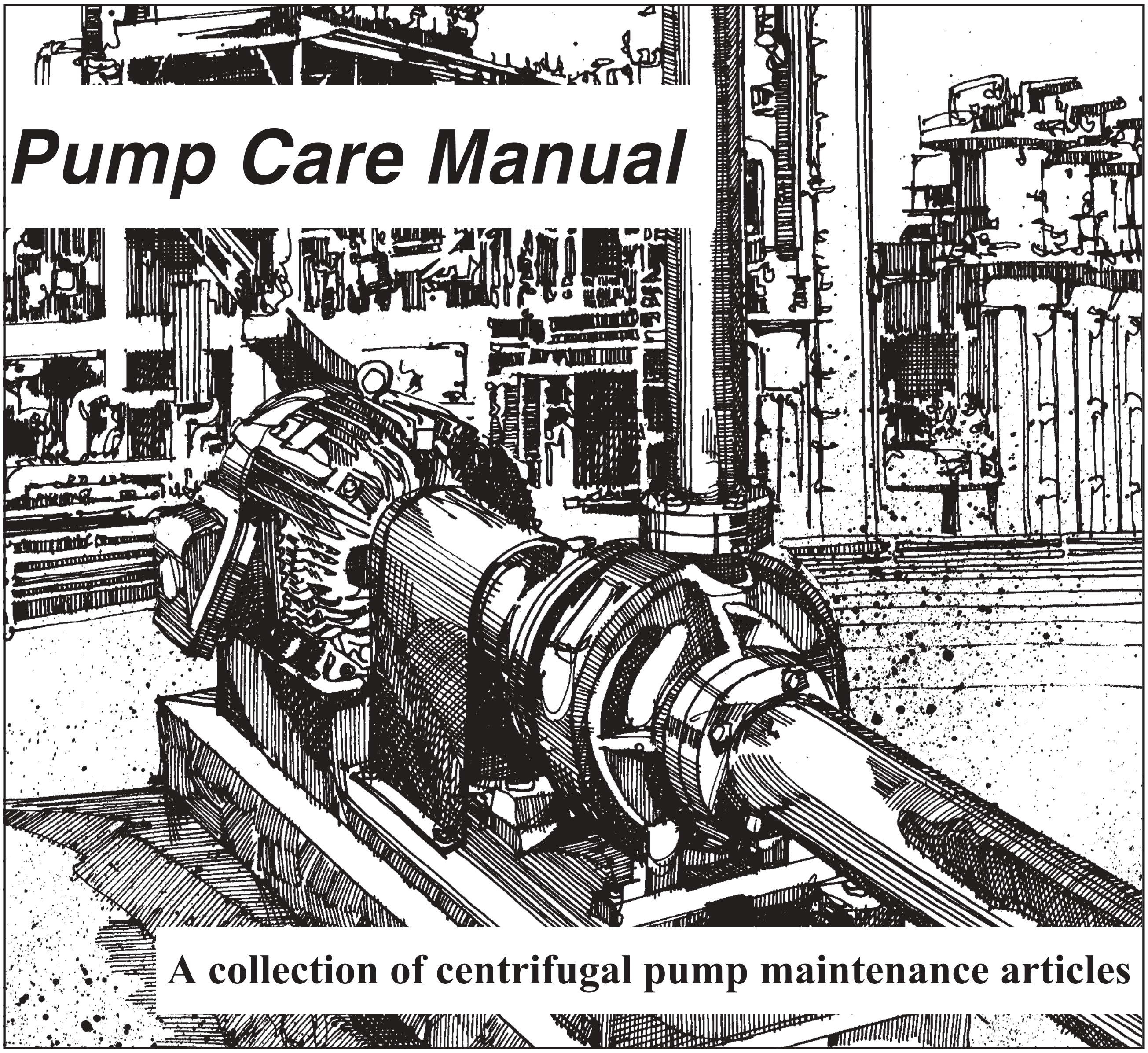
GOULDS PUMPS

Pump Care Manual



A collection of centrifugal pump maintenance articles

Goulds Pumps

ITT Industries

Engineered for life

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INSTALLATION

Very often problems with pump and motor bearings, shaft breakage and difficulty in aligning units stem from the improper installation of the pumping equipment. Inadequate foundations, bent or twisted baseplates, and excessive pipe strain can all work together to make an otherwise simple installation an operational and maintenance headache for the life of the unit.

Here are some tips on how you can prevent these things from causing increased downtime and higher than expected maintenance costs.

YOU NEED A GOOD FOUNDATION

The importance of a good foundation cannot be overemphasized. The foundation must be heavy enough to provide rigid support for the pump, motor and base combination and to resist the normal forces encountered when the unit is in service.

A concrete foundation firmly attached to a well designed floor system would be the most satisfactory arrangement. When site conditions make it necessary to support the pump on a metal structure, the structural members should be selected and designed to provide adequate stiffness and mass to support the unit and dampen any vibration that may be present. Unsupported wood floors and thin concrete pads should be avoided.

BASEPLATE INSTALLATION

Re-alignment of the pump and driver is required prior to final grouting of the baseplate. Even though the unit may have been aligned at the factory prior to shipment, components tend to shift in transit, during handling, or because of uneven foundation surfaces. Factory alignments cannot be depended upon during or after installation.

With the baseplate installed on the foundation you need to adjust the baseplate as necessary to level it and provide a flat surface for the pump and driver to rest. In some installations there are strict baseplate flatness guidelines which need to be adhered to. A good rule of thumb is to level the base to within .005” per foot in all directions. Other types of installations may require the base to be level to within .002” per foot. This is usually done to avoid problems later with a condition known as “soft foot” and to assure that oil lubricated bearings receive equal and adequate lubrication. It is usually necessary to remove the pump and motor from the base to attain this level of flatness.

The baseplate is leveled using shims or wedges at various locations around the base close to where the anchor bolts are located. Some bases may be equipped with leveling screws which make this

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process somewhat easier. The wedges, shims or leveling screws are alternately adjusted as necessary to bring the base to within the specified limit. Once done the components are re-installed on the base.

At this point, before grouting the unit, it is important to check the pump and motor for “Soft Foot” and to check the “Rough Alignment” of the components. If these checks are not made now and the unit is grouted in place, any changes that need to be made later become much more difficult and expensive. Taking a few minutes at this point to check and correct any problems that exist may save many hours of work later.

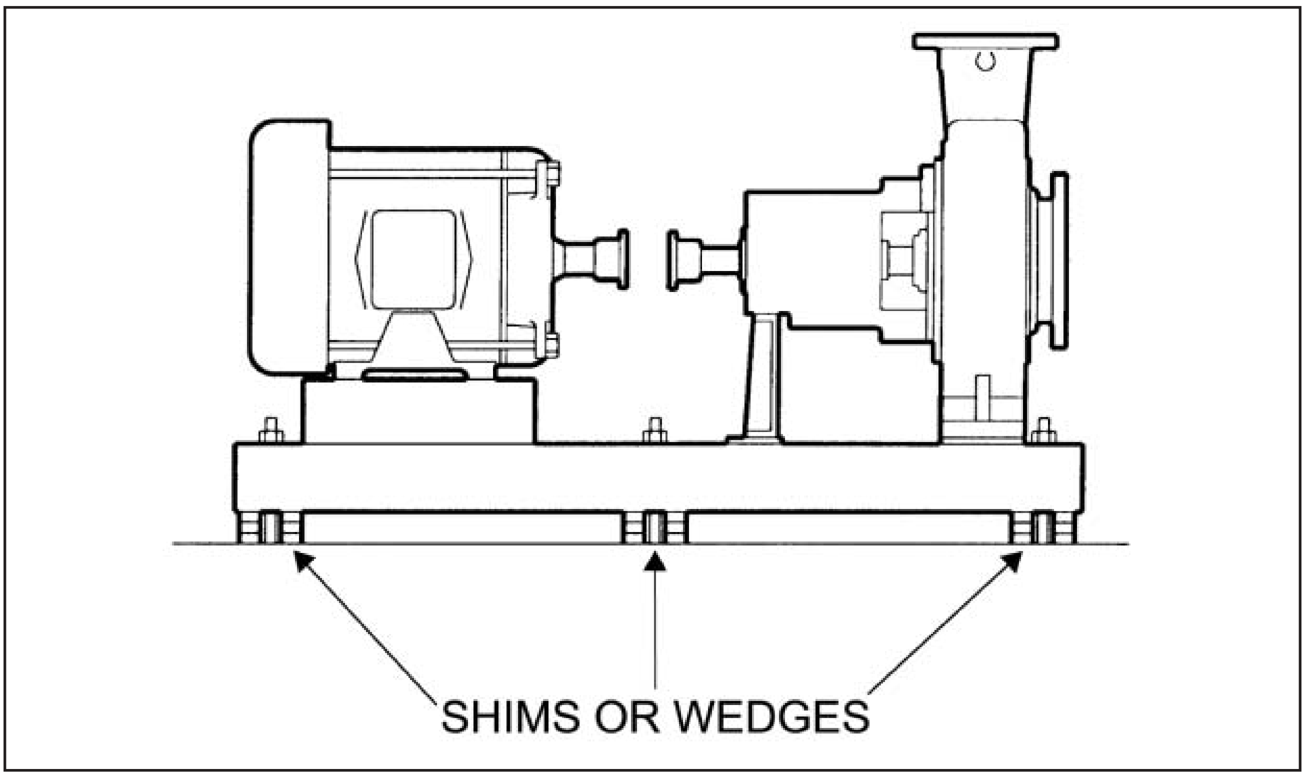

Figure 1

BASEPLATE GROUTING

Once the baseplate is properly installed, it must be grouted. Grout is a concrete material or multi-part epoxy compound that is used to fill the cast iron or fabricated steel baseplate so that when hardened, the baseplate and foundation become a unit. We recommend the use of a non-shrink grout for this purpose. The grout is mixed to a water-like consistency, and when poured should be puddled to insure that it flows evenly throughout the under portion of the

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baseplate, completely filling the base. It is recommended that the cementatious grout be allowed to cure for at least 48 hours. Refer to the epoxy grout manufacturer's instructions for the proper mixing and application of that product. Refer to your equipment instruction manual for further details.

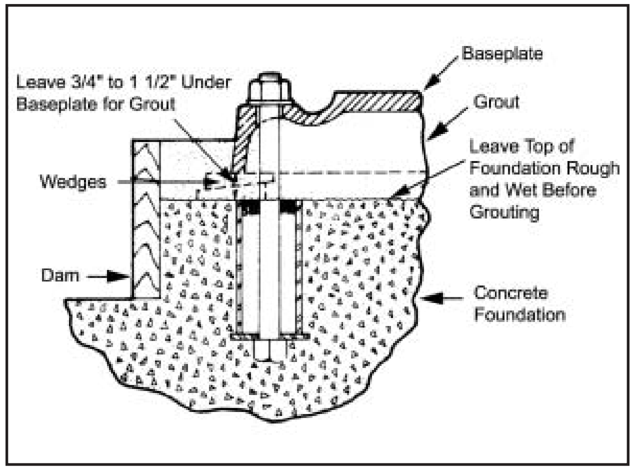

Figure 2

INITIAL ALIGNMENT

The proper alignment of your pump and driver is important to assure trouble-free mechanical operation. Noisy operation, vibration, reduced bearing life, shaft and coupling failures and wasted energy may result from faulty alignment. Before proceeding, you must know the thermal rise estimates of your equipment. Refer to the pump and driver manuals for these estimates.

The most common methods of measuring misalignment are:

Straight Edge and Feeler Gauges – This is the easiest and least expensive method of doing alignment but is also the least accurate. Used primarily for very small pump/motor combinations where there is not enough room to use more accurate, but larger alignment systems. The straight edge is laid across the flanges of the coupling hub and the feeler gauges are used between the faces of the coupling hubs. Shim changes are estimated and the alignment is attained through a process of “Trial & Error”.

Usually you cannot attain the equipment manufacturer's alignment specifications through the use of a Straight Edge and Feeler Gauges.

Dial Indicators – Used either singly or in pairs, as with the Reverse Indicator Method, the actual misalignment is measured by the indicator and through a few arithmetical calculations shim changes are determined.

Lasers – Somewhat more complicated to set up but can be more accurate if properly used. The laser especially lends itself to aligning shafts that are separated by more than a few inches. Since the machines have the capability of calculating the shim changes required, once the operator becomes familiar with the set-up he or she can do the alignment of a pump / motor combination fairly quickly and accurately. The primary drawback of the Laser is its cost and, in some cases, its size, although continuous advances in technology are making the machines both smaller and less expensive.

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Angular – With a non-spacer coupling, measure the space or gap between the hubs from top-to-bottom and from side-to-side using a feeler gauge. The measurements obtained should be very nearly the same. If they are not, shim or move the motor as necessary to satisfy the tolerance stated in your instruction manual for angular alignment. With spacer type couplings, a dial indicator should be used to obtain these readings.

Mount the indicator base on the driver coupling hub so that the indicator reads on the face of the pump hub. Rotate both shafts and read the indicator at 4 points, 90° apart. The readings obtained will tell you how far out of alignment your equipment is from side-to-side as well as from top-to-bottom. Add or remove shims from under the driver to correct the alignment. The amount of shim adjustment needed depends on the amount of misalignment present and distance from the point of measurement to location of the shims.

Parallel — Using a straight edge rest the edge on top of the coupling hub and at 90° to the top. The amount of misalignment is shown by the gap present between the straight edge and the hub. On a spacer type coupling mount a dial indicator on the pump coupling hub so that the indicator reads on the motor coupling hub. Rotate both shafts and read the indicator at 4 points, 90° apart. The readings obtained will tell you how far out of alignment your equipment is from side-to-side as well as top-to-bottom (elevation). Correct the top-to-bottom alignment first by shimming the driver as needed one half the amount shown on the indicator. Correct the side-to-side alignment by moving the driver as needed, again, by one half the amount shown on the indicator. Recheck the top-to-bottom (elevation) to make sure that it did not change during the process. Be sure to allow for the thermal rise estimates of both pump and driver.

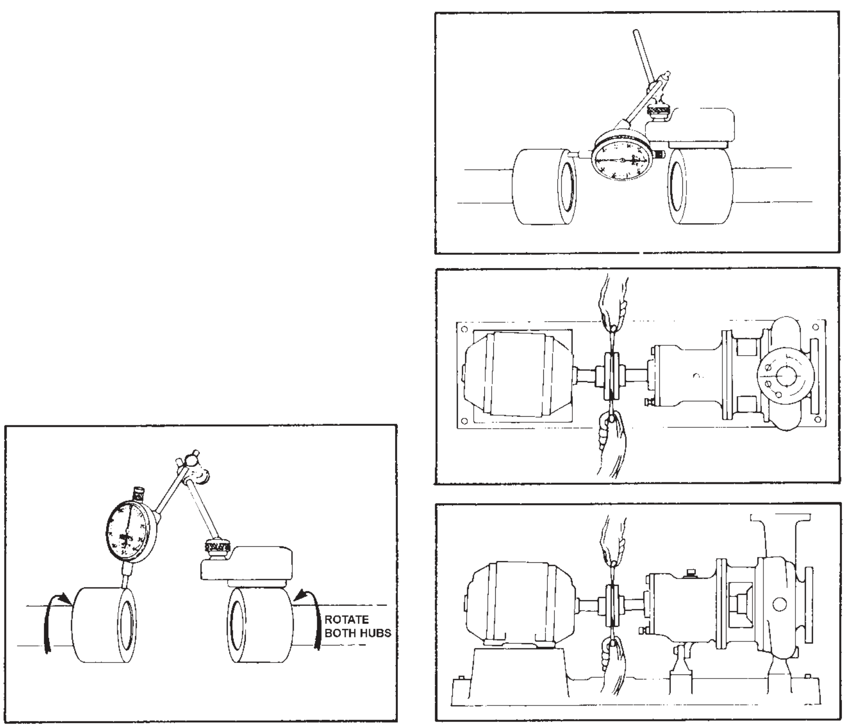

Figure 3

ALIGN PIPE FLANGES TO PUMP FLANGES

Never run pipe to the pump - always run pipe from the pump to a point several feet away where the final pipe connection can be made. This will help to minimize excessive pipe strain on the pump nozzles. The piping must be independently supported with an adequately designed pipe hanger system that will not allow the pump to carry the weight of the pipes or the liquid in them.

The system should also be designed to accommodate whatever thermal expansion or contraction is anticipated. After the piping has been made up, the alignment must be rechecked. By comparing this check with the initial alignment figures, the installer can determine the amount of pipe strain which has been put on the pump nozzles. The piping should be adjusted if there is any significant change in the alignment readings.

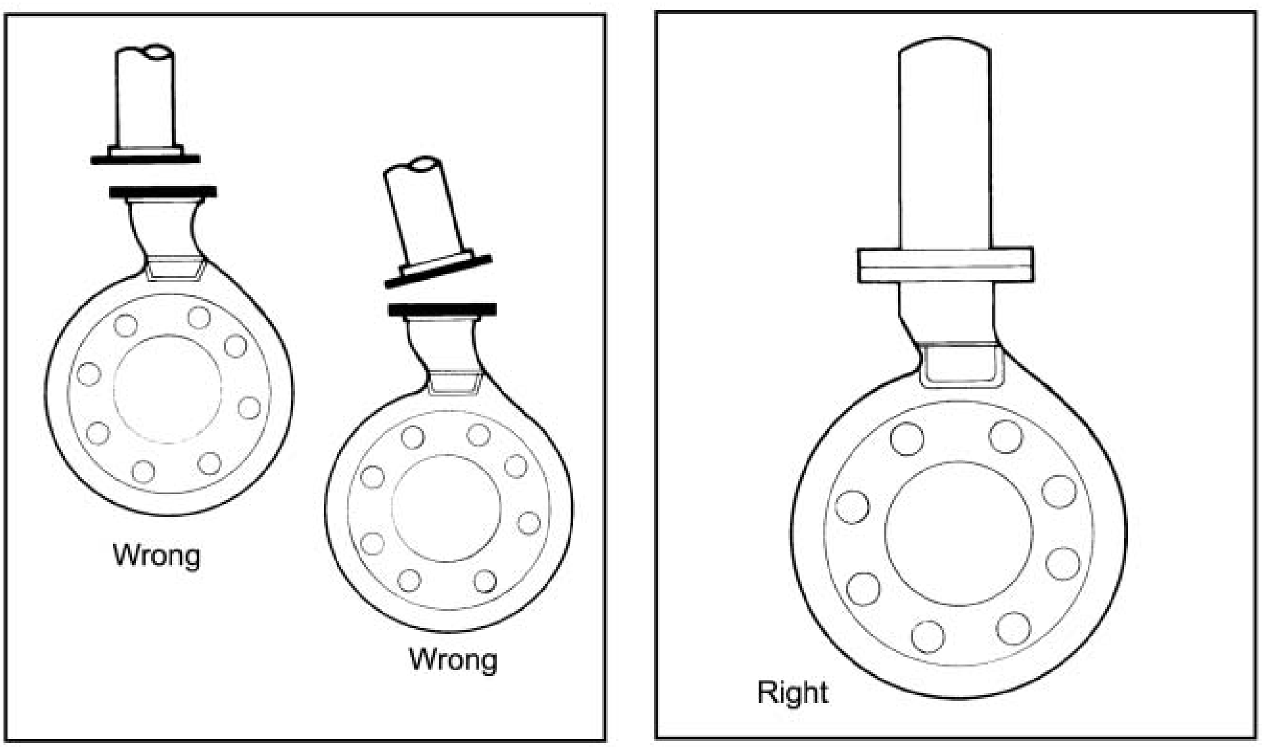

Figure 4

RAISED FACE AND FLAT FACE FLANGES (MATING COMBINATIONS)

Pumps of cast iron construction are furnished with 125 or 250 lb. flat face (F.F.) flanges. Since industry normally uses fabricated steel piping, the pumps are often connected to 150 or 300 lb. 1/16" raised face (R.F.) steel flanges.

Difficulty can occur with this flange mating combination. The pump flange tends to pivot around the edge of the raised face as the flange bolts are tightened. This can cause the pump flange to break allowing leakage at the joint (Fig. 5).

A similar problem can be encountered when a bronze pump with F.F. flanges is connected to R.F. steel flange (Fig. 6). Since the materials are not of equal strength, the bronze flange may distort, resulting in leakage.

To avoid problems when attaching bronze or cast iron F.F. pump flanges to R.F. steel pipe flanges, the following steps should be taken (refer to Fig. 7):

1. Machine off the raised face on the steel pipe flange.

2. Use a full face gasket.

If the pump is steel or stainless steel with F.F. flanges, no problem arises since materials of equal strength are being connected. Many customers, however, specify R.F. flanges on steel pumps for mating to R.F. companion flanges. This arrangement is technically and practically not required.

The purpose of an R.F. flange is to concentrate more pressure on a smaller gasket area and thereby increase the pressure containment capability of the joint. To create this higher gasket load, it is only necessary to have one-half of the flanged joint supplied with a raised face - not both. The following illustrations show 4" steel R.F. and F.F. mating flange combinations and the gasket loading incurred in each instance.

Assuming the force (F) from the flange bolts to be 10,000 lbs. and constant in each combination, the gasket stress is:

$$P \text{ (Stress)} \quad \frac{\text{Bolt Force (F)}}{\text{Gasket Area}}$$

$$P_1 \text{ (Fig. 8)} \quad \frac{10{,}000 \text{ lbs.}}{49.4 \text{ sq. in.}} \quad 203 \text{ psi}$$

$$\begin{array}{l} P_2 \text{ (Fig. 9)} = \\ P_3 \text{ (Fig. 10)} = \end{array} \quad \frac{10{,}000 \text{ lbs.}}{15.9 \text{ sq. in.}} \quad 630 \text{ psi}$$

It can be readily seen that the smaller gasket, used with a raised face flange, increases the pressure containment capability of a flanged joint. However, it can also be noted that there is no difference in pressure capability between R.F. to R.F. and R.F. to F.F. flange combinations.

In addition to being technically unnecessary to have a R.F. to R.F. mating combination, the advantages are:

1. The elimination of the extra cost for R.F. flanges.

2. The elimination of the extra delivery time required for a non-standard casing.

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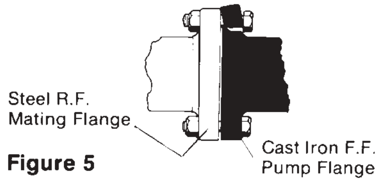


Figure 5

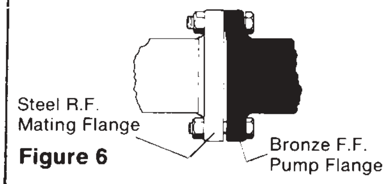


Figure 6

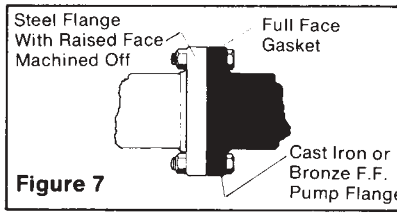


Figure 7

Gasket Area 49.4 sq. in.

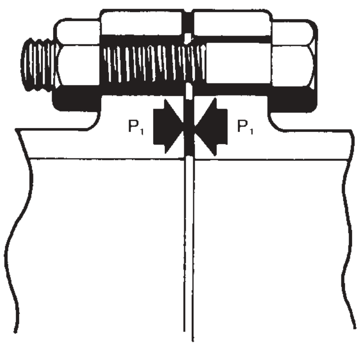


Figure 8 F.F. to F.F.

Gasket Area 15.9 sq. in.

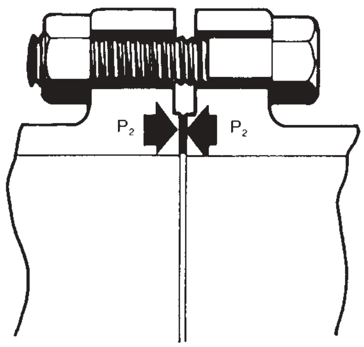


Figure 9 R.F. to R.F.

Gasket Area 15.9 sq. in.

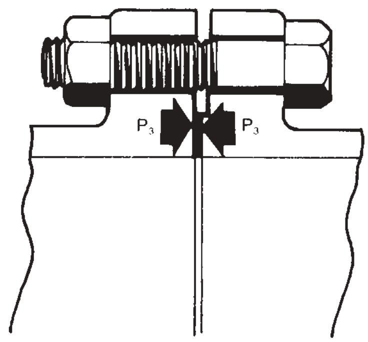


Figure 10 F.F. to R.F.

PUMP START-UP

All of the factors that contribute to successful pump operation must be checked carefully before starting a new or rebuilt pump. The majority of failures or serious problems at start up can be avoided by just taking a few minutes to verify that all of the components and associated systems are in operational condition.

Prior to coupling the unit, two checks which should always be made are: first, the unit turns freely by hand; and, second, the motor has been "bumped" and checked for the correct hand of rotation.

Operating some pumps in the wrong direction of rotation can severely damage the unit and can lead to personal injury for those nearby. A pump that does not turn freely, with consideration given to the drag in the stuffing box or mechanical seal area, usually requires some attention prior to starting.

If the pump shaft turned freely during the initial phases of the alignment process then something since that procedure has changed. Going back through each of the steps that followed the alignment in reverse will usually turn up the cause.

Causes can include pipe strain imposed due to the weight of the liquid in the piping, faults in the foundation or grouting, foreign material getting into the unit before the piping flanges were made up, etc. Whatever the cause, it must be identified and corrected before proceeding further.

Just prior to start up always make sure the correct type, quantity and quality of lubrication and / or cooling is applied to the components as recommended by the manufacturer. This includes motor bearings, couplings, mechanical seals, packing and of course, the pump bearings.

Once these checks have been made the Suction Valve on the pump may be fully opened while the air inside the pump is vented. This fills the unit with liquid, covering the impeller eye and makes it ready for start up. If the unit is in a "Suction Lift" type installation where the level of the liquid on the suction side of the pump is below the centerline of the pump casing, a vacuum priming system may be needed to evacuate the air from the pump casing or volute. In the case of "Self Priming" pumps the casing needs to have some liquid added to it before hand for the "Self Priming" feature to work.

TURBINE DRIVEN UNITS

When new or rebuilt turbine driven units are being started, it is important to protect the pump from damage related to an improperly adjusted turbine. All turbine controls should be checked and verified prior to coupling it to the pump. Consult the turbine manufacturer's instruction for details. When started, the unit should be brought up to speed and flow as quickly as possible in order to flush the new or recently opened piping system and minimize the chance of pump failure caused by small amounts of dirt or foreign material in the sytem. Suction strainers are used to provide some protection against this, but normally the openings are much larger than the running clearances of the rotating parts in the pump.

KEEP AIR OUT OF YOUR PUMP

Conventional Centrifugal Pumps are not designed to handle a mixture of liquid and gases. Pumping liquids containing a significant amount of entrained gas can lead to serious mechanical and hydraulic problems. A mixture of only 2% gas by volume will cause a 10% reduction in capacity and 4% will cause a reduction of over 43%. In addition to the loss of efficiency and wasted power the pump will probably be noisy and may vibrate excessively. Entrained gases can cause shaft breakage, seal failures and in some cases accelerate corrosion.

Air may be present in the liquid due to leaky suction lines on suction lift applications or a variety of other reasons. A free falling discharge into a tank or pit will also cause excessive gas entrainment and may cause problems for a pump drawing from that tank.

There are Centrifugal Pumps designed specifically for applications where entrained gases are encountered.

AIR POCKETS IN SUCTION PIPE

Air pockets can be a source of trouble on pump installations involving suction lift. They can cause a loss of prime on start-up as well as restrict flow in the suction pipe to the extent that a reduction in capacity is experienced. When installing piping, suction lines should always pass under interfering piping and where reducers are used, eccentric rather than straight reducers must be used.

Eccentric reducers are always installed with the horizontal side on top. Suction valves should be installed with the stems horizontal so that no air pockets are formed at the top of the valve near the bonnet. Figure 11 shows both the correct and incorrect method of installing piping.

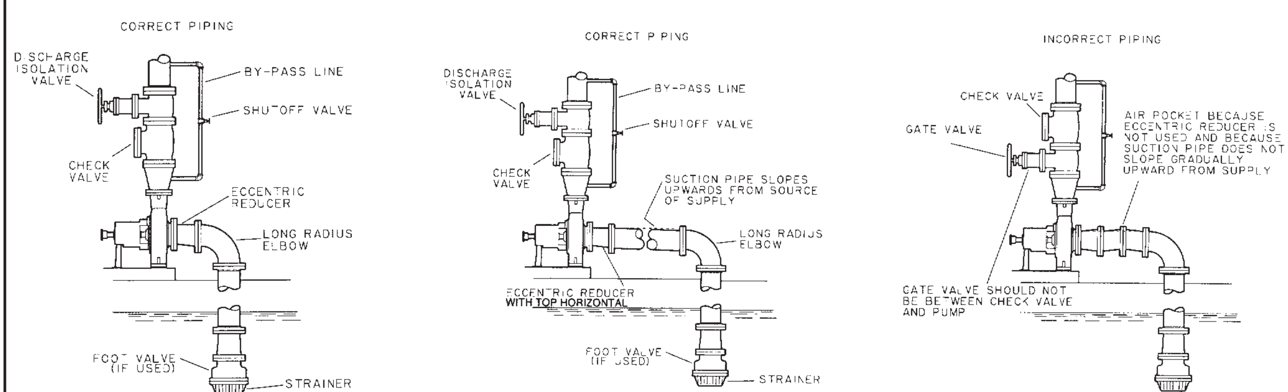


Figure 11

TESTING FOR AIR IN CENTRIFUGAL PUMPS

The amount of air which can be handled with reasonable pump life varies from pump to pump. However, in no case is it expected that a pump will give better life with air present than it would if the liquid were entirely air-free. The elimination of air has greatly improved the operation and life of many troublesome pumps. When trouble occurs, it is common to suspect everything but air, and to consider air last, if at all.

If air is present, the pump is likely to operate with a certain amount of internal noise. This noise can be described as a "gravel noise" - sounds very much must as though the pump were handling water full of gravel. This is the same type of noise generally associated with cavitation.

In many cases a great deal of time, inconvenience, and expense can be saved by making a simple test for the presence of air. We will assume that calculations have already been made to assure that the NPSH available is greater than that required by the pump (the noise is not a result of cavitation). The next step should be to check for the presence of entrained air in the suction.

When the source of suction supply is below the centerline of the pump, check for the conditions covered previously in this manual.

When the source of suction supply is above the centerline of the pump, a check for air leaks can be made by collecting a sample in a "bubble bottle" as illustrated in Figure 12. Since the pressure at the suction chamber of the pump is above atmospheric pressure, a valve can be installed in one of the tapped openings at the high point in the chamber and liquid can be fed into the "bubble bottle". The presence of air or vapor will show itself in the "bubble bottle".

Obviously, the next step is to eliminate the source of air since quantities present in sufficient amount to be audible are almost certain to cause premature mechanical failure.

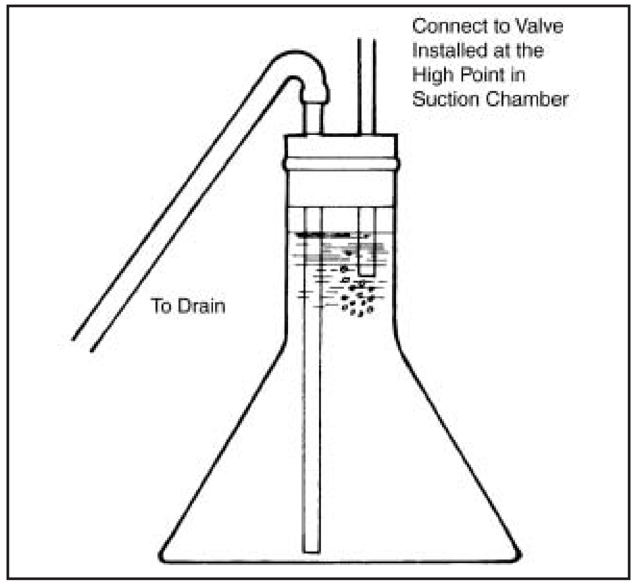


Figure 12

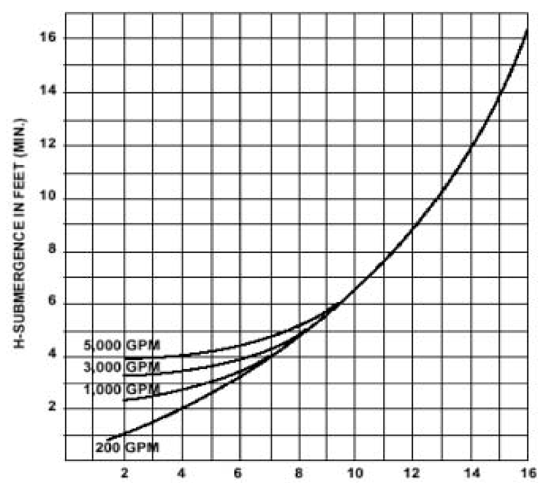
IMPORTANCE OF PROPER SUCTION PIPE SUBMERGENCE

If velocity of the water at the suction pipe entrance is too high, a vortex (commonly referred to as a whirlpool) is created, through which air flows downward to the end of the suction pipe and into the pump casing. Air in the pump casing causes rough and noisy operation and severe vibration that in the course of a short time can result in a broken shaft as well as other damage. If sufficient air reaches the pump to completely air bind it, dry operation will cause metal seizure at the impeller hub and wearing ring. Unless corrective action is taken, the rotating parts will be damaged beyond repair. These difficulties can be avoided by selecting and installing a suction pipe through which the fluid handled enters at a velocity not exceeding that shown on the graph. A lower velocity and more submergence are good insurance against this type of air trouble.

Sump design for large double suction pumps can become more complex, and appropriate sources such as the "Hydraulic Institute Standards" should be consulted.

In many cases, centrifugal pumps function improperly because air gets into the pump due to:

1. The installation of a suction pipe that is too small in diameter.
2. The end of the suction pipe not being submerged deeply enough.
3. Both of the above undesirable conditions.



VELOCITY IN FEET PER SEC. = $\frac{\text{QUAN. (G.P.M.)} \times .321}{\text{AREA (inches)}^2}$ OR $\frac{\text{G.P.M.} \times .4085}{D^2}$

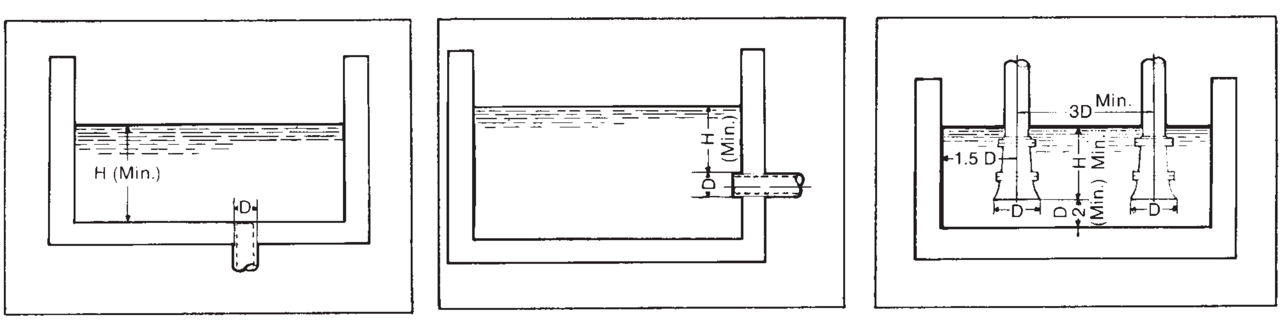


Figure 13

WATERFALL EFFECT

Liquid falling directly on the foot valve in Figure 14 would carry air into the suction line and could result in the pump losing its prime and seizing while running dry. This can also happen when a pump has a flooded suction as in Figure 15. In either case, the pump would be noisy.

The free fall shown in Figures 14 and 15 is serious, but a line discharging vertically downward under pressure, such as an orifice return line to a feed water storage tank, has even more effect, as shown in Figure 16.

If the returned liquid is heavily laden with air, or the sump is small, baffles may also be required as shown in Figure 14 to allow the air to separate before entering the pump suction.

The continued presence of entrained air and vapor in any liquid being pumped is extremely serious and should be corrected immediately. Any return line should be extended far enough away from the pump suction to prevent trouble, and below the surface of the water to minimize air entrainment. Check your installation today and make sure YOUR pump is not suffering from entrained air.

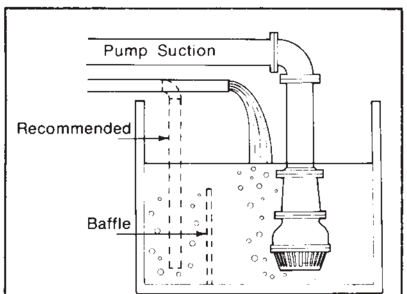


Figure 14

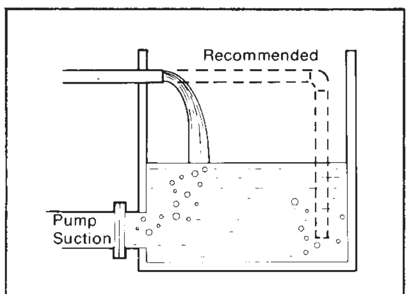


Figure 15

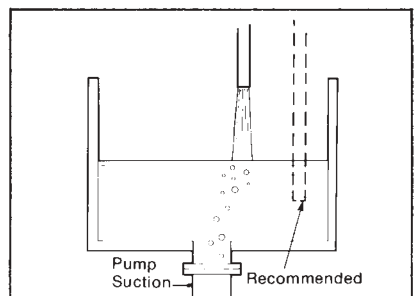


Figure 16

EXCESSIVE THROTTLING CAN SHORTEN THE LIFE OF YOUR PUMP

A centrifugal pump should never be operated continuously near shutoff or zero capacity. To do so may shorten the life of the pump and greatly increase down time and maintenance.

The difference between input horsepower and water horsepower is transferred to the liquid in the pump as heat. When only a small percentage of rated flow is allowed through the pump, the casing may be unable to radiate the heat generated, and the liquid and the pump may rise to a dangerously high temperature.

Hydraulic radial thrust is unbalanced when the pump is operating near shut-off, and this subjects the shaft to abnormal deflection. The pump will be noisy, will

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vibrate excessively, and may break shafts frequently.

Pumps are frequently rated for capacities sufficient to handle maximum requirements. They are also occasionally selected to handle capacities required under emergency operation or at some future predicted flow far in excess of present demands. This procedure will minimize ultimate plant investment, duplication, and replacement with larger equipment at a future date. However, it may require excessive throttling of the discharge valve, resulting in severe punishment to the pump.

Fortunately, there is a simple method to relieve the pump of undue strain. Simply extend a by-pass line from the pump discharge back to the source of supply. A throttle valve or an orifice plate should be placed in the by-pass line, and sufficient flow returned to allow the pump to operate at a capacity reasonably near its rating.

The bypassed liquid should never be returned to the suction line immediately upstream from the pump, but should always be returned back to the source of supply and discharged below the liquid level to avoid air entrainment.

If excessive throttling at the discharge valve is required with the by-pass line open, actual system head and capacity requirements should be reviewed and a new rating selected.

Excessive maintenance and shortened life can always be expected when a pump is operated continuously at or near shut-off. A return line by-passing part of the flow back to the source of supply will allow the pump to operate near rated capacity and extend the life of your pumping equipment.

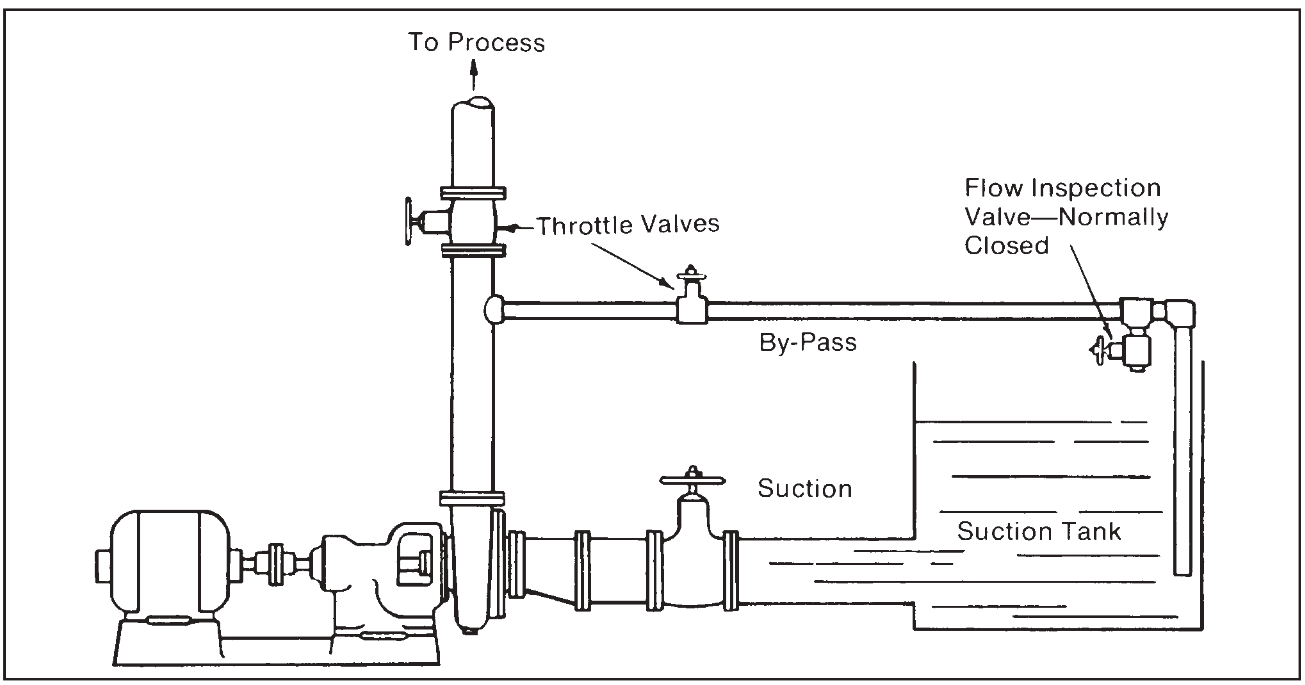

Figure 17

THROTTLING ACCELERATES ERROSION WHEN PUMPING LIQUIDS CONTAINING ABRASIVES

When pumping liquids containing abrasive solids some errosion of the pump impeller and other parts is to be expected. The useful life of the unit is reduced not only by the quantity of abrasives present, but also the physical characteristics of the solids and the velocity of the mixture through the pump. Throttling a unit accelerates the wear even more by causing localized higher velocities and re-circulation within the impeller and casing. With a pump operating at its best efficiency point, the majority of solids make only one trip through the unit. Throttling the unit increases the internal re-circulation and allows the abrasive particles to scrape against the inside of the pump a number of times before finally being discharged.

Often a plant design calls for pumps to be oversized to accommodate future requirements or the occasional upset condition. Again, a discharge by-pass line should be considered to help allow the pump to operate at a better point on the curve and at better efficiency and reduced internal re-circulation. Often the re-circulation line is directed back into the tank or sump to provide additional agitation and mixing.

ELBOWS AT PUMP SUCTION CAN CAUSE TROUBLE UNLESS PRECAUTIONS ARE TAKEN

Properly installed suction piping is of extreme importance in obtaining trouble-free operation in any double-suction type centrifugal pump.

If an elbow is required at the pump suction flange, it should always be in a *vertical* position. Where, because of space limitations or other peculiarities of the installation, an elbow must be used near the pump suction flange in any position except vertical, it should never be installed unless there is a minimum of two diameters of straight pipe between the elbow and the pump flange (see Figures 18 and 19).

Without this straight run of pipe, the incoming liquid, because of centrifugal force, tends to pile up on the outside of the elbow. If the liquid enters the pump suction directly in this condition, it will not be evenly distributed between the two inlets of the double suction impeller. This unbalance will cause noise and generally unsatisfactory operation of the pump (see Figure 20).

A device for relieving this condition is illustrated in Figure 21. It is a simple baffle that can be made in any shop. Inserted in the straight length of pipe adjacent to the pump flange, it will

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equalize the flow and assist materially in improving pump performance.

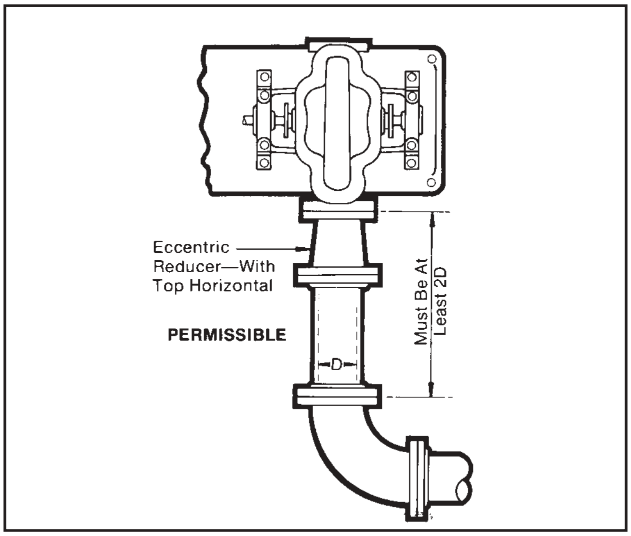


Figure 18

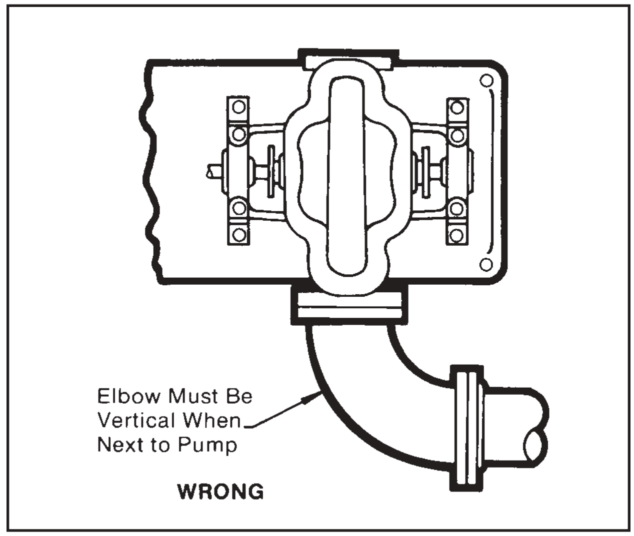


Figure 19

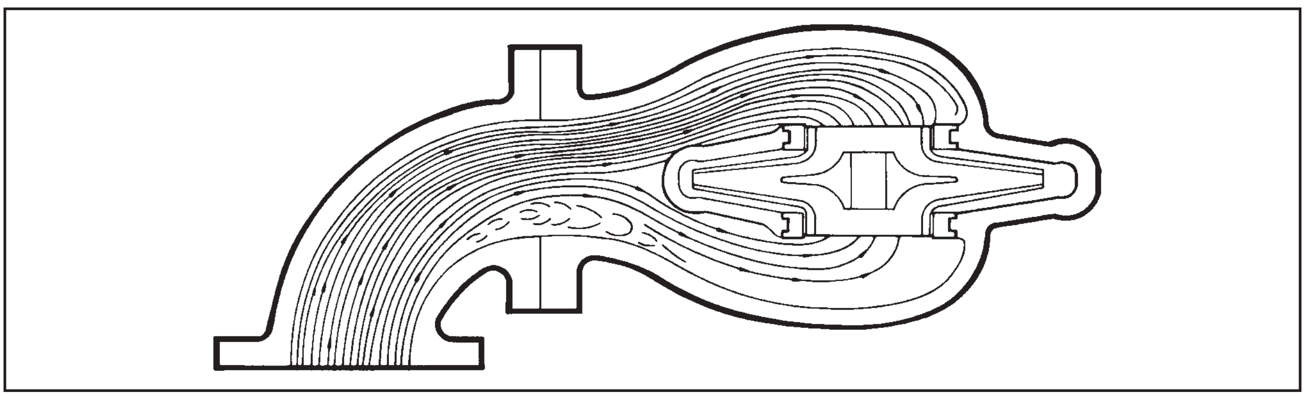


Figure 20

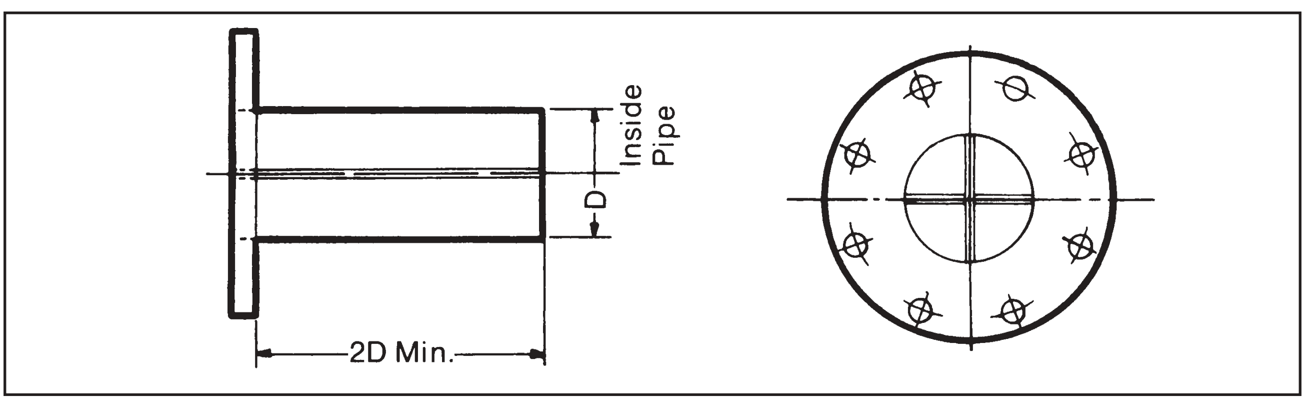


Figure 21

USE A THERMOMETER TO MEASURE BEARING TEMPERATURES

Pump bearing temperatures are frequently estimated by placing a hand lightly against the bearing housing or shell. If the bearing is “cool” or “warm,” we feel assured. If it feels “hot,” we may become concerned and spend considerable time and effort to reduce the temperature until the housing feels only “warm” without any clear idea of the actual operating temperature.

Unfortunately, the human hand is not an accurate thermometer and can give a danger signal falsely. A temperature which feels “hot” varies from 120° to 130°F depending on the individual. Above this temperature the human hand is worthless in estimating temperature. Grease lubricated ball and roller bearings can be operated safely at temperatures up to at least 200°F. In fact, the upper operating limit on anti-friction bearings is determined solely by the temperature at which the lubricant fails and begins to carburize. Bearing temperatures up to 160°F are extremely safe. Operation of bearings at this temperature is not at all undesirable, as a better flow of lubricant can be expected. This gives a clear indication of why a thermometer is necessary to determine bearing temperature.

All bearings operate at some temperature above that of the surrounding atmosphere, unless cooled. Heat is generated within the bearing due to rolling friction of the balls or rollers, and by the drag of the ball or roller separator cage. Some heat may be added by conduction along the pump shaft from the liquid within the pump and from external sources in close proximity. The amount of heat which can be dissipated is dependent on the cooling area of the bearing housing and the temperature and motion of the surrounding air. The net result is a stable operating temperature. Once this temperature has been established, it will remain constant until one or more of the variables changes. A stable temperature, no matter how hot it may feel to the human hand, is not necessarily an indication of danger so long as it does not exceed the upper limit of the lubricant. The temperature should be established accurately by thermometer and recorded in a convenient location.

A pronounced increase in temperature is an indication of danger and a signal to investigate. One shot of grease should be added to the bearing, but if this does not reduce the temperature immediately, no additional grease should be added. The unit should be checked for unnecessary loads, such as coupling misalignment, or improper packing adjustment. A temperature increase may not be an indication of impending bearing failure or excessive load, but could be due entirely to an increase in temperature of the liquid being pumped, or to an increase in the surrounding temperature during the warm summer months. Heat transfer due to increased liquid temperature can be minimized by connection of a quenching gland with cold water from the plant supply line.

Bearing temperatures are frequently increased by lubrication practices. Excessive or over-greasing which results in complete packing of the bearing shell with grease greatly increases the resistance to rotation and the amount of

(cont'd)

heat generated. Removal of excessive grease from the bearing housing will lower the bearing temperature considerably. Most pumps handling hot liquids are equipped with quenching glands and water cooled bearings so that the actual operating temperature within the bearing may be lower than the ambient temperature of the surrounding air. High temperature greases should not be used with water cooled bearings as poor grease fluidity and inadequate lubrication may likely result.

Occasionally when pumps are first started the bearings seem to run extremely hot. This high temperature is frequently caused by the grease seals not the bearings. As soon as the seals are seated, the temperature will drop to a normal level.

NPSH

The most maligned of all pump criteria is actually a simple physical phenomenon. NPSH, net positive suction head, is defined by the Hydraulic Institute as “the total suction head in feet of liquid absolute, determined at the suction nozzle and corrected to datum, less the vapor pressure of the liquid in feet absolute.”

The key to the problem is vapor pressure. Every liquid exerts a pressure on its surroundings which is dependent on temperature. When this pressure is equal to the pressure of its environment, the liquid is said to boil or turn to vapor. For example, at sea level a pan of water on a stove will boil at 212°F. This means that the vapor pressure of water at 212°F is equal to the atmospheric pressure 14.7 psia.

The amount of space a liquid occupies is very small compared to the amount of space occupied by the equivalent amount of liquid converted into vapor. For example, 1 pound of water at 50° F occupies a space equal to about a pint (16 oz.) of soda. One pound of water vapor at 50° F would occupy a space of over 1,700 cubic feet, nearly the size of a 15’ x 15’ x 8’ room. One pound of water vapor occupies about 10,000 times the volume of an equal mass of water.

Figure 22 is a profile and end view of a typical open impeller. An impeller is essentially a spinning wheel and as such centrifugal force creates a low pressure area at the center point “A”. In addition, pressure is lost due to liquid friction between the suction nozzle and the entrance to the vane. One other source of pressure loss is due to the shock of the vane hitting into the liquid stream.

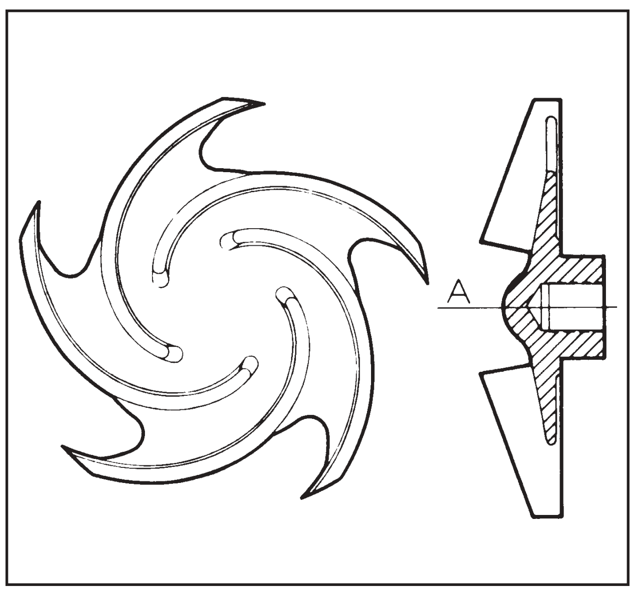


Figure 22

(cont'd)

If the pressure at point “A” is lower than the vapor pressure of the liquid being pumped, the liquid boils or vaporizes. When this occurs, the vapor occupies many times the volume of the liquid, consequently the performance in terms of mass flow is greatly decreased.

When the vapor enters the vane area, condensation or collapse of the vapor bubble occurs as soon as the pressure is higher than the vapor pressure. Since the pressure is generated by the vane working on the liquid or vapor, the collapse always happens at the surface of the vane. The crackling noise associated with cavitation is this collapse taking place.

When the vapor bubbles collapse on the vane surface, energy is released in the form of an implosion which will take particles of metal out of the surface. This causes the familiar pitting of the vane.

Thus, NPSH required is simply the amount of pressure necessary to keep the liquid from vaporizing at the “eye” or center of the impeller. It is made up of the losses due to friction and shock plus the natural pressure reduction due to centrifugal force.

PACKED STUFFING BOX LUBRICATION

Shaft sealing is the necessary evil to centrifugal pump operation. Mechanical seals and packing are the two methods used to accomplish a seal.

It must be understood that while these sealing devices act in different ways to do the same job, there is work being done which generates friction and heat. Because of this, they require both cooling and lubrication.

Packing does not and should not completely eliminate leakage, but rather throttles it to a tolerable limit. There should always be some drippage from the gland for good packing lubrication.

The choice of stuffing box lubricant depends on the pumpage, temperature and pressure. Obviously, if the liquid pumped is a poor lubricant or contains abrasive solids, the packing will be severely damaged. Should the liquid be at or near its vapor pressure, the heat generated in the bearing surface area may cause flashing with the resultant loss of lubricating properties.

Figure 23 illustrates the most common method of supplying lubrication to the packing. The pumpage is simply by-passed from the discharge volute into the stuffing box. This is used when the liquid and operating conditions are such that the pumpage will provide adequate lubrication. Part A in Figure 23 is the cage ring. Its function is to provide an annular channel for even distribution of the lubricant.

Figure 24 illustrates a method of lubrication used when the pumpage is heavy or thick, such as paper stock. In this case, the cooling fluid is supplied to a cage or lantern ring which is placed in the back of the stuffing box. This method prevents the pumpage from entering the throat bushing and will minimize wear and clogging. The cooling fluid is forced into

(cont'd)

the stuffing box at a higher pressure than is in the area between the back of the impeller and the stuffing box with a portion entering the pumpage through the throat bushing. The remainder of the cooling fluid is forced along the shaft to cool and lubricate the packing.

Care must be exercised in the choice of external lubricant. The primary concern will be compatibility with the pumpage. In most cases water will suffice for lubrication, but serious product damage may result if a chemical reaction occurs.

If water is undesirable, grease oil or a liquid with good lubricity and chemical compatibility should be selected.

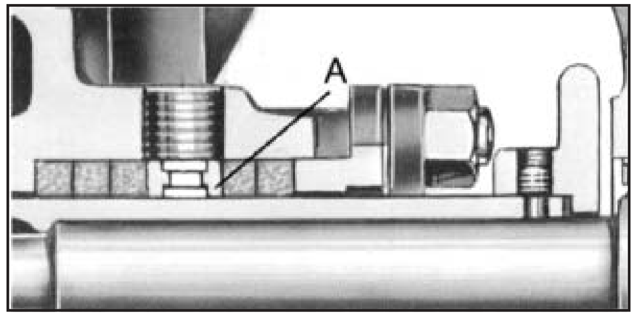


Figure 23

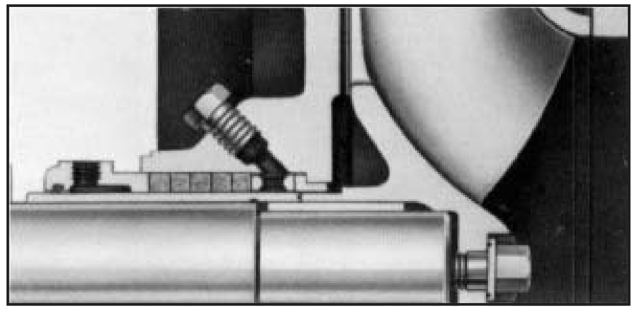


Figure 24

ARE YOU INCREASING YOUR STUFFING BOX PACKING PROBLEMS?

Normal stuffing box leakage usually ranges from 40 to 60 drops per minute which is adequate to cool and lubricate the packing. Stuffing boxes for pumps with very large shaft diameters may need to leak as much as a small stream to provide sufficient cooling and lubrication. On units handling clear liquids, the pumpage itself becomes the coolant and lubricant. A seal flush line is piped up from a point on the pump casing to the seal water inlet. As the unit is operated, pumpage flows naturally to the stuffing box. An external sealing liquid is needed if the pressure at the stuffing box is negative, as in a Suction Lift situation, or if the liquid being pumped has suspended solids, or is highly volatile with poor lubricating qualities.

If the pressure at the stuffing box is negative, air in the atmosphere will be drawn into the packing. If enough air enters the pump through the packing it is possible for the unit to lose its prime or seize the pump altogether. A liquid barrier such as seal water, helps to prevent this.

The sealing liquid is piped externally to the stuffing box and distributed between the rows of packing through a seal cage or lantern ring. Some of the sealing liquid will be drawn into the pump, and the remainder will trickle out of the stuffing box. In this case, the sealing liquid acts as a coolant, a lubricant and a barrier against the entrance of air as well. If the liquid pumped has solids in suspension, the stuffing box should be sealed with clean water from the plant water supply or some other source. The use of plant water assures a positive seal on the stuffing box during the priming period.

(cont'd)

If the pressure at the stuffing box is positive and the liquid is clean, the natural flow of liquid trying to escape to the atmosphere will cool and lubricate the packing. In this case it is not necessary to connect either a flush line from the pump casing or an external supply of seal water. The additional pressure of the supply will increase the pressure at the stuffing box unnecessarily and require the packing to be tightened further to control the leakage.

If the pumpage contains suspended solids special consideration needs to be given to both the type of packing being used and the supply of seal water. If allowed to leak, the solids in the pumped liquid will collect on the rings of packing and score the shaft sleeve. External sealing with clean plant water is recommended.

The pressure of the seal water supply, at the stuffing box connection, should be between 5 and 10 PSIG above the pressure acting on the stuffing box. This will insure a flow of liquid along the shaft sleeve and into the pump helping to prevent the entrance of solids.

If the pumpage is toxic, volatile or cannot be contaminated, a non-soluble lubricant may be introduced at the seal cage or lantern ring. A quench type gland is often used to dilute and carry away any leakage of the pumped liquid which might escape past the packing.

Recommendations for stuffing box sealing liquids are tabulated below. They should be used as a guide for new pump installations and when reviewing existing installations.

STUFFING BOX PACKING

Packing is generally a rope-like material containing some type of lubricant, usually grease and graphite, Teflon™, mica or some other lubricating substance. The initial impregnation is intended to serve as the primary lubricant for the start-up and break-in period. After start-up other lubrication must be provided to prolong packing life. This lubrication is most often provided by the pumpage.

Packing supplied by ITT Industries Fluid Technologies can be broken down into two primary classifications: general and special.

General

Packing supplied for general service applications is braided, long filament vegetable fiber material impregnated with grease and graphite flakes. This packing is best suited for low speed, low temperature and low pressure applications on cold water and general services. It has a pH range of 4 to 10 which enables it to withstand mildly acidic or alkaline solutions. It's the least expensive of the packing types available.

Special

A. Chemical and Solvent Applications

A wide variety of packing is available for severe chemical or solvent applications. Typical types include Teflon™ impregnated vegetable fiber packing, pure Teflon™, Grafoil™ and pure graphite, Kevlar™, lead packing, various combinations of these and other materials. Your choice of packing should

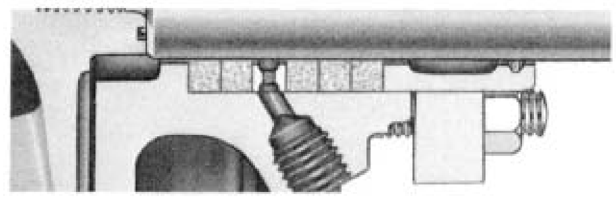
(cont'd)

be based on the nature and concentration of the chemicals or solvents being pumped.

B. High Pressure or Temperature Applications

Packings of varied materials and arrangements can be supplied for most applications involving high pressures and/or temperatures. Combination sets of various types of foil, plastic or braided natural and man-made fibers are common for use with these applications. Your choice should be based on shaft speeds and the severity of the pressures and temperatures involved.

Often consultation with your ITT Industries Sales Representative or local packing supplier can provide alternative solutions to what may be considered an impossible situation.



Goulds Model 3196 with non-cooled stuffing box w/quench type gland. Standard configuration with 5 rings of packing and lantern ring illustrated.

Figure 25

STUFFING BOX COOLING FOR HIGH TEMPERATURE APPLICATIONS

High temperature applications present certain inherent problems which do not arise when pumping lower temperature fluids. Special attention must be given to the stuffing box and packing arrangement to assure a long and useful life.

In centrifugal pumps fitted with packing, there are high temperature options available, such as jacketed stuffing boxes, to help extend the life of the packing. As the temperature and pressures increase or as the pumpage becomes more volatile, corrosive or toxic, packing becomes more of a problem. Since stuffing boxes need to leak to provide the cooling and lubrication for the packing, this leakage can cause serious safety and health issues in the plant. In some applications the few drops of leakage can vaporize harmlessly into the atmosphere. In others, there can be no leakage to the environment at all.

Because they eliminate this leakage, mechanical seals have largely replaced packing for these sorts of applications. However, when using mechanical seals, care must be taken to be sure that there be liquid film between the seal faces at all times. If the liquid separating the seal face surfaces were to flash to vapor and allow the faces to touch, seal failure is nearly guaranteed.

There are several ways to help keep the seal faces and sealing fluid cool:

1. Cooling liquid can be circulated through a Jacketed stuffing box which is an option for many of our models, including the 3196. The chart on the next page shows the cooling water required for various application temperatures.

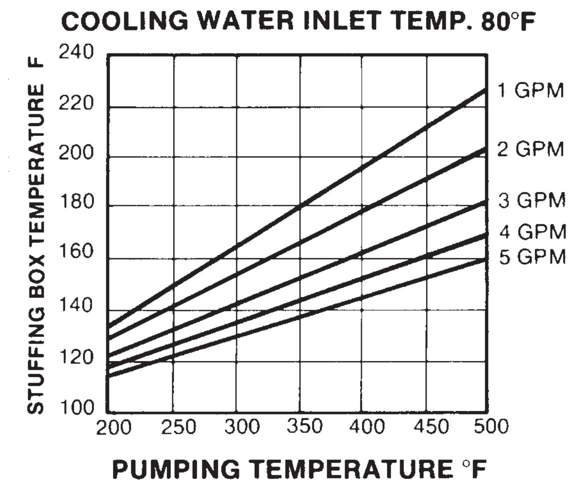
2. A cool compatible liquid from an outside source can be injected directly into the seal chamber.

(cont'd)

3. The pumpage can be cooled in a heat exchanger and returned to the seal chamber.

Either (2) or (3) can be used by themselves or in combination with (1).

Temperature of liquid in pump stuffing box for various pumping temperatures



The above chart indicates the rate of flow of cooling water required to be circulatea in the stuffing box jacket to cool the insulated or "dead-ended" liquid around the seal.

Figure 26

TYPE 4
TaperBore™ PLUS Seal Chamber

TYPE 5
Jacketed BigBore™ Seal Chamber

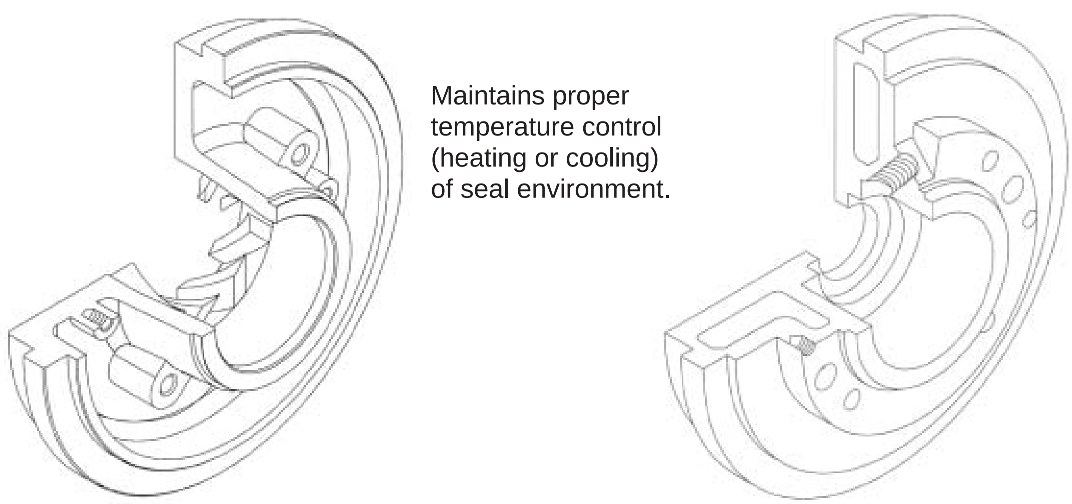


Figure 27

MECHANICAL SEALS . . . PART I - THE BASIC SEAL

A mechanical seal is a sealing device that provides a seal between rotating and stationary parts, thus keeping the liquid being pumped inside the pump. Today, the design of liquid handling equipment with rotating parts usually include the features necessary for the use of a mechanical seal. Some of the advantages mechanical seals offer over conventional packing include:

1. Reduced friction and power losses because of lower drag in the stuffing box.
2. Reduced leakage from the stuffing box.
3. Reduced shaft or sleeve wear.
4. Reduced maintenance.

The wide variety of styles and designs together with extensive experience allows the use of seals on practically any sealing problem.

A mechanical seal must seal at three points:

1. Static seal between the stationary part and the housing.
2. Static seal between the rotary part and the shaft.
3. Dynamic seal between the rotating seal face and the stationary seal face.

Figure 28 shows a basic seal with these components:

1. Stationary seal part-positioned in the housing with preload on the O-ring to effect sealing and prevent rotation.
2. Rotation seal part-positioned on the shaft by the O-ring. The O-ring seals between it and the shaft and provides resiliency.
3. The mating faces - The faces are precisioned lapped for a flatness of 3 light bands and a surface finish of 5 microinches.
4. Spring Assembly - Rotates with the shaft and provides pressure to keep the mating faces together during periods of shut down or lack of hydraulic pressure.
5. Driving member positions the spring assembly and the rotating face. It also provides the positive drive between shaft and the other rotating parts.
6. End movement - As wear takes place between the mating faces, the rotating face must move along the shaft to maintain contact with stationary face. The "O" ring must be free to move.

The principle of operation of a mechanical seal is fairly straight forward: Liquid pressure in the seal chamber along with some type of spring arrangement forces the faces together. Due to capillary action, a thin film of lubricant separates them and keeps them from touching under normal circumstances. The pressure of the liquid, the spring pressure and this very thin layer of liquid between the faces reduces the leakage of the pumped fluid past the seal to a point near zero. These basic components are a part

(cont'd)

of every seal, secondary sealing systems consisting of “O” rings, “V” rings, gaskets, etc. complete the assembly.

The form, shape, style and design will vary greatly depending upon the service and manufacturer. The basic theory, however, remains the same.

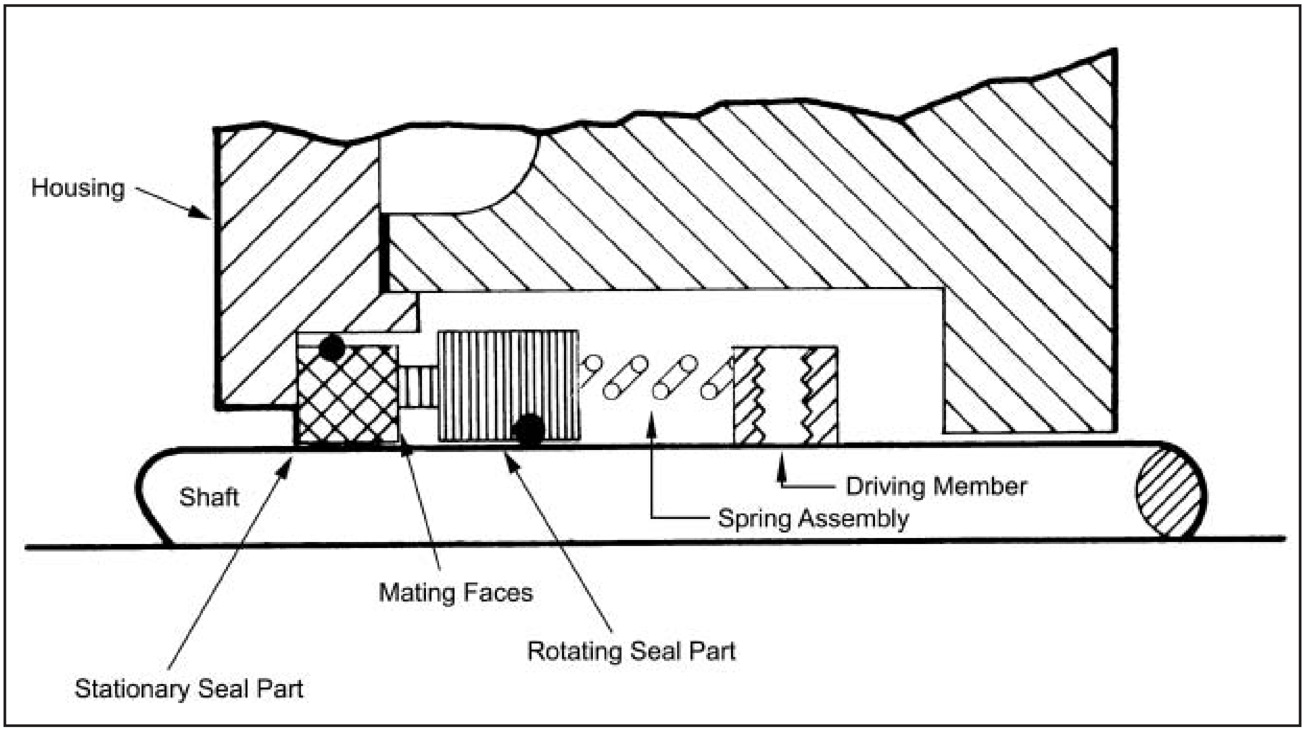


Figure 28

MECHANICAL SEALS . . . PART II - TYPES

Mechanical seals can be classified into the general types and arrangements shown below. Understanding these classes provides the first step in proper seal selection.

Single Seals
Inside, outside, unbalanced, balanced

Double seals
Unbalanced or balanced

Single Seals, Inside Unbalanced (Figure 29)

The single inside seal mounts on the shaft or sleeve within the stuffing box housing. The pumpage is in direct contact with all parts of the seal and provides the lubrication for the faces. The full force of pressure in the box acts on the faces providing good sealing to approximately 200 PSIG. This is the most widely used type for services handling clear liquids. A circulation or by-pass line connected from the volute to the stuffing box provides continual flushing of the seal chamber.

Single Seals, Outside Unbalanced (Figure 30)

This type mounts with the rotary part outside of the stuffing box. The springs and drive element are not in contact with

(cont'd)

the pumpage, thus reducing corrosion problems and preventing product accumulation in the springs. Pressures are limited to the spring rating, usually 35 PSIG. Usually the same style seal can be mounted inside or outside. The outside seal is easier to install, adjust and maintain.

Single Seals, Balanced (Figure 31)

Balancing a seal varies the face loading exerted by the box pressure, thus extending the pressure limits of the seal. A balanced rotating part utilizes a stepped face and a sleeve. Balanced seals are used to pressures of 200 PSIG. Their use is also extensive on light hydrocarbons which tend to vaporize easily.

Balanced outside seals allow box pressure to be exerted toward the seal faces, thus allowing pressure ranges to above 150 PSI as compared to the PSIG limit for the unbalanced outside seal.

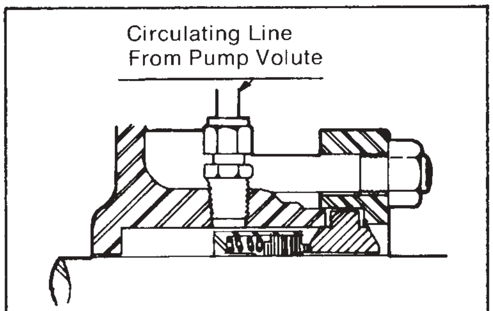


Figure 29

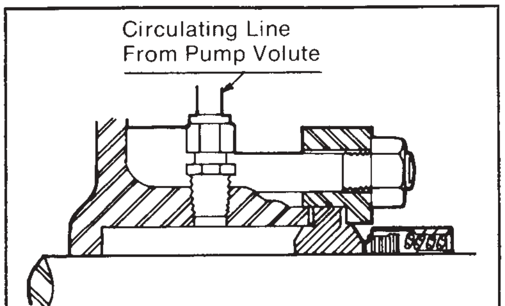


Figure 30

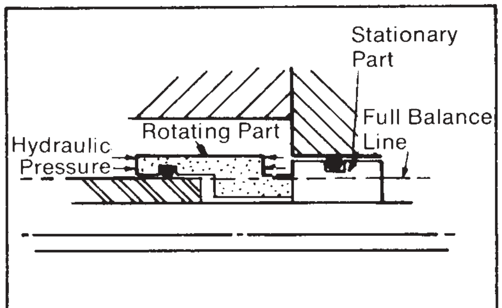


Figure 31

Double Seals

Double seals use two seals mounted back to back in the stuffing box. The box is pressurized with a clear liquid from an outside source. This arrangement is often used on slurry service or where there can be no release of the pumpage to the environment because the condition of the inner seal can be monitored by checking the characteristics of the seal flush liquid.

Balanced and unbalanced designs are available to meet specific pressures.

Cartridge Seals

Recent advancements in sealing technology have made Cartridge Type Seals the preferred type for many applications. The advantages include that they are available in all of the previously mentioned configurations, they are self contained, completely assembled on their own shaft sleeve, can accommodate pump impeller adjustments and are significantly easier to install. The problems of having to assemble the pump, mark the seal chamber location on the shaft or sleeve, disassemble the unit, install the seal and re-assemble the pump are eliminated. A further advantage is that you do not need to handle the delicate sealing faces or other components.

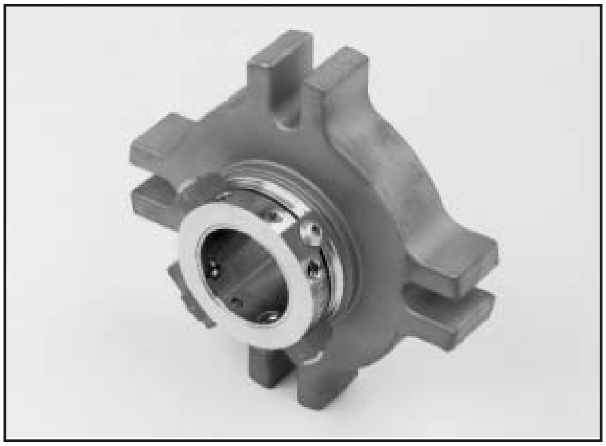


Figure 32

Split Seals

Sealing technology has advanced even further in the area of Split Seals. Split seals are usually a cartridge type seal with the added advantage of being split horizontally at the centerline. Since the seal can be separated into halves, placed around the shaft and reassembled, taking the pump apart to install it is unnecessary. The primary disadvantages are that the seal components need to be very accurately aligned and, since you are handling many of the individual parts of the seal, extra care needs to be taken to prevent damaging any of the components.

Seal Chambers

Seal life can be significantly extended by the proper selection of the seal environment. By providing better cooling and lubrication by increasing the circulation near the seal faces, the life of the seal is greatly increased. The design of the chamber can be as simple as just increasing the bore diameter in the area of the seal to increase the liquid capacity or as complicated as adding a taper to the sides and including ribs, vanes or fins to further induce circulation around the faces.

Manufacturers offer different configurations for different applications. A seal chamber suitable for a clear liquid application maybe totally unacceptable for a viscous application or one containing abrasive solids. Usually as the effectiveness of the seal chamber in cooling and lubricating the seal goes up, so does the cost of that design. That added cost can easily be more than offset by a reduction in the number of seal failure in the pump’s life time.

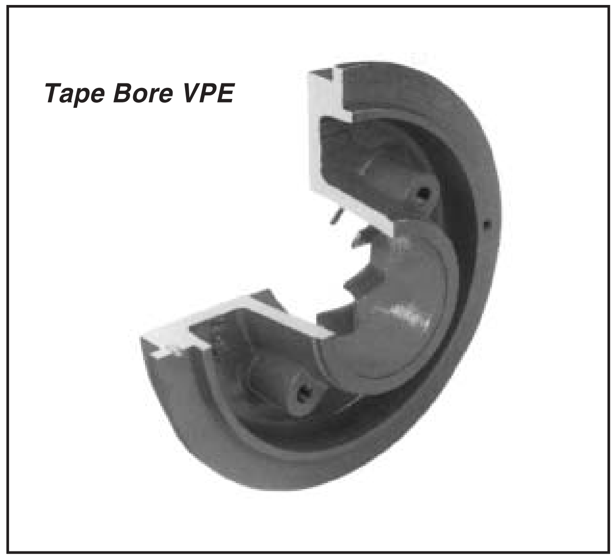


Figure 33

MECHANICAL SEALS . . . PART III - SELECTION

The proper selection of a mechanical seal can be made only if the full operating conditions are known. These conditions are as follows:

1. Liquid
2. Pressure
3. Temperature
4. Characteristics of liquid
5. Pump model

Liquid – Identification of the exact liquid to be handled provides the first step in seal selection. The metal parts must be resistant to the effects of the pumpage. These parts are available in steel, bronze, stainless steel, Hastelloy and a wide variety of other alloys to meet specific needs.

The seal faces must be able to resist both corrosion and wear. Carbon, various types of ceramics, Stellite, silicon carbide and tungsten carbide are a few of the variety of materials which offer both excellent wear properties and corrosion resistance.

The elastomers in the seal can be made of any number of different compounds including Teflon™, Viton, Buna N which completes the proper materials selection.

Pressure – the proper type of seal configuration, unbalanced or balanced, is based on the pressure that the seal will be subjected to. Unbalanced seals are used in applications with pressures up to approximately 200 PSI. Balanced seals are usually required for pressures above 200 PSI.

Temperature – The temperature will, in part, determine the material of the other sealing members. Synthetic rubbers are used to approximately 250°F, Teflon™ to 500°F. and other, more exotic elastomers to 750°F and above. Cooling the liquid in the seal chamber by use of cooling jackets or heat exchangers to cool flushing liquid , often extends seal life and allows for a wider selection of materials.

Characteristics of Liquid – Liquids containing abrasive solids can create excessive wear and shorten seal life. Double seals, very hard faces and clear liquid flushing from an external source allows the use of mechanical seals on these difficult applications. On light hydrocarbons, balanced seals are often used even though pressures are low to promote longer seal life.

Pump Model – Mechanical seal selection, installation and application varies with the pump model. Not all seals fit all pumps. Seal chamber configuration, pump discharge pressures and temperatures may not present ideal design conditions for the seal. Consult your ITT/FTC Pump Sales Engineer for selection of the proper pump and seal for the service.

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MECHANICAL SEALS ... PART IV - TROUBLESHOOTING

When discussing mechanical seal problems, the following analogy may be made: Mechanical seals behave very much like bearings. Conditions that lead to bearing failure also lead to mechanical seal breakdown. Let's consider these conditions.

Misalignment - One of the main causes of mechanical seal failure is misalignment. This commonly occurs when seals are installed by personnel not trained in proper mechanical seal installation practices. If the seal is installed so that the mating faces are not parallel to each other and perpendicular to the shaft, the rotating face will try to square itself with the seal causing uneven wear on the mating faces.

To prevent this problem, check to be sure that the seal is properly installed. The rotary unit must run parallel with the shaft and the stationary seat must be perpendicular to the shaft. Be sure the gland is drawn up evenly. This will eliminate the possibility of the improperly aligned stationary seat. Do not overtighten the gland - little more than finger tight is all that is generally needed in low pressure applications. Make sure all gaskets and/or O-Rings are properly installed.

Lack of Lubrication - To operate properly, the mechanical seal mating faces must run on a liquid film. If there is no film, the seal will overheat and fail. Because this failure can occur in a matter of seconds, it is extremely important that the liquid film be present whenever the pump is in operation or about to be put into operation. Lack of lubrication at the seal faces is commonly experienced when handling hot water or light hydrocarbons near their vapor pressure. To insure lubrication and dissipate heat, flushing must be provided at the seal faces.

Be sure seal flush piping is used when employing balanced seals - the type usually recommended for liquids having low specific gravities. The slight additional cost of bypass piping is a good investment when handling liquids near their vapor pressure.

Overheating - In some types of mechanical seal failures, overheating and lack of lubrication go hand in hand. This is especially true during the hard face heat checks. On examining a hard face that has heat checked you can see that the stress cracks are confined to the surface in direct contact with the carbon mating face. These surface cracks are caused when there is an intense heat buildup on the surface of the hard metal, while the subsurface of the part remains at ambient temperature. Since the surface metal is not allowed to expand in a lineal direction because of the subsurface temperature, the surface buckles upward and cracks. The edges of the cracks, which are slightly raised above the face, begin to shave material from the carbon mating face.

Other symptoms of overheating are damaged elastomers (O-rings). This condition is easily recognized because the elastomers will show cracking on the surface and harden.

To eliminate this type of failure, be sure the seal is operated in an environment having a lower temperature than the limits of any of its component parts.

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Abrasive Damage - Abrasive damage is a common cause of seal failure. When minute abrasive particles work their way between the seal faces, they can groove the mirror finish of the faces or they can leave deposits from evaporation of the liquid. When these deposits build up on the shaft they will freeze the sealing elements to the shaft and eliminate all seal flexibility. This condition can be particularly acute when teflon sealing wedges are incorporated in the seal. Abrasive particles may become imbedded in the teflon, and can cause shaft wear directly under the sealing member. Abrasive liquids must be kept out of the stuffing box if at all possible.

Corrosion - Corrosion failure is easily recognized in most cases. Metal parts show signs of pitting, or the sections are totally corroded away. We will not attempt to go into a discussion of corrosion, but it should be remembered that the metal sections of mechanical seals are usually much smaller than the metal section of a pump. Therefore, they can withstand much less corrosion attack. It is a good rule not to downgrade the metallurgy of the seal from that of the pump. When in doubt, select the more noble metal for the seals. (This is a general rule and does not hold true for all applications.)

QUENCH GLANDS

A gland is a device used to compress the pump packing or provide support for a mechanical seal face in the stuffing box of rotating or reciprocating machinery.

In particular, on centrifugal pumps, the primary function of a gland is to exert the necessary pressure on soft packing to hold it in place and to control the leakage of fluid from the stuffing box. It's also used to prevent air from entering the pump casing through the stuffing box. In the case of a mechanical seal, the gland acts as a clamp holding the stationary seal face in concentric alignment with the shaft axis and in perpendicular alignment with the rotating seal face.

The quench gland has a hollow portion adjacent to the shaft outside the stuffing box. This hollowed area can serve several purposes.

The gland can be used to remove heat from the shaft. Water, oil or other cooling is passed through the hollow portion and as it passes, it collects heat from the shaft and packing. Quenching in this manner permits pumping liquids at temperatures much higher than those normally allowed by the limits of bearing and packing lubrication. The quench lowers the temperature of the shaft at the packing and where it enters the bearing to permissible limits.

The same quench gland may be used to prevent the escape of toxic or volatile liquid into the environment. Again water, oil or some other suitable fluid is fed through the gland, flushing away the undesirable leakage to a waste receiver.

In some cases the gland is just used as a collector for the stuffing box leakage. There is no flushing connection - only a drain line to carry away any leakage from the stuffing box.

Because of the quenching, flushing or collecting function, a quench gland must be made to fit the shaft closely on the

(cont'd)

outboard end. If not close fitting, liquid would be allowed to spray or splash out along the shaft. Three separate methods may be used to achieve the close fit. First, the clearance between shaft and gland may be established by holding the I.D. of the outboard end of the gland to a close tolerance in comparison to the O.D. of the shaft in that area. This type of clearance cannot be restored without using a complete new gland and possibly a new shaft or new sleeve.

Second, the clearance may be established by using a replaceable bushing machined as above and pressed into a bored fit in the gland. This will allow restoring the clearance by changing only the bushing but would not compensate for any wear on the sleeve or the shaft. Third, auxiliary packing may be used. This may consist of a single ring of packing, a series of “V” rings and/or “O” rings - these are normally held in place by fitting them into a machined groove - or it may consist of several rings held in place by an auxiliary gland.

CONSTANT LEVEL OILERS - OPERATION AND ADJUSTMENT

ITT Industries oil lubricated pumps come with a variety of methods of distributing the oil to the bearings. Some, such as the “X series” power frames use a flood oil arrangement with a sight glass mounted in the side of the bearing frame for checking the level. Others use the sight glass and oil rings to pick the oil up from the sump and distribute it to the bearings. Still some current models and many of the older models use a bottle - type constant level oiler.

A cutaway of these oilers is shown in Figure 34. The oil stays in the bottle as long as the oil level in the bearing housing is level with the mouth of the bottle. When the oil level drops, air enters the bottle allowing oil to flow out until the oil level in the bearing housing is again up to the mouth of the bottle. The oil level in the bearing housing therefore stays constant.

The oiler bottle rests on a leveling bar that can be screwed up or down and locked in place. Raising the leveling bar raises the oiler bottle mouth and therefore the oil level in the bearing housing. The oil setting must be checked in the field. To check setting remove oiler bottle – dust cap assembly, and lift leveling bar assembly from oiler body (see Figure 35). Setting should be the same as that shown in the pump instruction book. The oiler is usually set so that the oil either:

1. covers half of the lowest ball in a flood oil lubricated ball bearing, or
2. is ¼” above the bottom of the oil ring in a ring oiled bearing.

The level must be set carefully. Too much oil is almost as bad as no oil at all.

Be sure the oiler is clean when it is installed in the pump. Piping from the oiler to the pump must be level. If the oiler sags, the oil level in the housing will drop which could possibly ruin the bearing. Fill the oiler bottle with the proper oil, and place in the oiler body. The bearing housing is filled when the

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bubbles of air entering the oiler stop and the level in the bottle remains constant. Never pour oil directly into the oiler body since you could easily overfill it.

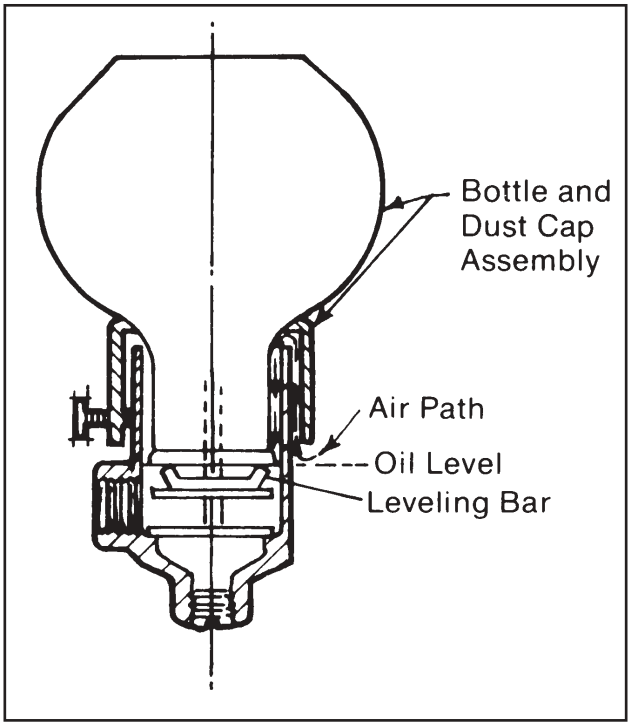

Figure 34

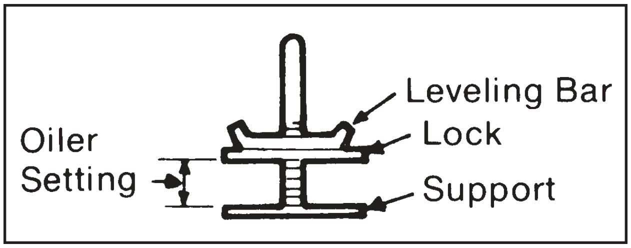

Figure 35

BALL BEARINGS – HANDLING, REPLACEMENT AND MAINTENANCE SUGGESTIONS

Ball and roller bearings are carefully designed and made to watch-like tolerances. They give long, trouble-free service when properly handled and maintained. They will not stand up to abuse.

Keep Clean

Dirt getting into the bearing during installation causes probably 90% of early bearing failures. Cleanliness is a must when working with bearings. Some things that will help are:

1. Do not open bearing housings unless absolutely necessary.

2. Spread clean plastic, newspapers or rags on the work benches and at the pump. Set your tools and the bearings on these covered surfaces only.

3. Wash your hands. Wipe dirt, chips and grease off tools.

(cont'd)

4. Keep the bearings, housings, and shaft covered with clean plastic or cloths whenever they are not being worked on.

5. Do not open the boxes or unwrap new bearings until you're ready to install them.

6. Flush the shaft and housings with clean solvent before reassembly.

Pull Bearings Carefully

1. Use a sleeve or puller which contacts just the inner race of the bearing. (The only exception to this is some double suction pumps which use the housing to pull the bearing.)

2. Never press against the balls or ball cages, only against the races.

3. Take care not to cock the bearing. Use a sleeve which is cut squarely, or puller which is adjusted to draw the bearing off the shaft squarely.

Inspect Bearings and Shaft

1. Examine the bearing carefully. Scrap it if there are any flat spots, nicks or pits on the balls or races. Bearings should be in perfect condition to be reused.

2. Turn the bearing over slowly by hand. It should turn smoothly and quietly. If not, scrap the bearing.

3. Whenever in doubt about the condition of a bearing, scrap it. The relatively few dollars invested in new bearings may prevent serious loss from downtime and pump damage. In severe or critical services, replace the bearings at each overhaul.

4. Check the condition of the shaft. Bearing surfaces should be smooth and free from burrs. Smooth burrs with emery cloth and polish with crocus cloth. Shaft shoulders should be square, free from nicks and burrs and smooth at the radii where smaller sections join larger sections.

Check New Bearings

Be sure the replacement bearing is of correct size and type. An angular contact bearing may be dimensionally interchangeable with a deep groove bearing and may fit perfectly in the pump. However, the angular contact bearing is not suitable for end thrust in both directions, and will probably fail quickly. Also check to see that the shields (if any) are the same as in the original unit. Refer to the pump bulletins and instruction book for the proper bearing to use and any notes regarding installation.

Install Carefully

1. Oil the bearing surfaces on the shaft lightly.

2. Shielding, if any, must face in the proper direction. Angular contact bearings, on pumps where they are used, must also be oriented in the proper direction. Duplex bearing arrangements must be mounted with the proper faces together. Mounting arrangements vary from model to model. Consult the bulletin and instruction book for specific pump model.

3. Press the bearing on the shaft squarely. Be sure that the sleeve used to press the bearing on is clean, free from burrs, is square cut and contacts the inner race only.

4. Press bearing firmly against shaft shoulder. The shoulder helps locate the bearing on the shaft, support the inside race and square the bearing.

5. Be sure the snap rings, if used, are properly installed, flat side against bearing, and that the gap at the ends of the ring align with the oil return channel in the bearing housing to provide an open path for the oil to travel back into the housing. Make sure the lock nuts are tight.

6. Lubricate properly. Consult the instruction manual for specific information regarding lubrication requirements.

GREASE LUBRICATION

ITT Industries grease lubricated pumps are designed for simple foolproof maintenance. Only one lubricant is required for a large variety of services. The correct lubricant is a premium ball, roller and plain bearing grease, which is oxidation resistant, has a wide temperature range and has a lithium soap base. The oil viscosity should be 200 – 250 SSU at 100°F. This would correspond to NLGI Grade 2 Grease. “Filled” greases such as those with molybdeum, graphite, Teflon™ and other fillers should be avoided.

This one type of grease will suffice for nearly all applications; hot and cold; wet and dry; clean and dirty.

Grease should be added to the bearing after each 2000 hours of operation, or about every three months. More frequent lubrication can lead to shorten bearing life due overheating.

The normal bearing temperature depends on many variables, the ambient temperature in the area of the unit, the temperature of the liquid being pumped, the speed of the shaft and the amount of air flow around the pump are only a few. It is expected that the bearing temperature will stabilize at some level between 130° and 180°F which is perfectly acceptable for a grease or oil lubricated assembly. For most people anything above 120°F is uncomfortable to touch, so it is not unusual to have a properly lubricated, aligned, and operating bearing assembly which is too hot to hold your hand on.

On the other hand, a sudden temperature rise without a corresponding rise in ambient or pumpage temperature can be a warning sign that there is something going wrong.

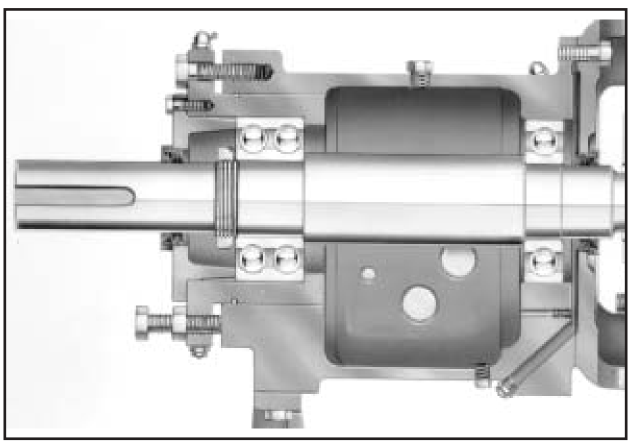

Figure 36

FLEXIBLY MOUNTED BASEPLATES

The flexibly mounted baseplate was first used in chemical plants in the mid 1950s. At that time, special extra heavy bases were used to provide adequate rigidity. Historically, pump and other rotating equipment suppliers have recommended a grouted base to provide a suitable platform to insure a permanent means of establishing and maintaining alignment between the rotating machine and its driver.

The permanent grouted base is still recommended for large units where flange loads can be adequately limited, and where the base can be suitably maintained. In many chemical plants atmospheric corrodents make baseplates and foundations expensive and difficult to maintain. As a result, after a short time, the baseplate can deteriorate to a point where it no longer is able to maintain suitable alignment.

Under adverse environmental conditions a well engineered flexibly mounted base can provide an economical solution to difficult problems. The flexible mounting system offers the following advantages: **Reduced installation costs** – no tiled or acid proof brick foundation required, no grouting required, the pump can easily be adjusted to line up with piping. **Lower maintenance costs** – uniform flange loads will move the pump, base and motor as a unit reducing both the possibility of coupling misalignment and the seal problems normally associated with excess flange loading. The raised base can be easily hosed to remove corrosives and debris for simplified housekeeping and longer baseplate life. A successful installation requires that the base have sufficient rigidity to permit initial alignment and to maintain alignment over the operational life of the equipment. Goulds offers the flexibly mounted base arrangement as an option on our ANSI Family units and many other pump types. The standard cast iron or structural steel base normally furnished on the ANSI Family X-Series units have proved satisfactory under actual plant conditions up to 100 HP at 3550 RPM.

Users report that the elimination of the expensive foundation enables them to use a standard horizontal ANSI type pump at the same installation cost as a vertical inline unit.

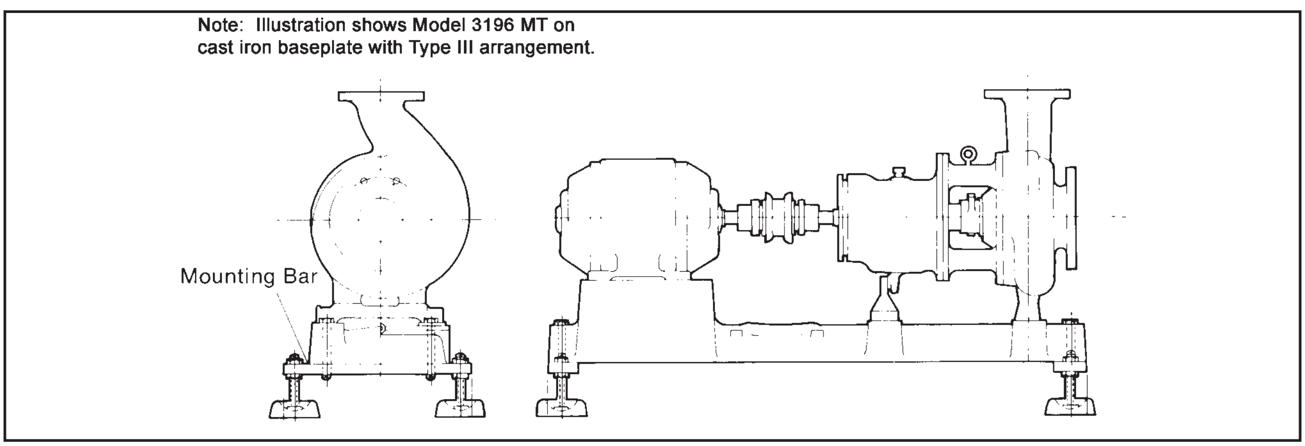

Figure 37

IMPELLER CLEARANCE

The open impeller centrifugal pump offers several advantages over units equipped with other types of impellers. It is particularly suited to applications where the liquid contains abrasive solids.

Abrasive wear on an open impeller is distributed over the diametrical area swept by the vanes. The resulting total wear has less effect on performance than the same total wear concentrated on the radial ring clearance of a closed impeller.

Because of the impeller adjustment feature, the open impeller permits restoration of nearly "new pump" running clearance after wear has occurred without expensive parts replacement.

A well designed open impeller pump will feature a simple positive means for axial adjustment without necessity of disassembling the unit to add shims or gaskets.

Figure 38, with these typical instructions, shows one method employed on ITT Goulds open impeller centrifugal pumps for making this adjustment. Other methods using Dial Indicators, etc, may be used. Since the actual impeller clearance varies with pump type and temperature, consult the specific Operating Instruction for your pump for the proper settings. On units equipped with mechanical seals, follow the seal manufacturers instructions for the care of the seal during the impeller adjustment process.

1. Loosen the jam nuts and bolts (370D) located at the bearing housing (111).
2. Tighten the bolts (370C) evenly while slowly rotating shaft, until the impeller (101) just contacts the casing (100).
3. Snug all the bolts (370C) against the bearing housing (111). Loosen each bolt (370C) until a .015" feeler gauge can be placed between the bearing housing and the underside of the head of the bolt (370C).
4. Be sure the jam nuts on the bolts (370D) are loose. Tighten each bolt (370D) one flat at a time until the bearing housing is tight against the bolts (370C). Be sure all the bolts (370C and 370D) are tight. Tighten the jam nuts on bolts (370D).
5. The rotating element and consequently the impeller has been moved .015" away from the casing thus giving the required clearance between these two parts.

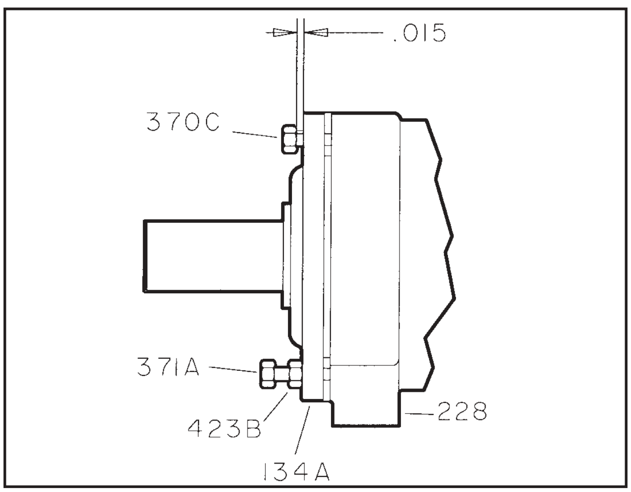

Figure 38

(cont'd)

PUMP VIBRATION ANALYSIS

Figure 39

Pump users have become increasingly aware of vibration and the use of vibration analysis in detecting problems, predicting failures and scheduling equipment outages.

The vibration analyzer shown in Figure 39 is used to measure and indicate the amplitude, frequency and phase values of vibration. Furthermore, when vibration occurs at several frequencies, it separates one frequency from another so that each individual vibration characteristics can be identified and evaluated.

The vibration pickup, or transducer, (A) senses the motion of the machine and converts it into an electrical signal. This signal may represent the amount of movement, which would be displacement, the velocity at which the machine is moving as it vibrates or the acceleration required to produce the particular velocity. The analyzer receives this signal and converts it to the corresponding amplitude and frequency.

Depending on the instrument and the settings used, the amplitude may be displayed in terms of peak-to-peak vibration, which would be the maximum values present or various methods of averaging such as RMS. Displacement is indicated in mils where 1 mil equals 1 one-thousandth of an inch, Velocity in Inches per Second and Acceleration in "G's".

Many machines today have graphical displays for an immediate indication of both the amplitude and frequency of the vibration present and some have printers built into them to print out the information right on the spot. Often the units include a down loading feature so that the information collected may be saved on a PC or disk for later analysis.

Most instruments are equipped with a device to indicate frequency which gives a direct readout of the predominant frequency of the vibration. Other instruments have tunable filters (C) which allow scanning the frequency scale and

(cont'd)

reading amplitudes at any particular frequency.

The strobe light (D) is used to determine the phase of the vibration which can be of great value when dealing with complicated vibration signatures and in depth analysis. The strobe can be made to flash at the predominant frequency of the vibration or at any frequency setting using an internal oscillator to fire it.

A reference mark on a rotating part viewed under the strobe flashing at the vibration frequency may appear as a single mark, multiple marks and either stationary or rotating. The number of marks viewed and whether they are rotating or stationary is useful in determining the source of the vibration. The relationship between the location of the mark on the part and where it is illuminated by the strobe can be determined and that information can be used when balancing rotating parts.

The first step in vibration analysis is to determine the severity of the vibration, then, if the vibration is serious, a complete set of vibration readings should be taken before attempting to analyze the cause. Figure 40 is a severity chart based on amplitude, frequency and velocity. Figure 41 is a general guide for centrifugal pumps as published by the Hydraulic Institute.

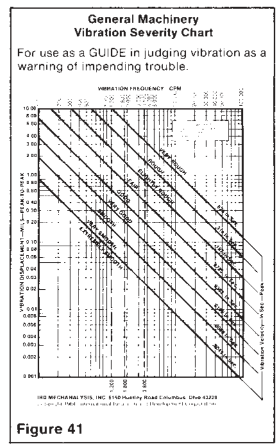

Figure 40

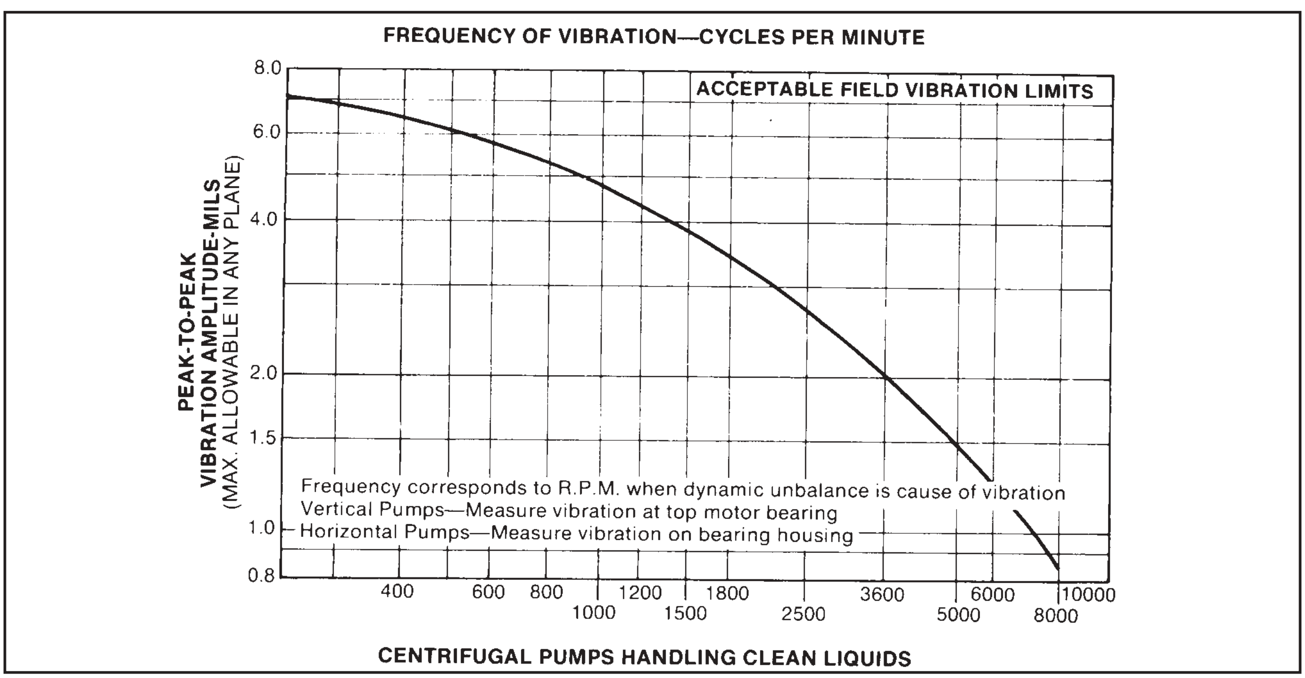

Figure 41

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The severity of vibration is a function of amplitude and frequency; however, it should be noted that a change in severity over a period of time is usually an indication that the condition of the machine is deteriorating. This change is often more important than the actual vibration values themselves.   Vibration levels in the “slightly rough” or “rough” ranges that do not change with time maybe be perfectly acceptable. The opposite is also true; a machine that exhibits vibration in the “Very Smooth” area but moves to the “Very Good” area in a short period of time may be very close to failure.

Complete pump vibration analysis requires taking vibration readings at each bearing in three planes (horizontal, vertical and axial). Readings at the pump suction and discharge flanges may also be useful in some cases.

After all data has been tabulated, it can be analyzed to determine the most likely cause or causes of the vibration. Figure 42 lists the most common causes of vibration and the identifying characteristics of each.

By analyzing the tabulated vibration data with the reference to the identification chart, one or more causes may be found. Each possibility must be checked, usually starting with the most likely cause or easiest or least expensive to correct.

For example, assume that axial vibration is 50% or more of the horizontal or vertical vibration and the predominant frequency is the same as the RPM of the pump. The chart indicates the most probable cause is misalignment or a bent shaft. Coupling misalignment is probably the most common single cause of pump vibration and is one of the easiest to check. If, after checking, the alignment is found to be within acceptable limits, then an inspection for excessive flange loading is warranted. Flange loading beyond the limits allowed can cause distortion and misaligment inside the pump leading to higher than expected vibration levels. If that inspection does not reveal the source then the next step is to check for a bent pump shaft.

Cavitation in a pump can cause serious vibration. Vibration caused by it usually appears at random frequencies but it is most often identified by a loud crackling noise in the suction area of the pump rather than by any characteristic vibration. Vibration at random frequencies can also be caused by hydraulic disturbances in poorly designed suction or discharge systems.

Vibration analysis, while a relatively new field, has rapidly gained acceptance in industry as a valuable tool in preventive and predictive maintenance and as a general troubleshooting tool.

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Vibration Identification Chart				
Cause	**Amplitude**	**Frequency**	**Phase**	**Remarks**
Unbalance	Largest in radial direction. Proportional to unbalance.	1 X RPM	Single reference mark	
Misalignment of coupling or bearings and bent shaft	Axial direction vibration 50% or more of radial	1 X RPM normally	Single, double, or triple	Easily recognized by large axial vibration. Excessive flange loading can contribute to misalignment.
Bad anti-friction bearings	Unsteady	Very high. Several times RPM.	Erratic	Largest high-frequency vibration near the bad bearing.
Mechanical looseness		2 X RPM	Two reference marks. Slightly erratic	Check grouting and baseplate bolting.
Bad drive belts	Erratic or pulsing	1, 2, 3, & 4 X RPM of belts	Unsteady	Use strobe light to freeze faulty belt.

Figure 42

PUMP MAINTENANCE RECORDS PAY DIVIDENDS

Too often plant records of pumps are little more than an inventory record for accounting and insurance purposes listing not much more than the pump size and manufacturer. In other plants, maintenance records may be kept as part of a pump lubrication schedule. In either case complete pump records could be kept with very little additional effort. With the wide spread use of PC's in industry and the development of inexpensive Data Base programs more and more organizations are going to Maintenance Tracking Systems in an effort to reduce downtime, lost production and maintenance costs. Complete maintenance records, filed in an accessible location, are invaluable in diagnosing pump failure, in ordering repair parts and in establishing lubrication and maintenance schedules. In addition these maintenance record cards are helpful in determining a pump's suitability for new requirements due to process changes. Notations of pump failures and the repairs required may be used to define the optimum period of any given pump before complete inspection and overhaul is required. If a unit has been on an annual overhaul program and the pump Maintenance record indicates that no failures have occurred over a period of years, perhaps a longer period between inspections would be warranted.

On other pumps, the records may indicate frequent failures. This may suggest that semi-annual inspections and repairs are required. It may also suggest that there is a need to take a closer look at this unit. Something may be wrong with the unit itself, its assembly, the application, the operation or the installation.

The chronological notation of pump repairs can be used to develop a proper spare parts inventory and may, in some cases reduce inventory and the costs associated with it. Frequent replacement of a part subject to abrasive wear may indicate a different material selection would be more economical.

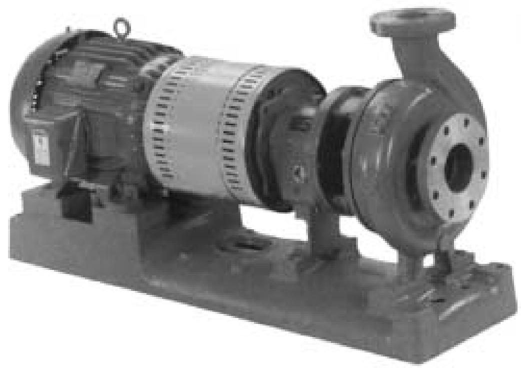
Simple maintenance records for each pump, in both primary and secondary service, are much less expensive than trial and error solutions.

ITT INDUSTRIES / FTC OFFERS A COMPLETE LINE OF PUMPS TO MEET EVERY INDUSTRIAL NEED

ANSI Process

Model 3196

This is the original ANSI pump that has become the standard of the industry. Over 600,000 installations attest to the remarkable performance of the 3196.



Magnetic Drive

Model 3298

The 3298 is designed specifically to handle moderate to severe corrosives with or without solids. As a sealless design, it's an effective alternative to pumps with mechanical seal problems. Meets strictest EPA regulations.

Abrasive Slurry

Model SRL

SRL rubber lined pumps are designed specifically for handling abrasive and corrosive slurries found in the mining and mineral process industries, and in pulp and paper, aggregate, pollution control and chemical applications.

In-Line ANSI Process

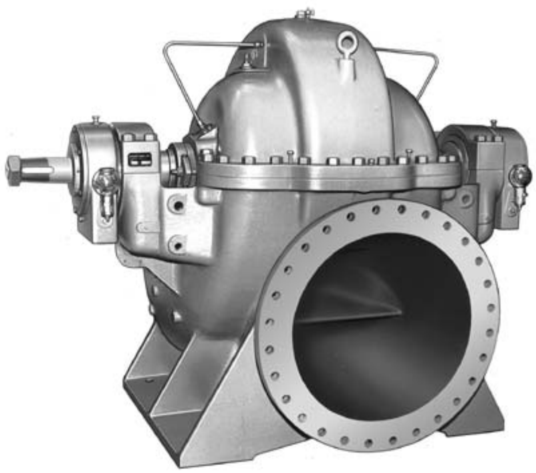
Model 3996

For corrosives, abrasives and high temperatures. Fully open impeller, back pull-out design, heavy duty construction. Field alignment not required.

Splitcase Double Suction

Model 3420

For a wide range of industrial, municipal, marine services .. cooling tower, raw water supply, booster, fan pump, high/low lift, fire pump, pipelines.



Vertical Industrial

Model VIT (Turbine)

A wide range of hydraulic conditions allows meeting requirements of virtually every pump service. These pumps are designed to meet custom user specifications. Model VIC can-type turbine meets API-610 specifications.

Model VIC (Can-type)

Paper Stock Process

Model 3180

All customer requirements were considered in the design of this paper stock/process pump. The result is a true world class pump line. . .a product without compromise.

Vertical Sump and Process

Model 3171

Vertical industrial pumps for wide range liquid handling capabilities. For drainage and general sump, tank unloading and transfer, process circulation, outside tank mounting, high temperature service.

ISO Lined Sealless

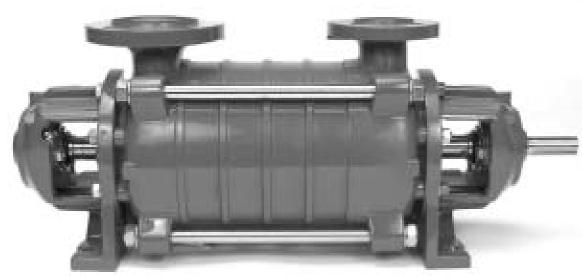
Model Richter PCK

The ICK is designed to handle moderate to sever corrosives, providing excellent service in a variety of process plants worldwide.

General Industry

Model 3355

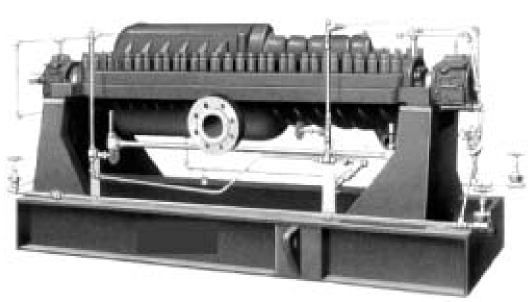
Goulds Model 3355 is a multi-stage ring section pump designed for high-pressure services including: boiler feed, reverse osmosis, shower service, pressure boosting, plus much more. Its modular design and multiple configurations make it ideal for your system.



Multi-Stage

Model 3600

Advanced design with proven operating history. Axially split, with many enhanced features that make it an extremely reliable, high performance, pump well suited to a wide range of services.



API-610 (8th Edition) Process

Model 3700

High temperature and high pressure process pumps designed to meet the requirements of API-610 (8th Edition). Centerline support for high temperature stability, maximum rigidity. Tangential discharge for maximum hydraulic efficiency.

Pump Category	Goulds Model	Pump Type	Chemical	Pulp & Paper	Mining & Minerals	Power Generation	Oil Refining & Gas Processing	Primary Metals	Water & Wastewater	Food & Beverage	Nature of Pumpage: Corrosive	Nature of Pumpage: High Temperature 500°F (260°C) or Greater	Solids: Abrasive	Solids: Non-Abrasive	Solids: Fibrous/Stringy
PumpSmart®	Process Pumping System		z	q	m					&	x	x	x	x	x
PRO Services™	Repair & Service of all Rotating Equipment Types of Manufacture														
Chemical Process	3196	ANSI Chemical Process	z	q	m	@	/	—	o	&	x	x		x	
	LF 3196	Low Flow ANSI Process	z	q		@	/			&	x	x		x	
	IC	ISO Chemical Process	z	q	m	@	/	—	o	&	x	x		x	
	NM 3196	FRP Process	z	q				—			x	x			
	3796	Self-Priming Process	z	q	m		/	—	o		x	x		x	
	3996	In-Line Process	z	q	m	@	/	—	o	&	x	x		x	
	3296	Magnetic Drive Process	z	q			/				x	x			
	3298/SP3298	Tefzel® Lined Sealless	z	q			/	—	o	&	x				
	V 3298	Tefzel® Lined Sealless	z	q			/	—	o	&	x				
	3299	PFA Lined Sealless	z	q			/	—	o	&	x			x	
	MPB/F	Peripheral with Magnetic Drive	z	q			/	—	o	&	x			x	
	MNK	Lined Sealless	z	q			/	—	o	&	x			x	
	MDK	Lined Sealless	z	q			/	—	o	&	x				
	PCK/SCK	Lined/Heavy Duty Lined Process	z	q			/	—	o	&	x			x	
	3198	PFA Teflon® Process	z	q			/	—	o	&	x			x	
	3200	Light Duty Sealless	z	q			/	—	o	&	x			x	
	CV 3196	Non-Clog Process	z	q	m		/	—	o		x	x	x		x
	AF	Axial Flow	z	q		@	/		o		x		x		
	3171	Vertical Sump & Process	z		m	@	/		o		x				
	NM 3171	FRP Vert. Sump/Process	z						o		x				
Paper Stock/ Process	3175	Paper Stock/Process	z	q	m	@	/	—	o	&	x			x	x
	3180/3185	Paper Stock/Process	z	q	m	@	/	—	o	&	x			x	x
	3181/3186	High Temperature	z	q							x	x		x	
	3500	Heavy Duty Paper Stock		q										x	x
API Process	3700/3710	API-610 Process	z	q		@	/		o		x	x			
	3910	API 610-In-Line	z			@	/		o		x	x			
	3620	High Temp. Double Suction				@	/					x			
	3640	High Temp. Two-Stage					/					x			
	CPR	API Process	z	q		@	/		o	&	x	x			
Abrasive Slurry/Solids Handling	JC	Medium Duty Abrasive Slurry	z	q	m	@		—	o		x		x		
	SRL	Rubber-Lined Abrasive Slurry	z	q	m	@		—	o		x		x	x	x
	5000/RX	Side Suction Abrasive Slurry			m	@		—	o		x		x	x	
	Slurry Pro	Heavy Duty Abrasive Slurry			m	@		—	o		x		x	x	
	5500	Severe Duty Abrasive Slurry			m	@		—	o		x		x	x	
	HS	Non-Clog Solids Handling	z	q	m	@	/		o	&	x		x	x	x
Multi-Stage & Double Suction	3310H	High Pressure Multi-Stage	z	q	m	@	/		o					x	
	3600	Heavy Duty Multi-Stage			m	@	/	—						x	
	3335	Diffuser Type Multi-Stage	z	q	m	@	/		o	&				x	
	3935		z	q	m	@	/		o	&				x	
	3316	Two-Stage	z	q	m	@	/		o					x	
	3408	Single-Stage, Double Suction	z	q	m	@	/	—	o	&				x	
	3409		z	q	m	@	/	—	o	&				x	
	3410		z	q	m	@	/	—	o	&				x	
	3415		z	q	m	@	/	—	o	&				x	
	3420		z	q	m	@	/	—	o	&				x	
	3498		z	q	m	@	/	—	o	&				x	
	MP	Multi-Stage	z	q	m	@	/	—	o	&	x				
	P	Multi-Stage	z	q	m	@	/	—	o	&	x				
Sump/ Abrasives/ Solids Handling	Whirl-Flo	Vortex	z	q	m			—	o	&	x		x		x
	Prime Line	Industrial Self-Priming	z	q	m	@	/		o	&	x		x	x	
	Trash Hog	Solids Handling, Self-Priming	z	q	m				o	&	x		x		x
	VHS	Vertical Cantilever	z	q	m	@	/	—	o		x		x	x	x
	5100		z	q	m	@	/	—	o		x		x	x	x
	5150		z	q	m	@	/	—	o		x		x	x	x
	VJC		z	q	m	@	/	—	o		x		x	x	x
	HSU	Submersible	z	q	m	@	/		o	&	x		x	x	x
	HSUA		z	q	m	@	/		o	&	x		x	x	x
	HSUL		z	q	m	@	/		o	&	x		x	x	x
	JCU		z	q	m	@	/		o	&	x		x	x	x
General Service/Vertical Turbine	3100	General Service	z	q	m			—	o	&	x		x	x	
	VIT & VIC	Vertical Turbine/Can Type	z	q	m		/	—	o	&	x			x	
	VMF/VAF	Mixed Flow/Axial Flow	z	q					o					x	
	VMP	Vertical Marine							o					x	
	VIS	Vertical Submersible			m		/		o					x	

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