (20 bar), Sealol recommends DOUBLE-PLY Inconel bellows for added strength. Inconel and Hastelloy bellows are used for corrosive, light ends applications.

For hydrocarbons with specific gravities of .4 to .55, pressures usually range around 300 psi (20 bar). To keep these fluids in a liquid form, the seal chamber pressure must be 25 to 30 psi (2 bar) above the vapor pressure of the fluid in the seal area. This may require a close clearance bushing in the bottom of the seal chamber.

Narrow face diameters and higher face balance increase seal stability. Along with the need for pressure, flow of the flush fluid also required to take away heat generated by turbulence and the frictional heat at the seal faces. This can be accomplished with API Plan 11.

Tandem seals in methane, ethylene, and ethane require a moisture drying process, using methanol, to prevent ice formation in both the pump and seal chamber before start-up. Ethylene glycol and water mixture, automatic transmission fluid, methanol, or silicon-based lubricants can be used as barrier fluids for start-up and operation.

Sealol is doing extensive research on light ends sealing to prevent seal face convergence or tipping under high pressures. This research has resulted in a new design of bellows seals for this application.

X. APPLICATION ENGINEERING PROCEDURES

To properly select a seal for a particular application you must consider the following group of questions. The use of a data sheet will ensure these questions are answered as well as provide the necessary information to the seal supplier for proper seal recommendation. API 610 and Sealol provide data sheet formats for this purpose.

A. Single Seal?
   1. Inside
   2. Outside
B. Double Seal?
   1. Back-to-Back
   2. Face-to-Face
C. Tandem?
D. Component or Cartridge Seal?
E. Rotating or Stationary Seal Head?
F. Pusher or Bellows Seal?
G. Balanced or Unbalanced?
H. Secondary Seals?
I. Face Combinations?
J. Materials of Construction?
K. Seal Protection Systems?

The answers to these questions should greatly enhance your chances for selecting a seal that will perform well for your specific applications.

Glossary

A

ANSI/ASME: American National Standards Institute/ American Society of Mechanical Engineers. Develops chemical process pump standards and specifications.

anti-rotation pin: A device used to prevent rotation of one seal component (usually the stationary seat) relative to an adjacent component (usually a gland plate or seal housing).

API: American Petroleum Institute. Develops standards and specifications for centrifugal pumps for general refinery service.

ASLE: American Society of Lubrication Engineers.

asperities: Minute imperfections in a seal face that are the result of normal surface finishing.

axial movement: Movement along the axis parallel to the center line of the shaft, usually occurring in both directions.

B

balance diameter: That diameter of a face seal at which the resultant force is considered to be acting. This force is obtained from area integration of the pressure profile which exists on the fore and aft surfaces of the axially moveable portion of the seal assembly. For a balanced seal, the secondary seal and diameter and the balance diameter are the same.

balanced seal: A mechanical seal designed to divert a portion of a system's hydraulic pressure to substantially reduce or balance the hydraulic pressure acting to close the seal faces.

balance holes: Openings near the hub of an impeller of a centrifugal pump that reduce the seal chamber pressure by equalizing the pressure behind the impeller with the suction pressure of the pump.

barrier fluid: A fluid which is introduced between two seal elements to completely isolate the pumped product from the environment.

bellows convolution: In a welded bellows, an assembly of two single-ply or multiple-ply formed plates, welded at the I.D.

bellows pitch: The distance between convolutions.

bellows plate: A single metallic plate or disc. When adjacent plates are alternately welded together at their inner and outer edges, they form the bellows core.

bellows seal: A type of mechanical seal which utilizes bellows for providing secondary sealing and spring-type loading.

bellows span: The distance between the I.D. and O.D. of the bellows core.

bushing: A square or rectangular cross section device used to restrict fluid flow.

bypass flush: An environmental control whereby a line is taken from the pump discharge and is piped into the seal chamber area to provide fluid flow to the seal chamber product. A bypass flush can also be attached from the seal chamber area to the pump suction.

C

cartridge seal: A completely self-contained assembly including seal, gland, sleeve, mating ring, etc., usually needing no installation measurement.

cavitation: A condition in which vapor or gas bubbles form locally in liquids, as a result of an abrupt decrease in pressure. The subsequent collapse of these bubbles causes high local impact pressure which can contribute to equipment wear and reduced seal life.

circulation connections: The inlet and outlet ports of the seal chamber or seal gland that receive the piping for circulating liquid in close proximity to the mechanical seal.

closing force: The sum of the seal chamber fluid pressure and the mechanical seal spring force acting together to maintain seal face contact in opposition to the opening force.

closing mechanism: In a mechanical face seal, that device (either springs or a bellows) that provides an axial force to push the faces together.

coking: The severe oxidation of a hot hydrocarbon into a hard, black carbon deposit. Coking is usually encountered on the atmospheric side of high temperature seals and can hang up seal components.

collar: A band or ring, usually attached by set screws to a shaft, that acts to set and attach the rotary unit of a mechanical seal to the shaft.

Glossary

compression packing: A material such as carbon, flax, TFE, or synthetic fiber designed to be packed into the seal chamber to reduce product leakage. It is usually held in place by an adjustable packing gland.

compression set: The difference between the thickness of a gasket or static seal before the seal is compressed and after it is released from compression. Compression set is normally expressed as a percentage of the total compression.

D

dead-ended: A term used to describe a seal chamber that has no circulation.

differential pressure: The difference in pressure between two points in a system, e.g. the difference between the discharge pressure and suction pressure in the pump.

diffuser: Stationary vanes surrounding the impeller to channel flow. Used to convert velocity into pressure energy.

DIN: Deutsches Institut für Normung e.V. German industry standard.

double seal: Two mechanical seals designed to permit a liquid or gas barrier fluid between the seals mounted back-to-back or face-to-face.

dry running: A term describing a mechanical seal that is in operation without liquid lubrication between its faces.

durometer reading: An index which is used for ranking the relative hardness of rubber and plastics.

dynamic seal: A seal that moves due to axial or radial movement of the unit.

E

externally pressurized seal: A seal that has pressure acting on the seal parts from an external independent source of supply.

extrusion: The displacement of part of a seal's secondary element (such as an O-ring) into a gap under the action of fluid pressure.

eye: The inlet center of a centrifugal pump's rotating impeller.

F

face pressure: The face load (computed as the sum of the pneumatic or hydraulic force and the spring force) divided by the contacting area of the sealing face or lip. For lip seals and packings, the force load also includes the interference load.

face seal: A device that prevents leakage of fluids along rotating shafts. Sealing is accomplished by a stationary primary seal ring bearing against the face of a mating ring mounted on a shaft. Axial pressure maintains the contact between the seal ring and the mating ring.

film thickness: In a dynamic seal, the distance separating the two surfaces which form the primary seal of a mechanical seal.

flashing: A rapid vaporization of a liquid. This frequently occurs when frictional energy is added to a liquid as it passes between the primary seal faces or when the pressure of the liquid falls below the liquid's vapor pressure because of a pressure drop across the sealing faces.

flatness: In sealing technology, the degree to which the surface of a seal face deviates from a perfect plane. Expressed in helium light bands where one helium light band is equal to 11.6 millionths of an inch (.0000116").

flexible members: That portion of the seal pertaining to the springs or bellows.

flush: A small amount of fluid introduced into the seal chamber in close proximity to the sealing faces, usually to cool or protect the seal faces.

formed bellows: A thin, solid metal cylinder formed into parallel corrugations, that is able to flex to permit loading and movement of seal faces and to prevent leakage.

free length: The uncompressed axial length of a face seal assembly. The term also applies to a spring or bellows.

fretting: A combination of corrosion and wear that occurs when a secondary seal continuously wipes the protective oxide coating from a shaft or shaft sleeve.

friction drive: The use of O-rings or a rubber bellows to transmit motion from the shaft to the rotary unit of a mechanical seal.

Glossary

G

gasket: A deformable material used between two static surfaces to prevent leakage.

gland plate: An end plate that connects the non-rotating assembly of a mechanical seal to the seal chamber.

H

hard face: Either a stationary or rotating seal face which is made of, or coated with, a material of high hardness. Common hard face materials are silicon carbide, tungsten carbide, ceramic, and Stellite.

harmonic resonance: A rhythmic harmonic motion having a specific vibration frequency.

head: the height of a column or body of fluid above a given point, expressed in linear units.

heat check: Heat checking is a condition of minute radial cracks on and beneath the surface of a hard face caused by highly localized thermal stresses. This phenomenon is usually associated with some discoloration on hard metals.

hydraulic balance: The ratio of two areas: The area of the sealing face that is bound by the balance diameter and the I.D. of the sealing face, and the area that is bound by the outer and inner diameters of the seal face contact area.

I

impeller: A rotating element of a centrifugal pump driven by a motor or turbine. The impeller has vanes or grooves to impart rotary velocity to the product.

insert: The seal face that is located in the shell of a bellows assembly.

Inside-mounted seal: A mechanical seal located inside the seal chamber with the pumped product's pressure at its O.D.

L

lantern ring: A device used to separate rings of compression packing within a pump seal chamber. It permits the injection, under pressure, of a fluid to provide cooling and lubrication.

lapping: A finishing operation using small, free-floating abrasives in a paste on a flat surface or diamond-charged plates. Used to produce an extremely smooth and flat surface on a mechanical seal face.

leakage rate: The quantity of fluid passing through a seal in a given length of time.

lip seal: An elastomeric or metallic seal that prevents leakage in dynamic and static applications by a scraping or wiping action at a controlled interference between itself and the mating surface.

M

magnetic seal: A seal that uses magnetic material (instead of springs or a bellows) to provide the closing force that keeps the seal faces together.

mating ring: A disc or ring-shaped member mounted on the shaft or in a gland. It provides the primary seal when in proximity to the face of an axially adjustable face seal assembly.

metal fatigue: A condition usually caused by bending and flexing of a metal part, characterized by a decrease of physical properties such as tensile strength. Metal failure due to repeated or fluctuating stresses which are frequently below the yield strength of the material.

monochromatic light: A device used in conjunction with an optical flat to determine the flatness of a seal face. The helium lamp gives off a yellowish/orange color light having a wavelength of 23.3 millionths of an inch. The wedge of air between the optical flat and the seal face is measured in half-wavelengths or light bands equal to 11.6 millionths of an inch.

O

opening force: The portion of the seal chamber fluid pressure found between the mating seal faces. Opening force partially balances the closing force and reduced seal face wear.

operating length: For an installed seal face assembly, the axial distance from the seal face to a reference plane.

optical flat: A transparent disc (usually of fused quartz) that has been lapped flat and polished less than one light band. It is used to measure seal face flatness in conjunction with a monochromatic light source.

Glossary

O-ring: An elastomeric, doughnut-shaped sealing device used as a gasket or as a secondary seal in mechanical seals.

O-ring groove: The space into which an O-ring is inserted and retained.

outside-mounted seal: A mechanical seal with its seal head mounted outside the seal chamber that holds the fluid to be sealed. Outside seals have the pumped fluid's pressure at their I.D.

P

packing gland: A non-rotating device attached to the seal chamber. It is used to apply an adjustable pressure to compression packing to create a seal against a rotating or reciprocating shaft.

pilot or piloting: A feature of a stationary insert, seal chamber face, or gland face used to center them around the shaft.

pitting: Surface voids usually caused by mechanical erosion, chemical corrosion or cavitation.

positive drive: The use of pins, lugs, bellows and other devices to transmit motion from the shaft to the rotary unit of a mechanical seal.

primary seal: The closure at the mating faces.

product: This is the key component of a process stream. Also referred to as pumped product.

pressure (absolute) PSIA: The sum of gauge pressure and atmospheric pressure.

pressure (atmospheric): The force exerted on a unit by the weight of the atmosphere. The pressure at seal level is 14.7 psi.

pressure (gauge) PSIG: Corrected pressure: the difference between a given pressure and that of the atmosphere.

pumping ring: A device attached to a rotating shaft within a seal chamber. It circulates fluid through a closed loop for cooling purposes.

pump shaft rotation: Either clockwise or counterclockwise as viewed from the driver end.

pusher seal: A mechanical seal in which the secondary seal is pushed along the shaft or sleeve to compensate for misalignment and face wear.

P-V factor: An arbitrary term which is the product of face pressure and relative sliding velocity. The units customarily used are pounds per square inch and feet per minute. The term is normally considered to provide some measure of severity of service and thus relates to a seal's wear life.

Q

quench: A fluid that is introduced on the atmospheric side of the seal to 1) dilute any fluid that may have leaked across the seal faces; 2) to cool the fluid coming across the faces; 3) to help control the temperature of the seal.

R

radial movement: Movement perpendicular to the shaft.

rotary seal: A mechanical seal which rotates with a shaft and is used with a stationary mating ring.

rotary unit: The portion of a mechanical seal that rotates with the shaft.

run-in: The initial period of operation of a seal or compression packing, when the rate of wear and leakage is at its highest.

runnout: Twice the distance that the center of a shaft is displaced from the axis of rotation.

S

seal: A device designed to prevent the movement of fluid from one area to another, or to exclude contaminants.

seal assembly: A group of parts, or a unitized assembly, that includes sealing surfaces, provisions for initial loading, and a secondary sealing mechanism that accommodates the radial and axial movement necessary for installation and operation.

seal chamber: The area between the seal chamber bore and a shaft in which a mechanical seal is installed.

seal face: It is either of the two lapped surfaces in a mechanical seal assembly forming the primary seal.

seal face width: The radial distance from the inside edge to the outside edge of the sealing face.

Glossary

sealing fluid: A fluid used to lubricate the seal faces, usually the pumped product.

secondary seal: Any O-ring, V-ring, wedge, gasket or bellows used to form seals between seal components, shafts and glands.

shaft deflection: The distance the shaft is displaced from its design centerline. A dynamic condition resulting from radial hydraulic forces acting on the impeller.

shaft diameter: The outside diameter of the shaft.

shaft sleeve: A cylindrical device placed over the shaft to protect it from wear and corrosion. It may also be used to provide spacing for the impeller, or as a device to provide a step in the shaft to achieve balance or as a device upon which a cartridge seal can be mounted.

sintered: Fired at high temperature. A technique to bond materials such as carbon, silicon carbide, and tungsten carbide.

slinger: Resembling a washer, a device that is mounted on the shaft to impart radial movement to liquids and contaminants to isolate them from seals and bearings. Also called a "flinger".

specific gravity: The relative measure of the fluid's density as compared with water. The specific gravity of water at 60°F is 1.0.

spiral-wound gasket: A closure device formed by winding a metal and filler into a spiral configuration. It is used as a high temperature gland gasket.

spring: A machined element which is capable of storing energy and releasing it, as required. The most commonly used type is the coil spring. In face seals, its principal use is to keep the faces of the primary sealing elements together. When used for such purpose, it is considered to be a part of the seal head.

spring free length: The length of the spring when not compressed.

spring rate: The force required for extending or compressing the working length of a spring one unit of distance.

starting torque: The torque required to initiate rotary motion.

static seal: A seal between two surfaces which have no relative motion.

stationary ring: A ring that is mounted in or on the non-rotating assembly. Normally, it is the main sealing member that loads against the rotating seal ring.

stationary seal: A mechanical seal in which the flexible members do not rotate with the shaft.

stick-slip: A function phenomenon which can be described as a jerky motion which sometimes results when one surface is dragged against another. Normally it is associated with a non-lubricated or boundary-lubricated condition.

seal chamber: An enclosure forming a cylindrical cavity around a rotating or reciprocating shaft. Packing material is stuffed and compressed in this cavity to prevent or limit product leakage. The term is often used to describe the seal cavity or stuffing box.

surface finish: The smoothness of a shaft, sleeve, housing, gland, etc., expressed in root mean square (RMS).

T

tandem seal: An arrangement of two seals mounted one after the other, with the faces of the seal heads oriented in the same direction.

TFE (Tetrafluoroethylene): Most commonly referred to as Teflon, DuPont's brand name for its PTFE product.

throat bushing: A rectangular cross-section device used to restrict flow between the seal chamber and the pump volute.

torque: In sealing technology, the force is required to rotate a shaft against the resistance caused by the frictional drag at the seal faces. It is normally expressed in foot/pound or inch/pound units.

U

U-cup: A U-shaped secondary seal usually made of TFE or rubber..

Glossary

unbalanced seal: A mechanical seal arrangement wherein the full hydraulic pressure of the seal chamber acts to close the seal faces.

uni-directional seal: A seal that prevents the passage of fluid from one direction only.

V

vacuum: Refers to the pressures below atmospheric.

vaporization: The process by which a liquid becomes a gas.

vapor pressure: Also called the boiling point. A pressure at a given temperature below which a liquid will convert to a gas. It is measured in pounds per square inch absolute (psia) and is a function of the temperature of the liquid.

vent connection: A connection in the gland plate outboard of the seal through which leakage may be vented. Also the inlet connection for quench fluids.

vibration dampener: A frictional device added to a metal bellows seal to prevent the seal from going into harmonic vibration.

viscosity: The properly of a fluid which causes it to offer resistance to shear stress.

volute: That portion of a centrifugal pump casing which houses the impeller.

V-Ring: A V-shaped secondary seal, usually made of TFE.

W

wave spring: A disc washer-type of spring that has been deformed to have a multiple wave pattern in a plane perpendicular to its axis. Since it utilizes little axial space, it is frequently used to produce compact seal assemblies.

wear track: The mark or path worn on the mating ring seal faces during use.

wedge-type seal: A type of secondary seal, or wedge-shaped cross sections sometimes used in mechanical seals, usually made of TFE.

weepage: A minute amount of liquid leakage across a seal face. It has rather arbitrary limits, but is commonly considered to be a leakage rate of less than one drop of liquid per minute.

welded metal bellows: A seal assembly bellows fabricated by welding together a series of thin metal plates to form an accordion-type structure. When assembled to other components in the seal assembly, it acts as both the secondary seal and the spring loading mechanism.

wipe: Any excess radial movement of the seal faces across each other.

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