Nomenclature and Definitions

BY PAT FLACH

Vapor pressure, cavitation, and NPSH are subjects widely discussed by engineers, pumps users, and pumping equipment suppliers, but understood by too few. To grasp these subjects, a basic explanation is required.

VAPOR PRESSURE

Knowledge of vapor pressure is extremely important when selecting pumps and their mechanical seals. Vapor pressure is the pressure absolute at which a liquid, at a given temperature, starts to boil or flash to a gas. Absolute pressure (psia) equals the gauge pressure (psig) plus atmospheric pressure.

Let's compare boiling water at sea level in Rhode Island to boiling water at an elevation of 14,110 feet on top of Pikes Peak in Colorado. Water boils at a lower temperature at altitude because the atmospheric pressure is lower.

Water and water containing dissolved air will boil at different temperatures. This is because one is a liquid and the other is a solution. A solution is a liquid with dissolved air or other gases. Solutions have a higher vapor pressure than their parent liquid and boil at a lower temperature. While vapor pressure curves are readily available for liquids, they are not for solutions. Obtaining the correct vapor pressure for a solution often requires actual laboratory testing.

CAVITATION

Cavitation can create havoc with pumps and pumping systems in the form of vibration and noise. Bearing failure, shaft breakage, pitting on the impeller, and mechanical seal leakage are some of the problems caused by cavitation.

When a liquid boils in the suction line or suction nozzle of a pump, it is said to be "flashing" or "cavitating" (forming cavities of gas in the liquid). This occurs when the pressure acting on the liquid is below the vapor pressure of the liquid. The damage occurs when these cavities or bubbles pass to a higher pressure region of the pump, usually just past the vane tips at the impeller "eye," and then collapse or "implode."

NPSH

Net Positive Suction Head is the difference between suction pressure and vapor pressure. In pump design and application jargon, $NPSH_A$ is the net positive suction head available to the pump, and $NPSH_R$ is the net positive suction head required by the pump.

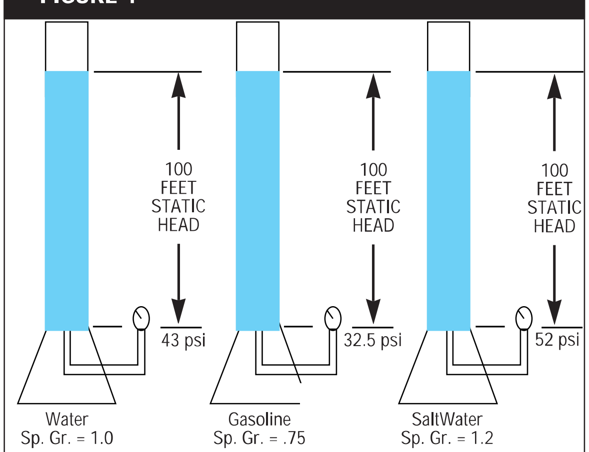
The $NPSH_A$ must be equal to or greater than the $NPSH_R$ for a pump to run properly. One way to determine the $NPSH_A$ is to measure the suction pressure at the suction nozzle, then apply the following formula:

$$NPSH_A = P_B - V_P \pm Gr + h_v$$

where P_B = barometric pressure in feet absolute, V_P = vapor pressure of the liquid at maximum pumping temperature in feet absolute, Gr = gauge reading at the pump suction, in feet absolute (plus if the reading is above barometric pressure, minus if the reading is below the barometric pressure), and h_v = velocity head in the suction pipe in feet absolute.

$NPSH_R$ can only be determined during pump testing. To determine it, the test engineer must reduce the $NPSH_A$ to the pump at a given capacity until the pump cavitates. At this point the vibration levels on the pump and system rise, and it sounds like gravel is being pumped. More than one engineer has run for the emergency shut-down switch the first time he heard cavitation on the test floor. It's during these tests that one gains a real appreciation for the damage that will occur if a pump is allowed to cavitate for a prolonged period.

FIGURE 1



Static head using various liquids.

CENTRIFUGAL PUMPING

Centrifugal pumping terminology can be confusing. The following section addresses these terms and how they are used:

Head is a term used to express pressure in both pump design and system design when analyzing static or dynamic conditions. This relationship is expressed as:

$$\text{head in feet} = \frac{(\text{pressure in psi x } 2.31)}{\text{specific gravity}}$$

Pressure in static systems is referred to as **static head** and in a dynamic system as **dynamic head.**

To explain static head, let's consider three columns of any diameter, one filled with water, one with gasoline, and one with salt water (Figure 1). If the columns are 100 ft tall and you measure the pressure at the bottom of each column, the pressures would be 43, 32.5, and 52 psi, respectively. This is because of the different specific gravities, or weights, of the three liquids. Remember, we are measuring pounds per square inch at the bottom of the column, not the total weight of the liquid in the column.

The following four terms are used in defining pumping systems and are illustrated in Figure 2.

Total static head is the vertical distance between the surface of the suction source liquid and the surface level of the discharge liquid.

Static discharge head is the vertical distance from the centerline of the suction nozzle up to the surface level of the discharge liquid.

Static suction head applies when the supply is above the pump. It is the vertical distance from the centerline of the suction nozzle up to the liquid surface of the suction supply.

Static suction lift applies when the supply is located below the pump. It is the vertical distance from the centerline of the suction nozzle down to the surface of the suction supply liquid.

Velocity, friction, and pressure head are used in conjunction with static heads to define dynamic heads.

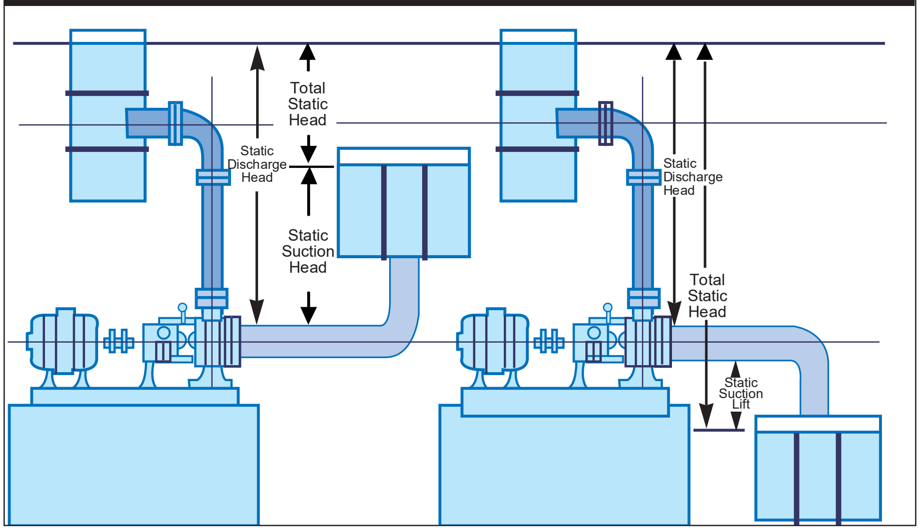
Velocity head is the energy in a liquid as a result of it traveling at some velocity V. It can be thought of as the vertical distance a liquid would need to fall to gain the same velocity as a liquid traveling in a pipe.
This relationship is expressed as:

$$h_V = V^2/2g$$

where V = velocity of the liquid in feet per second and g = 32.2 ft/sec^2.

Friction head is the head needed to overcome resistance to liquid flowing in a system. This resistance can come from pipe friction, valves, and fittings. Values in feet of liquid can be found in the Hydraulic Institute Pipe Friction Manual.

FIGURE 2



Total static head, static discharge head, static suction head, and static suction lift.

Pressure head is the pressure in feet of liquid in a tank or vessel on the suction or discharge side of a pump. It is important to convert this pressure into feet of liquid when analyzing systems so that all units are the same. If a vacuum exists and the value is known in inches of mercury, the equivalent feet of liquid can be calculated using the following formula:

$$\text{vacuum in feet} = \frac{\text{in. of Hg x 1.13}}{\text{specific gravity}}$$

When discussing how a pump performs in service, we use terms describing dynamic head. In other words, when a pump is running it is dynamic. Pumping systems are also dynamic when liquid is flowing through them, and they must be analyzed as such. To do this, the following four dynamic terms are used.

Total dynamic suction head is the static suction head plus the velocity head at the suction flange minus the total friction head in the suction line. Total dynamic suction head is calculated by taking suction pressure at a pump suction flange, converting it to head and correcting to the pump centerline, then adding the velocity head at the point of the gauge.

Total dynamic discharge head is the static discharge head plus the velocity head at the pump discharge flange plus the total friction head in the discharge system. This can be determined in the field by taking the discharge pressure reading, converting it to head, and correcting it to the pump centerline, then adding the velocity head.

Total dynamic suction lift is the static suction lift minus the velocity head at the suction flange plus the total friction head in the suction line. To calculate total dynamic suction lift, take suction pressure at the pump suction flange, convert it to head and correct it to the pump centerline, then subtract the velocity head at the point of the gauge.

Total dynamic head in a system is the total dynamic discharge head minus the total dynamic suction head when the suction supply is above the pump. When the suction supply is below the pump, the total dynamic head is the total dynamic discharge head plus the total dynamic suction lift.

Centrifugal pumps are dynamic machines that impart energy to liquids. This energy is imparted by changing the velocity of the liquid as it passes through the impeller. Most of this velocity energy is then converted into pressure energy (total dynamic head) as the liquid passes through the casing or diffuser.

To predict the approximate total dynamic head of any centrifugal pump, we must go through two steps. First, the velocity at the outside diameter (o.d.) of the impeller is calculated using the following formula:

$v = (rpm \times D)/229$

where v = velocity at the periphery of the impeller in ft per second, D = o.d. of the impeller in inches, rpm = revolutions per minute of the impeller, and 229 = a constant.

Second, because the velocity energy at the o.d. or periphery of the impeller is approximately equal to the total dynamic head developed by the pump, we continue by substituting v from above into the following equation:

$H = v^2/2g$

where H = total dynamic head developed in ft, v = velocity at the o.d. of the impeller in ft/sec, and g = 32.2 ft/sec^2.

A centrifugal pump operating at a given speed and impeller diameter will raise liquid of any specific gravity or weight to a given height. Therefore, we always think in terms of feet of liquid rather than pressure when analyzing centrifugal pumps and their systems. ■

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Pumping Terms

Have you had a momentary (or continuing) problem with converting gallons per minute to cubic meters per second or liters per second? Join the crowd. Though the metric or SI system is probably used as the accepted system, more than English units, it still presents a problem to a lot of engineers.

Authors are encouraged to use the English system. Following is a list of the common conversions from English to metric units. This is far from a complete list. It has been limited to conversions frequently found in solving hydraulic engineering problems as they relate to pumping systems.

PUMPING UNITS

FLOW RATE

(U.S.) gallons/min (gpm) x 3.785 = liters/min (L/min)
(U.S.) gpm x 0.003785 = cubic meters/min (m^3/min)
cubic feet/sec (cfs) x 0.028 = cubic meters/sec (m^3/s)

HEAD

feet (ft) x 0.3048 = meters (m)
pounds/square inch (psi) x 6,895 = Pascals (Pa)

POWER

horsepower (Hp) x 0.746 = kilowatts (kW)

GRAVITATIONAL CONSTANT (g)

32.2 ft./s^2 x 0.3048 = 9.81 meters/second^2 (m/s^2)

SPECIFIC WEIGHT

lb/ft^3 x 16.02 = kilogram/cubic meter (kg/m^3)

VELOCITY (V)

ft/s x 0.3048 = meters/second (m/s)

VELOCITY HEAD

$V^2/2g$ (ft) x 0.3048 = meters (m)

SPECIFIC SPEED (N_S)

(gpm–ft) x 0.15 = Ns(m3/min–m)
$N_S = N(rpm)[(gpm)^{0.5}/(ft)^{0.75}]$

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TABLE 1. ENGLISH TO METRIC CONVERSION

Basic Units	Multiply English	x Factor	= Metric
Length	Feet	x 0.3048	= Meter (m)
Mass	Pound	x 0.454	= Kilogram (Kg)
Force	Pound	x 4.448	= Newton (N)
Pressure	Pound/Square In. (psi)	x 6,895	= Pascal (Pa)
Time	Seconds	x 1	= Seconds (s)
Gallon (US)	Gallon	x 0.003785	= Meter Cubed (m^3)
Gallon (US)	Gallon	x 3.785	= Liter (L)

Centrifugal and Positive Displacement Pumps in the Operating System

BY ROSS C. MACKAY

In the many differences that exist between centrifugal and positive displacement pumps, one which has caused some confusion is the manner in which they each operate within the system.

Positive displacement pumps have a series of working cycles, each of which encloses a certain volume of fluid and moves it mechanically through the pump into the system, regardless of the back pressure on the pump. While the maximum pressure developed is limited only by the mechanical strength of the pump and system and by the driving power available, the effect of that pressure can be controlled by a pressure relief or safety valve.

A major advantage of the positive displacement pump is that it can deliver consistent capacities because the output is solely dependent on the basic design of the pump and the speed of its driving mechanism. This means that, if the required flow rate is not moving through the system, the situation can always be corrected by changing one or both of these factors.

This is not the case with the centrifugal pump, which can only react to the pressure demand of the system. If the back pressure on a centrifugal pump changes, so will its capacity.

This can be disruptive for any process dependent on a specific flow rate, and it can diminish the operational stability, hydraulic efficiency and mechanical reliability of the pump.

CENTRIFUGAL PUMP PERFORMANCE CURVE

The interdependency of the system and the centrifugal pump can be easily explained with the use of the pump performance curve and the system curve.

A centrifugal pump performance curve is a well known shape which shows that the pressure the pump can develop is reduced as the capacity increases. Conversely, as the capacity drops, the pressure it can achieve is gradually increased until it reaches a maximum where no liquid can pass through the pump. Since this is usually a relatively low pressure, it is rarely necessary to install a pressure relief or safety valve.

When discussing the pressures developed by a centrifugal pump, we use the equivalent linear measurement referred to as "head," which allows the pump curve to apply equally to liquids of different densities.

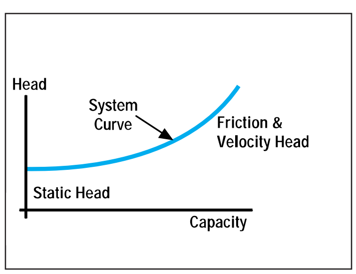
[Head (in feet)=Pressure (in p.s.i.) x 2.31+ Specific Gravity of the liquid]

SYSTEM CURVE

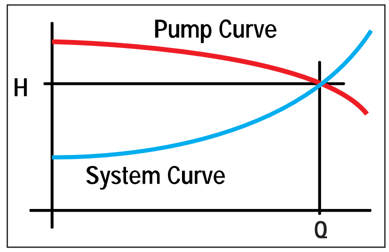
The system curve represents the pressures needed at different flow rates to move the product through the system. To simplify a comparison with the centrifugal pump curve, we again use the 'head' measurement. The system head consists of three factors:

- static head, or the vertical elevation through which the liquid must be lifted
- friction head, or the head required to overcome the friction losses in the pipe, the valves and all the fittings and equipment
- velocity head, which is the head required to accelerate the flow of liquid through the pump (Velocity head is generally quite small and often ignored.)

As the static head does not vary simply because of a change in flow rate, the graph would show a straight line. However, both the friction head and the velocity head will always vary directly with the capacity. The combination of all three creates the system curve.



When the pump curve is superimposed on the system curve, the point of intersection represents the conditions (H,Q) at which the pump will operate.



Pumping conditions change ONLY through an alteration in either the pump curve or the system curve.

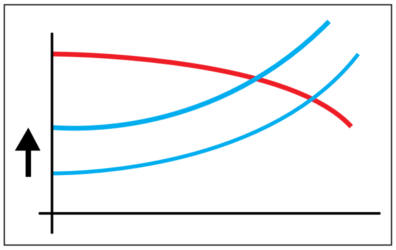
When considering possible movements in these curves, it should be noted that there are only a few conditions which will cause the pump curve to change its position and shape:

- wear of the impeller
- change in rotational speed
- change in impeller diameter
- change in liquid viscosity

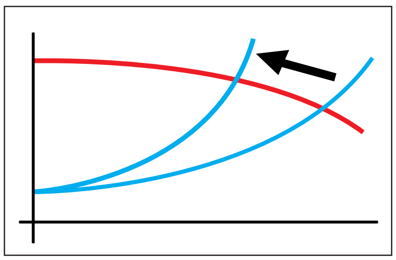
Since these conditions don't normally develop quickly, any sudden change in pumping conditions is likely to be a result of a movement in the system curve, which means something in the system has changed.

Since there are only three ingredients in a system curve, one of which is minimal, it follows that either the static head or the friction head must have changed for any movement to take place in the system curve.

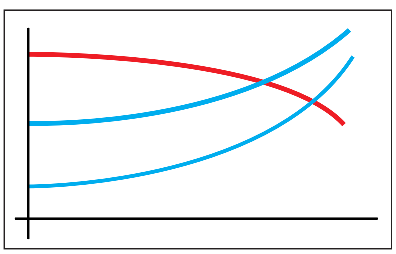
A change in the static head is normally a result of a change in tank level. If the pump is emptying a tank and discharging at a fixed elevation, the static head against which the pump must operate will be gradually increasing as the suction tank empties. This will cause the system curve to move upwards as shown.



An increase in friction head can be caused by a wide variety of conditions such as the change in a valve setting or build-up of solids in a strainer. This will give the system curve a new slope.



Both sets of events produce the same result: a reduction of flow through the system. If the flow is redirected to a different location (such as in a tank farm), it means that the pump is now operating on an entirely new system which will have a completely different curve.



Thus, it is clear that regardless of the rated capacity of the centrifugal pump, it will only provide what the system requires. It is important to understand the conditions under which system changes occur, the acceptability of the new operating point on the pump curve, and the manner in which it can be moved.

When the operating conditions of a system fitted with a centrifugal pump change, it is helpful to consider these curves, focus on how the system is controlling the operation of the pump, and then control the system in the appropriate way. ■

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Cavitation and NPSH in Centrifugal Pumps

BY PAUL T. LAHR

Cavitation is the formation and collapse of vapor bubbles in a liquid.

Bubble formation occurs at a point where the pressure is less than the vapor pressure, and bubble collapse or implosion occurs at a point where the pressure is increased to the vapor pressure. Figure 1 shows vapor pressure temperature characteristics.

This phenomenon can also occur with ship propellers and in other hydraulic systems such as bypass orifices and throttle valves—situations where an increase in velocity with resulting decrease in pressure can reduce pressure below the liquid vapor pressure.

CAVITATION EFFECTS

BUBBLE FORMATION PHASE

Flow is reduced as the liquid is displaced by vapor, and mechanical imbalance occurs as the impeller passages are partially filled with lighter vapors. This results in vibration and shaft deflection, eventually resulting in bearing failures, packing or seal leakage, and shaft breakage. In multi-stage pumps this can cause loss of thrust balance and thrust bearing failures.

BUBBLE COLLAPSE PHASE

1. Mechanical damage occurs as the imploding bubbles remove segments of impeller material.
2. Noise and vibration result from the implosion. Noise that sounds like gravel being pumped is often the user's first warning of cavitation.

NET POSITIVE SUCTION HEAD

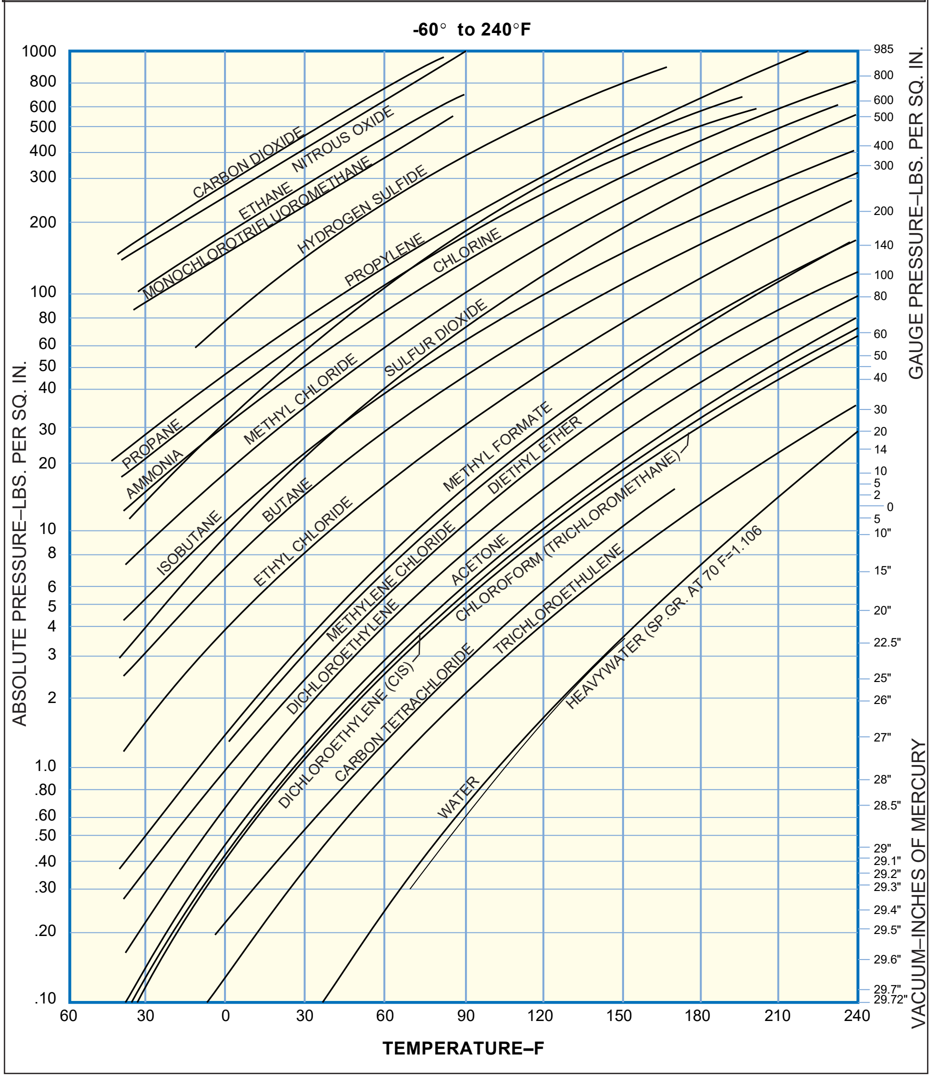
When designing a pumping system and selecting a pump, one must thoroughly evaluate net positive suction head (NPSH) to prevent cavitation. A proper analysis involves both the net positive suction heads available in the system ($NPSH_A$) and the net positive suction head required by the pump ($NPSH_R$).

$NPSH_A$ is the measurement or calculation of the absolute pressure above the vapor pressure at the pump suction flange. Figure 2 illustrates methods of calculating $NPSH_A$ for various suction systems. Since friction in the suction pipe is a common negative component of $NPSH_A$, the value of $NPSH_A$ will always decrease with flow.

$NPSH_A$ must be calculated to a stated reference elevation, such as the foundation on which the pump is to be mounted.

$NPSH_R$ is always referenced to the pump impeller center line.

FIGURE 1



Vapor pressures of various liquids related to temperature.

It is a measure of the pressure drop as the liquid travels from the pump suction flange along the inlet to the pump impeller. This loss is due primarily to friction and turbulence.

Turbulence loss is extremely high at low flow and then decreases with flow to the best efficiency point. Friction loss increases with increased flow. As a result, the internal pump losses will be high at low flow, dropping at generally 20–30% of the best efficiency flow, then increasing with flow. The complex subject of turbulence and $NPSH_R$ at low flow is best left to another discussion.

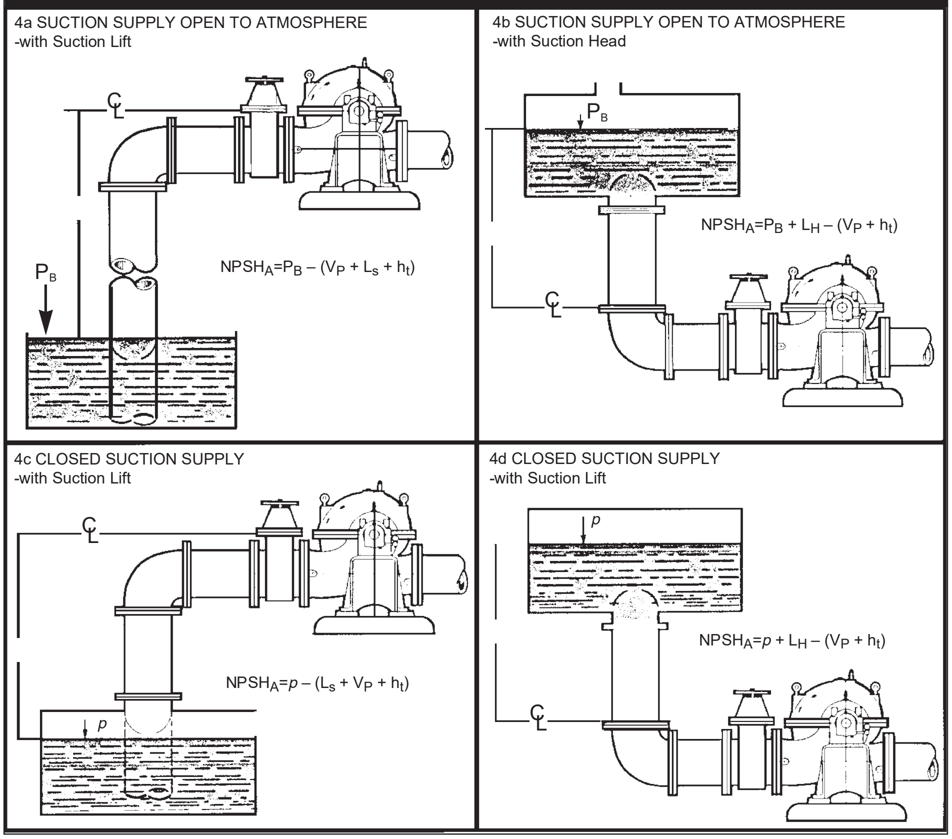
Figure 3 shows the pressure profile across a typical pump at a fixed flow condition. The pressure decrease from point B to point D is the $NPSH_R$ for the pump at the stated flow.

The pump manufacturer determines the actual $NPSH_R$ for each pump over its complete operating range by a series of tests. The detailed test procedure is described in the *Hydraulic Institute Test Standard 1988 Centrifugal Pumps 1.6.* Industry has agreed on a 3% head reduction at constant flow as the standard value to establish $NPSH_R$. Figure 4 shows typical results of a series of $NPSH_R$ tests.

The pump system designer must understand that the published $NPSH_R$ data established above are based on a 3% head reduction. Under these conditions the pump is cavitating. At the normal operating point the $NPSH_A$ must exceed the $NPSH_R$ by a sufficient margin to eliminate the 3% head drop and the resulting cavitation.

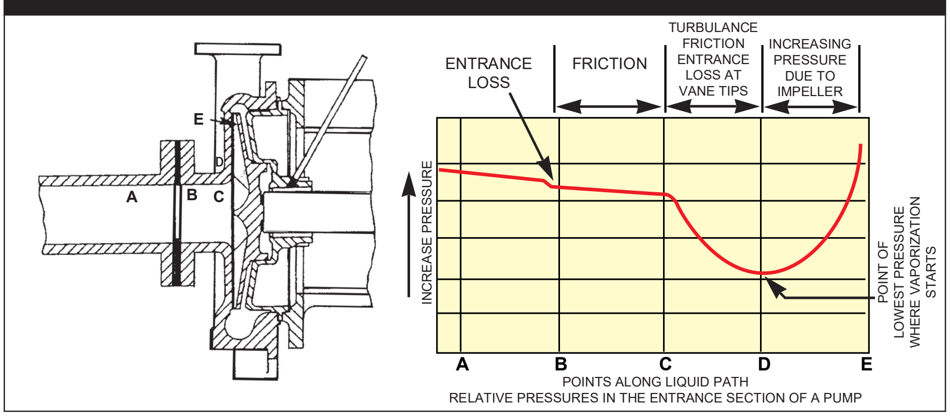
The $NPSH_A$ margin required will vary with pump design and other factors, and the exact margin cannot be precisely predicted. For most applications the $NPSH_A$ will exceed the $NPSH_R$ by a significant amount, and the NPSH margin is not a consideration. For those applications where the $NPSH_A$ is close to the $NPSH_R$

FIGURE 2



Calculation of system net positive suction head available ($NPSH_A$) for typical suction conditions. P_B = barometric pressure in feet absolute, V_P = vapor pressure of the liquid at maximum pumping temperature in feet absolute, p = pressure on surface of liquid in closed suction tank in feet absolute, L_s = maximum suction lift in feet, L_H = minimum static suction head in feet, h_f = friction loss in feet in suction pipe at required capacity.

FIGURE 3



The pressure profile across a typical pump at a fixed flow condition.

(2–3 feet), users should consult the pump manufacturer and the two should agree on a suitable NPSH margin. In these deliberations, factors such as liquid characteristic, minimum and normal $NPSH_A$, and normal operating flow must be considered.

SUCTION SPECIFIC SPEED

The concept of suction specific speed (S_s) must be considered by the pump designer, pump application engineer, and the system designer to ensure a cavitation-free pump with high reliability and the ability to operate over a wide flow range.

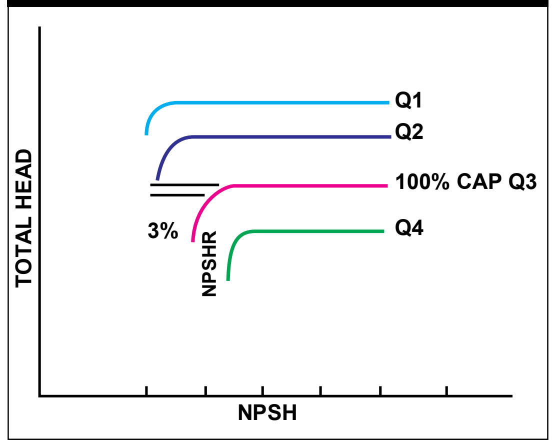
$$S_s = \frac{N \times Q^{0.5}}{(NPSH_R)^{0.75}}$$

where N = pump rpm

Q = flow rate in gpm at the best efficiency point

$NPSH_R$ = $NPSH_R$ at Q with the maximum impeller diameter

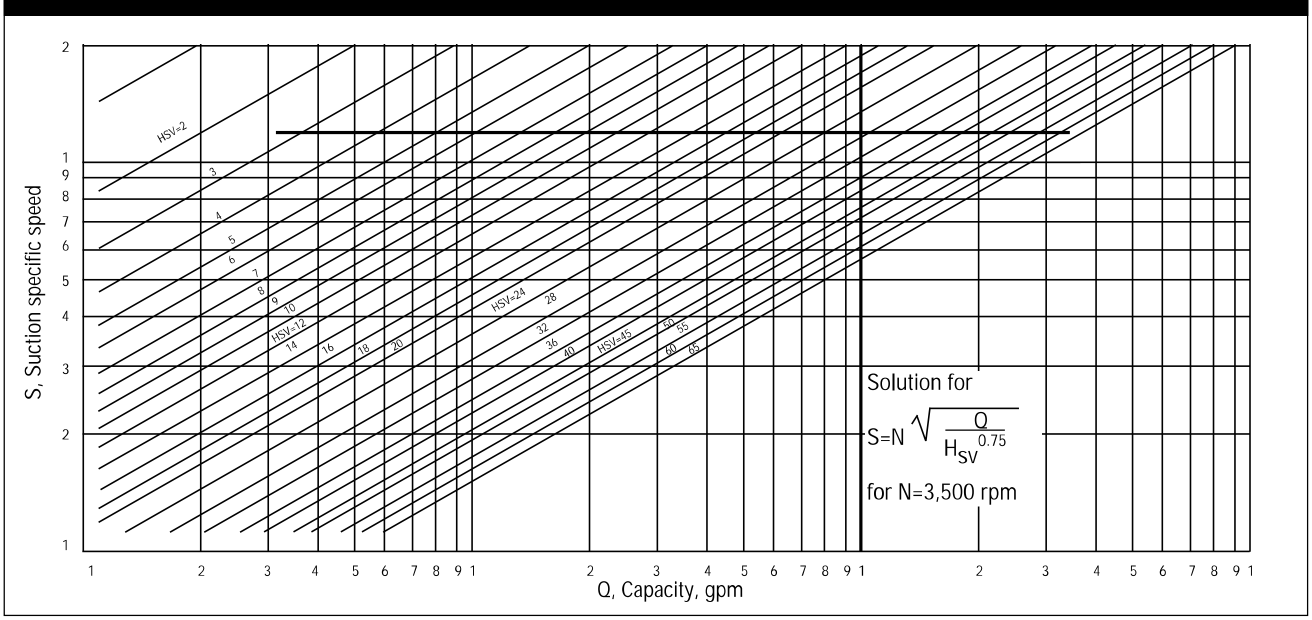
The system designer should also calculate the system suction specific speed by substituting design flow rate and the system designer's $NPSH_A$. The pump speed N is generally determined by the head or pressure required in the system. For a low-maintenance pump system, designers and most user specifications require, or prefer, S_s values below 10,000 to 12,000. However, as indicated above, the pump S_s is dictated to a great extent by the system conditions, design flow, head, and the $NPSH_A$.



Typical results of a four-point net positive suction head required ($NPSH_R$) test based on a 3% head drop.

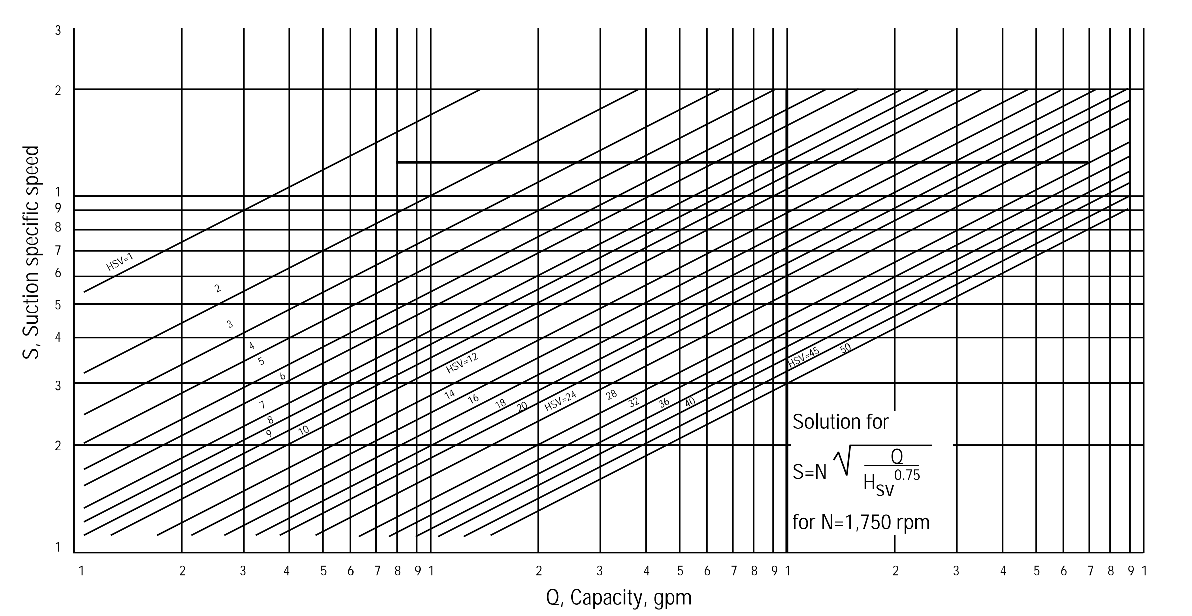
Figures 5 and 6 are plots of S_s versus flow in gpm for various $NPSH_A$ or $NPSH_R$ at 3,500 and 1,750 rpm. Similar plots can be made for other common pump speeds.

Using curves from Figure 5 and Figure 6 allows the system designer to design the system S_s, i.e., for a system requiring a 3,500 rpm pump with 20 feet of $NPSH_A$, the maximum flow must be limited to 1,000 gpm if the maximum S_s is to be maintained at 12,000. Various options are available, such as reducing the head to allow 1,750 rpm (Figure 7). This would allow flows to 4,000 gpm with 20 feet of $NPSH_A$.



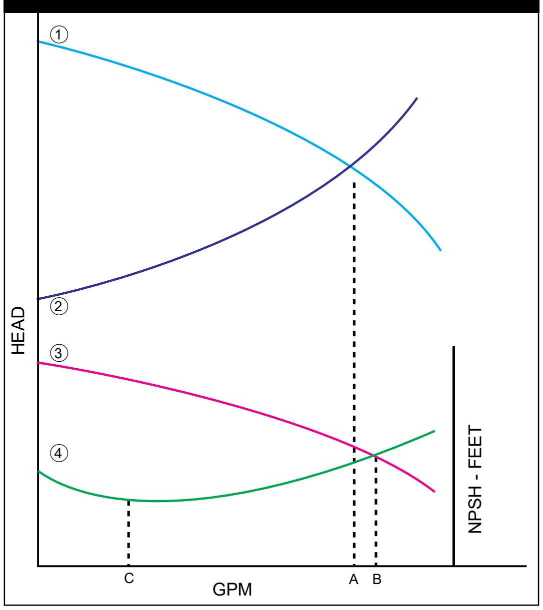
A plot of suction specific speed (S_s) versus flow in gallons per minute (gpm) for various $NPSH_A$ or $NPSH_R$ at 3,500 rpm. (Single suction pumps. For double suction use 1/2 capacity). H_{sv}= $NPSH_R$ at BEP with maximum impeller diameter.

FIGURE 6



A plot of suction specific speed (S_S) versus flow in gallons per minute (gpm) for various $NPSH_A$ or $NPSH_R$ at 1,750 rpm. (Single suction pumps. For double suction use 1/2 capacity.) H_{SV}= $NPSH_R$ at BEP with the maximum impeller diameter.

FIGURE 7



A typical plot of the suction and discharge systems. Curve 1 = pump head capacity performance, curve 2 = total system curve, curve 3 = suction system curve $NPSH_A$, and curve 4 = pump $NPSH_R$.

It is important for the pump user to understand how critical the system design requirements are to the selection of a reliable, trouble-free pump.

Matching the system and pump characteristics is a must. Frequently, more attention is paid to the discharge side. Yet it is well known that most pump performance problems are caused by problems on the suction side.

Figure 7 is a typical plot of the suction and discharge systems.

It is important that points A, B, and C be well established and understood. A is the normal operating point. B is the maximum flow for cavitation-free operation. C is the minimum stable flow, which is dictated by the suction specific speed.

As a general rule, the higher the suction specific speed, the higher the minimum stable flow capacity will be. If a pump is always operated at its best efficiency point, a high value of S_S will not create problems. However, if the pump is to be operated at reduced flow, then the S_S value must be given careful consideration. ■

REFERENCES

1. Goulds Pump Manual.
2. Durco Pump Engineering Manual.
3. Hydraulic Institute Test Standards—1988 Centrifugal Pumps 1.6.

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Pump Suction Conditions

BY ROSS C. MACKAY

If a wide receiver has the right speed and good hands, all that's needed from the quarterback is to throw the ball accurately, and the team will probably gain good yardage, maybe even a touchdown.

Believe it or not, much the same is true of a pump and its suction conditions. If it has the right speed and is the right size, all that's required from the quarterback is to deliver the liquid at the right pressure and with an even laminar flow into the eye of the impeller.

If the quarterback's pass is off target, badly timed, or the ball's turning end over end in the air, the receiver may not be able to hang on to it, and there's no gain on the play. When that happens, the quarterback knows he didn't throw it properly and doesn't blame the receiver. Unfortunately, that's where the comparison ends. The engineering "quarterbacks" tend to blame the pump even when its their delivery that's bad!

Just as there are techniques a quarterback must learn in order to throw accurately, there are rules which ensure that a liquid arrives at the impeller eye with the pressure and flow characteristics needed for reliable operation.

RULE #1.
PROVIDE SUFFICIENT NPSH

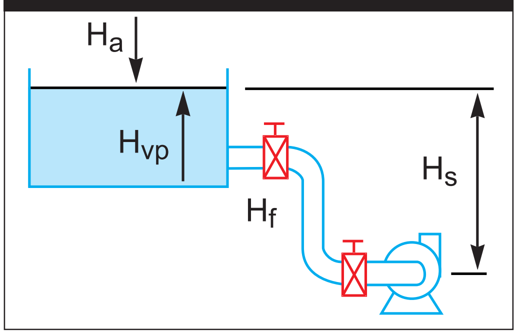
Without getting too complicated on a subject about which complete books have been written, let's just accept the premise that every impeller requires a minimum amount of pressure energy in the liquid being supplied in order to perform without cavitation difficulties. This pressure energy is referred to as Net Positive Suction Head Required.

The NPSH Available is supplied from the system. It is solely a function of the system design on the suction side of the pump. Consequently, it is in the control of the system designer.

To avoid cavitation, the NPSH available from the system must be greater than the NPSH required by the pump, and the biggest mistake that can be made by a system designer is to succumb to the temptation to provide only the minimum required at the rated design point. This leaves no margin for error on the part of the designer, or the pump, or the system. Giving in to this temptation has proved to be a costly mistake on many occasions.

In the simple system as shown in Figure 1, the NPSH Available can be calculated as follows:

FIGURE 1



$$NPSH_A = H_a + H_s - H_{vp} - H_f$$

where

H_a= the head on the surface of the liquid in the tank. In an open system like this, it will be atmospheric pressure.

H_s= the vertical distance of the free surface of the liquid above the center line of the pump impeller. If the liquid is below the pump, this becomes a negative value.

H_{vp}= the vapor pressure of the liquid at the pumping temperature, expressed in feet of head.

H_f= the friction losses in the suction piping.

The NPSH Available may also be determined with this equation:

$$NPSH_A = H_a + H_g + V^2/2g - H_{vp}$$

where

H_a= atmospheric pressure in feet of head.

H_g= the gauge pressure at the suction flange in feet of head.

$\frac{V^2}{2g}$= The velocity head at the point of measurement of Hg. (Gauge readings do not include velocity head.)

RULE #2.
REDUCE THE FRICTION LOSSES

When a pump is taking its suction from a tank, it should be located as close to the tank as possible in order to reduce the effect of friction losses on the NPSH Available. Yet the pump must be far enough away from the tank to ensure that correct piping practice can be followed. Pipe friction can usually be reduced by using a larger diameter line to limit the linear velocity to a level appropriate to the particular liquid being pumped. Many industries work with a maximum velocity of about 5ft./sec., but this is not always acceptable.

RULE #3.
NO ELBOWS ON THE SUCTION FLANGE

Much discussion has taken place over the acceptable configuration of an elbow on the suction flange of a pump. Let's simplify it. There isn't one!

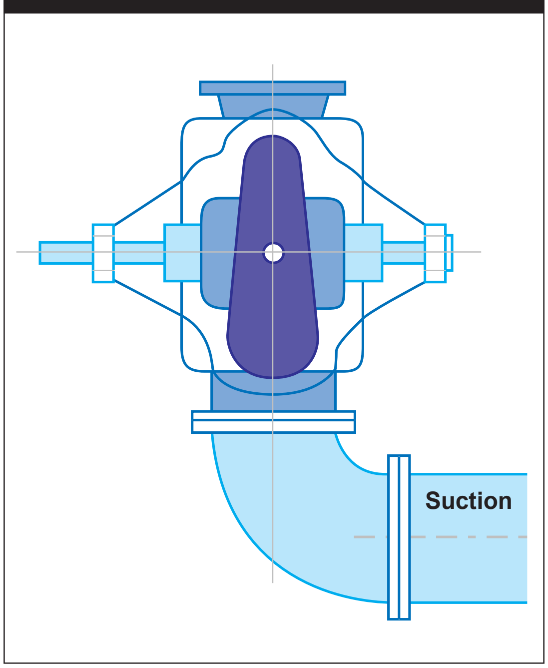
There is always an uneven flow in an elbow, and when one is installed on the suction of any pump, it introduces that uneven flow into the eye of the impeller. This can create turbulence and air entrainment, which may result in impeller damage and vibration.

When the elbow is installed in a horizontal plane on the inlet of a double suction pump, uneven flows are introduced into the opposing eyes of the impeller, upsetting the hydraulic balance of the rotating element. Under these conditions the overloaded bearing will fail prematurely and regularly if the pump is packed. If the pump is fitted with mechanical seals, the seal will usually fail instead of the bearing-but just as regularly and often more frequently.

The only thing worse than one elbow on the suction of a pump is two elbows on the suction of a pump— particularly if they are positioned in planes at right angles to each other. This creates a spinning effect in the liquid which is carried into the impeller and causes turbulence, inefficiency and vibration.

A well established and effective method of ensuring a laminar flow to the eye of the impeller is to provide the suction of the pump with a straight run of pipe in a length equivalent to 5-10 times the diameter of that pipe. The smaller multiplier would be used on the larger pipe diameters and vice versa.

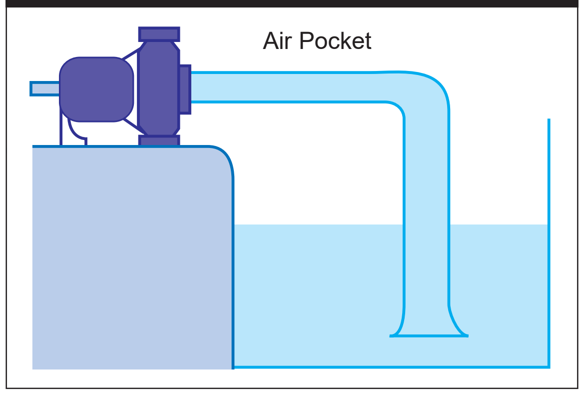
FIGURE 2



RULE #4. STOP AIR OR VAPOR ENTERING THE SUCTION LINE

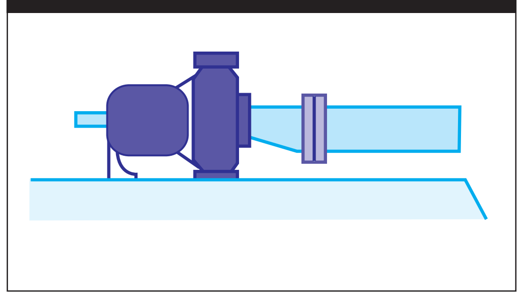
Any high spot in the suction line can become filled with air or vapor which, if transported into the impeller, will create an effect similar to cavitation and with the same results. Services which are particularly susceptible to this situation are those where the pumpage contains a significant amount of entrained air or vapor, as well as those operating on a suction lift, where it can also cause the pump to lose its prime. (Figure 3)

FIGURE 3



A similar effect can be caused by a concentric reducer. The suction of a pump should be fitted with an eccentric reducer positioned with the flat side uppermost. (Figure 4).

FIGURE 4



If a pump is taking its suction from a sump or tank, the formation of vortices can draw air into the suction line. This can usually be prevented by providing sufficient submergence of liquid over the suction opening. A bell-mouth design on the opening will reduce the amount of submergence required. This submergence is completely independent of the NPSH required by the pump.

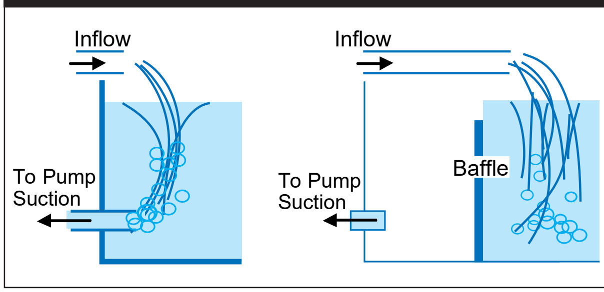
It is worthwhile noting that these vortices are more difficult to troubleshoot in a closed tank simply because they can't be seen as easily.

Great care should be taken in designing a sump to ensure that any liquid emptying into it does so in such a way that air entrained in the inflow does not pass into the suction opening. Any problem of this nature may require a change in the relative positions of the inflow and outlet if the sump is large enough, or the use of baffles. (Figure 5)

RULE #5. CORRECT PIPING ALIGNMENT

Piping flanges must be accurately aligned before the bolts are tightened and all piping, valves and associated fittings should be independently supported, so as to place no strain on the pump. Stress imposed on the pump casing by the piping reduces the probability of satisfactory performance.

FIGURE 5



Under certain conditions the pump manufacturer may identify some maximum levels of forces and moments which may be acceptable on the pump flanges.

In high temperature applications, some piping misalignment is inevitable owing to thermal growth during the operating cycle. Under these conditions, thermal expansion joints are often introduced to avoid transmitting piping strains to the pump. However, if the end of the expansion joint closest to the pump is not anchored securely, the object of the exercise is defeated as the piping strains are simply passed through to the pump.

RULE #6. WHEN RULES 1 TO 5 HAVE BEEN IGNORED, FOLLOW RULES 1 TO 5.

Piping design is one area where the basic principles in-volved are regularly ignored, resulting in hydraulic instabilities in the impeller which translate into additional shaft loading, higher vibration levels and premature failure of the seal or bearings. Because there are many other reasons why pumps could vibrate, and why seals and bearings fail, the trouble is rarely traced to incorrect piping.

It has been argued that because many pumps are piped incorrectly and most of them are operating quite satisfactorily, piping procedure is not important. Unfortunately, satisfactory operation is a relative term, and what may be acceptable in one plant may be inappropriate in another.

Even when "satisfactory" pump operation is obtained, that doesn't automatically make a questionable piping practice correct. It merely makes it lucky.

The suction side of a pump is much more important than the piping on the discharge. If any mistakes are made on the discharge side, they can usually be compensated for by increasing the performance capability from the pump. Problems on the suction side, however, can be the source of ongoing and expensive difficulties which may never be traced back to that area.

In other words, if your receivers aren't performing well, is it their fault? Or does the quarterback need more training? ■

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Elements of Minimum Flow

BY TERRY M. WOLD

Minimum flow can be determined by examining each of the factors that affect it. There are five elements that can be quantified and evaluated:

1. Temperature rise (minimum thermal flow)
2. Minimum stable flow
3. Thrust capacity
4. NPSH requirements
5. Recirculation

The highest flow calculated using these parameters is considered the minimum flow.

TEMPERATURE RISE

Temperature rise comes from energy imparted to the liquid through hydraulic and mechanical losses within the pump. These losses are converted to heat, which can be assumed to be entirely absorbed by the liquid pumped. Based on this assumption, temperature rise ΔT in °F is expressed as:

$$\Delta T = \frac{H}{778 \times C_p} \times \frac{1}{\eta - 1}$$

where

H = total head in feet

C_p = specific heat of the liquid, Btu/lb x °F

η = pump efficiency in decimal form

778 ft–lbs = energy to raise the temperature of one pound of water 1°F

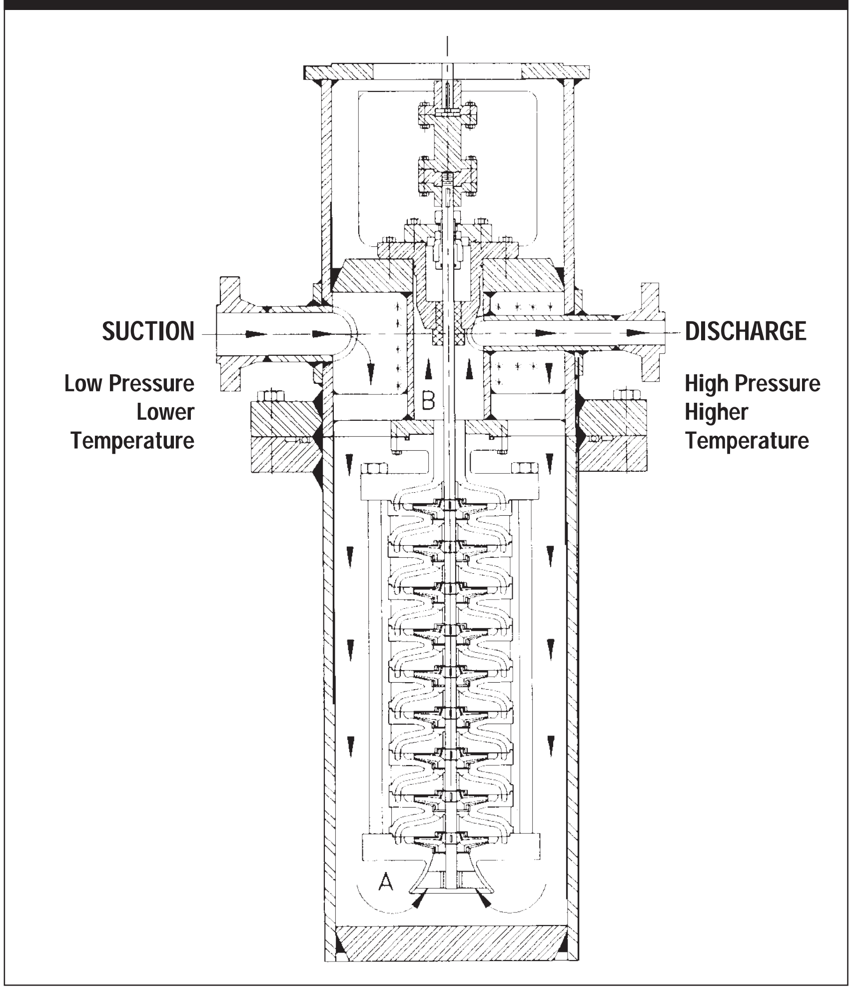
To calculate this, the specific heat and allowable temperature rise must be known.

The specific heat for water is 1.0, and other specific heats can be as low as 0.5. The specific heats for a number of liquids can be found in many chemical and mechanical handbooks.

What is the maximum allowable temperature rise? Pump manufacturers usually limit it to 15°F. However, this can be disastrous in certain situations. A comparison of the vapor pressure to the lowest expected suction pressure plus NPSH required ($NPSH_R$) by the pump must be made. The temperature where the vapor pressure equals the suction pressure plus the $NPSH_R$ is the maximum allowable temperature. The difference between the allowable temperature and the temperature at the pump inlet is the maximum allowable temperature rise. Knowing ΔT and C_p, the minimum flow can be determined by finding the corresponding head and efficiency.

When calculating the maximum allowable temperature rise, look at the pump geometry. For instance, examine the vertical can

FIGURE 1



A high-pressure vertical pump. Asterisks (*) denote where low-temperature fluid is exposed to higher temperatures. Flashing and vaporization can occur here. Temperature increases as fluid travels from A towards B.

pump in Figure 1. Although pressure increases as the fluid is pumped upward through the stages, consider the pump inlet. The fluid at the inlet (low pressure, low temperature) is exposed to the temperature of the fluid in the discharge riser in the head (higher pressure, higher temperature). This means that the vapor pressure of the fluid at the pump inlet must be high enough to accommodate the total temperature rise through all the stages. If this condition is discovered during the pump design phase, a thermal barrier can be designed to reduce the temperature that the inlet fluid is exposed to.

Some books, such as the *Pump Handbook* (Ref. 5), contain a typical chart based on water ($C_p = 1.0$) that can be used with the manufacturer's performance curve to determine temperature rise. If the maximum allowable temperature rise exceeds the previously determined allowable temperature rise, a heat shield can be designed and fitted to the pump during the design stage. This requirement must be recognized during the design stage, because once the pump is built, options for retrofitting the pump with a heat shield are greatly reduced.

MINIMUM STABLE FLOW

Minimum stable flow can be defined as the flow corresponding to the head that equals shutoff head. In other words, outside the "droop" in the head capacity curve. In general, pumps with a specific speed less than 1,000 that are designed for optimum efficiency have a drooping curve. Getting rid of this "hump" requires an impeller redesign; however, note that there will be a loss of efficiency and an increase in $NPSH_R$.

What's wrong with a drooping head/capacity curve? A drooping curve has corresponding heads for two different flows. The pump reacts to the system requirements, and there are two flows where the pump can meet the system requirements. As a result, it "hunts" or "shuttles" between these two flows. This can damage the pump and other equipment, but it will happen only under certain circumstances:

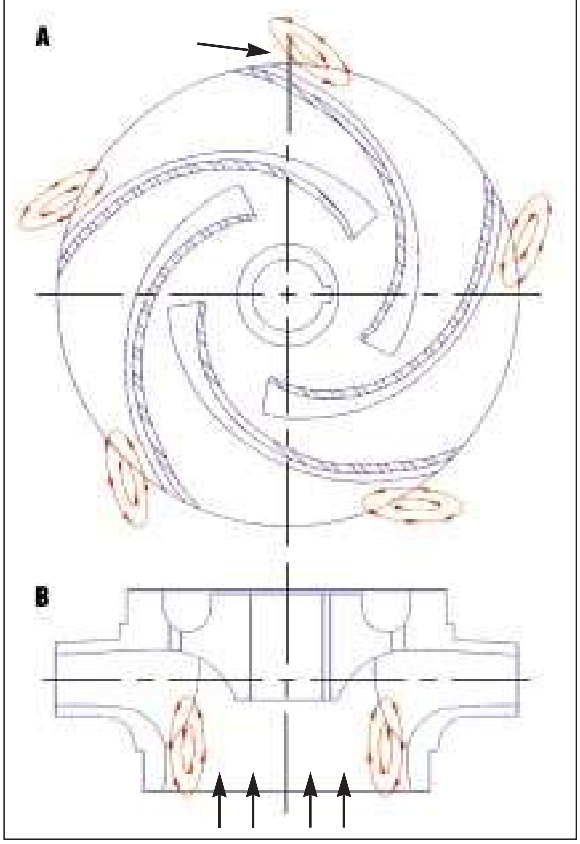
1. The liquid pumped must be uninhibited at both the suction and discharge vessels.
2. One element in the system must be able to store and return energy, i.e., a water column or trapped gas.
3. Something must upset the system to make it start hunting, i.e., starting another pump in parallel or throttling a valve.

Note: All of these must be present at the same time to cause the pump to hunt.

Minimum flow based on the shape of the performance curve is not so much a function of the pump as it is a function of the system where the pump is placed. A pump in a system where the above criteria are present should not have a drooping curve in the zone of operation.

Because pumps with a drooping head/capacity curve have higher efficiency and a lower operating cost, it would seem prudent to investigate the installation of a minimum flow bypass.

FIGURE 2



Recirculation zones are always on the pressure side of the vane. A shows discharge recirculation (the front shroud has been left out for clarity). B shows inlet recirculation.

THRUST LOADING

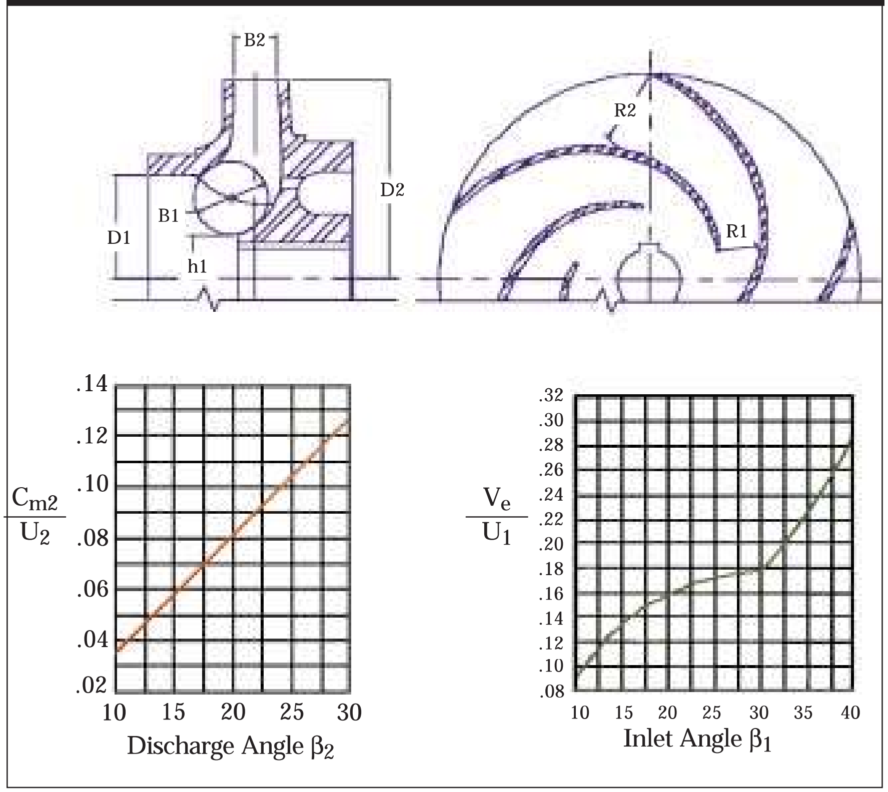
Axial thrust in a vertical turbine pump increases rapidly as flows are reduced and head increased. Based on the limitations of the driver bearings, flow must be maintained at a value where thrust developed by the pump does not impair bearing life. Find out what your bearing life is and ask the pump manufacturer to give specific thrust values based on actual tests.

If a problem exists that cannot be handled by the driver bearings, contact the pump manufacturer. There are many designs available today for vertical pumps (both single and multistage) with integral bearings. These bearings can be sized to handle the thrust. Thrust can be balanced by the use of balanced and unbalanced stages or adding a balance drum, if necessary. These techniques for thrust balancing are used when high thrust motors are not available. It is worth noting that balanced stages incorporate wear rings and balance holes to achieve lower thrust; therefore, a slight reduction in pump efficiency can be expected, and energy costs become a factor.

NPSH REQUIREMENTS

How many pumps have been oversized because of NPSH available ($NPSH_A$)? It seems the easiest solution to an NPSH problem is to go to the next size pump with a larger suc-

FIGURE 3



Incipient recirculation. Minimum flow is approximately 50% of incipient flow, while minimum intermittent flow is approximately 25% of incipient flow. See text under "Recirculation Calculations" for details

tion, thereby reducing the inlet losses. A couple of factors become entangled when this is done. A larger pump means operating back on the pump curve. Minimum flow must be considered. Is the curve stable? What about temperature rise? If there is already an NPSH problem, an extra few degrees of temperature rise will not help the situation. The thrust and eye diameter will increase, possibly causing damaging recirculation. When trying to solve an NPSH problem, don't take the easiest way out. Look at other options that may solve a long-term problem and reduce operating costs.

RECIRCULATION

Every pump has a point where recirculation begins. But if this is the case, why don't more pumps have problems?

Recirculation is caused by oversized flow channels that allow liquid to turn around or reverse flow while pumping is going on (Figure 2 shows recirculation zones). This reversal causes a vortex that attaches itself to the pressure side of the vane. If there is enough energy available and the velocities are high enough, damage will occur. Suction recirculation is reduced by lowering the peripheral velocity, which in turn increases NPSH. To avoid this it is better to recognize the problem in the design stage and opt for a lower-speed pump, two smaller pumps, or an increase in $NPSH_A$.

Discharge recirculation is caused by flow reversal and high velocities producing damaging vortices on the pressure side of the vane at the outlet (Figure 2). The solution to this problem lies in the impeller design. The problem is the result of a mismatched case and impeller, too little vane overlap in the impeller design, or trimming the impeller below the minimum diameter for which it was designed.

Recirculation is one of the most difficult problems to understand and document. Many studies on the topic have been done over the years. Mr. Fraser's paper (Ref. 1) is one of the most useful tools for determining where recirculation begins. In it he describes how to calculate the inception of recirculation based on specific design characteristics of the impeller and he includes charts that can be used with a minimum amount of information. An example of Fraser calculations, which show the requirements to calculate the inception of suction and discharge recirculation, is shown in Figure 3.

RECIRCULATION CALCULATIONS

Figure 3 indicates the user-defined variables and charts required to make the Fraser calculations for minimum flow. Information to do the detailed calculations include:

- Q = capacity at the best efficiency point
- H = head at the best efficiency point
- $NPSH_R$ = net positive suction head required at the pump suction
- N = pump speed
- N_S = pump specific speed
- N_{SS} = suction specific speed
- Z = number of impeller vanes
- h_1 = hub diameter ($h_1 = 0$ for single suction pumps)
- D_1 = impeller eye diameter
- D_2 = impeller outside diameter
- B_1 = impeller inlet width
- B_2 = impeller outlet width
- R_1 = impeller inlet radius
- R_2 = impeller outlet radius
- F_1 = impeller inlet area
- F_2 = impeller outlet area
- β_1 = impeller inlet angle
- β_2 = impeller outlet angle

The above information is obtained from the pump manufacturer curves or impeller design files. The impeller design values are usually considered proprietary information.

K_{Ve} and K_{Cm2} can be determined from the charts in Figure 3.

With all of the above information at hand, suction recirculation and the two modes of discharge recirculation can be determined.

As previously mentioned, Fraser has some empirical charts at the end of his paper that can be used to estimate the minimum flow for recirculation. A few of the design factors of the impeller are still required. It is best to discuss recirculation with your pump manufacturer before purchasing a pump, in order to reduce the possibility of problems with your pump and system after installation and start-up.

SUMMARY

Minimum flow can be accurately determined if the elements described above are reviewed by the user and the manufacturer. Neither has all the information to determine a minimum flow that is economical, efficient, and insures a trouble-free pump life. It takes a coordinated effort by the user and the manufacturer to come up with an optimum system for pump selection, design, and installation.

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Effects of Oversizing

BY: IGOR J. KARASSIK

One of the greatest sources of power waste is the practice of oversizing a pump by selecting design conditions with excessive margins in both capacity and total head. It is strange on occasion to encounter a great deal of attention being paid to a one-point difference in efficiency between two pumps while at the same time potential power savings are ignored through an overly conservative attitude in selecting the required conditions of service.

POWER CONSUMPTION

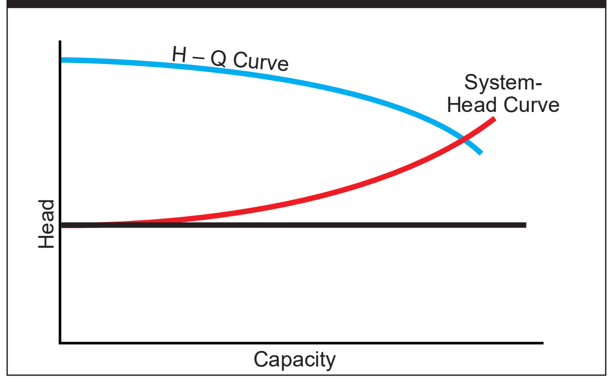
After all, we are not primarily interested in efficiency; we are more interested in power consumption. Pumps are designed to convert mechanical energy from a driver into energy within a liquid. This energy within the liquid is needed to overcome friction losses, static pressure differences and elevation differences at the desired flow rate. Efficiency is nothing but the ratio between the hydraulic energy utilized by the process and the energy input to the pump driver. And without changing the ratio itself, if we find that we are assigning more energy to the process than is really necessary, we can reduce this to correspond to the true requirement and therefore reduce the power consumption of the pump.

It is true that some capacity margin should always be included, mainly to reduce the wear of internal clearances which will, with time, reduce the effective pump capacity. How much margin to provide is a fairly complex question because the wear that will take place varies with the type of pump in question, the liquid handled, the severity of the service and a number of other variables.

A centrifugal pump operating in a given system will deliver a capacity corresponding to the intersection of its head-capacity curve with the system-head curve, as long as the available NPSH is equal to or exceeds the required NPSH (Figure 1). To change this operating point in an existing installation requires changing either the head-capacity curve or the system-head curve, or both. The first can be accomplished by varying the speed of the pump (Figure 2), or its impeller diameter while the second requires altering the friction losses by throttling a valve in the pump discharge (Figure 3). In the majority of pump installations, the driver is a constant speed motor, and changing the system-head curve is used to change the pump capacity. Thus, if we have provided too much excess margin in the selection of the pump head-capacity curve, the pump will have to operate with considerable throttling to limit its delivery to the desired value.

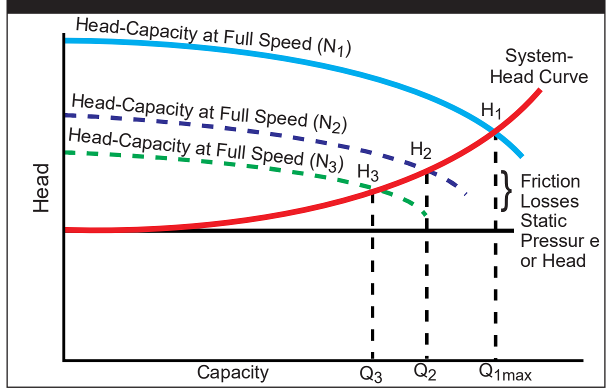
If, on the other hand, we permit the pump to operate unthrottled, which is more likely, the flow into the system will increase until that capacity is reached where

FIGURE 1



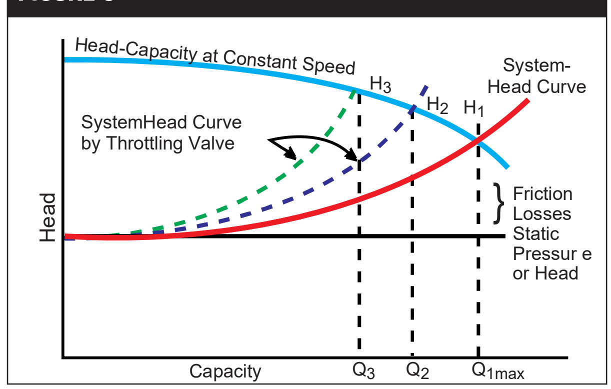
Pump H-Q curve superimposed on system-head curve

FIGURE 2



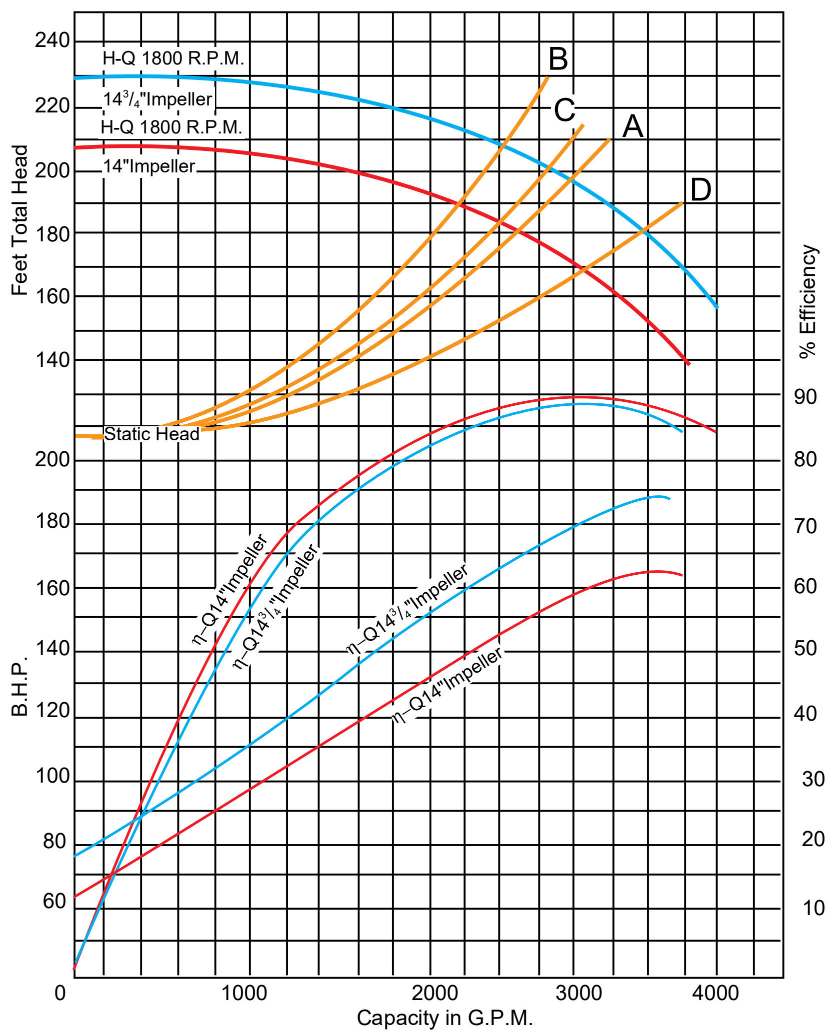
Varying pump capacity by varying speed

FIGURE 3



Varying pump capacity by throttling

FIGURE 4



Effect of oversizing a pump

the system-head and head-capacity curves intersect.

EXAMPLE

Let's use a concrete example, for which the maximum required capacity is 2700 gpm, the static head is 115 ft and the total friction losses, assuming 15-year old pipe, are 60 ft. The total head required at 2700 gpm is therefore 175 ft. We can now construct a system-head curve, which is shown on curve A, Figure 4. If we add a margin of about 10% to the required capacity and then, as is frequently done, we add some margin to the total head above the system-head curve at this rated flow, we end up by selecting a pump for 3000 gpm and 200 ft. total head. The performance of such a pump, with a 14-3/4 in. impeller, is superimposed on the system-head curve A in Figure 4.

The pump develops excess head at the maximum required capacity of 2700 gpm, and if we wish to operate at that capacity, this excess head will have to be throttled. Curve "B" is the system-head curve that will have to be created by throttling.

If we operate at 3000 gpm, the pump will take 175 bhp, and we will have to drive it with a 200 hp motor. If we operate it throttled at the required capacity of 2700 gpm, operating at the intersection of its head-capacity curve and curve B, the pump will require 165 bhp.

The pump has been selected with too much margin. We can safely select a pump with a smaller impeller diameter, say 14 in., with a head-capacity curve as shown on Figure 4. It will intersect curve A at 2820 gpm, giving us about 4% margin in capacity, which is sufficient. To limit the flow to 2700 gpm, we will still have to throttle the pump slightly and our system head curve will become curve C. The power consumption at 2700 gpm will now be only 145 bhp instead of the 165 bhp required with our first overly conservative selection. This is a very respectable 12% saving in power consumption. Furthermore, we no longer need a 200 hp motor. A 150 hp motor will do quite well. The saving in capital expenditure is another bonus resulting from correct sizing.

Our savings may actually be even greater. In many cases, the pump may be operated unthrottled, the capacity being permitted to run out to the intersection of the head-capacity curve and curve A. If this were the case, a pump with a 14-3/4 in. impeller would operate at approximately 3150 gpm and take 177 bhp. If a 14 in. impeller were to be used, the pump would operate at 2820 gpm and take 148 bhp. We could be saving more than 15% in power consumption. Tables 1 and 2 tabulate these savings.

And our real margin of safety is actually even greater than I have indicated. Remember that the friction losses we used to construct the system-head curve A were based on losses through 15-year old piping. The losses through new piping are only 0.613 times the losses we have assumed. The system-head curve for new piping is that indicated as curve D in Figure 4. If the pump we had originally selected (with a 14-3/4 in. impeller) were to operate unthrottled, it would run at 3600 gpm and take

TABLE 1. COMPARISON OF PUMPS WITH 14 3/4 IN. AND 14IN. IMPELLERS, WITH THE SYSTEM THROTTLED		
Throttled to 2700 GPM		
Impeller	14 3/4"	14"
Curve	"B"	"C"
BHP	165	145
Savings		20 hp or 12.1%

TABLE 2. COMPARISON OF PUMPS WITH THE SYSTEM UNTHROTTLED		
Unthrottled, on Curve "A"		
Impeller	14 3/4"	14"
GPM	3150	2820
BHP	177	148
Savings		29 hp or 16.4%

TABLE 3. EFFECT OF DIFFERENT SIZE IMPELLERS IN SYSTEM WITH NEW PIPE AND RESULTING SAVINGS NEW PIPE (UNTHROTTLED OPERATION, CURVE "D")			
Impeller	14 3/4"	14"	13 3/4"
GPM	3600	3230	3100
BHP	187.5	156.5	147
Savings		31 hp 16.5%	40.5 hp 21.6%

187.5 bhp. A pump with only a 14 in. impeller would intersect the system-head curve D at 3230 gpm and take 156.6 bhp, with a saving of 16.5%. As a matter of fact, we could even use a 13-3/4 in. impeller. The head-capacity curve would intersect curve D at 3100 gpm, and the pump would take 147 bhp. Now, the savings over using a 14-3/4 in. impeller becomes 21.6% (See Table 3).

Clearly, important energy savings can be achieved if, at the time of the selection of the conditions of service, reasonable restraints are exercised to avoid incorporating excessive safety margins into the rated conditions of service.

EXISTING INSTALLATIONS

But what of existing installations in which the pump or pumps have excessive margins? Is it too late to achieve these savings? Far from it! As a matter of fact, it is possible to establish more accurately the true system-head curve by running a performance test once the pump has been installed and operated. A reasonable margin can then be selected and several choices become available to the user:

1. The existing impeller can be cut down to meet the more realistic conditions of service.
2. A replacement impeller with the necessary reduced diameter can be ordered from the pump manufacturer. The original impeller is then stored for future use if friction losses are ultimately increased with time or if greater capacities are ever required.
3. In certain cases, there may be two separate impeller designs available for the same pump, one of which is of narrower width than the one originally furnished. A replacement narrow impeller can then be ordered from the manufacturer. Such a narrower impeller will have its best efficiency at a lower capacity than the normal width impeller. It may or may not need to be of a smaller diameter than the original impeller, depending on the degree to which excessive margin had originally been provided. Again, the original impeller is put away for possible future use. ■

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Fluid Viscosity Effects on Centrifugal Pumps

BY: GUNNAR HOLE

When sizing a pump for a new application or evaluating the performance of an existing pump, it is often necessary to account for the effect of the pumped fluid's viscosity. We are all aware that the head-capacity curves presented in pump vendor catalogs are prepared using water as the pumped fluid. These curves are adequate for use when the actual fluid that we are interested in pumping has a viscosity that is less than or equal to that of water. However, in some cases—certain crude oils, for example—this is not the case.

Heavy crude oils can have viscosities high enough to increase the friction drag on a pump's impellers significantly. The additional horsepower required to overcome this drag reduces the pump's efficiency. There are several analytical and empirical approaches available to estimate the magnitude of this effect. Some of these are discussed below.

Before beginning the discussion, however, it is vital to emphasize the importance of having an accurate viscosity number on which to base our estimates. The viscosity of most liquids is strongly influenced by temperature. This relationship is most often shown by plotting two points on a semi-logarithmic grid and connecting them with a straight line. The relationship is of the form:

$$\mu = Ae^{B/T}$$

where

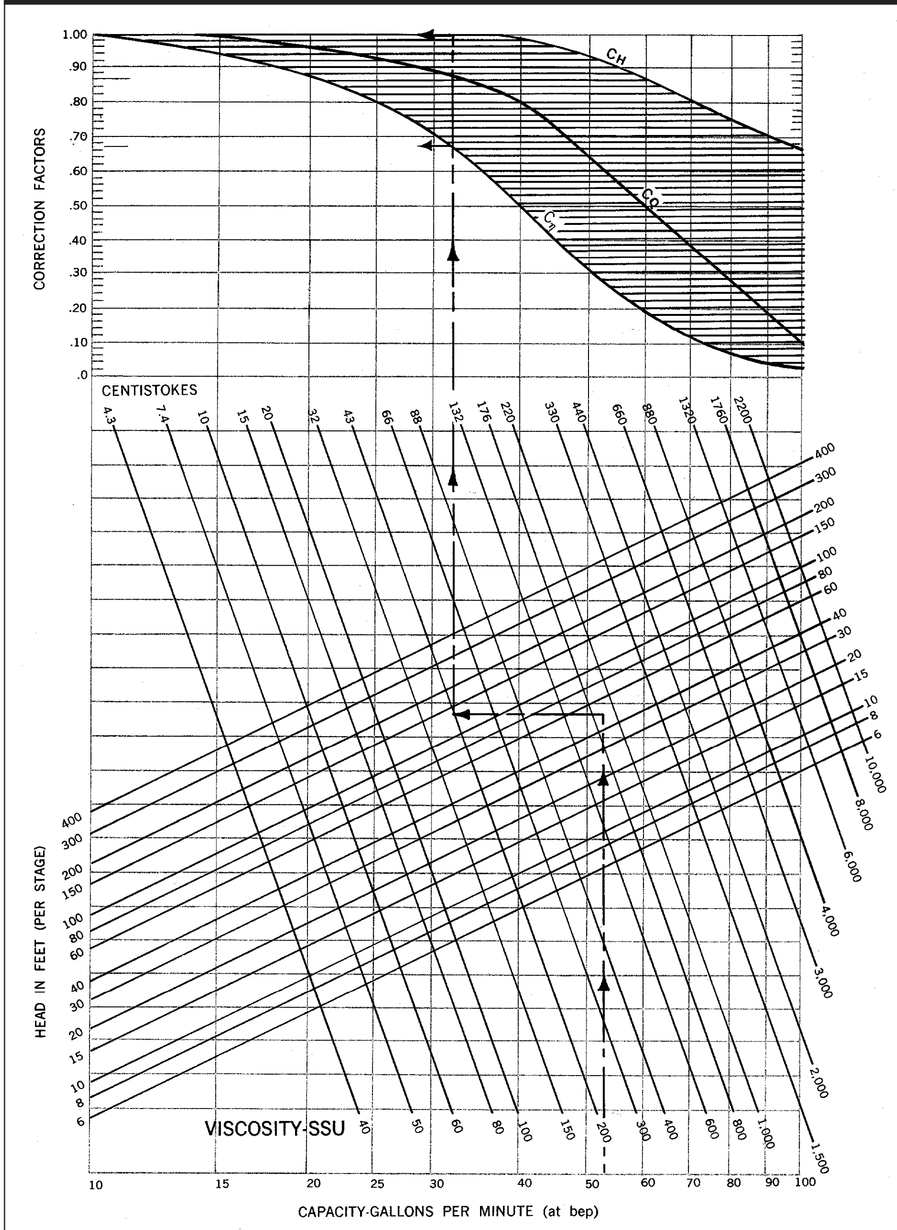
μ = the absolute viscosity of the fluid

A and B = constants

T = the absolute temperature of the fluid

Plotting this relationship requires knowledge of two data points, and using them effectively requires some judgement as to the normal operating temperature as well as the minimum temperature that might be expected during other off-design conditions such as start-up.

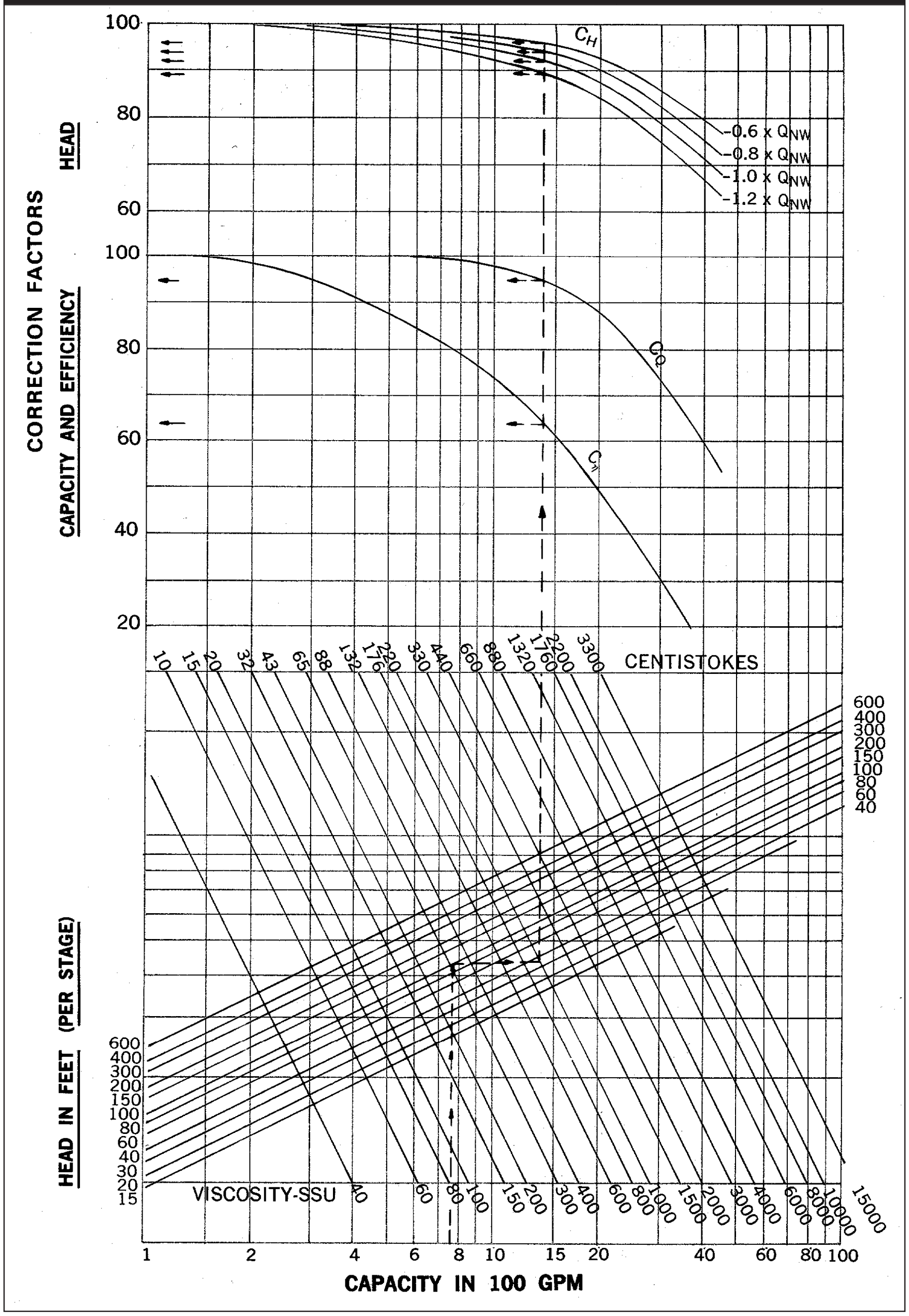
FIGURE 1



Reproduced from the Hydraulic Institute Standards (Figure 71)

The effect of pressure on the viscosity of most fluids is small. For mineral oils, for example, an increase of pressure of 33 bars (≈480 psi) is equivalent to a tem-

FIGURE 2



Reproduced from the Hydraulic Institute Standards (Figure 72)

perature drop of 1°C.

The following definitions are used when discussing fluids and viscosity. There are five basic types of liquid that can be differentiated on the basis of their viscous behavior; they are:

NEWTONIAN

These are fluids where viscosity is constant and independent of shear rate, and where the shear rate is linearly proportional to the shear stress. Examples are water and oil.

NON-NEWTONIAN

These are fluids where the shear rate-shear stress relationship is nonlinear. They can be divided into four categories:

- Bingham-plastic fluids are those in which there is no flow until a threshold shear stress is reached. Beyond this point, viscosity decreases with increasing shear rate. Most slurries have this property, as does America's favorite vegetable, catsup.
- Dilatant fluids are those of which viscosity increases with increasing shear rate. Examples are candy mixtures, clay slurries, and quicksand.
- Pseudo-plastic fluids are similar to Bingham-plastic fluids, except there is no definite yield stress. Many emulsions fall into this category.
- Thixotropic fluids are those of which viscosity decreases to a minimum level as their shear rate increases. Their viscosity at any particular shear rate may vary, depending on the previous condition of the fluid. Examples are asphalt, paint, molasses, and drilling mud.

There are two other terms with which you should be familiar:

- Dynamic or absolute viscosity is usually measured in terms of centipoise and has the units of force time/length^2.
- Kinematic viscosity is usually measured in terms of centistokes or ssu (Saybolt Seconds Universal). It is related to absolute viscosity as follows:

kinematic viscosity =
absolute viscosity/mass density

The normal practice is for this term to have the units of length^2/time. Note:

1 cSt = cP x sp gr

1 cSt = 0.22 x ssu – (180/ssu)

1 cP = 1.45E^{-7} lbf – s/in^2

1 Reyn = 1 lbf – s/in^2

TABLE 1. WATER-BASED AND VISCOUS PERFORMANCE

Water Curve-Based Performance	% of BEP Capacity			
	60%	80%	100%	120%
Capacity, gpm	450	600	750	900
Differential Head, ft.	120	115	100	100
Efficiency	0.70	0.75	0.81	0.75
Horsepower	18	21	21	27
Viscous (1,000 ssu) Performance				
Capacity, gpm	423	564	705	846
Differential Head, ft.	115	108	92	89
Efficiency	0.45	0.48	0.52	0.48
Horsepower	25	29	28	36

Note: Pumped fluid specific gravity = 0.9

The process of determining the effect of a fluid's viscosity on an operating pump has been studied for a number of years. In the book *Centrifugal and Axial Flow Pumps*, A.J. Stepanoff lists the losses that affect the performance of pumps as being of the following types:

- mechanical losses
- impeller losses
- leakage losses
- disk friction losses

Of all external mechanical losses, disk friction is by far the most important, according to Stepanoff. This is particularly true for pumps designed with low specific speeds. Stepanoff gives a brief discussion of the physics of a rotating impeller and emerges with a simple equation that summarizes the drag force acting upon it:

$$(hp)_d = Kn^3D^5$$

where

K = a real constant

n = the pump operating speed

D = the impeller diameter

The explanation further describes the motion of fluid in the immediate neighborhood of the spinning impeller. There Stepanoff mentions the experimental results of others demonstrating that, by reducing the clearance between the stationary casing and the impeller, the required power can be reduced. He also writes about the details of some investigations that demonstrate the beneficial effect of good surface finishes on both the stationary and rotating surfaces. Included is a chart prepared by Pfleiderer, based on work by Zumbusch and Schultz-Grunow, that gives friction coefficients for calculating disk friction losses. The chart is used in conjunction with the following equation:

$$(hp)_d = KD^2\gamma u^3$$

where

K = a constant based on the Reynolds number

D = impeller diameter

γ = fluid density

u = impeller tip speed

Like most of Stepanoff's writing, this presentation contains great depth with considerable rigor. It makes interesting reading if you are willing to put forth the time. Those of us who need a quick answer to a particular problem may need to look elsewhere for help.

In the book, *Centrifugal Pumps*, V. Lobanoff and R. Ross discuss the effect of viscous fluids on the performance of centrifugal pumps. They make the point that because the internal flow passages in small pumps are proportionally larger than those in larger pumps, the smaller pumps will always be more sensitive to the effects of viscous fluids. They also introduce a diagram from the paper "Engineering and System Design Considerations for Pump Systems and Viscous Service," by C.E. Petersen, presented at Pacific Energy Association, October 15, 1982. In this diagram, it is recommended that the maximum fluid viscosity a pump should be allowed to handle be limited by the pump's discharge nozzle size. The relationship is approximately:

$$viscosity_{max} = 300(D_{outlet\ nozzle} - 1)$$

where

viscosity is given in terms of ssu

D is measured in inches

With respect to the prediction of the effects of viscous liquids on the performance of centrifugal pumps, Lobanoff and Ross direct the reader to the clearly defined methodology of the *Hydraulic Institute Standards*. This technique is based on the use of two nomograms on pages 112 and 113 of the 14th edition (Figures 71 and 72). They are reproduced here as Figures 1 and 2. They are intended

TABLE 2. POLYNOMIAL COEFFICIENTS

Correction Factor						
	D_{x1}	D_{x2}	D_{x3}	D_{x4}	D_{x5}	D_{x6}
C_η	1.0522	-3.5120E-02	-9.0394E-04	2.2218E-04	-1.1986E-05	1.9895E-07
C_Q	0.9873	9.0190E-03	-1.6233E-03	7.7233E-05	-2.0528E-06	2.1009E-08
$C_{H0.6}$	1.0103	-4.6061E-03	2.4091E-04	-1.6912E-05	3.2459E-07	-1.6611E-09
$C_{H0.8}$	1.0167	-8.3641E-03	5.1288E-04	-2.9941E-05	6.1644E-07	-4.0487E-09
$C_{H1.0}$	1.0045	-2.6640E-03	-6.8292E-04	4.9706E-05	-1.6522E-06	1.9172E-08
$C_{H1.2}$	1.0175	-7.8654E-03	-5.6018E-04	5.4967E-05	-1.9035E-06	2.1615E-08

for use on pumps with BEPs below and above 100 gpm, respectively, which permits the user to estimate the reduction of head, capacity, and efficiency that a viscous fluid will produce on a pump curve originally generated with water. A variation on this technique is described below.

The following example is taken from pages 114-116 of the *Hydraulic Institute Standards* section on centrifugal pump applications. There, the use of Figure 72, "Performance Correction Chart For Viscous Liquids," is discussed. Table 1 was calculated using polynomial equations developed to replace the nomogram presented in Figure 72. The results of the calculation are within rounding error of those presented in the standard. And the approach has the additional benefit of being more convenient to use, once it has been set up as a spreadsheet.

In the course of curve-fitting Figure 72, it was convenient to define a term known as pseudocapacity:

$$\text{pseudocapacity} = 1.95(V)^{0.5}[0.04739(H)^{0.25746}(Q)^{0.5}]^{-0.5}$$

where

V = fluid viscosity in centistokes

H = head rise per stage at BEP, measured in feet

Q = capacity at BEP in gpm

TABLE 3. CORRECTION FACTOR COMPARISON

	C_η	C_Q	$C_{H0.6}$	$C_{H0.8}$	$C_{H1.0}$	$C_{H1.2}$
Per Table 7 of HI Standards	0.635	0.95	0.96	0.94	0.92	0.89
Per Polynomial Expressions	0.639	0.939	0.958	0.939	0.916	0.887

Pseudocapacity is used with the following polynomial coefficients to determine viscosity correction terms that are very close to those given by Figure 72 in the *Hydraulic Institute Standards*. These polynomials have been checked throughout the entire range of Figure 72, and appear to give answers within 1.0% of those found using the figure.

The polynomial used is of the form:

$$C_x = D_{x1} + D_{x2}P + D_{x3}P^2 + D_{x4}P^3 + D_{x5}P^4 + D_{x6}P^5$$

where

C_x is the correction factor that must be applied to the term in question

D_{xn} are the polynomial coefficients listed in Table 2

P is the pseudocapacity term defined above

For comparison, the correction factors for the example above (tabulated in Table 7 of the *Hydraulic Institute Standards*) and those calculated using the polynomial expressions above are listed in Table 3.

The problem of selecting a pump for use in a viscous service is relatively simple once the correction coefficients have been calculated. If, for example, we had been looking for a pump that could deliver 100 feet of head at a capacity of 750 gpm, we would proceed as follows:

$$H_{water} = H_{viscous\ service}/C_{H1.0}$$

$$Q_{water} = Q_{viscous\ service}/C_Q$$

The next step would be to find a pump having the required performance on water. After determining the efficiency of the pump on water, we would correct it for the viscous case as shown above:

$$\eta_{viscous\ service} = \eta_{water} \times C_\eta$$

The horsepower required by the pump at this point would be calculated as follows:

$$hp_{viscous\ service} = \frac{(Q_{viscous\ service} \times H_{viscous\ service} \times sp\ gr)}{(3{,}960 \times \eta_{viscous\ service})}$$

As with water service, the horsepower requirements at off-design conditions should always be checked. ■

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Pump Balancing Criteria

BY GUNNAR HOLE

The subject of balancing rotors is one of the fundamentals of rotating equipment engineering. A number of balancing standards have been developed over the years to meet the requirements of pump manufacturers and users, and the idea of balancing is simple. Unfortunately, the definitions and mathematics used in describing balancing problems can be confusing. This article compares these criteria so the end user can use consistent reasoning when making balancing decisions.

TABLE 1. BALANCING CRITERIA

	Unbalanced Force Method As per API 610 6th Edition	Specified Eccentricity Method As per AGMA 510.02	Specified Circular Velocity Method As per API 610 7th Edition
Residual Unbalance (RUB), in.–oz where: W_j = rotor weight per balance plane, lbf; N = rpm; ε = eccentricity, in.	$\frac{56347}{N^2} W_j$	$16 \varepsilon W_j$	$4 \frac{W_j}{N}$
Eccentricity (ε) or Specific Unbalance in.–oz/lbm	$\frac{56347}{N^2}$	16ε	$\frac{4}{N}$
in.–lbm/lbm where RUB = εW_j	$\frac{3522}{N^2}$	ε see Table 2	$\frac{0.25}{N}$
Unbalance Force (UBF), lbf where: UBF = $\varepsilon M \omega^2$ and $M = W_j/386$ lbf–s^2/in. $\omega = 2 \pi N/60$ rad/s	$0.10 W_j$	$\frac{\varepsilon W_j N^2}{35200}$	$\frac{W_j N}{140800}$
Circular Velocity (CV), in./s	$\frac{368}{N}$	$\frac{\varepsilon N}{9.54}$	0.26
mm/s where CV = $\varepsilon \omega$	$\frac{9347}{N}$	$2.66 \varepsilon N$	0.665
ISO Standard 1940 Balance Grade	$G - \frac{9347}{N}$	$G - 2.66 \varepsilon N$	$G - 0.665$

Presented below is a description of the problem, definitions of some of the more important terms used, and references that can be consulted for a more thorough review. A table also compares three of the most common balancing criteria used in the pump industry.

Perhaps the least controversial comment that can be made to an experienced equipment specialist is that "accurate rotor balancing is critical to reliable operation." I could add some spice to the conversation by giving my opinion on how good is good enough, but I would rather address the standards used in the pump industry and show how they take different approaches to resolve the problem of balancing rotors.

I use the term rotor repeatedly in this discussion. For the purpose of this article, I include partially and fully assembled pump shaft/sleeve/impeller assemblies as well as individual pump components installed on balancing machine arbors in this definition.

The three major criteria used will be referred to as the Unbalanced Force Method (UFM), the Specified Eccentricity Method (SEM), and the Specified Circular Velocity Method (SCVM).

In the UFM the allowable unbalance permitted in a rotor is the amount that will result in a dynamic force on the rotor system equal to some percentage of the rotor's static weight. This allowable unbalance is therefore related to the operating speed of the rotor. An example of this method can be found in API Standard 610 6th Edition, where the unbalance force contributed to a rotor system by a rotating unbalance is limited to 10% of the rotor's static weight.

The SEM attempts to specify balance quality by limiting the distance by which the center of mass of the rotor can be offset from the center of rotation of the rotor. This method is used in AGMA Standard 515.02, which is commonly referenced by flexible coupling vendors. It has the advantage of being conceptually simple. For the gear manufacturers who developed this standard, it allowed the use of manufacturing process tolerances as balancing tolerances. In Paragraph 3.2.7, API 610 7th Edition suggests that couplings meeting AGMA 515.02 Class 8 should be used unless otherwise specified.

The SCVM is based on considerations of mechanical similarity. For geometrically similar rigid rotors running at equal peripheral speeds, the stresses in the rotor and bearings are the same. This method is described in ISO Standard 1940—Balance Quality of Rigid Rotors. It also forms the basis of API Standard 610 7th Edition's very stringent 4W/N balancing requirement. Standards based on this methodology are becoming more common.

In Table 1 the three balancing criteria discussed above are compared with respect to their effect on the various parameters involved in balancing. The terms used in the table are defined as follows:

RESIDUAL UNBALANCE

This is the amount of unbalance present or allowed in the rotor. It has the units of mass and length. It is computed by taking the product of the rotor mass (per balance plane) times the distance from the rotor's center of mass to its center of rotation. Note that 1 in.–oz is equivalent to 72.1 cm–g.

ECCENTRICITY

This is the distance that the center of mass of the rotor is displaced from the rotor's center of rotation. It has the unit of length. It can also be considered as a measure of specific residual unbalance, having the units of length–mass/mass. This term is the basic criterion of SEM balancing rules (see Table 2). Note that 1 in. is equivalent to 25.4 mm.

TABLE 2. BALANCE QUALITY CLASSES

Note: AGMA 515.02 refers to several Balance Quality Classes. They are summarized as follows:

AGMA Class	ε, μ-in.	Equivalent ISO Balance Quality Grade 1,800 rpm	3,600 rpm
8	4,000	19.2	38.3
9	2,000	9.6	19.2
10	1,000	4.8	9.6
11	500	2.4	4.8
12	250	1.2	2.4

UNBALANCE FORCE

This is the force that is exerted on a rotor system as a result of the non-symmetrical distribution of mass about the rotor's center of rotation. The units of this term are force. This term is the basic criterion of UFM balancing rules. Note that 1 lbf is equivalent to 4.45 Newton.

CIRCULAR VELOCITY

This is the velocity at which the center of mass of the rotor rotates around the center of rotation. You can think of it as a tangential velocity term. It has the units of length per unit time. It forms the basis for balancing rules based on the ISO Standard 1940 series. In fact, the Balancing Grades outlined in ISO 1940 are referenced by their allowable circular velocity in millimeters per second. The balance quality called for in API 610 7th Edition is better than the quality that ISO 1940 recommends for tape recorder drives and grinding machines. ISO 1940 recommends G–6.3 and G–2.5 for most pump components, where API 610 calls for the equivalent of G–0.67. Note that 1 in./s is equivalent to 25.4 mm/s.

RIGID ROTOR

A rotor is considered rigid when it can be balanced by making mass corrections in any two arbitrarily selected balancing planes. After these corrections are made, the balance will not significantly change at any speed up to the maximum operating speed. With the possible exception of home ceiling fans, I believe that two-plane balancing is the minimum required for rotating equipment components.

FLEXIBLE ROTOR

The elastic deflection of flexible rotors sets up additional centrifugal forces that add to the original unbalance forces. Such rotors can be balanced in two planes for a single speed only. At any other speed they will become unbalanced. Balancing the rotor to allow it to run over a range of speeds involves corrections in three or more planes. This process is called multi-plane balancing.

One important point is that the pump/coupling/driver system must be considered as a whole when evaluating balance quality. A simple pump rotor can be balanced to meet API 610 7th Edition's 4W/N criteria in a modern balancing machine without too much trouble. An electric motor rotor may be even easier to balance due to its simple construction. But the coupling connecting them can be a completely different matter.

The coupling will likely have more residual unbalance than either the pump or the motor. And every time you take the coupling apart and put it back together you take the chance of changing its balance condition. As written, API 610 7th Edition allows a coupling to have a specific residual unbalance nearly 60 times higher than for a 3,600 rpm pump. This can be a significant problem if you use a relatively heavy coupling.

These balancing methods are primarily intended for use on rigid rotors—those operating at speeds under their first critical speed. Flexible rotors, which operate above their first critical speed, are considerably more complicated to balance. The process of balancing flexible rotors is discussed in ISO Standard 5406–The Mechanical Balancing of Flexible Rotors and ISO Standard 5343–Criteria for Evaluating Flexible Rotor Unbalance.

The basic concepts of rigid and flexible rotor balancing are the same. The main difference is that with rigid rotor balancing we are only concerned with the rigid body modes of vibration. With a flexible rotor, we have to consider some of the higher modes of vibration as well. In these cases the deflection of the rotor affects the mass distribution along its length. In general, each of the modes has to be balanced independently.

Appendix I of API 610 7th Edition briefly discusses some of the implications of operating a rotor near a critical speed. The guidelines given there recommend separation margins that specify how far away from a critical speed you can operate a rotor. These margins depend on the system amplification factors (also known as magnification factors), which are directly related to the damping available for the mode or resonance in question. The net result of these recommendations is to limit the maximum operating amplification factor to a maximum of about 3.75. The amplification factor can be thought of as a multiplier applied to the mass eccentricity, ε, to account for the effect of system dynamics. Algebraically, the physics of the situation can be represented as follows:

$$x = X \sin(\omega t - \Phi)$$

$$X = \varepsilon \frac{(\omega/\omega_n)^2}{([1 - (\omega/\omega_n)^2]^2 + (2\zeta\omega/\omega_n)^2)^{0.5}}$$

$$\Phi = \tan^{-1} \frac{2\zeta\omega/\omega_n}{1 - (\omega/\omega_n)^2}$$

where

- x is the displacement of a point on the rotor
- X is the magnitude of the vibration at that point
- ε is the mass eccentricity
- ω is the operating speed or frequency of the rotor
- Φ is the phase angle by which the response lags the force
- ζ is the damping factor for the mode of vibration under consideration
- X/ε is the amplification factor
- ω/ω_n is the ratio of operating speed to the critical speed under consideration

A more detailed discussion on the topic of damped unbalance response (or whirling of shafts) can be found in any introductory vibration textbook. ■

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Bearing Basics

BY RAY RHOE

Antifriction bearings, which can utilize either balls or rollers, are used to transfer radial and axial loads between the rotating and stationary pump and motor assemblies during operation. Even under the best of installation, maintenance, and operating conditions, bearing failures can and will occur. The purpose of this article is to provide a working-level discussion of bearings, the types of failures, and how bearings should be installed and maintained for optimum life expectancy.

Due to space limitations, we cannot address all the different sizes and types of bearings available, or all the constraints currently utilized in design. However, because electric motors are used more often to drive centrifugal pumps, our discussion will be based on bearings typically used in quality motors. These bearings usually include a single radial bearing and a matched set of duplex angular contact bearings (DACBs). Together, these bearings must:

- allow the unit to operate satisfactorily over long periods of time with minimum friction and maintenance
- maintain critical tolerances between rotating and stationary assemblies to prevent contact and wear
- transmit all variable radial and axial loads in all operating conditions, which include reverse rotation, startup, shutdown, maximum flow, and maximum discharge pressure

Each bearing has a specific purpose. The radial bearing, which is located at one end of the motor, only transfers radial loads such as minor unbalanced rotor loads—and the weight of the rotor itself in the case of horizontally oriented components. The DACBs must transfer radial loads at the other end of the motor, and they must transfer all axial loads. Photo 1 shows several typical radial bearings, and Photo 2 shows DACBs.

Photo 1. Typical radial bearings

DIFFERENT BEARING CONFIGURATIONS

Radial bearings may be provided with either 0, 1, or 2 seals or shields that are effectively used to prevent entry of foreign material into the bearings. If the bearing is equipped with one seal or shield, the installer should determine which end of the motor the seal or shield should face. Failure to install radial bearings properly in the correct orientation may result in the blockage of grease or lubricant to the bearings during routine maintenance.

The orientation of DACBs is more complex, DACBs must be installed in one of four configurations, as determined by design:

1. face-to-face
2. back-to-back
3. tandem: faces toward the pump
4. tandem: faces away from the pump

The “face” of the DACB is that side that has the narrow lip on the outer race. The “back” of the bearing has the wider lip on the outer race and usually has various symbols and designators on it. Photo 2 shows two pairs of DACBs. The pair on the left is positioned face-to-face while the pair on the right is back-to-back. Note that the lip on the outer races of the first pair is narrower than on the second pair. This distinguishing characteristic provides an easy identification of which side is the face or back. In tandem, the narrow lip of one bearing is placed next to the wide lip on the other. In other words, all bearing faces point either toward the pump or away from it.

To facilitate the installation of DACBs, the bearing faces should be marked with a black indelible marker showing where the burnished alignment marks (BAMs) are on the back. This is because the four BAMs, two on each bearing, must be aligned with their counterparts, and not all BAMs are visible during installation. For example, when the first bearing is installed in a face-to-face configuration, the BAMs are on the back

Photo 2. Two pairs of DACBs, with the pair on the left positioned face-to-face, the pair on the right back-to-back

side, hidden from the installer. Marking the face of each bearing allows the installer to see where the BAMs are, so that all four BAMs may be aligned in the same relative position, such as 12 o'clock.

BEARING PRELOAD

Under certain operating conditions (hydraulic forces, gravity, and movement of the pump and motor foundation such as on a seagoing vessel), the rotor may be loaded in either direction. If this occurs, the balls in a DACB with no preload could become unloaded. When this happens, the balls tend to slide against the races (ball skid) rather than roll. This sliding could result in permanent damage to the bearings after about five minutes.

To prevent ball skid, bearing manufacturers provide bearings that have a predetermined clearance between either the inner or outer races. Face-to-face bearings have this clearance between the outer races. When the bearings are clamped together at installation (the outer races are clamped together), the balls are pressed between the inner and outer races, causing the preload. Back-to-back bearings have the clearance on the inner races, which are usually clamped together with a bearing locknut.

By increasing the clearance between races, the preload can be increased from zero to a heavy load. This way, when conditions cause the rotor loads to change direction or be eliminated, the bearing balls will still be loaded and ball skid should not occur.

One disadvantage of using preloaded bearings is that bearing life will be reduced due to the increased loading. Preloaded bearings should not be used unless design conditions require them.

If uncertain about the need for preload, users should contact manufacturers.

BEARING INSTALLATION

Once the proper bearings have been obtained and the correct orientation determined, installation is relatively simple.

The shaft and especially the shaft shoulder should be cleaned and any welding or grinding operations secured. The bearings must be installed in a clean environment, and the shaft must be free of nicks and burrs that may interfere with installation.

1. RADIAL BEARINGS

To install radial bearings, they should be heated in a portable oven to 180–200°F. Then, using clean gloves and remembering the correct orientation, quickly slide each bearing over the shaft and firmly onto the shaft shoulder. Do not drop or slap them into position. Experience indicates that you have about 10 seconds after removing the bearing from the oven before it cools and seizes the shaft. If it seizes the shaft out of location, remove it and scrap it. The bearing cannot be hammered into position or removed and reused because it will be destroyed internally by these actions.

2. DACBs

Installation of DACBs follows the some procedure, except that additional care must be taken to position the bearings properly, line up the burnished alignment marks, and not erase the indelible marks added on each bearing face. After the first bearing has been installed, rotate the rotor (if necessary) so the alignment mark on the inner race is at 12 o'clock, then rotate the outer race so it too is at 12 o'clock. Before proceeding with the second bearing, mentally walk through the procedure. Remember which direction the face goes and that the burnished alignment marks must be in the same position as the first bearing marks. Also remember you have about 10 seconds before the bearing seizes the shaft.

The purpose of aligning the four burnished alignment marks is to minimize off-loading (fight) and radial runout loads that will occur if the true centers of the bearings are not lined up. Minor imperfections will always occur, and they must be minimized. Failure to align the marks will result in the bearings loading each other.

DACBs come only in matched pairs—they must be used together. To verify that a pair is matched, check the serial number on the bearing halves—they should be the same, or properly designated, such as using bearing "A" and bearing "B."

NEW BEARING RUN-IN

After new bearings have been installed, they should be run in while monitoring their temperature, noise, and vibration. Run-in is often called the "heat run" or "bearing stabilization test."

To perform this test, first rotate the pump and driver by hand to check for rubbing or binding. If none occurs, operate the

Photo 3. Radial bearing disassembly

unit at the design rating point and record bearing temperatures every 15 min. Bearing temperatures should increase sharply and then slowly decline to their normal operating temperature, usually 20–60°F above ambient.

During the heat run care should be taken to ensure that the temperature does not exceed the value specified by the manufacturer. If it does, the unit should be secured and allowed to cool to within 20°F of ambient, or for 2 hours. The unit may then be restarted and the test repeated as necessary until the bearing temperature peaks and begins to decline.

If, after repeated attempts, bearing temperatures do not show signs of stabilization, too much grease may be present. The bearing should be inspected and corrective actions taken as necessary.

Now let's cover some basic rules to follow when working with bearings:

1. Never reuse a bearing that has been removed using a gear puller, even if it is new. The bearing has been internally destroyed in the removal process. See "True Brinelling" under "Bearing Failures."
2. DACBs must be installed in the correct orientation. If not, they may experience reverse loading and fail. See "Reverse Loading" under "Bearing Failures."
3. Bearings must be installed in a clean environment. Contamination is a leading cause of premature failures.
4. Do not pack the bearing and bearing cups full of grease. Excessive grease will cause overheating and ball skid. See "Excessive Lubrication" under "Bearing Failure."

BEARING FAILURES

Failure to follow these four basic rules will result in premature bearing failures. These and other failures will occur for the following reasons:

True Brinelling (failure to follow Rule 1): This type of bearing failure occurs when removing bearings with a gear puller. The force required to remove a bearing from a shrink-fit application is great enough that when it is transferred through the balls to the inner races, the balls are pressed into the inner and outer races forming permanent indentations.

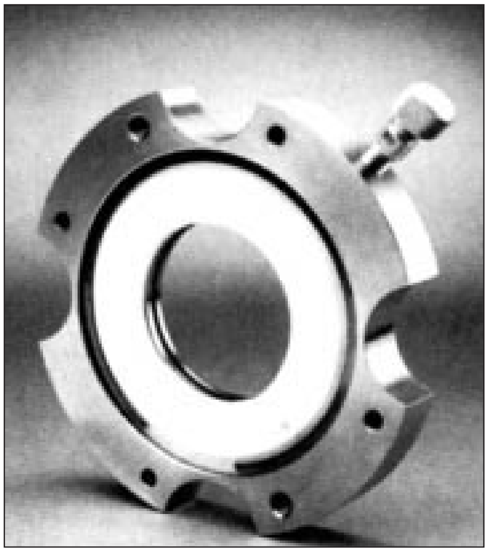
Reverse Loading (failure to follow Rule 2): Reverse loading occurs when DACBs are improperly installed and the balls ride on the ball ridge located on the outer race. Evidence of reverse loading appears as "equator" bands around the balls.

Contamination (failure to follow Rule 3): Contamination of bearings almost always occurs during installation, but can also occur when liquids or other constituents from the pump leak or are present in the surrounding environment. If contamination is found in a new bearing before installation, the bearing should be carefully cleaned and repacked. Evidence of solid contamination in used bearings usually appears as very small, flat dents in the races and balls.

Excessive Lubrication (failure to follow Rule 4): Too much grease in a bearing may cause the balls to "plow" their way through the grease, resulting in increased friction and heat. If the bearings and bearing caps are packed full of grease, ball skid could occur. When it does, the balls do not roll, but actually slide against the races. Experience shows that the bearings may be permanently damaged after more than five minutes of ball skid. Finding packed bearings and bearing caps is a good indication that too much grease caused the bearing to fail. Bearing manufacturers usually recommend that bearings have 25-50% of their free volume filled with grease.

Excessive Heat: Failure to provide adequate heat transfer paths, or operating the component at excessive loads or speeds may result in high operating temperatures. Evidence of excessive temperature usually appears as silver/gold/brown/blue discoloration of the metal parts.

False Brinelling: False brinelling occurs when excessive vibrations cause wear and breakdown of the grease film between the balls and the races. This condition may be accompanied by signs of corrosion. A good example of how false brinelling could occur would be when a horizontally positioned component is shipped



Zero-leakage magnetic liquid seal developed to retrofit process pumps

across the country and not cushioned from a rough road surface. The load of the rotor is passed through the bearing balls, which wear away or indent the races. Evidence of false brinelling looks similar to true brinelling, but may be accompanied by signs of corrosion where the grease film has not been maintained. Correction simply involves protecting the unit from excessive vibration and using specially formulated greases where past experience demonstrates the need.

Fatigue Failure: Even when all operating, installation, and maintenance conditions are perfect, bearings will still fail. In this case, the bearings have simply reached the end of their useful life, and any additional use results in metal being removed from the individual components. Evidence of fatigue failure appears as pits.

BEARING DISASSEMBLY FOR INSPECTION

Now that we know what to look for in failed bearings, let's see how we disassemble a bearing for inspection. Before disassembling any bearing, however, turn it by hand and check it for rough performance. Note its general condition, the grease (and quantity thereof), and whether there is any contamination. If solid contamination is present, the particles should be collected using a clean filter bag as follows:

1. Partially fill a clean bucket or container with clean diesel fuel or kerosene.
2. Insert a clean filter bag into the kerosene container. This ensures that no contamination from the container or the kerosene gets into the filter bag.
3. Using a clean brush, wash the grease and contamination out of the bearing. The grease will dissolve and any contamination will be collected in the filter bag for future evaluation.

RADIAL BEARING DISASSEMBLY

After removing the grease and any contamination, you should disassemble radial bearings by removing any seals or shields, which are often held in place by snap rings. Then, to remove a metal retainer, drill through the rivets and remove both retainer halves. Then the bearing should again be flushed (in a different location) to remove any metal shavings that may have fallen between the balls and races when drilling out the rivets. If the bearing does not freely turn by hand, some metal particles are still trapped between the balls and races.

Next, place the bearing on the floor as shown in Photo 3 with the balls packed tightly together on the top. Insert a rod or bar through the inner race and press down, hard if necessary. Note: If the balls are not packed tightly together, disassembly will not occur.

DACB DISASSEMBLY

To disassemble DACBs, support the face of the outer race and press down against the inner race. The back of the bearing must be on top.

HANDLING, TRANSPORTATION, AND STORAGE

Common sense applies in handling, storing, or transporting precision bearings. They should not be dropped or banged. They should be transported by hand in cushioned containers, or on the seat of vehicle—not in a bike rack. They should be stored in a cool, clean, dry environment.

Because nothing lasts forever, including bearing grease, bearings should not be stored for more than a few years. After this, the grease degrades and the bearings may become corroded. At best, an old bearing may have to be cleaned and repacked, using the correct type and amount of grease.

MAINTENANCE

Routine maintenance of bearings usually involves periodic regreasing (followed by a heat run) and monitoring bearing vibrations, which will gradually increase over long periods of time .

To maintain pumps and drivers that are secured for long periods of time, simply turn the rotor 10–15 revolutions every three months by hand. This will ensure than an adequate grease film exists to prevent corrosion of the bearing. If this action is not taken, the bearings may begin to corrode due to a breakdown in the grease film. ■

Ray W. Rhoe, PE, has a BSCE from The Citadel and 15 years' experience with pumps, testing, and hydraulic design.

Centrifugal Pump Efficiency

BY DAVID CUMMINGS

The efficiency of centrifugal pumps of all sizes is becoming more important as the cost and demand for electricity increases. Many utilities are emphasizing conservation to reduce the number of new generating facilities that need to be built. Utilities have increased incentives to conserve power with programs that emphasize demand side conservation. These programs often help fund capital equipment replacements that reduce electrical consumption. Demand side management programs make replacing old pumping equipment more feasible than ever before.

DETERMINING PUMP EFFICIENCY

The efficiency of a pump η_p is ratio of water horsepower (whp) to brake horsepower (bhp). The highest efficiency of a pump occurs at the flow where the incidence angle of the fluid entering the hydraulic passages best matches with the vane angle. The operating condition where a pump design has its highest efficiency is referred to as the best efficiency point (BEP).

η_p = whp/bhp

The water horsepower (whp) may be determined from the equation:

whp = QHs/3,960

where Q = capacity in gallons per minute, H = developed head in feet, and s = specific gravity of pumped fluid.

Preferably, the brake horsepower supplied by a driver can be determined using a transmission dynameter or with a specially calibrated motor. Brake horsepower determined in the field by measuring kilowatt input and multiplying by the motor catalogue efficiency can be inaccurate. If motor power is determined in the field, data should be taken at the motor junction box, not at the motor control center.

Overall pump motor efficiency η_o may be determined from the equation:

η_o = whp/ehp

where ehp = electrical power in horsepower.

EFFICIENCY LOSSES

Pump efficiency is influenced by hydraulic effects, mechanical losses, and internal leakage. Each of these factors can be controlled to improve pump efficiency. Any given design arrangement balances the cost of manufacturing, reliability, and power consumption to meet users needs.

Hydraulic losses may be caused by boundary layer effects, disruptions of the velocity profile, and flow separation. Boundary layer losses can be minimized by making pumps with clean, smooth, and uniform hydraulic passages. Mechanical grinding and polishing of hydraulic surfaces, or modern casting techniques, can be used to improve the surface finish, decrease vane thickness, and improve efficiency. Shell molds, ceramic cores, and special sands produce castings with smoother and more uniform hydraulic passages.

Separation of flow occurs when a pump is operated well away from the best efficiency point (BEP). The flow separation occurs because the incidence angle of the fluid entering the hydraulic passage is significantly different from the angle of the blade. Voided areas increase the amount of energy required to force the fluid through the passage.

Mechanical losses in a pump are caused by viscous disc friction, bearing losses, seal or packing losses, and recirculation devices. If the clearance between the impeller and casing sidewall is too large, disc friction can increase, reducing efficiency. Bearings, thrust balancing devices, seals, and packing all contribute to frictional losses. Most modern bearing and seal designs generate full fluid film lubrication to minimize frictional losses and wear. Frequently, recirculation devices such as auxiliary impellers or pumping rings are used to provide cooling and lubrication to bearings and seals. Like the main impeller, these devices pump fluid and can have significant power requirements.

Internal leakage occurs as the result of flow between the rotating and stationary parts of the pump, from the discharge of the impeller back to the suction. The rate of leakage is a function of the clearances in the pump. Reducing the clearances will decrease the leakage but can result in reliability problems if mating materials are not properly selected. Some designs bleed off flows from the discharge to balance thrust, provide bearing lubrication, or to cool the seal.

EXPECTED EFFICIENCIES

The expected hydraulic efficiency of a pump design is a function of the pump size and type. Generally, the larger the pump, the higher the efficiency. Pumps that are geometrically similar should have similar efficiencies. Expected BEPs have been plotted as a function of specific speed and pump size. A set of curves that may be used to estimate efficiency is provided in Figure 1. The specific speed (N_s) of a pump may be determined from the equation:

$N_s = NQ^{0.5}/H^{0.75}$

where N = speed in rpm, Q = capacity in gpm, and H = developed head in feet.

Using a pump performance curve, the highest efficiency can be determined and the specific speed calculated using the head and capacity at that point. Using the specific speed and the pump capacity, the expected efficiency can be estimated. If the pump has bearings or seals that require more power, such as tilting pad thrust bearings or multiple seals, this should be considered when calculating the expected efficiency.

MOTOR EFFICIENCIES

Efficiencies for new "premium efficiency motors" are provided in Figure 2. Using these values, with anticipated pump efficiencies in Figure 1, the expected power consumption for a well designed pump and motor can be determined. The calculated power consumption can be compared with an existing installation to determine the value of improving pump performance or replacing the unit.

EXAMPLE CALCULATION OF PUMP EFFICIENCY

A single-stage end-suction process pump will be used as an example for an efficiency calculation. The pump uses a mechanical seal and an angular contact ball bearing pair for thrust. The pumped fluid is water with specific gravity of 1.0. The pump operates at its BEP of 2,250 gpm, developing 135 feet of head. The pump speed is 1,750 rpm (note: with the new motor the speed may change, but to simplify the example it will be assumed the new and old motor both operate at 1,750 rpm). The expected power consumption for a new unit can be calculated.

First the pump specific speed will be calculated:

$$N_s = \frac{[1{,}750 \text{ rpm x } (2{,}250 \text{ gpm})^{0.5}]}{(135 \text{ ft})^{0.75}}$$

N_s = 2,096

Figure 1 can be used at N_s = 2,100 and interpolated for 2,250 gpm.

The expected efficiency is 86%.

The water horsepower is:

$$\text{whp} = \frac{(2{,}250 \text{ gpm x } 135 \text{ ft x } 1.0)}{3{,}960}$$

whp = 76.7 Hp

The expected brake horsepower is:

bhp = 76.7 Hp/0.86

bhp = 89.2 Hp

TABLE 1. EXPECTED EFFICIENCY FOR "PREMIUM EFFICIENCY MOTORS"

Motor	Minimum Acceptable Efficiency		
Horsepower	1200 rpm	1800 rpm	3600 rpm
5	88.0	88.0	87.0
10	90.2	91.0	90.2
15	91.0	92.0	91.0
20	91.7	93.0	91.7
25	92.4	93.5	92.0
30	93.0	93.6	92.4
40	93.6	94.1	93.0
50	93.6	94.1	93.0
75	94.5	95.0	94.1
100	94.5	95.0	94.5
125	94.5	95.4	94.5
150	95.0	95.4	94.5
200	95.0	95.0	95.0
Over 200	95.0	95.4	95.0

The antifriction bearings and typical mechanical seal do not require a power adjustment. However, if a tilting pad thrust bearing or other device, such as a special seal, that used more power was part of the design, the correction would be made here by adding the additional horsepower to the calculated value.

Using Figure 2, the minimum efficiency for a 100 Hp motor is 95%. The efficiency value may change slightly for the operating condition and should be verified with a motor manufacturer. The 95% efficiency will be used in this example. The expected electrical horsepower is:

ehp = 89.2 Hp/0.95

ehp = 93.9 Hp

or

ehp = 93.9 Hp x .7457 kw/Hp

ehp = 70.0 kw

The last time this pump was rebuilt and put in service, power was measured at 79.5 kw. The difference in power consumption between the existing unit and a new unit can be calculated:

ehp = 79.5 kw - 70.0 kw

ehp = 9.5 k.

If power costs 8 cents a kilowatt hour and the pump operates continuously, the savings of replacing this unit on an annual basis can be calculated:

cost = 9.5 kw x $0.08 per kw hr x 8,760 hr/yr

cost = $6,658 year

This figure can be used to determine if the additional power consumption justifies replacing the pump. If a replacement pump and motor of this size can be purchased and installed for $40,000, and the electric utility offers a 50% rebate program, the net cost of $20,000 for the user is certainly worth considering.

SUMMARY

Remember, for any centrifugal pump to operate efficiently it needs to be properly applied. When processes require changing flow rates frequently, variable speed drives can be a solution. A pump operating far from its BEP will be neither efficient nor reliable. Many times changing the pump size to better match the system will reduce power costs dramatically. ■

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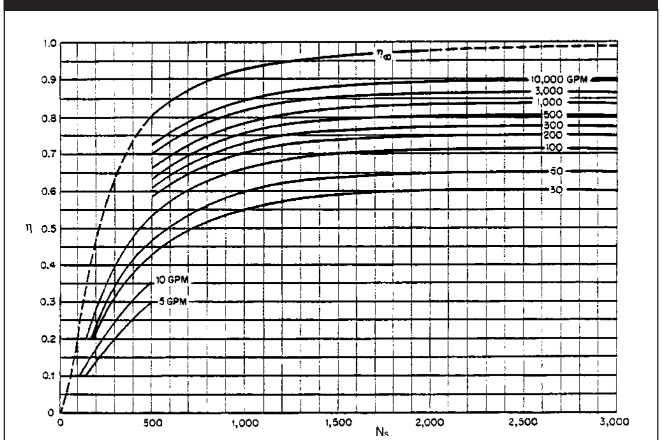
FIGURE 1



Efficiency of various pumps sizes and specific speeds

Motor Size Selection for Centrifugal Pumps

BY ROBERT J. HART

Q: How do I select the proper motor size for my centrifugal pump applications? In some applications we have experienced driver overload, while other applications appear satisfactory using the same selection method.

A: What seems to be a straightforward requirement of selecting a pump motor too frequently results in a major problem when commissioning the pumping system and the installed motor overloads and is tripped off the line. Correcting the problem can be as simple as adjusting a manual valve. And it can be as complex and time consuming as replacing the motor, motor starters, and service wire with a larger size, and altering the flow control system.

A simple rule of thumb of supplying a motor size that exceeds the pump manufacturers rated point brake horsepower by some fixed margin, or of supplying a motor size equal to the brake horsepower at the end of the selected impeller diameter curve, may not work in all cases. The person selecting the motor must have a thorough knowledge of the pumping system and its characteristics, pump industry practices, and limitations of the generalized data provided by the pump manufacturer with the quotation.

GENERALIZED PERFORMANCE CURVE LIMITATIONS

The brake horsepower values published by the pump manufacturer on the generalized hydraulic performance curves (TDH, efficiency, BHP vs. capacity) are the basis for the rated point BHP returned to a potential customer when responding to a request for quotation for a specific application. The data is as accurate as practical for the designated equipment design, but does have a tolerance range which may be experienced for any specific pump for such characteristics as the Total Dynamic Head (TDH) at a specific flow rate.

Brake horsepower is related to the TDH by the following formula:

$$\text{BHP} = \frac{(\text{TDH}) \times \text{Flow} \times \text{Specific Gravity}}{3960 \times \text{Efficiency} \times \text{Viscosity Correction Factors*}}$$

**See Hydraulic Institute Standards for Values*

Most pump users will not accept a lower than specified rating point TDH, and the manufacturer is frequently required to test the equipment prior to shipment to confirm that the pump meets the specified requirements. If the installed impeller should produce less TDH than specified, the manufacturer must replace the impeller with a larger diameter. If high alloy materials are involved, there may be considerable expense and delay involved. Hence the practice in the pump industry is to publish a performance curve (TDH vs. Capacity) for a given impeller diameter that is somewhat less than can actually be achieved by the specified diameters. Should a test then be specified and the impeller TDH test higher than allowed tolerances at a given flow rate, the impeller can be reduced to a smaller diameter to provide the required values without replacing it.

One of the current pump industry acceptance test criteria, the Hydraulic Institute Standards, permits the TDH to exceed the design point requirements by as much as 8%. Sometimes pump impellers will exceed the published data by as much as 20% when first tested. This is not usually the case, but there is a range of performance, especially on relatively new or infrequently tested pump designs.

The user should be informed of this potential variation if the impeller requires replacement due to normal operating wear of the pump, especially if it is to be purchased from a source other than the original equipment manufacturer, which may not have historical test records of the original hydraulic design.

Industry practice is to guarantee only the TDH (with a tolerance range) at a specified flow rate and the pump efficiency. The resulting brake horsepower is guaranteed only by the same tolerance, and then only if the pump is tested.

MECHANICAL SEAL HORSEPOWER LOSSES

The performance curve published by the pump manufacturer and described above does not provide allowances for the power required to turn a mechanical seal that is loaded to typical process conditions. For a high suction pressure, double mechanical shaft seal pump installation, this can be a measurable amount and must be added to the horsepower required to move the liquid. An allowance of one to two horsepower, for example, may be required for some ANSI style pump designs to compensate for seal losses. Hence, if the generalized performance curve rating point results in a BHP of 7.5 Hp, a motor of 10 Hp may be considered for the application if no allowance is given for factors like seal drag.

As part of the equipment quotation, an estimate for the seal horsepower drag should be requested for all pumps requiring mechanical seals. If a double mechanical seal has been specified with a buffer fluid pressurized flush, the buffer fluid pressure must be specified by the seal and pump manufacturer and observed by the user to assure the estimated seal drag horsepower is not exceeded. Over pressurizing the double mechanical seal buffer system at the site can result in a motor overload condition not anticipated during the motor selection phase.

Seal horsepower losses typically have a greater impact on the installations at or below 25 Hp, but they should be considered for all installations.

FLOW CONTROL

Since the BHP of most pump designs increases with increasing flow through the pump, it is the user's responsibility to assure that the actual system flow does not exceed the rated flow originally specified when the pump was purchased. Pumping systems that limit flow only by the resistance of installed piping have a tendency to be sized with safety factors to "assure" the pump selected will provide adequate flow (see the above comments regarding generalized performance curves). A motor may overload when the pump operates at a higher flow rate than anticipated and requires a greater horsepower. Should the actual system curve extend beyond the end of the published pump curve and not intersect the pump curve, the actual horsepower will be greater than the "end of curve" horsepower frequently used as basis for motor selection. With adequate NPSH available to the pump, the performance curve and corresponding horsepower may extend to greater than published values (Figure 1).

FLUID CHARACTERISTICS

Both specific gravity and viscosity can affect the required pump brake horsepower (see equation above). Motors are normally selected on the basis of rated conditions of head, flow, specific gravity, temperature, and viscosity.

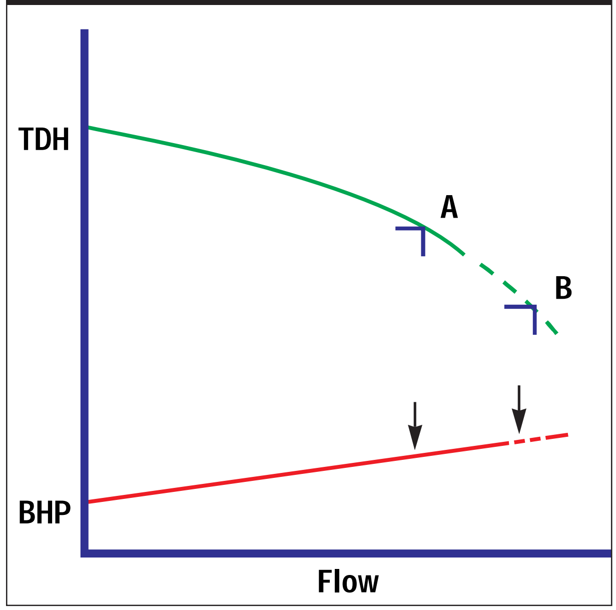
The off-design conditions of these characteristics should be examined and those fluid characteristics which affect brake horsepower evaluated before selecting a driver.

As examples:

Is there an alternate start-up or shutdown flush liquid required which has a higher specific gravity liquid than the rated flow material?

What is the actual liquid viscosity at a lower temperature than rated conditions, and will it increase the BHP of the pump? Even though the pump and piping is well insulated, without heat tracing the system will be at ambient temperature during a start-up. This will cool the incoming liquid below the continuous on-line conditions that would exist once the piping system is in operation and at equilibrium.

FIGURE 1



Pump performance curve. A= calculated system curve with safety factors, B= actual system curve.

PUMP WEAR

A certain amount of internal recirculation takes place inside a centrifugal pump casing at all times. As internal clearances change due to wear, the rate of this circulation increases. If the system demands down stream of the pump remain constant and the system is designed to maintain process flow, the pump must flow at a higher rate to compensate for this recirculation. Because of this, it may require a corresponding higher horsepower. See I.J. Karassik's recent article "When to Maintain Centrifugal Pumps" (Hydrocarbon Processing, April 1993) for additional information on this topic.

SUMMARY

Motor selection for centrifugal pumps involves many considerations, some of which are beyond the control of the pump manufacturer. There is no simple rule of thumb.

Oversizing motors to compensate for all of the conditions that may or may not exist on every installation can be a major additional expense when considering the total electrical system. This article has illustrated the variables that must be taken into account. ■

Robert J. Hart is a Senior Consultant at the DuPont Company. He also serves on the Pumps and Systems *Editorial Advisory Board.*

Setting the Minimum Flows for Centrifugal Pumps

BY: IGOR J. KARASSIK

Q: I am presently involved in replacing a newly purchased pump. It was accepted by the purchaser, but the shop test was noisy. The manufacturer said this was due to the poor suction piping. The field test was unacceptable and noisy, and there was disagreement about whether the noise was due to improperly placed elbows in the suction piping or if the pump was inappropriately selected. The pump, probably designed for flows much greater than system requirements, was recirculating. The noise was a very low frequency, random banging. The single-stage, double-suction, twin-volute design had four times more NPSH than required. How would your experts have diagnosed this costly problem?

What witness shop test should be conducted so that the pump purchaser can be assured of a safe continuous flow as quoted in the proposal? What measurements, observations (both audible and visual), and instrumentation should be used to detect the onset of suction and/or discharge recirculation? If the pump does not perform as quoted, are minor shop alterations conceivable?

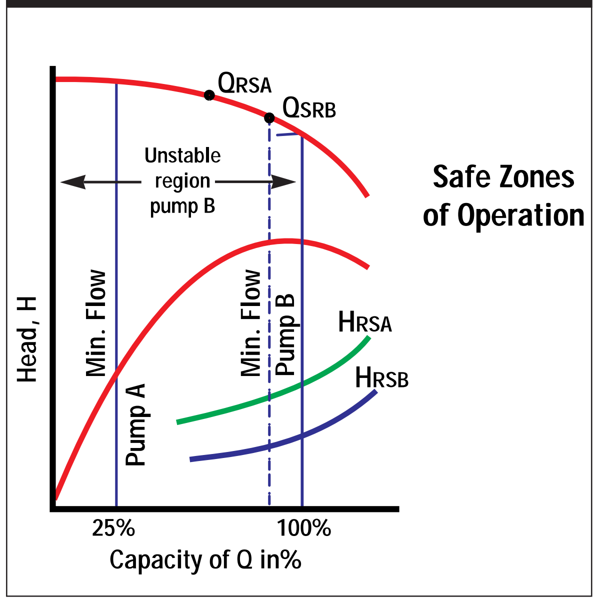
I would like to suggest to the Hydraulic Institute that a minimum nonrecirculating flow test be added to the standards. Your thoughts?

J. P. Messina, Professional Engineer, Pump and Hydraulics Consultant, Springfield, NJ.

A: Until about 25 years ago, there were only four factors to consider when setting an acceptable minimum flow for centrifugal pumps:

- higher radial thrust developed by single volute pumps at reduced flows
- temperature rise in the liquid pumped
- desire to avoid overload of drivers of high specific-speed axial-flow pumps
- for pumps handling liquids with significant amounts of dissolved or entrain-ed air or gas, the need to maintain sufficiently high fluid velocities to wash out this air or gas along with the liquid

FIGURE 1



Comparison of safe zones of operation for normal and for high S value impellers

Since then, a new phenomenon has been discovered that affects the setting of minimum flows. At certain reduced flows, all centrifugal pumps are subject to internal recirculation, both in the suction and discharge areas of the impeller. This produces pulsations at both the suction and discharge, and the vibration can damage impeller material in a way similar to classic cavitation, although taking place in a different area of the impeller.

Each of these effects may dictate a different minimum operating capacity. Clearly, the final decision must be based on the greatest of the individual minimums. The internal recirculation usually sets the recommended minimum, which appears to be what happened in your case.

You've actually raised two questions:

1. How can one determine the onset of recirculation?
2. After determining the onset, what should be the recommended minimum flow in relation to the recirculation flow?

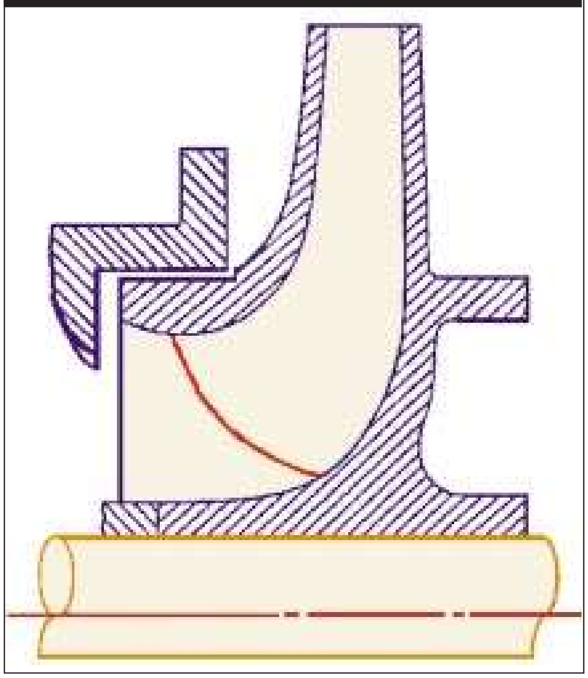
Unfortunately, the answers to both questions remain in the realm of strong controversy. Accept my comments as a personal opinion.

One theoretical method exists to predict the onset of recirculation (Ref. 1 and 2). The results of this method have been verified by many tests, with actual pumps and plastic transparent models where the onset could be observed with a strobe light. The results corresponded within no more than 5% deviation from the predictions.

Assuming that the pump is properly furnished with the necessary instrumentation, such as flowmeter, pressure gauges with sufficient sensitivity to show pulsations, and vibration and noise monitoring devices, an experienced test engineer should be able to pinpoint the onset of internal recirculation.

But it is the setting of the minimum flow which—for the time being—remains controversial. Obviously, any material damage cannot serve as a standard, because by the time the correctness of the deci-

FIGURE 2



“Bulk-head ring” construction used to eliminate unfavorable effects of excessively large impeller eye diameter

sion has been verified, it is too late. Therefore, the magnitudes of pressure pulsation, noise and vibration are the only criteria for establishing the minimum flow.

Regarding vibration, the Hydraulic Institute Standards includes a chart, plotting maximum permissible peak-to-peak amplitudes against frequency, and it is applicable “when the pump is operating at rated speed within plus or minus 10% of rated capacity.” This could create a serious problem whenever a pump meets these limits but is subject to considerably higher vibrations when operated below the recirculation flows. The API-610 Standard is more specific, defining the minimum continuous stable flow at which the pump can operate without exceeding the noise and vibration limits imposed by the Standard. These limits are expressed in inches per second rather than mils of displacement.

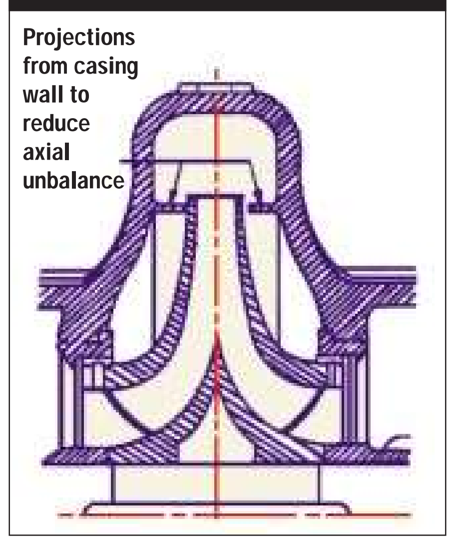
The Hydraulic In-stitute should probably set up rules for establishing a minimum flow test. But obtaining a consensus about the acceptable limits of vibration and noise will be difficult. The choice of a minimum flow is much more subjective if it is based on problems arising from internal recirculation than when the temperature rises, and radial thrust and overload of drivers of high specific-speed pumps are concerned. In these situations, the effect of operating at any given low capacity can be quantified. Even the effect of handling liquids laden with air or gas is easy to determine since at some given flow the noncondensible content of the liquid will not be washed away, and will accumulate within the pump, which will stop pumping.

Another major obstacle to overcome in achieving a consensus is to define what is continuous service and what is intermittent. When Warren Fraser (who did all the seminal work on internal recirculation) and I tried to produce a quantitative value that would distinguish between these two, we first tried to define 25% of the time as the breaking point between them. At first this seemed reasonable, but we soon realized that we had another problem to face. There was a difference between running a pump for six hours per day at or below an arbitrary flow and running it for three months out of a year for some strictly climatic conditions. So, we decided to avoid making any formal distinction between continuous and intermittent.

I admit that I do not have a definite and final answer to offer on the subject of selecting a minimum flow standard. I continue to use a guideline that I established some years ago (Ref. 3). Because the choice of required NPSH affects the onset of internal recirculation, for high suction specific speeds the minimum flow should correspond to the onset of the recirculation. For cold water, this refers to values over 10,500. And for more conservative S values, such as 8500 to 9500, set the minimum flow at 25% of the best efficiency capacity (Figure 1).

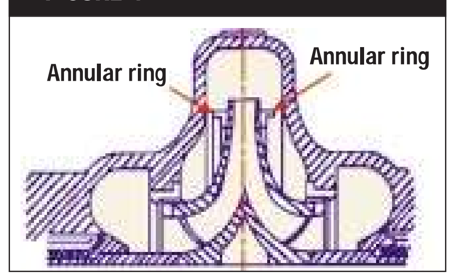
These comments represent my personal opinion. I am aware that some users may be more conservative and insist that the minimum flow should never be less than the recirculation capacity. In that case, users should specify this restriction in their

FIGURE 3



Projections from casing wall provided to reduce problems created by discharge side recirculation

FIGURE 4



Addition of two annular rings to impeller shrouds to reduce axial movement of rotor caused by internal recirculation at discharge

requests for bids. But this could only be acceptable if guidelines are published on how to conduct a test for recirculation or a formula becomes widely accepted on how to calculate the onset of internal recirculation.

Regarding your question about what minor alterations may be made if the pump does not perform satisfactorily in this connection, there are two possible solutions:

1. For suction recirculation, you can reduce the minimum acceptable flow by incorporating a "bulk-head ring" with an apron overhanging the eye of the impeller (Figure 2). Of course, this does increase the required NPSH and can only be done if there is the necessary margin between available and required NPSH.
2. If the problem is caused by discharge side recirculation, you can achieve some relief by providing projections from the casing wall (Figure 3). Alternately, annular rings can be fitted to the outer shrouds of the impeller (Figure 4).

I hope these comments will serve to open a dialogue between pump users and manufacturers. Such a discussion should lead to the undertaking of a series of tests that will shed additional light on the problem of acceptable minimum flows. These tests, in turn, could permit the Hydraulic Institute to include guidelines in its standards. ■

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Estimating Maximum Head in Single – and Multi-Stage Pump Systems

BY JAMES NETZEL

The maximum head or discharge pressure of a centrifugal pump can be easily estimated if the impeller diameter, number of impellers used, and rpm of the driver are known.

Q: How can you estimate the maximum (shutoff) head that a centrifugal pump can deliver?

A: The maximum pressure a centrifugal pump delivers should be known in order to ensure that a piping system is adequately designed. Any pump that operates at a high flow rate could deliver significantly more pressure at zero (0) gpm flow, such as when the discharge valve is closed, than it delivers at operating flow.

The maximum head or discharge pressure of a centrifugal pump, which usually occurs at shutoff conditions (0 gpm), can be easily estimated if the impeller diameter, number of impellers used, and rpm of the driver (electric motor, gas engine, turbine, etc.) are known.

Let's say we have a single-stage pump with a 10-in. diameter impeller and an 1,800 rpm driver. To determine the head in feet, simply take the impeller diameter in inches and square it. Our 10-in. impeller at 1,800 rpm would yield 10^2, or 100 ft of head. An 8-in. impeller would yield 8^2, or 64 ft of head, while a 12-in. impeller would yield 12^2, or 144 ft of head.

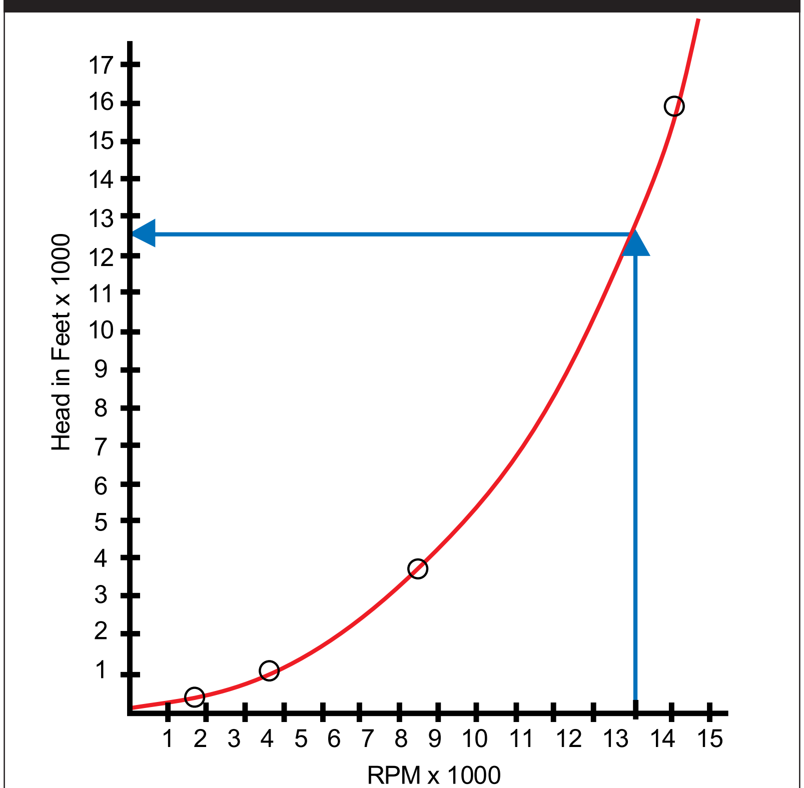
Now let's assume that our 10-in. diameter impeller is driven by a 3,600 rpm motor. We first determine the head at 1,800 rpm, but then multiply this value by a factor of four. The basic rule is that every time the rpm changes by a factor of two, the head changes by a factor of four. The head at 3,600 rpm for our 10-in. impeller is therefore 10^2 x 4, or 400 ft of head. Our 8-in. impeller at 3600 rpm would give us 8^2 x 4, or 256 ft of head, and our 12-in. impeller would give us 12^2 x 4, or 576 ft of head.

For multiple stages (more than one impeller), simply multiply the final head for one impeller by the total number of impellers in the pump. For a pump with three 10-in. impellers and a speed of 3,600 rpm, we get (10^2 x 4) x 3 = 400 x 3 = 1,200 ft. of head.

Now what happens if we reduce the speed below 1,800 rpm? The same rule still applies: a change in speed by a factor of two changes the head by a factor of four. Therefore, a 10-in. diameter impeller spinning at 900 rpm delivers only one fourth the head it would at 1,800 rpm: $10^2/4$ = 25 ft.

Plotting several head-versus-rpm points on a curve will allow the user to estimate the maximum

FIGURE 1



Rotations per minute (rpm) vs. head in feet to estimate maximum head

head at any given speed. Let's say we have a turbine-driven pump that injects water into the ground to raise the subterranean oil reserves to the surface for processing. The vendor tells you that the maximum head is classified, but you have been requested to resolve system problems that you believe are pressure related. The vendor tells you that the pump has four 8-in. diameter impellers and is driven by the turbine at 13,000 rpm. You would estimate the maximum head as follows:

Step 1 Determine the head at 1,800 rpm:

8^2 x 4 stages = 256 ft

Step 2 Multiply the head at 1,800 rpm by four to get the head at 3,600 rpm:

256 x 4 = 1,024 ft

Step 3 Multiply the head at 3,600 rpm by 4 to get the head at 7,200 rpm:

1,024 x 4 = 4,096 ft

Step 4 Multiply the head at 7,200 rpm by 4 to get the head at 14,400 rpm:

4,096 x 4 = 16,384 ft

Step 5 Plot the rpm-versus-head points to obtain the curve shown in Figure 1.

As you can see, the estimated head at 13,000 rpm is 12,500 ft. To convert head in feet to psi, simply divide the head by 2.31 to get 5,411 psi.

Ray W. Rhoe, PE, has a BSCE from The Citadel and 15 years' experience with pumps, testing, and hydraulic design.

Q: What different types of seal lubrication exist?

A: A mechanical seal is designed to operate in many types of fluids. The product sealed becomes the lubricant for the seal faces. Many times the fluid being sealed is a poor lubricant or contains abrasives that must be taken into account in the seal design. The design of the seal faces, materials of construction, and seal lubrication play an important role in successful operation. Achieving a high level of reliability and service life is a classic problem in the field of tribology, the study of friction, wear, and lubrication.

The lubrication system for two sliding seal faces can be classified as follows: 1) hydrodynamic, 2) elastohydrodynamic, 3) boundary, and 4) mixed film.

Hydrodynamic conditions exist when the fluid film completely separates the seal faces. Direct surface contact between seal faces does not take place, so there is no wear, and heat generation from friction is zero. The only heat generation occurs from shearing of the fluid film, which is extremely small. A hydrodynamic seal may rely on design features such as balance factors, surface waviness, or spiral grooves to separate the seal faces. The Society of Tribologists and Lubrication Engineers (STLE) guideline in "Meeting Emissions Regulations with Mechanical Seals" lists hydrodynamic seals as a technology to control emissions.

Elastohydrodynamic lubrication (EHD) is also found in sliding surfaces, but more often this involves rolling surfaces separated by an oil film. Here the moving surfaces form an interface region that deforms elastically under contact pressure. This deformation creates larger film areas and very thin films. Such lubrication systems are normally used to control wear in rolling element bearings. In seals where the viscosity of the fluid sealed increases with increasing pressure, elastohydrodynamic lubrication occurs.

Boundary lubrication is important for seal faces that are moving very slowly under heavy load. Here, hydrodynamic and elastohydrodynamic lubricant pressures are insufficient to separate the seal faces. The sliding surfaces are protected by the tribological properties of the materials of construction. An example of a seal operating within this lubrication system is a dry-running agitator seal.

Mixed-film lubrication, a combination of all the previous systems discussed, occurs in all contact seals. Here the fluid film becomes very thin and is a combination of both the liquid and the gas phases of the fluid sealed. Asperities from one surface may penetrate the lubricating film and contact the opposite surface. The seal face load is then supported partially by the fluid film and partially by solid contact. If the generated head at the seal faces is not removed, surface wear and damage can occur. For applications where the seal face load is too high or the fluid viscosity is too low, designs of seal faces can be changed through balance and face geometry to improve seal performance. ■

James Netzel is Chief Engineer at John Crane Inc. He serves on the Editorial Advisory Board for Pumps and Systems.

Tips on Pump Efficiency

BY WILLIAM E. (ED) NELSON

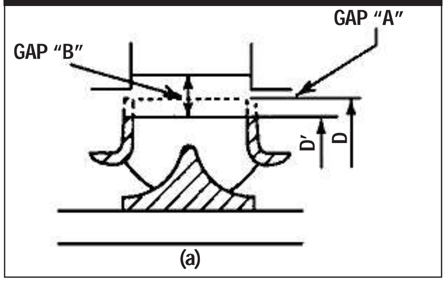
Q: When trimming a pump impeller to change the flow and head, I sometimes get too much of a reduction. What is the problem?

A: With a constant rotational speed, as is the case with most pumps, the "Affinity Laws" commonly used for calculating the trim do not accurately reflect the relationship between the change in impeller diameter and the hydraulic performance achieved by the pump. The calculations generally dictate more of a cut than required to affect the desired head and flow reduction. The "Affinity Law" errors can be on the order of 20 percent of the calculated reduction. If the calculated reduction trimming calls for a 10 percent reduction in diameter, only seven or eight percent reduction should be made. The lower the specific speed of the impeller cut, the larger the discrepancy. This subject is covered in only a few pump handbooks. The subject is well covered on pages 18 and 19 of *Centrifugal Pumps - Design and Application*, First Edition, by Val Lobanoff and Robert R. Ross. There are several reasons for the actual head and flow being lower than that calculated:

1. The "Affinity Laws" assume that the impeller shrouds are parallel. In actuality, the shrouds are parallel only in lower specific speed pumps.
2. The liquid exit angle is altered as the impeller is trimmed, so the head curve steepens slightly.
3. There is increased turbulent flow at the vane-tips as the impeller is trimmed, if the shroud-to-casing clearance (Gap "A") is not maintained.

All of these effects contribute to a reduced head development and flow. Pumps of mixed flow design are more affected than the true radial flow impellers found in higher head pumps. More caution has to be exercised in altering the diameter of a mixed flow impeller.

FIGURE 1. SHROUD & VANE REDUCTION



Q: What is the effect of trimming an impeller on pump efficiency?

A: It depends on the *specific speed* of the impeller. The specific speed index classifies the hydraulic features of pump impellers according to their type and proportions. Most refinery pumps fall between about 900 and 2,500 on this index. Some vertical multistage pumps are in the 4,000 to 6,000 range.

For radial designs, impeller diameter should not be reduced more than 70 percent of the maximum diameter design. Reductions in pump impeller diameters also alter outlet channel width, blade exit angle and blade length and may significantly reduce the efficiency. The greater the impeller diameter reduction from maximum diameter and the higher the specific speed *(not suction specific speed)*, the more the pump will decrease with the trimming of the impeller.

Q: What is the effect of impeller trimming on NPSH required?

A: Small reductions in impeller diameter will increase the required NPSH only slightly. Diameter reductions greater than about five to 10 percent will increase NPSH required, which occurs because specific vane loading is raised by the reduced vane length, affecting velocity distribution at the impeller inlet. Not all pump companies consistently show their pump curves the increase NPSHR with reduced impeller diameters. Attention must be paid to this factor when the margin between NPSHR and NPSHA is very narrow or the NPSHR for a pump is extremely low.

Q: What effect does trimming an impeller have on axial vibration?

A: Excessive impeller shroud-to-casing clearances (Gap "A") and suction recirculation cause eddy flows around the impeller, which in turn cause low frequency axial vibrations. Flow disturbances related to suction recirculation and cavitation are always present in both diffuser and volute type pumps. As the

FIGURE 2. OBLIQUE CUTS OF VANE

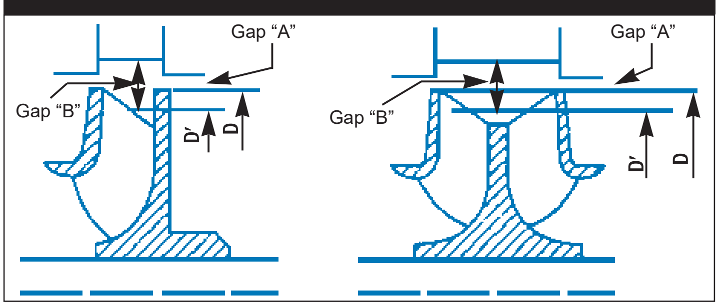
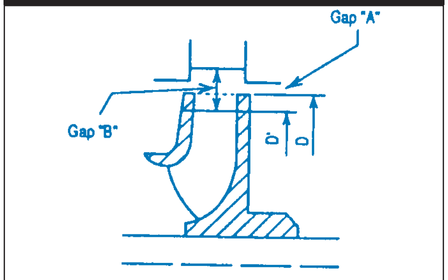


FIGURE 3. TERMINATING VANES ONLY



impeller diameters are reduced, the flow distribution pattern across the exit width of the impeller becomes more unstable. The tendency for the high-pressure liquid to return to the low pressure side and create tip recirculation is greatly increased. Again, the higher energy level pumps are of major concern (above 200 HP and 650 feet of head per stage).

Q: What are the effects of trimming an impeller on radial vibration?

A: Careful machining of the volute or diffuser tips to increase Gap "B" while maintaining Gap "A" has ben used for a number of years to greatly reduce the vane-passing frequency vibration. The pulsating hydraulic forces acting on the impeller can be reduced by 80 to 85 percent by increasing the radial Gap "B" from 1 percent to 6 percent. There is no loss of overall pump efficiency when the diffuser or volute inlet tips are recessed, contrary to the expectations of many pump designers. The slight efficiency improvement results from the reduction of various energy-consuming phenomena: the high noise level, shock, and vibration caused by vane-passing frequency, and the stall generated at the diffuser inlet.

Table 1 gives recommended dimensions from Dr. Elemer Makay for radial gaps of the pump impeller to casing. Note that if the number of impeller vanes and the number of diffuser/volute vanes are both even, the radial gap must be larger by about 4 percent.

Q: When trimming an impeller from its maximum diameter to adjust the head and flow developed by a centrifugal pump, what is the best way to cut the impeller? Is it best to trim the impeller vanes *and* the shrouds or just the vanes?

A: No hard and fast guidelines for the mechanical aspects of impeller trimming exist, but there are several pump construction and hydraulic design factors to consider while making the decision of what to trim.

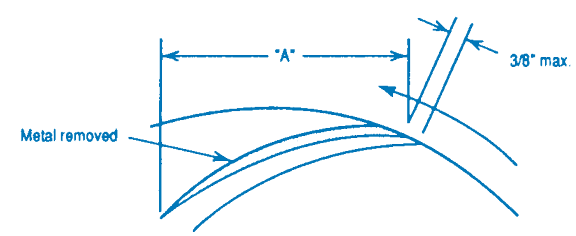
How the impeller is trimmed will greatly influence the hydraulic performance of the pump as well as the vibration levels experienced. You must evaluate the hydraulic characteristics before you decide how to trim the impeller.

For volute type pumps, the entire impeller, vanes and shrouds *may* be cut as shown in Figure 1. However, in some pumps, this method will alter Gap "A" (shroud-to-case clearance), leading to uneven flow distribution at the impeller exit area, which can cause axial vibration and other problems. The double suction impeller type pump is especially sensitive to problems caused by increased Gap "A", so trimming the entire impeller is not a good choice. It is best to cut the impeller vanes obliquely (Figure 2), which leaves the shrouds unchanged or to cut the vanes only (Figure 3). Trimming the vanes only tends to even out the exit flow pattern and reduce recirculation tendencies at the exit area. Gap "A" should be about 0.050 inch (radial) for minimum vibration due to vane-passing frequency.

In most diffuser type pumps, it is best to trim only the vanes (Figure 3) to control tip recirculation and the ill effects of an increased Gap "A". This cut yields a more stable head curve. The uniform flow reduces the tendency for tip recirculation, and the possibility of suction recirculation is greatly reduced at the exit area.

Structural strength of the shrouds is a factor in the decision in how to trim the impeller. There may be too much unsupported shroud left after a major reduction in diameter. The

FIGURE 4. IMPELLER VANE OVERFILING



Length of blend for over filing	
Impeller diameter, in	"A" distance of blend, in
10 & below	1 1/2
10 1/16 through 15	2 1/2
15 1/16 through 20	3 1/2
20 1/16 through 30	5
30 & larger	6

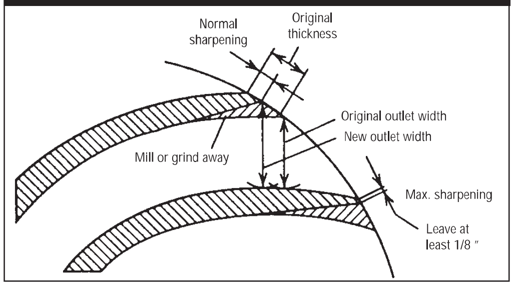
TABLE 1. RECOMMENDED RADIAL GAPS FOR PUMPS

Type of Pump Design	Gap "A"	Gap "B" +/- percentage of impeller radius		
		Minimum	Preferred	Maximum
Diffusers	50 mils	4%	6%	12%
Volute	50 mils	6%	10%	12%

$$*B = \frac{100\,(R^3-R^2)}{R^2}$$

where R^3 = Radius of diffuser of volute inlet
and R2 = Radius of impeller

FIGURE 5. SHARPENING OF IMPELLER VANES



oblique cut leaves the shrouds unchanged and solves the structural strength problem as well as improving the exit flow pattern.

Q: I frequently encounter "vane-passing" frequencies during vibration analysis of a pump. What are some of the methods that can be used to reduce this problem?

A: The most effective method of reducing vane-passing frequencies is to carefully maintain proper Gap "A" and Gap "B" clearances to reduce impeller-casing interaction. Sometimes, impellers manufactured with blunt vane tips cause disturbances in the impeller exit area and in the volute area by generating hydraulic "hammer" even when the impeller O.D. is the correct distance from the cut water (Gap "B"). Corrections can be achieved by two methods:

1. Overfiling: This disturbance may be partly or entirely eliminated by tapering the vanes by "overfiling" or removal of metal on the *leading* face of the vanes (Figure 4). This technique has the additional advantage of restoring the vane exit angle to near that of the maximum impeller design (i.e., before the diameter was reduced).

2. Underfiling: Sharpening the underside of the *trailing* edge of the vane (Figure 5) can enlarge the outlet area of the liquid channel. This will generally result in about five percent more head near the best efficiency point, depending on the outlet vane angle. At least 1/8 inch of vane tip thickness must be left. Sharpening the vanes also improves the efficiency slightly. Where there are high stage pressures, you must sharpen the vanes carefully because the vanes are under high static and dynamic stresses. ■

Ed Nelson is a consultant to the turbomachinery and rotating equipment industries. He serves on the Pumps and Systems Editorial Advisory Board.

Examining Pump Capacity Problems

BY WAYNE C. MICHELETTI

Q: We have a 1,200 gpm centrifugal pump that transfers water from a public reservoir up to our makeup reservoir inside the plant. The change in elevation is about 30 ft over a distance of roughly a mile. The pump does not operate continuously, rather it is turned on and off by plant staff who check the makeup reservoir level once per shift. According to operators, the pump seems to deliver "full flow" when first started, but is operating at a much lower capacity when checked later. What could cause this consistent decline in pump capacity?

A: Your problem could have a couple of causes. One cause might be air that has entered the system and accumulated in the pump. While it is possible for the reservoir water to be saturated with air that will come out of solution in the pump, most centrifugal pumps can handle a small amount of air (2–3% by volume), which will pass through as bubbles with the liquid. Instead, the introduction of air is more likely equipment related. Depending on the pump and piping system, air can get into the water in several ways.

For the pump, check the shaft sleeves to ensure that the seal between the sleeves and the impeller hub is adequate. Then examine the stuffing boxes. For pumps operating with a suction lift, lantern rings should be installed and have seal water under positive pressure.

Piping can be more difficult to examine because most of it will likely be underground. However, any surface piping should be inspected to assure that it is airtight. And the as-built drawings should also be studied to determine if there might be any irregularities (such as improper pitch or high spots) along the pipeline in which air pockets could form.

If air is the cause of reduced pump capacity, this can be confirmed by stopping the pump, opening and closing the vent valve on top of the volute, and immediately restarting the pump (which should run at full capacity). Do not open the vent valve while the pump is operating. Even if air is present in the pump during operation, it will be trapped near the center of the impeller while the heavier water will be forced to the outer edge (and out the vent valve if it is open) (see Ask the Experts, November 1993).

A second possible source of your difficulty is the intake at the public reservoir. From the information presented, the system probably has a submerged offshore intake with some form of screening to prevent the entrainment of unwanted materials. Underwater plants, particularly filamentous grasses, can be drawn into and entangled in the intake screening, blocking flow. When the pump is not operating, the natural underwater currents can clear some or all of the blockage so that full flow is temporarily restored at pump startup.

The intake can also contribute air to the system. If the level of the public reservoir has dropped, the distance between the surface of the water and the submerged intake might not be adequate to prevent the formation of vortices whenever the pump is operating. Such vortices can bring significant amounts of air into the system.

Q: We recently decided to move a relatively old but infrequently used standby service water system pump to an auxiliary cooling water application. The centrifugal pump was rated at 1,000 rpm for a suction lift of 15 ft against an 80 ft total head when running at 750 rpm. Using the rules governing the relation of capacity, head, and speed, we thought it should be possible to obtain 1,400 gpm against a total head of 150 ft by replacing the original motor with a larger, 1,050 rpm motor. However, in its new service, the pump has not provided anywhere near the anticipated capacity. What could be wrong?

A: Switching a pump from one service to another frequently appears to be an easy and cost-effective way of avoiding the purchase of a new pump designed for the desired use. Unfortunately, such switches can be tricky business (as discussed in this column in March, 1993). Yet since almost everyone will be tempted to engineer such a switch at least once during a career, it might be helpful to review key calculations that are needed in an effort to determine what went wrong in this case.

Summarizing the information you provided, we have one pump intended to handle streams of comparable quality (basically cold water) that has been operated with two different motors. Knowing the original design capacity and total head, we can quickly determine the same information for the new application by the following equations:

$$Q_2 = Q_1 \times (N_2/N_1)$$

and

$$H_2 = H_1 \times (N_2/N_1)^2$$

where

Q = pump capacity (gpm)

N = motor speed (rpm)

H = total head (ft)

As you expected, the corresponding capacity and total head for the new motor (at 1,050 rpm) should be:

$$Q_2 = 1{,}000 \times (1{,}050/750) = 1400 \text{ gpm}$$

$$H_2 = 80 \times (1050/750)^2 = 157 \text{ ft}$$

So far so good. According to these calculations, the pump should be able to provide the desired flow against the estimated head. But before a pump can transfer any fluid, the liquid must have enough outside energy to enter the pumping element at the velocity corresponding to the required pump flow rate. For a centrifugal pump, this energy must be great enough to make the fluid flow into the impeller eye with sufficient force to prevent the fluid pressure from dropping below its vapor pressure when passing the inlet vane edge.

This outside energy requirement is known as the Net Positive Suction Head Required or $NPSH_R$. Assuming that your system is at sea level, this value (for the original pump design) can be determined as follows:

barometric pressure (abs.) = 33.9 ft

– vapor pressure of water = 1.1 ft

– suction lift = 15.0 ft

$NPSH_R$ = 17.8 ft

For centrifugal pumps, the $NPSH_R$ can be correlated to pump capacity and motor speed by a value known as the suction specific speed (S) according to the following formula:

$$S = (N \times Q^{0.5})/(NPSH_R)^{0.75}$$

Using this equation and the original pump design data (Q = 1,000 gpm; N = 750 rpm), the value of S is 2,737. Since the suction specific speed is constant for a given pump, this equation can be rearranged to calculate the $NPSH_R$ for the pump's new application (Q = 1,400 gpm; N = 1,050 rpm). The increase in pump capacity and speed mean an increase in the $NPSH_R$ from 17.8 ft to 34.9 ft.

As a result, the conditions of the new application correspond to a suction head as opposed to a suction lift:

barometric pressure (abs.) = 33.9 ft

– vapor pressure of water = 1.1 ft

+ suction lift = 2.1 ft

$NPSH_R$ = 34.9 ft

If the NPSH available (the difference between the absolute suction pressure and the liquid vapor pressure) is reduced below the NPSH required, then the pump capacity is reduced, and the pump is likely to cavitate. Unless you can change the suction conditions for the auxiliary cooling water application, it would be better to buy a new pump than attempt this switch.

Q: As the result of a recently implemented water management program, several of our older, constant-speed centrifugal pumps now provide significantly more water than is required. What is the best approach for operating these pumps at reduced capacity?

A: Congratulations. Many would envy the problem produced by your success in water conservation. Fortunately, there are three well-proven solutions to reducing existing pump capacity.

If the system flow is expected to change frequently or irregularly, new adjustable-speed drives might be in order. For variable-torque applications (such as centrifugal pumps), solid-state AC or DC drives are usually best. They will allow the pump to respond quickly and efficiently to lower (and higher) flow demands, enabling you to conserve energy as well as water. However, electrical adjustable-speed drives can be expensive and should be thoroughly evaluated from an economic perspective for "older" pumps.

A second, more common approach is simply to throttle the discharge. Doing so will introduce a new artificial friction loss component to the head. This will shift the present system–head curve upward to intersect the pump head–capacity curve at a new operating point (corresponding to reduced capacity). It should also reduce the energy requirement slightly.

It is never advisable to throttle the pump suction. This approach (occasionally referred to as operating in the "break") changes the pump head–capacity curve through cavitation. The resulting operation is not only inefficient but potentially damaging to internal pump components.

The third option is more energy-efficient than throttling, but only suitable if a permanent reduction in pump capacity is acceptable. The pump impeller can be cut down, essentially lowering the pump head–capacity curve. However, before trimming an impeller, a number of other factors and resulting implications should be carefully considered. (These were discussed in this column in the January, 1993 issue.) ■

Wayne Micheletti is a water and wastewater consultant and a member of the Pumps and Systems *Editorial Advisory Board.*

Venting Pump Systems

BY MICHAEL D. SMITH

Pumps sometimes suffer damage unnecessarily because they are not 100% full of liquid before they are started. The systems in which they function either are not or cannot be completely vented. A common misconception is that a pump that produces discharge pressure immediately after start-up was sufficiently full of liquid. For some users, this is the working definition of the word "primed."

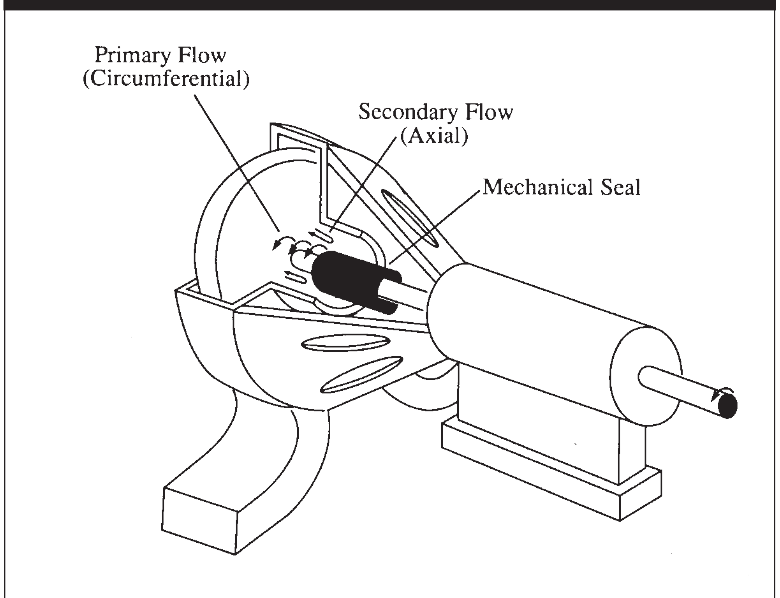
Igor J. Karassik, an internationally recognized authority on pump systems, has written for years about the need to remove all of the gas or vapor from pumps before starting them.

Widespread understanding of the problems trapped gasses can cause developed during 1991 from an effort to understand erosion problems with enlarged, tapered-bore seal chambers used on ANSI B73.1M chemical pumps. Testing, independently performed by and for a number of pump and seal companies, showed that the liquid in the seal chamber circulates around the chamber at a large fraction of shaft rotation speed. A secondary flow was observed heading away from the impeller, along the outside diameter of the seal chamber, and toward the impeller along the shaft. Together, these two flow patterns explained how erosion damage was occurring in a few cases where abrasive solids were present (Figure 1).

An unexpected byproduct of this testing was the realization that gas or vapor that is present in the seal chamber at the time the pump is started can be trapped there for several minutes by these same flow patterns. Worse, trapped vapors or gases tend to accumulate close to the shaft, near the rear of the seal chamber. For most single mechanical seal installations, that is where the seal faces are located.

Here are some common questions on venting pump systems:

FIGURE 1



Primary and secondary flow patterns can result in erosion damage.

Q: Why would gas end up close to the shaft?

A: Imagine the seal chamber is more than half full of liquid as the pump is started. As the pump shaft (and mechanical seal) picks up speed, viscous drag causes the liquid to begin to circulate around the chamber. Soon, centrifugal force overcomes gravity, and the liquid is thrown to the outside of the seal chamber. Any gas is forced inward by the denser liquid.

Q: What problems are caused by the gas?

A: The most common problem is mechanical seal damage. If the gas bubble is big enough to surround the seal faces, it can prevent the liquid in the seal chamber from cooling and lubricating the faces. Large pockets of gas can damage wear rings and bushings, but gas would tend to be swept out of these areas quickly.

Q: My pumps have flooded suctions. Won't they fill completely when I open the suction valve?

A: Probably not! While it is true most most modern pumps are designed to be completely self-venting, there is an assumption that there is someplace for the gas to go as the liquid enters. Unless the discharge valve is opened slightly *and* there is no discharge pressure, the gas has nowhere to go. When a horizontal end-suction pump is installed (or re-installed after a repair), and the suction valve is opened, it will often fill to the top of the suction pipe. When the gas (air, in this case) can no longer escape out the suction pipe, it will compress a small amount in response to the suction pressure. A very large gas pocket remains in the pump at this point, although the pump is probably "primed."

Q: Why has this cause of seal damage remained hidden?

A: A big reason why pockets of gas have not been a concern is that they don't always cause an immediate failure. Seal face damage progresses each time the pump is started while it is not full. Venting is not an issue in many pump starts because the pump was not drained since its last use. If a pump seal fails about once a year, we assume it has a one-year wear life. We don't even consider that it might be failing every third time the pump is started without being 100% full of liquid.

Q: How can the problem be avoided?

A: The operator must understand why it is important to fill the pump completely. The pump system must be capable of being completely vented. When the liquid can be released to the atmosphere, a vent valve is all that may be required. See the sidebar at the end of this article for a procedural solution to a common situation. While discussing system design, it should be noted that the suction line should not have any high points. The suction line should rise continuously either toward the pump or back to the source. If a local high spot is necessary, it will also have to be vented. I have seen many long suction lines that were designed to be level that still had local high spots several pipe diameters above the ends. This can be due to problems with the original installation or the shifting of pipe supports at a later time.

CONCLUSION

Whoever has responsibility for the design of the "system" will need information on the pump, the piping, and the operating conditions to assure that it can be vented. ■

Michael D. Smith is a Senior Consulting Engineer at the DuPont Company in Wilmington, DE.

Venting Your Pumps

It is common to have venting problems when a pump is connected to a system that is pressurized even when the pump is not running. These systems often employ a check valve in addition to a discharge valve. Some users drill a small hole in the check valve flapper to help vent the pump, but this technique is not effective for the most common operating strategies.

The following is a simple procedure that can be used to get more complete venting of these hard-to-vent systems. It assumes the pump is empty of liquid and both suction and discharge valves are closed.

- Open suction valve (pump fills part way).
- Close suction valve.
- Open discharge valve part way (once pressure equalizes, air will rise into discharge piping).
- Open suction valve.
- Start pump.

CAUTION:

The pump seal will be exposed to full discharge pressure using this procedure.

Never start a pump with the suction valve closed.

Installation and Start-Up Troubleshooting

BY JOHN W. DUFOUR AND LYNN C. FULTON

A lot of time and money are spent manufacturing and testing centrifugal pumps and developing purchasing specifications for bidding and selecting them. However, events after leaving the manufacturer may result in a pump that won't perform reliably or deliver the desired hydraulics.

SHIPPING AND HANDLING

Once the pump/driver/baseplate assembly leaves the factory, anything can happen if specific instructions on how it should be shipped, received, stored, and installed are not followed. A document that records what was and must be done, what must be approved and by whom, and when these events should happen is crucial. Without this, work will be missed or duplicated.

Manufacturers prepare products for shipping differently. Some mount pumps in custom-made crates, while others hang the shipping tag on auxiliary piping and bolt two-by-fours to the base. The purchaser should define special requirements. Will the pump be shipped overseas? Is long-term storage required? Is there lifting equipment at the site? These questions must be answered ahead of time.

In all cases, Material Safety Data Sheets should be included during shipping and installation. Everyone who comes in contact with the pump needs to know what's in it.

There are other questions, too. What form of transportation will be used? A dedicated truck or a common carrier? Who will receive the equipment? When? A dedicated truck usually has two drivers driving around the clock, directly from manufacturer to delivery site. This is costly but quick. A common carrier is less expensive but can take longer. For example, pumps from an East Coast manufacturer, destined for Texas, were loaded on a truck Friday afternoon. The pumps arrived 15 days later. With no one to receive them, the driver left them at a warehouse. It took two days to locate them, and they were delivered a week after that.

Pumps are easily damaged during transportation, storage, or installation. Most baseplates are designed to be lifted with an overhead device or moved by fork lift. Care must be taken to prevent damage to auxiliary piping from lifting slings or hooks. Storage facilities often don't have an overhead crane, so a forklift moves the assembly off the truck and around the storage area. Again care must be taken to balance the load before lifting and to avoid bumping or dropping the assembly (falling just an inch can crack the mechanical seal face ring). Never lift the pump by its shaft or auxiliary piping.

STORAGE

Sometimes the pump goes directly from truck to foundation, but the assembly is often stored for a time. Storage may be a graveled yard or a warehouse with overhead lifting equipment and a controlled environment. In any case, following three rules will help avoid problems:

1. Keep oil/grease in the bearings.
2. Keep water/moisture out of the case (seal, windings, etc.).
3. Protect the pump from abuse.

Check the pump over. To prevent baseplate distortion, place it level and out of traffic. See that all cover plates are bolted on. Be sure no auxiliary piping or components were lost or damaged in transit; replacing a part may delay start-up. Bearing housings should be filled with oil to the bottom of the shaft and rotated periodically to keep bearings coated. Document who turns it and when. Pumps stored long-term with oil mist lubrication should be hooked to a portable mist generator. Verify that the mechanical seal sleeve locking collar is tight and that the shaft turns freely.

Stored drivers may require extra care. Heaters on electric motors should be energized to keep windings dry. Steam turbines often have carbon rings and seals. Remove them to prevent corrosion under the rings, or continuously purge the case with dry nitrogen.

INSTALLATION

As mentioned, prepare a document to ensure proper installation. Outline specific requirements, in sequence, for each pump. Define tasks and inspections, who is responsible, and special procedures—grouting plans, cold alignment targets, pre-start-up checks, hot alignment checks, etc.

Vendors often give details on installation, and writings on the subject are available. Here is a list to aid installation:

GROUTING

- Prepare the foundation surface. Chip latence off, exposing aggregate. Remove loose material, grease, and water.
- Level the baseplate using jackbolts bearing on jackplates (Photos 1 and 2). Jackplates should have rounded corners. It's easier to slice sections from round stock than to cut plate.
- Remove pump and motor before installation; it's easier to level the baseplate and pour grout.
- Check the baseplate bottom for cleanliness. Verify that each compartment has grout and vent holes. Drill holes before lifting the baseplate onto the foundation.
- Don't grout around anchor bolts. Baseplates are grouted to provide uniform load distribution. Anchor bolts hold the pump down. To keep anchor bolts free to stretch, install sleeves around bolts.
- Install the baseplate, establishing correct elevation (within 1/8 in.) and pedestal level (within 0.002 in./ft). Some contractors like to put pumps back on the baseplate to shoot the nozzle elevation.

This is unnecessary and may distort the baseplate.

- Coat forms with furniture paste wax to ease removal. Fix forms to the foundation block at different elevations to avoid fracture lines from anchor studs. Drilled holes with screws look better after removing forms and eliminate potential impact cracks from hammering nails or using charged drivers.
- Tape or grease machined mounting surfaces for protection.
- Ensure that grout flows into all compartments by using a head box and vent tubes. The head box can be six-inch sonotubes RTV'd to the baseplate surrounding the pour holes. Vents can be plastic pipe. These should be at least six inches high to provide enough head to get all voids under the baseplate.
- Grout between 60 and 90°F (Photo 3). Cooler temperatures don't allow curing. Higher temperatures may cause fast curing and heat cracks. Grout should harden in 24 hours.
- As soon as the grout firms (not hardens), remove vent pipes and head boxes. Grout consistency should be like hard rubber, making it easy to trim.
- Forms can usually be removed after 48 hours. Remove jackbolts from baseplate and fill holes with RTV.

MOUNTING/ALIGNMENT

- Set pump on its pedestals, center bolts in their holes, and snug. This allows movement if side-to-side motor adjustment can't achieve alignment.
- Mount motor with a minimum of 1/8 in. stainless shims under the feet using the required distance between shaft ends (DBSE). This is usually found on the general arrangement or coupling drawing. With sleeve bearing motors, the magnetic center of the motor with respect to the stator must be determined and the DSBE set with the motor rotor in its magnetic center.
- Make sure the mechanical seal drive collar locking screws are tight, then roll locating cams out of the drive collar. Lock cams out of the way or remove them. Remember that future work will require cams to reset seal compression—don't loose them. The shaft should turn freely.
- Align motor to pump, free of any piping, using, as a minimum, the reverse indicator alignment method. To avoid soft foot, minimize shims under each support area. When alignment is achieved, tighten hold-down bolts and recheck.
- To check for soft foot, place a dial indicator on each mounting foot, then loosen the hold-down bolt. If the reading changes more than .001 in., reshim.

PIPING

Care in fabricating and aligning piping avoids problems that may require recutting, fitting, rewelding, and retesting the pipe or lead to premature pump failure. Good system design supports piping loads and forces along spring hangers and bracing that don't have to be removed during normal maintenance. The system should be fabricated starting at the pump flanges, working toward the pipe rack, using temporary braces/supports to avoid pump strain.

The most common piping fabrication error, producing the largest piping strain, is nonparallel flange faces. A feeler gauge helps detect this. If you see a difference in two facing flange planes, piping strain will result. For example, during installation of circulating water pumps in a refinery, suction piping was forced to the pump flange without checking for non-parallel faces. The resulting strain distorted the casing to the point where the shaft and impeller would not turn. Fortunately, no serious damage occurred. The cases were reclaimed after the piping was aligned and supported.

- After fabrication and pipe testing, remove temporary bracing and lock-pins from spring hangers and check strain.
- Remove flange covers and inspect the pump for debris. Clean out the case. Bring the piping to the pump flanges. Flange holes should drop through with no binding.
- Place dial indicators to monitor vertical and horizontal movement of pump shaft relative to driver shaft. Make up suction and discharge flanges separately, continuously observing indicator readings. If movement exceeds 0.001 in., piping strain is excessive. Readjust pipe, retighten, and retest.

PREOPERATIONAL CHECKS

The period from installation until full operation may be the most important phase of pump life. It's filled with activity and riddled with pitfalls that can complicate start-up and prevent establishing a reliable system. The rules above also apply here:

1. Keep bearings lubricated.
2. Keep moisture out of the case.
3. Protect the pump.

Drain and flush bearing housings with clean oil. Oil rings may have moved during handling, so look through the vent caps to verify that they're in position. On oil mist installations see that mist reclassifiers have been installed correctly. Directed oil mist fittings have a "V" at the orifice. This must be pointed towards the bearing. Insure that all mist lines are sloped so no low points cause liquid buildup and block flow.

Greased bearings should be repacked with the correct grease. Make sure all old grease is displaced by new. Different greases (lithium- vs. soda-based) have incompatible additives. Mixing two greases can give an inferior blend.

Bump check motors for proper rotation. Do not attempt this while the motor is coupled to the pump. Reverse rotation can cause the impeller to loosen or come off the shaft. If rotation is correct, run the motor alone for at least one half hour. Monitor bearing temperatures and motor bearing housing vibration. This should reveal any major problems. Most others will not be revealed until it is loaded and generating heat. Install the coupling spacer and guard and verify smooth assembly rotation.

Don't overlook small steam turbine drivers. Verify rotation direction by inspecting nozzle orientation. As soon as steam is available, check the operation of the governor and overspeed systems. Run the turbine solo at least one half hour.

Mechanical seals and bearings are easily destroyed during initial start-up on hot pumps where water is circulated for cooling. Install pressure gauges, temperature indicators, and valves so water flow can be regulated and adjusted. Throttling valves are typically installed on parallel outlet lines to adjust flow to each pump skid area. It's a good idea to flow most cooling water through the seal and bearing coolers initially.

Debris in the pump and seals is a problem during initial start-up. A welding rod lodged in an impeller eye can seize the pump. To prevent this, use suction screens. Insert temporary stainers if they're not built in. Pressure gauges on both sides of the strainer indicate when it is plugging.

INITIAL OPERATION

The electronic equipment adage, "If it works the first hundred hours it should work a lifetime," also applies to centrifugal pumps. Knowledge of the equipment and the system it will operate in are key to successful start-up. While each installation is different, this general procedure will help prevent problems:

- Close case drains and vents. Slowly open the suction line. Look for leaks at the case flanges, seal area, seal piping, drain piping, and inlet discharge piping. If there are leaks, return to a safe situation and repair them. If leaks occur around the shaft, determine if seal faces are leaking or if the leak is under the seal sleeve. Stop leaks between sleeve and shaft by adjusting the drive collar. Stop leaks around the seal flange by retorquing the bolting to clamp the stationary gasketing. It's important to know where the leak is before pulling the pump apart.

Good system design supports piping loads and forces along spring hangers and bracing that don't have to be removed during normal maintenance.

- If no leaks are seen, open the inlet valve 100%. Vent areas of the system that don't self-vent. Crack the discharge valve. Start-up horsepower is minimum to the left hand side of the pump curve. On systems pumping higher specific gravity liquids at start-up than during normal operation (typical of cold start-ups), the discharge system may have to be throttled to avoid motor overloading. Throttling to 50% BEP is acceptable in most cases, but more than that may cause seal problems.
- Start the pump. Slowly open the discharge valve. If a discharge control valve is installed and on automatic, the control valve will be wide open until the block valve opens enough for the control system to take over. If the pump cavitates, there may be too much flow. Start to pinch down on the discharge valving, preferentially using the control valve. Most systems have a flow meter. Flow can sometimes be determined from the meter directly, and the differential across the pump can be determined using pressure gauges on the pump. Using flow and differential head, determine where the pump is operating on the curve. Low flow, high head may indicate running too far back, leading to bearing or seal problems. High flow, low differential head means the pump is running out on its curve and could cavitate. Check differential across the inlet screen and use a spare pump before low suction pressure causes cavitation.
- Where flow can't be measured directly with a meter, estimate it using motor current, horsepower requirement of the pump, and plotting that point straight up to the performance curve. The intersection of the vertical line from the horsepower curve to the performance curve should be the capacity point as long as specific gravity is similar to horsepower curve specific gravity. Differential head on the pump should be similar to differential head on the curve at the capacity point determined from the horsepower calculation.

From Figure 1:

Example point 1.

$$\text{TDH} = 2.31\,\frac{P_2 - P_1}{\text{S.G.}} = 188 \text{ ft}$$

where

P_1 = 3 psig

P_2 = 60 psig

S.G. = 0.7

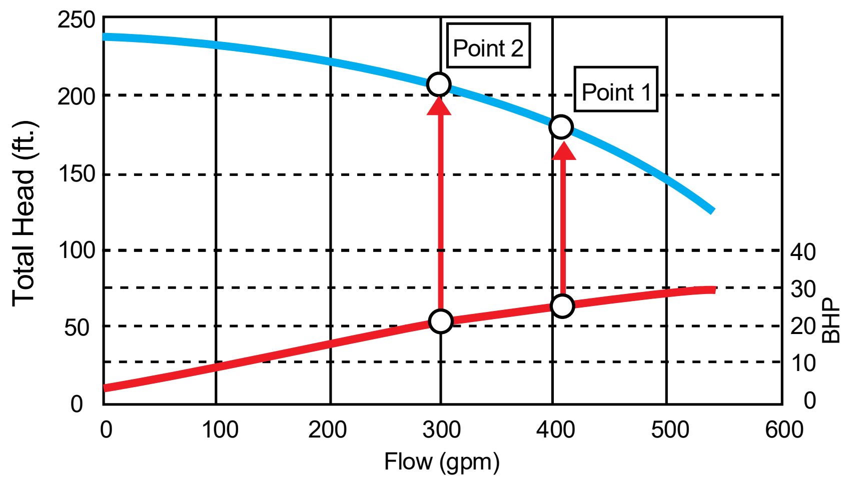
$$\text{M.H.O.} = \frac{\sqrt{3} \times I \times V \times \eta}{746} = 26.5 \text{ Hp}$$

where

M.H.O. = motor horsepower output

V = motor voltage = 460 V (30 Hp motor, 3ϕ, 460 V [line to line])

FIGURE 1



Head and BHP vs. Flow. Operating point 1: using the BHP vs. flow curve with horsepower calculation derived from measurement of current with voltage assumed to be 460 V, flow is found to be 405 gpm. The calculated 188 ft based on pressure differential confirms flow to be 405 gpm. Operating point 2: similar calculations for horsepower and head at operating point 2 also confirm the calculation method. See text under "Initial Operation" for calculations.

η = motor efficiency = 90% (2-pole motor, 90% efficiency)

I = phase amp measurement = 27.2 amps

Example point 2.

Throttling pump discharge

$$TDH = 2.31 \frac{P_2 - P_1}{S.G.} = 214.5 \text{ ft}$$

where

P_1 = 3 psig

P_2 = 68 psig

I = 20.8 amps

$$M.H.O. = \frac{\sqrt{3} \times I \times V \times \eta}{746} = 20 \text{ Hp}$$

where

M.H.O. = motor horsepower output

V = motor voltage = 460 V (30 Hp motor, 3ϕ, 460 V [line to line])

η = motor efficiency = 90% (2-pole motor, 90% efficiency)

I = phase amp measurement = 27.2 amps

- Check motor and pump vibration. Vibration levels should be below 0.15 in./sec. Most new equipment vibrates less than 0.1 in./sec. true peak.
- Compile documents for each pump and file them for reference. ■

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Upgrading Utility and Process Pumps

Improvements to make your equipment better than new.

BY KURT SCHUMANN

Upgrade (up + grade, v.): *to raise the grade of; to raise the quality of a manufactured product* (Webster's Third New International Dictionary).

A pump upgrade (also called a revamp or retrofit) involves changing mechanical or hydraulic design or materials to solve a problem or increase reliable run time. An upgrade is different than a repair, which attempts to duplicate original construction and design, whereas an upgrade improves the design beyond the original.

Rerates are a type of hydraulic upgrade, usually involving a change in pump head capacity. Repowering may involve repairs and/or upgrades. Philosophically, repowering is different from normal pump maintenance because the plant being repowered has decided to spend capital monies to extend the plant's useful life. Plants being repowered are candidates for pump upgrades because they are expected to run reliably with high capacity factors and can justify the additional cost (above and beyond normal repairs) to upgrade pumps.

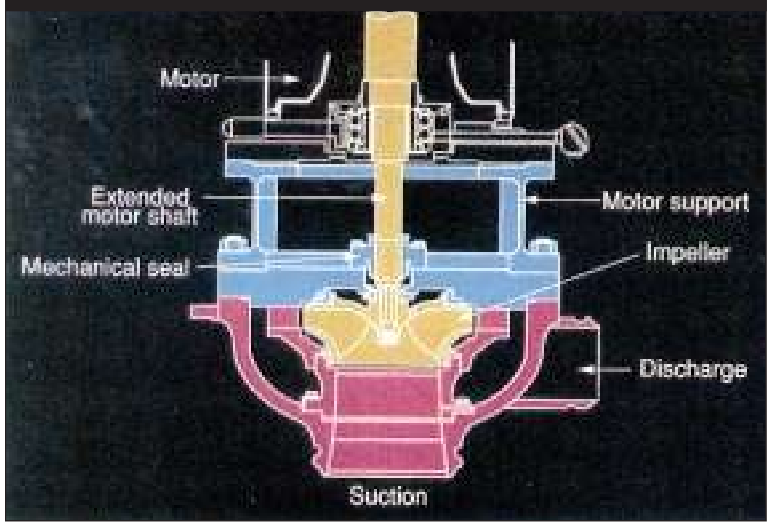
Pump upgrade goals include:

- decreasing plant operations and maintenance expenses
- increasing mean time between failures (MTBF)
- increasing pump and plant availability
- increasing pump efficiency
- complying with the latest legislative mandates (such as the Clean Air Act Amendments of 1990)
- minimizing the risk of fire or other safety hazards
- eliminating hazardous materials

Pump upgrades can be divided into major categories:

- mechanical design
- hydraulic design
- material
- ancillary/system

FIGURE 1

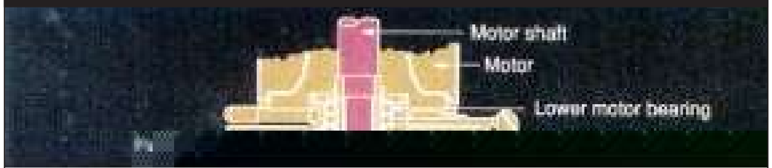


Original residual heat removal pumps in safety service at nuclear power plants. The design has high maintenance hours and exposure dosage due to short mechanical seal life and an overhung shaft design that makes seal maintenance difficult.

THE UPGRADE PROCESS

To identify upgrade candidates, pump users should review maintenance records to see which pumps were responsible for a disproportionate share of expenses or caused safety or reliability concerns. Once these pumps are identified, work with the upgrade supplier to identify and evaluate upgrades available for your particular pump. Provide the supplier

FIGURE 2



An upgrade of Figure 1. Spacer coupling between pump and motor allows seal access without disassembling the pump. A bearing above the seal limits shaft deflection. Conversion from non-cartridge mechanical seal to cartridge-type seal eases assembly. Benefits: increased seal life, decreased leakage, and decreased personnel exposure while changing seals.

with a maintenance history so problem areas can be addressed.

Plan for a future outage where the plant or process will be shut down long enough to complete design and hardware changes. This is important—upgrades take time to engineer and implement, and they must be planned in advance based on repair or outage schedules. This approach can also save money; upgrading worn out parts during the normal repair cycle (instead of replacing components that still have life left) minimizes incremental cost.

The following are upgrade examples from each of the areas above:

Mechanical design upgrades:

- install a stiffer shaft/rotor to reduce vibration
- modify structural elements to remove natural frequencies from the range of pump forcing frequencies (rotational frequency, blade pass frequency, etc.)
- eliminate threads (a source of breakage) on pump shafts
- modify components to make assembly/disassembly easier
- convert mechanical seals to further restrict or eliminate leakage

Hydraulic design upgrades:

- redesign first stage impellers to reduce cavitation damage
- redesign impellers to lower vibration for part load/peaking operation
- control “A” and “B” gaps to reduce pressure pulsations and vibration
- improve efficiency
- optimize blade number to reduce pressure pulsations and vibration
- increase pump head capacity to meet system requirements

Material upgrades:

- eliminate asbestos, an environmental hazard
- install impellers made of cavitation-resistant materials for longer life
- use hardened wear parts to increase MTBF
- eliminate leaded bronzes because of environmental problems with lead
- improve product-lubricated bearing materials
- change materials for seawater use
  - replace Monel components with austenitic stainless steel
  - replace nickel-aluminum-bronze parts with austenitic stainless steel
  - use other special alloys for critical parts
  - install non-metallic bearings

Ancillary/systems upgrades:

- install vibration monitoring and recording instrumentation
- improve lube oil system and instrumentation
- modify seal injection
- perform pump intake scale model testing

The following are examples of upgrades to improve pump operation:

RHR PUMP COUPLING MODIFICATION

These pumps are residual heat removal pumps, close coupled design in safety service at nuclear power plants (Figure 1). The original design resulted in considerable time, expense, and man-rem exposure for normal maintenance activities like seal change-out and motor thrust bearing replacement. There was also a high risk of damaged equipment from the difficulties of rigging in cramped quarters. Pump upgrade kits add a bearing and a spacer coupling. Installation of these kits allow seal removal without pump disassembly.

The original design frequently resulted in seal change-out times longer than the 72 hours permitted by most plant safety evaluations. The upgrade easily accommodates a seal or motor bearing change in 72 hours, without the high man-rem dosage involved in pump casing disassembly. This reduction in personnel exposure is an important benefit in any case, but it is especially so given the increased industry focus on the issue.

These coupling modifications have been supplied to several utility companies as bolt-on hardware kits installed during short outages (1 or 2 weeks). Other items like oil drain location and mechanical seal venting have also been improved in the design.

BOILER FEED PUMPS

Boiler feed pumps are at the heart of most power plants, and economical plant operation depends on reliable pump operation. Many pumps from the utility building boom of the 1950s–70s had larger capacities

overlap can be modified to current standards. By evaluating the cost of these modifications in light of expected benefits, users can choose the modification required to meet process requirements with the least cost.

6. Improved first-stage impeller inlet designs expand the stable operating range to lower flow rates without cavitation damage, vibration, or pressure pulsations. Material upgrades for first stage impeller service resist cavitation damage while maintaining ductility, corrosion resistance, and weld repairability.
7. Dry couplings (like the flexible disc or diaphragm-type) eliminate the need for periodic lubrication and the associated chance of failure. They are often lighter than gear couplings, resulting in lower vibration, and they tolerate more misalignment than gear-type couplings.
8. Instrumentation can be added to monitor and protect pumps. Possibilities range from simple vibration/temperature switches to complete monitoring of all key operation variables, including remote monitoring, diagnostics, etc. The most common items monitored include:
   a. shaft or bearing cap vibration
   b. lube oil temperature and pressure
   c. axial and radial bearing temperature
   d. casing (or barrel) temperature

Additional items include:

   e. pump suction condition (pressure, temperature)
   f. pump discharge condition (pressure, temperature)
   g. pump flow rate
   h. horsepower, efficiency
   i. balance drum leak off (temperature, flow rate)
   j. seal (drain temperature, mechanical seal face temperature, stuffing box temperature, etc.)

Monitoring can be stand-alone or can feed into the plant's control system.

9. Some older pumps have open vane diffusors. Vanes can fatigue due to unsteady hydraulic loads. Using a shrouded diffusor eliminates breakage problems and allows improved alignment.
10. Improved bearing designs are available, including ”high stability“ designs to eliminate half frequency (”oil whirl“) problems. Special attention is paid to individual plant operating modes (low-speed operation, turning gear, etc.) in recommending a particular bearing design.

CIRCULATING WATER PUMPS

Circulating water pump maintenance requirements vary greatly, depending on whether the pumps are used in freshwater or seawater.

For most freshwater applications, typical problems requiring pump maintenance are excessive vibration and premature bearing wear.

Vibrations can be analyzed using modal analysis or standard spectrum analysis techniques to identify the root cause of the vibration. If necessary, a finite element analysis (FEA) model of the pump can be built and correlated to the field data to verify the cause. It can also be used to help redesign the pump.

To improve bearing and sleeve life, upgrades are available to increase wear resistance through material selection and hardcoating.

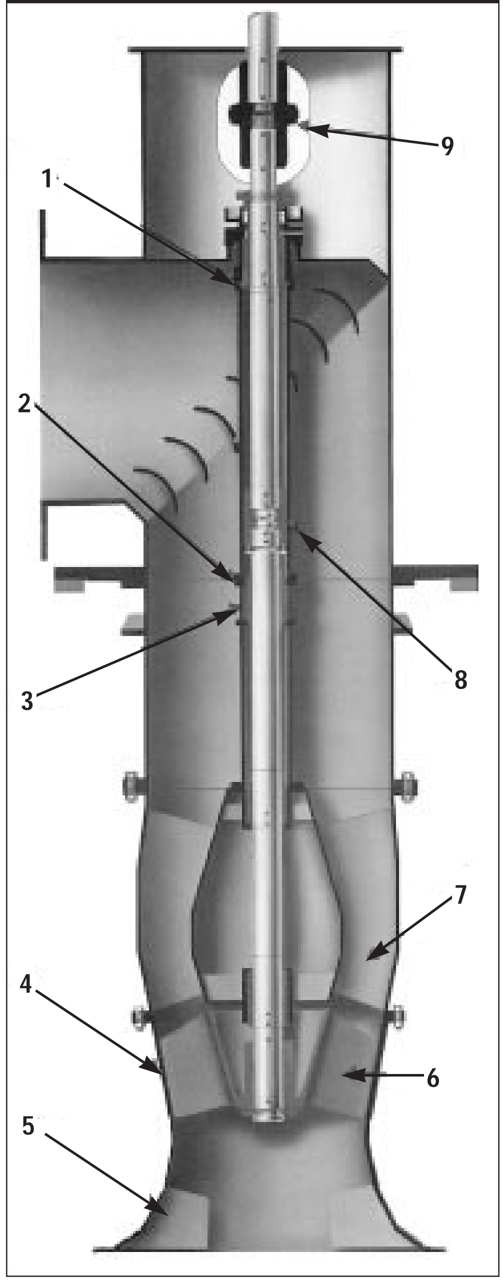
Circulating water pumps in seawater face additional problems due to corrosion. Material selection is critical, and the selection process must consider general corrosion as well as velocity effects, galvanic compatibility, and pitting resistance, plus manufacturability and cost.

The cost difference between materials can be significant because these pumps are large; care must be taken not to over-specify materials and inflate the price of equipment for marginal benefits. In some cases, lower-cost material may be more reliable. For example, 316 stainless steel has better pitting resistance than Monel in seawater, yet Monel is more expensive.

Upgrades for circulating water pumps include (Figure 4):

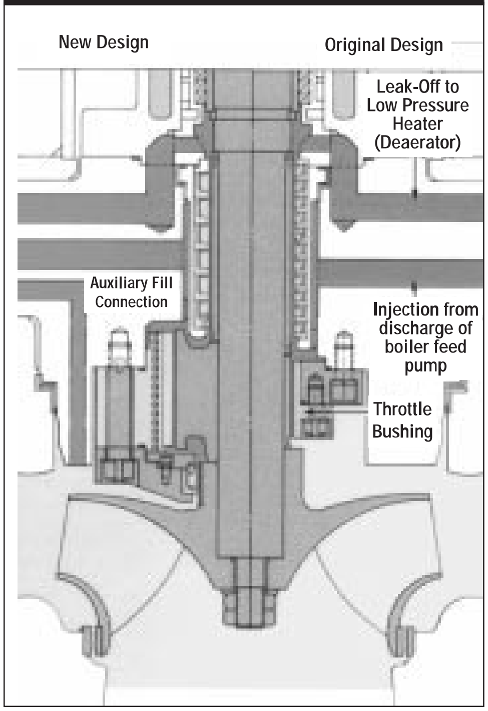
1. An inner column stop on pull-out style pumps to hold down the pump element during starting, stopping, unit trips, and other transients.
2. A flanged inner column with rabbet fits replaces screwed inner columns, resulting in better bearing alignment and easier disassembly.

FIGURE 4



Circulating water pump upgrades. See text under ”Circulating Water Pumps“ for details.

FIGURE 5



Right: Original boiler circulating pump design. The throttle bushing and throttle sleeve wear quickly, reducing floating seal ring life and shortening service intervals between pump rebuilds. Left: The upgraded design. The retrofit incorporates a water-lubricated carbon bearing and redesigned floating seals.

3. Bearing spiders provide stiffer bearing support.
4. Shroud metallurgy upgraded in high-velocity areas eliminates erosion and corrosion damage and extends efficient pump life.
5. Inlet bell modifications lower the required submergence and reduce vortexing. Intake studies can be performed to correct vortexing and other inlet problems and give uniform flows to the pump, resulting in stable operation.
6. Modified impellers optimize cooling water flow, increase plant output, or save pumping horsepower. Upgraded impeller materials resist erosion, corrosion, and cavitation damage.
7. Erosion in iron casing vanes can be repaired. Coatings can be applied to extend casing life.
8. Keyed shaft coupling improves shaft alignment and eliminates problems associated with removing threaded couplings. Shafts can usually be re-machined and re-used.
9. Rabbet-fit drive couplings replace body-bound bolt couplings and improve alignment repeatability.

BOILER CIRCULATING PUMPS

Boiler circulating pumps (BCPs) are in a particularly severe duty, handling 600°F water at over 3,000 psig.

Many of the original pumps supplied in the 1950s and 60s exhibited less-than-desirable life spans. In light of this, an upgrade program was developed that:

- adds a graphite-impregnated bearing
- improves the primary sealing device
- incorporates other reliability "lessons learned" (Figure 5)

To date, more than 200 BCPs have been upgraded to this new design, and the resulting MTBF is typically two to three times that of the pumps before upgrading.

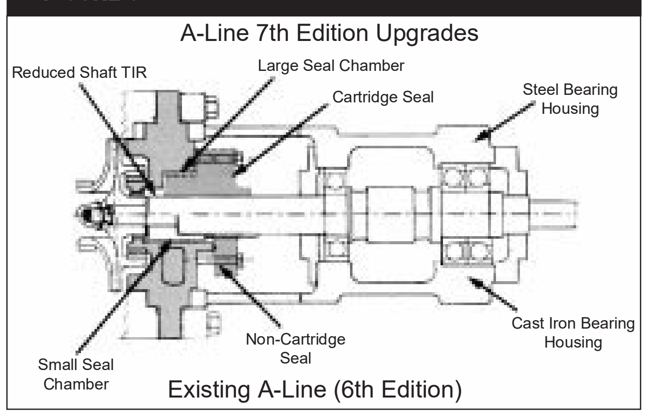
API PROCESS PUMPS

Process pumps may handle hazardous materials, and as a result seal leakage is critical. Industry standards (API 610 7th Edition), as well as federal legislation like the Clean Air Act, address mechanical seal reliability and pump maintenance.

There are upgrades available for process pumps to improve seal reliability and reduce emissions (Figure 6), including:

- using a heavy-duty rotor (shaft and bearings)
- enlarging the stuffing box bore for better seal cooling
- using a heavy-duty cartridge seal (single, double, or tandem)

FIGURE 6



Pump modification kits upgrade API 5th and 6th edition process pumps with the features of the 7th edition

Another option, if extremely low levels of emissions are required (for instance, pumping benzene, a carcinogen), is to use sealless (magnetic drive) technology. This can be accomplished by repowering (re-using the casing, bedplate, and driver, along with upgraded pump internals) or replacing the whole pump.

SUMMARY

These descriptions cover some typical upgrades. This article focused on specific types of pumps, but upgrades are available for most models and sizes. Pump companies are useful resources for aid in problem solving. They are usually anxious to apply new technology and gain field experience with new designs and materials. Most upgrade suppliers can customize upgrades for individual users. Review your maintenance problems and discuss them with your pump supplier.

Pump upgrades are a cost-effective way to improve plant performance within budget constraints. When upgrades are properly performed, an upgraded pump may well be "better than new." ■

Kurt Schumann is Manager of Pump Upgrades for the Engineered Pump Group of the Ingersoll-Dresser Pump Company, located in Phillipsburg, NJ. He has 18 years of experience in design engineering and field service of utility and process pumps.

Variable Speed Pumping

Variable speed pumping can save you money if you select and use systems wisely.

BY STEPHEN MURPHY

Most users operate their centrifugal pumps at a fixed speed and accomplish any required changes of flow by using a throttling valve. This practice is much like driving an automobile with the accelerator fully depressed and changing speed by stepping on the brake!

There is a better way to drive an automobile and there is a better way to accomplish variable flow for a centrifugal pump. Variable speed motors and associated electronic drives can be used to adjust pump speed to produce exactly the desired flow and head. By varying the speed of the pump, users can enhance performance, save energy, eliminate the need for throttling valves and reduce inputs of heat to the pumped liquid.

But to achieve these advantages, you must properly select the components of a variable speed system. And proper selection requires a thorough understanding of pump, motor and driver designs for variable speed operation.

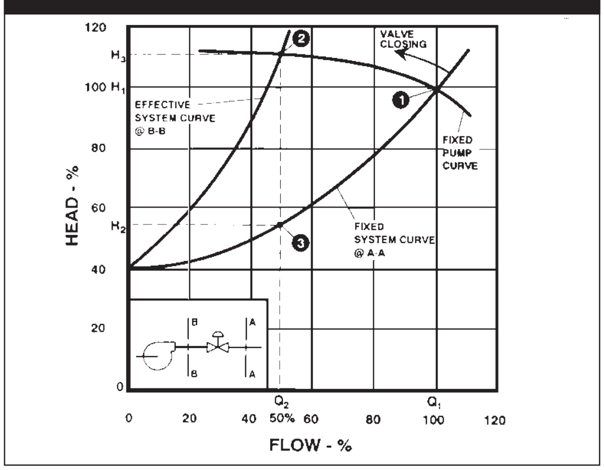
BEHAVIOR OF VARIABLE SPEED PUMPS

A good place to begin a discussion of variable speed pumping is the interaction between variable speed pumps and the fluid handling system. These interactions are different from those of a fixed speed pump.

For a fixed speed pump with flow controlled by a throttling valve, process demand depends on system back pressure and piping resistance, as shown by a fixed system curve (Figure 1). Pump performance is also represented by a fixed curve. With the discharge throttling valve fully opened, the pump seeks equilibrium with the system (point 1 in Figure 1: flow = Q_1 and head = H_1).

To change the flow to Q_2, the throttling valve is partially closed, changing the steepness of the system curve as seen at a point between the pump and the valve (at B-B in Figure 1). Closing the valve causes the pump to "run back" on its curve to point 2, producing flow Q_2 as desired. The pump, which can only operate on its fixed curve, produces head H_3 at point B-B. The pump thus produces H_3 at Q_2 but only H_2 at Q_2 is delivered to the system. The additional head (H_3 - H_2) is wasted across the valve in the form of heat and noise.

FIGURE 1



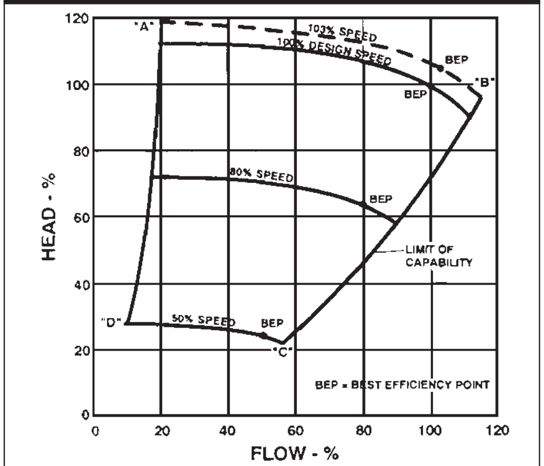
Fixed speed centrifugal pump operation

For a variable speed pump, flow is changed by varying speed. The variable speed pump retains its characteristic performance curve shape, changing flow and head in accordance with the well-known affinity laws (Figure 2). With varying speeds, pumps have wide rangeability and thus any head-flow combination within the envelope can be achieved. And with appropriate precautions, pumps can be operated at even higher or lower speeds than those shown on the curve.

The shape of the system curve influences the amount that flow will change with a change in speed. Flow is proportional to speed if no static lift exists but not proportional to speed if static lift exists (Figure 3). In systems with static lift, a minimum speed exists below which the pump will produce no flow.

Such behavior does not violate the affinity laws. It simply reflects the interaction of the shape of the system curve with those laws. In fact, it's this interaction that makes variable speed pumping advantageous (which also illustrates that users must understand these interactions).

FIGURE 2



Variable speed centrifugal pump operation

BENEFITS OF VARIABLE SPEED PUMPING

Because variable speed pumps can produce a desired head and flow over a broad range of hydraulic conditions, users do not have to be as certain of required flow when they select a pump. Instead of finding the exact fixed speed pump for the job, they can install a variable speed pump and adjust the speed to produce the exact conditions they require.

For example, one user required Pump A to produce 125 gpm flow at 2500 ft head in an upset condition and 100 gpm at 1500 ft under normal con-ditions and Pump B for a 125 gpm flow at 1500 ft head under normal conditions. The user needed an installed spare for each pump, for a total of four pumps. But by specifying variable speed pumps, the user required only three pumps: one for each duty level and a single spare which was valved to allow operation under either condition. Further savings were achieved for the main pumps since identical pumps were used (desired conditions were met by varying the speed). Parts were interchangeable and significantly less energy was required when running Pump A at the normal (i.e., low-head) condition.

In addition to covering a wide range of conditions, variable speed pumping can also eliminate the need for multiple stages. With increased speed, centrifugal pumps produce increased head and flow.

As mentioned above, variable speed pumping can also eliminate the need for a throttling valve. Also, bypass valves may no longer be necessary since minimal flow requirements for stable operation decrease with speed. Elimination of valves can reduce capital expense, maintenance costs, risk of leakage and pressure losses (pressure drop across the valve often accounts for 10 percent of total pressure rise required).

One user saved $20,000 by converting to variable speed pumping in an application involving injection of water into the combustion chamber of gas turbine engines. Since the system curve had relatively little static lift, the pump could be slowed to produce only the desired flow and head and still maintain good efficiency. A change from a fixed speed pump with throttling valve and bypass valve to variable speed eliminated the two valves, reduced the power requirement of the system from 100 hp to 75 hp and made the assembled skid of equipment smaller.

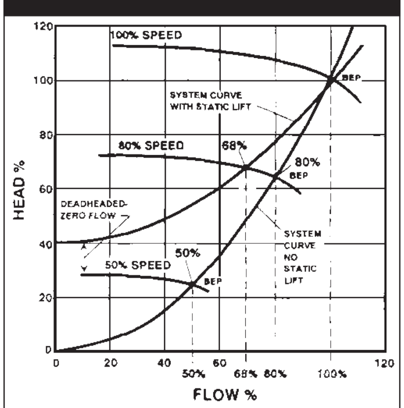
Dramatic power savings are available because of reduced head and flow points due to changing speed rather than by dis-charge throttling (Figure 4). For instance, by achieving 60 percent of design flow and head through variable speed, users can save 50 to 80 percent on energy costs compared to fixed speed pumping with a throttling valve.

Another advantage variable speed pumping offers is reduced heat to the pumped fluid. At constant speed, efficiency falls with reduced flow rate. The result of hydraulic inefficiencies is heat rise in the fluid. But variable speed pumps remain efficient at low flows (i.e., low speeds). Furthermore, horsepower levels are lower at low speeds, which means that heat input to the fluid is kept minimal. Variable speed pumping can thus be advantageous for light hydrocarbon and other volatile fluid applications.

SELECTING THE RIGHT SIZE PUMP

Like any pumping application, variable speed pumping requires proper sizing of pumps. But unlike constant speed pumps, variable speed pumps are not selected for a single design point. To select the correct size pump, you should construct the desired head versus flow range for all anticipated specific gravities. Then be sure to specify a pump that can cover that range (Figure 5 shows a pump that cannot reach point B).

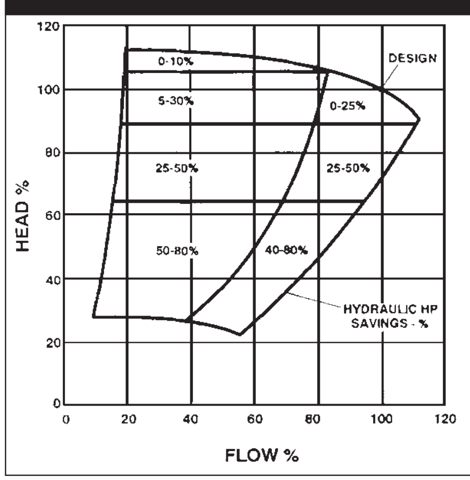
FIGURE 3



FLOW PROPORTIONAL TO SPEED - NO STATIC LIFT

Effects of changing speed of a centrifugal pump

FIGURE 4



Hydraulic HP savings for a centrifugal pump

You may need to specify a "fictitious" 100% speed point to ensure the pump has adequate range (Figure 6).

You must also ensure that NPSHA and motor horsepower are adequate for all combinations of flow and speed. NPSHR and efficiency vary approximately as the square of the speed (Figure 7). Since NPSHR increases with speed, in-ducers may be required to reduce NPSHR to available levels. Bearing loads and other pump characteristics must also be carefully examined.

MOTOR-VARIABLE FREQUENCY DRIVE BEHAVIOR

One of the most common methods of changing motor speed is the AC Variable Frequency Drive (VFD). VFDs are designed to take advantage of the fact that speed, torque and horsepower of an AC motor are all related to the frequency and voltage of the electric power supply:

$$\text{Nominal speed} \quad \frac{2 \times \text{hz} \times 60}{\text{\# of Poles}}$$

$$\text{Torque Capability} = F(\text{volts/hz})$$

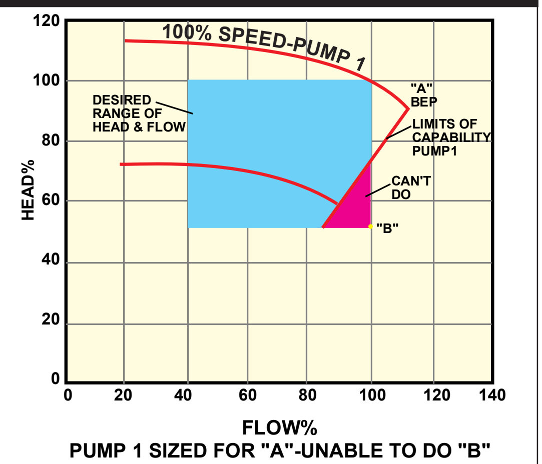
$$\text{HP Capability} = f(\text{Torque} \times \text{Speed})$$

VFDs convert incoming AC electrical power to DC then invert the DC power into variable frequency and voltage AC power. A number of technologies are available to switch the DC power through semiconductors to achieve the desired voltage or current pulses. The technologies differ in their ability to create optimal waveforms. Because the motor's torque and torque ripple are determined by the current, the VFD affect motor and pump operation. Thus, by knowing the characteristics of the VFD output, you can select a VFD suitable for your pump.

Most VFDs produce a constant volt/hz ratio, thus constant motor torque capability up to name-plate frequency (typically 60 hz or 3550 rpm for a two-pole motor — see Figure 8). Horsepower capability therefore rises from zero at zero speed to full horsepower at nameplate speed. Above nameplate speed, the VFD cannot provide increasing voltage, so torque falls due to the falling volts/hz ratio. Horsepower capability, however, remains constant since speed is increasing. Electrically, induction motors can be run at approximately 90 hz in this configuration. But mechanical constraints may limit the safe running speed to well below 90 hz.

VFDs can be used to provide extra motor horsepower above 60 hz. Recall that motor torque capability is proportional to the volts/hz ratio. If a motor is designed for a given volts/hz ratio, and that ratio can be maintained at a higher speed, torque capability will be constant. This technique can frequently be used with standard motors which are commonly wound for either 230 V or 460 V at 60 hz. By connecting for 230 V at 60 hz and operating to 460 V at 120 hz, both motor and horsepower capability and speed are doubled. Be sure to check with the motor manufacturer before using this technique. The motor may not have the thermal capacity or mechanical integrity to run at speeds considerably above 60 hz. Also, the motor may not be properly matched electrically to the VFD.

FIGURE 5



Improper sizing to meet required duty points

SELECTING THE MOTOR

VFDs are most frequently used with the familiar NEMA B squirrel cage AC induction motors. Some special considerations for selecting motors for use with VFDs include cooling, efficiency and operation in hazardous (e.g., explosive) environments.

Motors operated on VFDs operate at higher temperatures due to the irregular shape of the electrical waveforms produced by the VFD. To ensure that the motor will not overheat, the motors are typically derated at full load from 3 to 10 percent, depending on the type of VFD used.

This additional heat makes motors operated on VFDs less efficient than when operated across the line. Thus, many users specify high efficiency motors for use with VFDs. High efficiency is not a requirement, but the extra copper and other features are advantageous for VFD use.

Increased heat can lead to environmental hazards. Motors proposed for use in hazardous (e.g., explosive) environments must be designed differently or derated. The skin temperature of a standard motor operating on a VFD could exceed an area gas autoignition temperature at nameplate horsepower. Motors nameplated for use in Class I, Division I, Groups C and D environments, for example, are available for VFD use but must generally be purchased with a "matched" VFD from a single supplier.

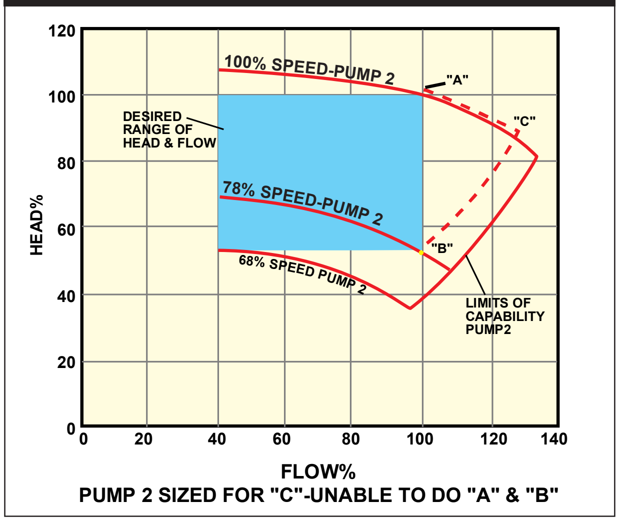
SELECTING A VFD

Important factors for selecting VFDs include power supply voltage and frequency, amperage requirements, torque requirements and motor and load characteristics.

VFDs must be selected to match the power supply and frequency. Many VFDs are switch selectable for a number of voltage/frequency combinations.

You can determine the amperage requirement of a motor using the equation:

FIGURE 6



Proper sizing to meet required duty points

$$\text{AMPS} = \frac{\text{HP} \times 746}{\text{Volts} \times 1.732 \times \text{Motor Efficiency} \times \text{Motor Power Factor}}$$

Nominal horsepower ratings are usually given by the VFD vendors but in some instances a VFD will only produce the stated nominal horsepower if a high efficiency motor is used. Unlike motors, VFDs generally have no continuous service factor. Momentary overloads, however, are permitted. VFDs generally exceed 97 percent efficiency at full load.

VFDs are designated constant torque or variable torque, depending on their current overload capacity. Variable torque VFDs can produce 110 percent of rated current for one minute. Constant torque VFDs can produce 150 percent of full load current for one minute and even more for shorter periods. Variable torque VFDs are generally used for centrifugal pumps.

VFDs must be matched to the load and motor characteristics. Certain VFDs, known as Current Source Inverters or CSIs, may require addition or deletion of capacitor banks to match the load and motor characteristics. The more commonly used Pulse Width Modulation and Six Step VFDs do not require this matching. They are suitable for a wide variety of motors. Most VFDs operate on 480 V input and produce a maximum of 480 V output. If a higher voltage motor is desired, you can install a step-up transformer between the VFD and the motor or use a higher voltage VFD.

APPLICATION CONSIDERATIONS

Be sure the motor will be capable of delivering enough torque to the pump. Motor torque capability (including breakaway or start-up torque) must exceed pump torque required at every speed. Generally, if the motor and VFD are properly sized for 100 percent speed, they will be adequate at lower speeds. However, in certain instances, such as applications with high suction pressure, motor and VFD sizing may be governed by start-up conditions. VFDs on positive displacement pumps must routinely be oversized to provide sufficient start-up torque.

Avoid lateral critical speeds. As an example, API Specification 610 states that depending on the unbalanced response amplification factor, a pump may not be operated between 85 percent and 105 percent of its critical speed. Adherence to these rules can block out a large portion of the allowable performance envelope of the variable speed pump (Figure 9). Fortunately, many pumps are of a stiff staff design and will operate below their first lateral critical speed. A vendor may be able to change the mechanical design to raise or lower the critical speed to provide full range speed adjustment.

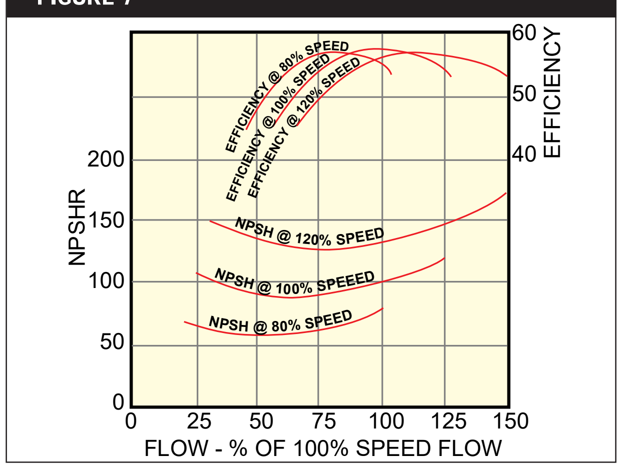
Be aware of torsional critical speed. Torsional critical speeds are resonant frequencies at which motor and driven equipment shafts can begin to oscillate with angular displacement as a result of torsional excitation. VFDs can cause torsional excitation problems known as torque ripple. For example, rather than delivering a continuous 295 ft-lb of torque, a VFD-driven, 200 HP motor may deliver torque cycling between 250 and 340 ft-lb at some 21,000 cycles per minute. This oscillation could be damaging. Clearly, careful analysis and selection of the VFD, motor, coupling and pump train are needed to avoid torsional problems.

ENVIRONMENTAL CONSIDERATIONS

To avoid potential problems in your application of VFDs, you must take a few precautions regarding their environment.

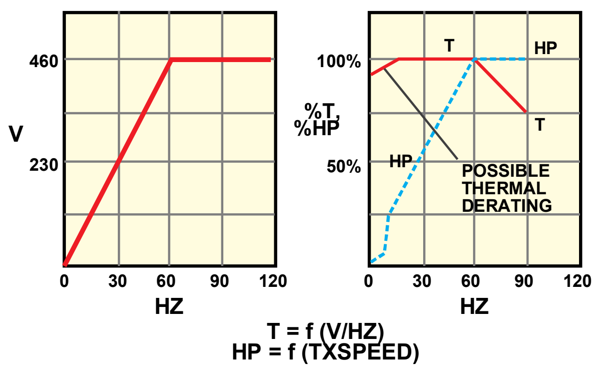
Locate VFDs indoors. Units can be placed outdoors with the proper enclosure, but the cost of the enclosure can run into thousands of dollars. Fortunately, the VFD can be up to several hundred feet

FIGURE 7



Centrifugal pump NPSHR and efficiency vs. speed

FIGURE 8



Performance of conventional variable speed motor

from the motor. So it can be indoors even if the motor is outdoors.

Derate for high temperatures and high elevations. If operated above 104º F, VFDs must be derated. They must also be derated if used at elevations above 3300 ft.

Be cautious of power supply. VFDs are sensitive to stiffness and irregularities in the electrical supply. You may need to install a line reactor or isolation transformer between the VFD and supply main if the feed transformer is very stiff (high KVA). Input line reactors or isolation transformers may also be necessary to prevent the VFD from feeding electrical noise back into the supply main. Such noise can distort instrument signals if they are fed from the same supply transformer as the VFD.

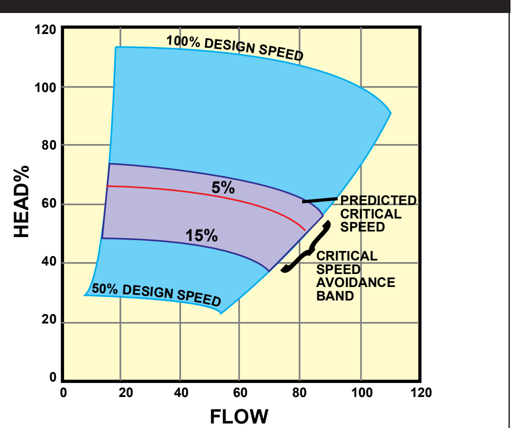
IS IT WORTH IT?

Despite the list of precautions, variable speed pumping can save you money. As shown, you can eliminate the need for throttling valves. You may be able to use one variable speed pump in place of two fixed speed pumps.

VFDs also eliminate the need for a motor starter. Variable speed pumping often reduces power requirements. And some electrical utilities provide rebates for companies that use energy saving devices such as VFDs. Rebates can be up to one-third the purchase price of the device. Other cost savings come through better process control due to lower heat inputs and fluid shear.

These savings frequently pay back the costs of utilizing variable speed pumping (such as the cost of the VFD, possibly extra costs for high-efficiency motors and possibly oversized pumps). Payback periods of as little as one year are typical when using variable speed pumping.

FIGURE 9



Avoiding lateral critical speeds

THE FUTURE

Variable speed pumping will become more popular as the technology establishes its track record. And as more system and plant engineers design for variable speed operation early in the development cycle, benefits beyond energy conservation will become apparent.

Advances in VFD technology will also increase user acceptance. New features such as greater adjustment in operating parameters will make VFDs easier to use and integrate into a system.

Improved reliability and fault tolerance will make VFDs easier to apply. You can expect manufacturers to add adjustment capabilities of output voltage and current waveforms to optimize motor efficiency and smoothness. Improvements in power semiconductors will provide higher efficiency and smoother output.

Sizes of VFDs will diminish as components on circuit boards are integrated into chips. Reduced size and improved efficiency will allow packaging to be more compact and environmentally rugged, which will allow placement even in hazardous environments.

Prices will come down, possibly by up to 25 percent over the next five years.

Even today, you can achieve greater flexibility, energy savings, equipment savings and extra head and flow through variable speed pumping, provided you take extra care in assembling an appropriate combination of pump, motor and VFD. With improvements in technology, more and more users will begin to take advantage of variable speed pumping. ■

Stephen P. Murphy is Senior Business Development Specialist for Sundstrand Fluid Handling in Arvada, CO.

Self-Priming Centrifugal Pumps

The ability to self-prime can be a cost effective solution for many applications.

BY TERRY W. BECHTLER

With greater global competition and increased environmental regulations, modern industrial applications over the years have evolved into sophisticated operations, demanding more control over their liquid handling processes. This is particularly evident on the "dirty" liquid side of a plant's manufacturing process, in the drainage, filtration/pollution control/wastewater areas. Self-priming centrifugal pumps are important in meeting this demanding challenge.

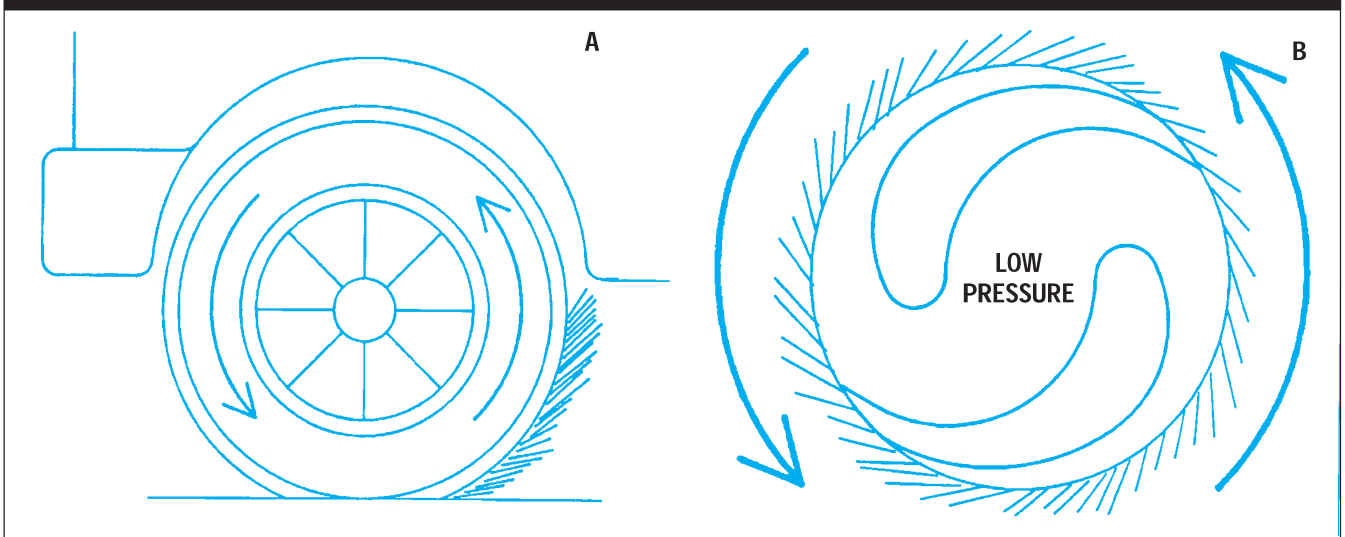
Single stage end suction centrifugal pumps may be divided by their designs into conventional or standard centrifugals and self-priming centrifugals. Centrifugal pumps incorporate a simple design with minimum moving parts - impeller, shaft and bearings. They are reliable, durable and relatively easy to maintain. To better understand the working principle of a self-priming centrifugal pump, let's first examine the centrifugal force principle and a standard or conventional centrifugal pump.

All centrifugal pumps incorporate the centrifugal force principle, which may be illustrated by a car running on a wet road (Figure 1). The tires pick up water and throw it by centrifugal force against the fender. Centrifugal pumps incorporate the same principle, but the tire is replaced by an impeller with vanes and the fender is replaced by the casing (Figure 1b). The liquid enters the center or eye of the impeller. As the liquid reaches the impeller vane, its velocity is greatly increased. Centrifugal force, created by the impeller blades or vanes, directs the liquid towards the outside diameter of the impeller. Once the liquid reaches the tip of the impeller vane it leaves the impeller at its greatest velocity. As the liquid leaves the impeller, its direction is controlled by the pump casing (the most common casing shapes are spiral or volute and circular).

The spiral or volute casing surrounds the impeller, beginning at the point where the liquid leaves the impeller. The liquid enters the casing and follows the rotation of the impeller to the discharge. Within the casing there is a section called the throat or cutwater.

The cutwater, also called the tongue, is a cast section of the volute casing, near the discharge that is positioned close to the maximum impeller diameter. As the liquid

FIGURE 1



reaches the cutwater it is diverted into the pump's discharge opening (Figure 2).

SELF-PRIMING

Self-priming centrifugal pumps incorporate all the above standard centrifugal pump design features and add the following internal modifications:

- A casing design that surrounds the volute and impeller and enables the pump to retain liquid in a built-in reservoir, or priming chamber. This reservoir is filled during the initial prime of the pump, and when the pump completes a pumping cycle and shuts down, the reservoir retains liquid for the next priming cycle.

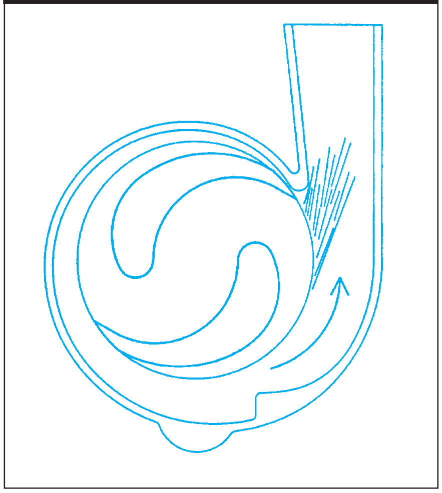
- An internal recirculation channel or port. This channel connects the pump's discharge cavity back to the suction reservoir internally, allowing the continuous recirculation of liquid from discharge back to suction during the priming, usually to the peripheral portion of the impeller (Figure 2B).

These two internal design features, the priming chamber and internal recirculation channel, are what distinguishes a self-priming centrifugal pump from a standard centrifugal pump. Self-priming can also be accomplished by a diffuser design centrifugal pump that is used primarily for clear liquids.

HOW IT WORKS

Self-priming centrifugal pumps can be placed above the liquid level of the source (Figure 3). Only the suction pipe enters the liquid being pumped. The pump is initially primed by adding liquid to the pump casing through a priming port, normally located near the discharge. The liquid fills the discharge reservoir, traveling into the eye of the impeller through the pump's recirculation channel. The suction line, itself, is not filled. A check valve is usually located just inside the suction reservoir. All connections must be airtight. During initial start-up, the impeller rotation causes the liquid in the pump reservoir to be directed to the discharge cavity via centrifugal force. Simultaneously, a lower pressure is formed in the suction reservoir. This draws the liquid from the discharge cavity back into the suction reservoir through the pump's internal recirculation channel. This is a continuing action during the priming cycle. While this is occurring, the air in the suction line is drawn by the lower pressure into the eye of the impeller with the priming liquid and travels through the volute into the discharge cavity. At this point velocities decrease, allowing the air and liquid mixture to separate. The air flows up and is ejected, and the priming liquid recirculates back into the impeller.

FIGURE 2



This process continues to draw all the air from the submerged suction line. In applications where the liquid level is at atmospheric pressure, that pressure on the liquid surface, coupled with the lower pressure in the suction pipe due to the evacuation of air, serves to push the liquid in the sump into the pump. When all air is evacuated liquid pumping automatically begins.

Note that the diffuser design self-prime principle incorporates an impeller rotating in a stationary multi-vane diffuser (Figure 4). During priming, the diffuser separates the air from the pumped liquid until priming is completed.

This priming action might seem somewhat complicated or mysterious, but it is actually a very easy task for a correctly installed self-priming centrifugal pump, and it happens automatically in a relatively short time (20 - 30 seconds for a normal 15 foot suction lift).

It's this feature that differentiates self-priming centrifugal pumps from standard centrifugal models. On a suction lift condition, a standard centrifugal pump, with only air in the casing and having no ability to separate air and liquid to create a vacuum, would have an impeller that simply spins, acting as a fan, because it has no way to lower the suction line pressure. By placing a foot valve on the end of a suction line and filling the pump and suction line with liquid, a standard centrifugal pump can be made to operate and pump in a conventional mode.

If the foot valve leaks and air enters the suction, such as under a shutdown condition, a standard centrifugal pump stands the risk of losing its prime and becoming air bound. Under suction lift conditions, self-priming centrifugal pumps are ideal for unattended use.

Standard centrifugal pumps are sometimes fitted with priming systems to fill the pump and suction line with liquid prior to starting. In such cases, a control device tells the pump when all air is evacuated and the unit is liquid filled to start.

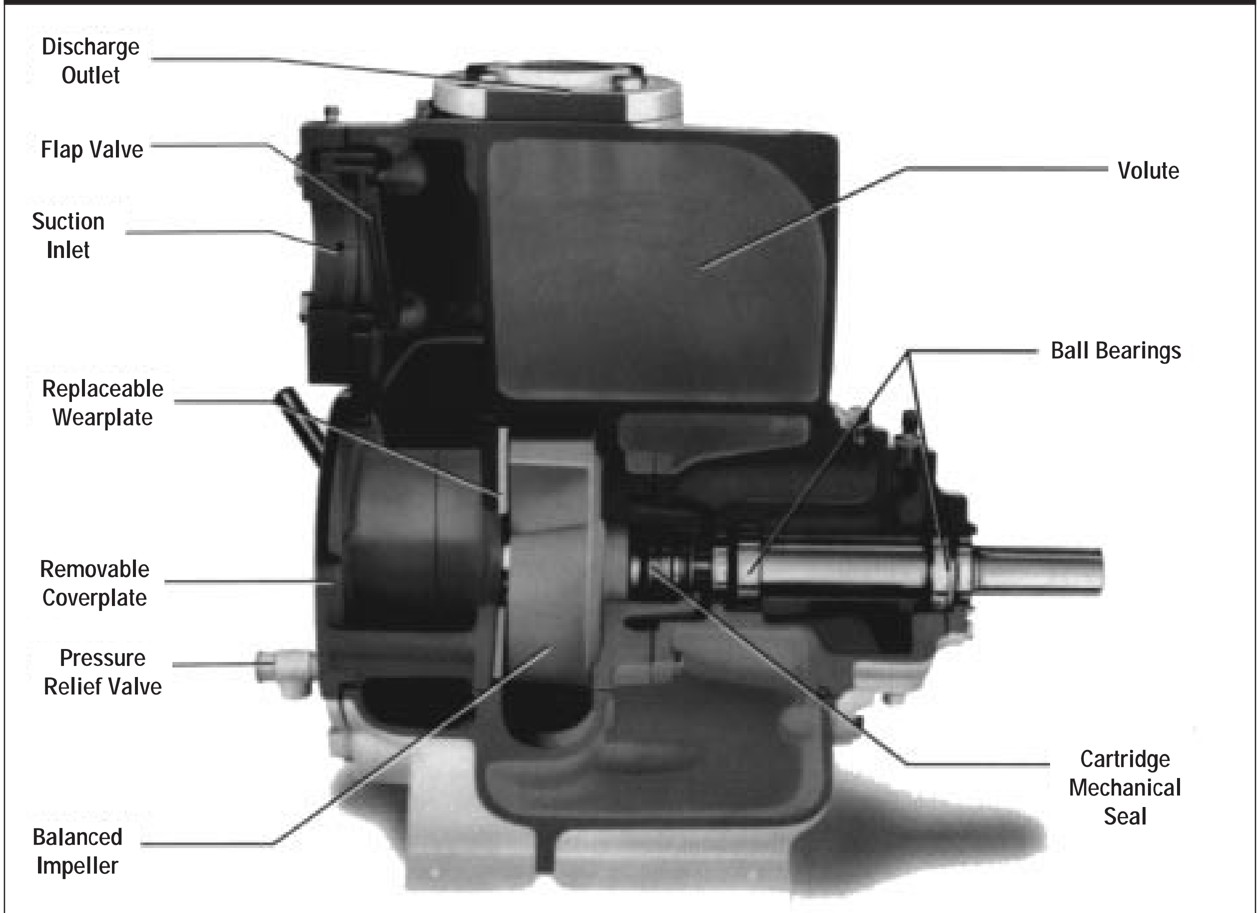
STYLES

Self-priming centrifugal pumps are usually classified into two groups: basic self-priming pumps and trash-handling self-priming pumps.

Basic self-priming pumps usually come with different impeller configurations, including fully enclosed and semi-open. Like all centrifugal pumps, the pressure developed is dependent on the impeller diameter and rpm.

- Fully enclosed impellers allow self-priming pumps to develop medium to medium-high discharge pressures, up to about 110 psi or 254 ft total dynamic head (TDH). Normal pump sizes range from 1 in. through 6 in. suction and discharge. Pumps with a fully enclosed impeller have a very limited solids handling capa-

FIGURE 2B



A cut-away view of a self-priming centrifugal pump designed to handle solids-laden liquids and slurries

bility, with sizes from 1 1/32 in. through 5/8 in. in diameter, depending on the size of the pump. This configuration is excellent for handling clear liquids, including processed hydrocarbons, along with general wash-down pressure applications.

- Semi-open multi-vane impellers are usually designed for slightly lower head conditions than fully enclosed impellers, but they have greater solids handling capabilities. Pump sizes usually range from 3/4 in. through 12 in. suction and discharge, with capacities to more than 5,500 gpm. Spherical solid sizes range from 3/4 in. through 3 in. in diameter, depending on the size of the pump. Basic self-priming pumps with semi-open impellers are sometime referred to as general-purpose self-priming pumps. They are excellent for handling dirty, contaminated liquids. Applications include extensive use in industrial filtration operations and a wide range of engine-driven models that serve the construction market.

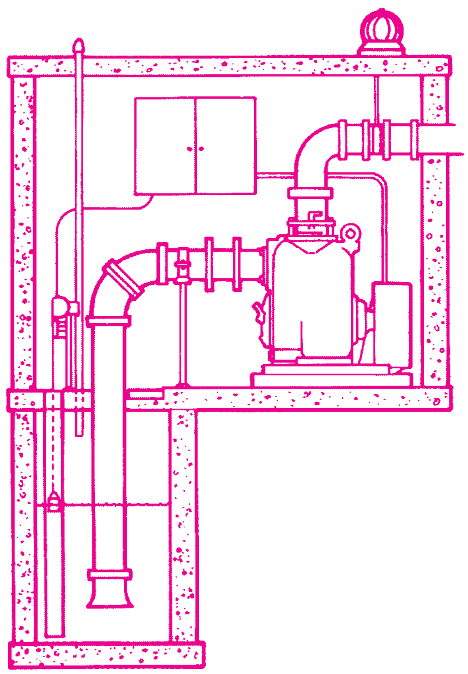
Trash handling self-priming pumps generally use a trash-type, semi-open, two-vane impeller that allows the pump to pass larger spherical solids.

- Trash handling self-priming pumps generate medium discharge pressures in the area of 62 psi or 145 ft TDH on electric motor drives and discharge pressures upwards of 75 psi or 173 ft TDH on engine-driven configurations, with capacities upwards of 3,400 gpm. Normal pump sizes range from 1-1/2 in. through 10 in. suction and discharge. The impeller design allows for excellent solids handling capability, ranging from l in. to 3 in. spherical solids diameter, depending on the pump size.

Trash handling self-priming pumps are often referred to as the workhorse of centrifugal pumps due to their rugged design and large solids handling capabilities. These pumps can be found on some of the most severe pumping applications within plants or on construction sites.

A desirable design feature of a trash handling self-priming pump is a removable cover plate, located

FIGURE 3



directly in front of the impeller on the suction side of the pump. Trash handling self-priming pumps may be applied in waste sump applications where they are exposed to various size solids. Any pump may clog trying to pump larger solids than it was designed to pass. The removable cover plate allows quick access to the suction side of the pump, expediting the removal of blockage. Some designs allow removal of the cover plate without disturbing the suction and/or discharge line.

SELECTION

As discussed, self-priming centrifugal pumps have a broad design range that allows them to serve a wide variety of applications. Many metallurgical choices and shaft seal configurations are available to best serve particular services.

Mechanical shaft seals can be single, double, or tandem. They are available as double grease lubricated for general purpose applications, oil lubricated with silicon carbide faces for industrial applications with abrasives, carbon against Ni-Resist faces for clean water or refined hydrocarbon applications, or Teflon fitted with carbon/ ceramic faces for chemical applications. Alloys available for pump construction also offer the same diversity. Cast iron and ductile iron are used for general purpose and refined hydrocarbons, hardened austempered ductile iron (ADI) is employed for abrasive applications, CD4MCu SS serves in corrosive and abrasive applications, and 316 SS, Alloy 20 SS, Hastalloy B, and Hi-Resin Epoxy Plastic are used for other special chemical applications.

APPLICATION GUIDELINES

The principal application area for self-priming pumps is where their ability to self-prime is a cost effective solution; and when it is more convenient and desirable to locate a pump ”high and dry“ above the liquid. Some general guidelines are in order:

The liquid being pumped should be of low viscosity (550SSU or less). Horsepower and efficiency corrections are needed for liquid viscosity above 550 SSU. If subjected to liquid freezing temperatures, the pump must be protected against freezing to avoid damage.

The vapor pressure of the liquid and the presence of high levels of entrained air are serious considerations in suction lift application.

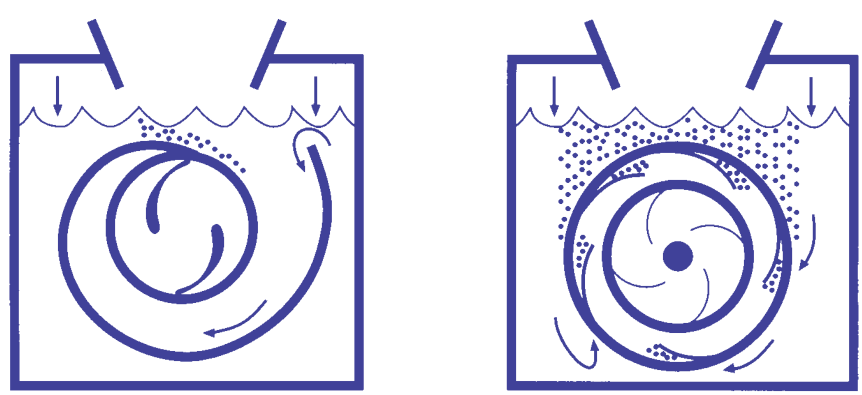
The NPSHA (net positive suction head available) must exceed the manufacturer’s published NPSHR (net positive suction head required) by a margin that accounts for the liquid properties.

Repriming time increases with suction lift. Suction lifts with water as the liquid at normal ambient temperature should be limited to 15 to 18 ft. best efficiency range, although maximum practical lifts are obtainable to 25 feet. For other liquids or liquid mixtures, the vapor pressure of the liquid or the most volatile components of a mixture must be considered. Reducing the speed of operation (rpm) significantly reduces the NPSHR. Suction line piping should be sized to velocities in the 5 to 7 ft. range at design flow. For self-priming pumps it is recommended that the suction piping should be the same size as the pump’s suction inlet.

The self-priming centrifugal pump offers a unique solution to many pumping applications. ■

Terry W. Bechtler has been Manager of Inside Sales for The Gorman-Rupp Co. in Mansfield, OH for four years.

FIGURE 4



Volute Priming **Diffuser Priming**

Centrifugal Pump Testing

Laboratory and on-site testing ensure pumps are up to their tasks.

BY LEO RICHARD

The test lab provides a tightly controlled environment and thereby generates the most accurate data.

As industry becomes increasingly competitive, pumps are being sized to precisely meet their duty requirements without oversizing. This allows users to maximize efficiency and minimize first capital costs. There is also a small but growing trend to question the economics of in-line spares and large spare parts inventories. These developments make it more critical than ever that rotating equipment precisely meets all hydraulic, material, and safety requirements. This is assured by thorough testing of pertinent parameters by manufacturers prior to shipment and by customers at their job sites.

The level of justifiable testing will depend on the nature of the service and significance of the parameter to be measured. For instance, a water transfer application can be served with a stock pump that has undergone the manufacturer's standard quality and performance checks. However, a corrosive, high pressure, or environmentally hazardous application may justify additional testing for material conformity and quality of construction.

In addition to the extent of testing, several other factors must also be considered. The first is location. A shop or laboratory test is typically conducted at the manufacturer's facility. The test lab provides a tightly controlled environment and thereby generates the most accurate data. In contrast, field tests sacrifice some accuracy, but they provide useful data under the actual conditions of service.

A reasonable split between the two approaches should be employed, depending on the nature of the evaluation and the user's ability to conduct on-site testing. Also, the user and manufacturer must agree to a set of guidelines such as those published by the Hydraulic Institute (HI). Among other things, HI standards generally define the methods and acceptable tolerances to be used. However, regardless of the standard employed, good laboratory practice requires that all instrumentation be calibrated prior to the test. For maximum accuracy the instruments should be located after straight runs of pipe where steady flow conditions exist. In addition, the local barometric pressure must be considered, especially in applications requiring suction lift. The data obtained should be recorded in a test log, and each round of evaluations must be identified in this document by the manufacturer's and user's serial/ equipment numbers.

Also, the question of user representation during testing should be clearly defined. This includes issues such as site location, the amount of advance notice prior to testing, and cost. Usually, the added cost and logistics problems make such witness testing inadvisable—unless the user has very limited experience with the manufacturer. This, as well as any other requirements, must be written into the specification prior to purchase.

A brief description of the most common performance and quality evaluations is given below. For simplicity, these tests have been characterized in terms of certifying

A technician attaches a mag drive pump to a test tank.

conformance in hydraulic capability, materials, or physical integrity.

HYDRAULIC CAPABILITY

The determination of hydraulic performance is the most basic and common category of testing. This typically involves performance, net positive suction head (NPSH), and power evaluation.

PERFORMANCE TESTING

The performance test of a specially ordered or job pump typically involves the generation of its head-versus-capacity curve at the rated impeller trim. Such pumps are shop tested on water at the manufacturer's site. If the HI standards are followed, the acceptance level must be defined. Level A requires that seven test points of head, flow, and efficiency be evaluated. Level B testing requires that five test points be checked. Each level of acceptance refers only to the head and capacity as specified by the customer for the service, also known as the rated or guarantee point. The defined tolerances for these parameters will vary depending on the size of the pumps and the level of testing required.

NPSH

The NPSH test is basically a measure of the suction head requirement necessary to prevent cavitation. The procedure typically involves holding the flow constant and reducing the suction head until a defined level of cavitation occurs. The data are used to generate a curve of the cavitation coefficient, Sigma, for the pump at the specified capacity. Sigma is defined as the net positive suction head available divided by the total pump head per stage. According to HI standards, the NPSH requirement of the pump is defined as the point at which a 3% head drop occurs on the Sigma curve. However, this criteria is somewhat controversial. The major issue is that incipient cavitation is well under way prior to the occurrence of the 3% head drop. In fact, some companies are considering an internal specification defining the NPSH requirement as only a 1% head drop on the Sigma curve.

POWER/EFFICIENCY TESTING

Power and efficiency testing is becoming increasingly critical as companies are closely evaluating power consumption during the pump selection process. Another relevant issue is the growing trend of retrofitting sealed applications with sealless designs. As is well known, due to magnetic coupling and viscous losses, sealless pumps inherently have slightly greater power requirements than their sealed counterparts. Therefore, a confirmation of the published power requirements may be in order, especially for installations where existing motors and starters are to be reused. Such tests are typically conducted on water using certified motors. Data are collected at several points, depending on the level specified as part of the performance test. This information can be used to generate both wire to water and hydraulic efficiencies.

MATERIAL CONFORMANCE

The usual considerations for material conformance testing are corrosion and erosion resistance. Again, the added cost for these procedures must be justified with regard to the particular application, as well as the consequences of process downtime and personnel or environmental exposure.

CERTIFICATE OF MATERIAL CONFORMITY

The most basic type of documentation is in the manufacturer's certificate of material conformity. This is a guarantee that the pump is made of the materials called out in the specification. This certificate is based solely on the standard quality tests performed by the manufacturer.

CHEMICAL ANALYSIS

This involves confirmation of the material of construction by chemically testing small samples from the pump. These tests range from sophisticated chemical analysis to a basic screening utilizing chemical test kits.

NUCLEAR ANALYSIS

This confirms materials used by means of a nuclear analyzer. This is a nondestructive test involving direct measurement on the surface to be analyzed. The composition of the material is determined by the equipment and matched with its internal database to generate an identification. Due to the high cost of this equipment, many sites utilize sub-contractors for this work.

HARDNESS TESTING

Hardness testing may be required by the user, especially for pumps in highly erosive services. Though the type of hardness test can vary, the Brinell hardness test is fairly common.

MILL CERTIFICATION

An extensive form of evaluation involves mill certification. Basically, the mill certs follow the pump along each step of the manufacturing process. This includes data from the initial pour at the foundry to the final checks of the finished components. The downside of mill certification is that it tends to be costly. Also, because additional data are required from the initial pour, stock pumps may not be used. Some parameters typically measured in mill certs include:

- Mechanical Test Certification, which includes tensile strength, proof stress, and elongation.
- Analysis certificates detailing the chemical composition.
- Intercrystallation corrosion and ultrasonic tests.

PHYSICAL INTEGRITY TESTING

As the name implies, this category of testing basically involves a confirmation of the pump's ability to maintain the liquid boundary under the conditions of service. The chief areas of concern prompting such testing are the integrity of welds and possible porosity of castings.

DYE PENETRANT TESTING

Dye penetrate testing involves the use of an extremely low surface tension liquid to detect possible leak paths in cast and welded surfaces. If the dye penetrates the surface, the piece is either rejected or weld repaired. If the component is repaired, the user is notified and the part retested to confirm the integrity of the weld.

RADIOGRAPHY

Radiographic testing is primarily used to confirm the integrity of welds in pressure-containing components. Procedures depend on the configuration and dimensions of the component, as well as the nature of the equipment being used. The test itself is somewhat costly and may impact delivery. For these reasons its use is most often limited to critical applications in the power industry.

GAS LEAK DETECTION

This involves pressurizing the pump with an inert gas such as arcton to detect any leak paths from the pump. Leaks are typically detected by means of a sniffer or mass spectrometer. This test is extremely sensitive and able to detect the slightest porosity in castings.

HYDROSTATIC TESTING

Hydrostatic pressure testing is a standard quality check. The procedure usually involves filling pressure-containing components with water and pressurizing to 1.5 times the rated working pressure. This pressure is held for a specified time, and the piece is inspected for leaks.

TESTING SEALLESS PUMPS

The testing procedures utilized to evaluate standard sealed centrifugals are commonly used for sealless configurations as well. However, due to the unique design of sealless pumps, some additional procedures may be considered. A complete discussion of this topic can be found in the Hydraulic Institute Standard for sealless centrifugal pumps. (HI 5.1–5.6, 1st edition, 1992)

The determination of hydraulic performance is the most basic and common category of testing.

MAGNETIC STRENGTH

The strength of the permanent magnets in a magnetically driven sealless pump can be evaluated with a Gaussmeter. This instrument directly measures the strength of the magnetic field in Gauss or Milligauss. Gauss testing is usually an overkill for new pumps for the following reasons:

- The relative uniformity of production magnets.
- The high safety factor incorporated into a magnetic coupling's power transmission capability. (A safety factor of 2.0 under full load conditions is typical.)
- The fact that the coupling is already inherently tested during generation of the hydraulic performance curve.

BREAKING TORQUE

A "low tech" but effective way of site testing synchronous magnetic couplings is by measuring the breaking torque. Breaking torque is simply the force required to break or decouple the two opposing halves of the magnetic coupling. This is accomplished by anchoring the inner rotating assembly and applying torque to the outer magnet ring (OMR). Force is applied and measured by a torque wrench fitted to the drive shaft of the pump (the drive shaft is mechanically coupled to the OMR). The data generated is then compared to the manufacturer's standards.

As in Gauss testing, this procedure is usually unnecessary for a new pump. However, it is a useful field tool for confirming the strength of the magnetic coupling. This is especially important during a rebuild after a dry run failure. During dry runs, the magnets are exposed to extreme temperatures that may reduce their strength. By utilizing the breaking torque procedure, maintenance personnel can pretest the magnetic coupling prior to reinstallation.

SECONDARY CONTAINMENT TESTING

CANNED MOTOR DRIVES

The stator housing in canned motor pumps is often used as a secondary containment vessel. Testing typically involves gas leak detection on the finished stators. For designs utilizing potting of the wire leads, confirmation of the integrity of the secondary containment chamber as the equipment ages may be in order. This is especially relevant in services with high temperature cycling, which may damage the potting compound.

MAG DRIVE DESIGNS

In some mag drive designs the coupling housing and an inboard magnetic seal are utilized for secondary containment. Testing usually involves a hydrostatic or gas leak detection of this assembly.

An A range pump hooked up for testing.

SITE TESTING

One of the most important and often overlooked opportunities for evaluating and documenting pump performance is the initial commissioning. Information gathered at this time is critical in verifying initial performance and providing a benchmark for future diagnostic and troubleshooting efforts.

It is suggested that, as a minimum, the following areas be evaluated:

- The total differential head generated by the pump. It is strongly recommended that both suction and discharge gauges be installed to facilitate measurement of this parameter. Once determined, it should be noted whether the actual operating point differs from the duty listed in the specification. If so, the user must first confirm proper operation of the pump and process. If these check out, an evaluation of potential problems associated with the new duty point must be evaluated. This includes a possible increase in the NPSH requirement and power consumption. Also note that continuous operation at extremely high or low flows will significantly increase dynamic loading on the impeller. Such loading can dramatically decrease the mean time between failures for the equipment.
- Evaluation of the operating point should be conducted for all conditions the pump will experience. For example, many pumps in transfer applications deliver liquid to various locations and are periodically operated in a recirculation mode. Each of these duty points must be determined and possible problems identified. If necessary, modifications in the pump and/or process should be made. Common corrective actions include resizing orifices, changing valve settings, and adjusting the impeller trim.
- The amp draw of the motor should be measured. This is then compared with the manufacturer's stated requirements to evaluate proper operation. Gross differences between these figures may indicate various conditions such as cavitation, operating to run out or shut in, or mechanical problems.
- The vibration level should be measured. This will confirm proper operation and serve as a benchmark for future testing.
- After the pump has achieved steady state, the bearing frame, process, and ambient temperatures should be monitored and recorded. This data will be used as an initial check as well as for future reference.
- Proper operation of all protective instrumentation should also be verified and any outputs recorded. For instance, many sealless pumps utilize a thermocouple temperature monitoring system to protect against dry runs. The initial temperature reading should be recorded in the commissioning data sheet.

SUMMARY

There are many options for testing the performance and integrity of centrifugal pumps. The use of such procedures depends on the significance of the service and the nature of the pumpage. Users will find that in most cases the standard compliment of manufacturing testing will be sufficient. However, critical services involving serious environmental or health risks may warrant the added assurance of supplemental testing. In either case, the user and manufacturer must work as partners to achieve the best engineering solution for the particular application. ■

Leo Richard is a Technical Service Manager with the Kontro Company, Inc. Mr. Richard has experience in process and project engineering with General Electric and W.R. Grace.

The Canned Motor vs. Magnetic Drive Debate

BY GREGORY ZIMMERMAN

Perhaps you've decided to purchase sealless centrifugal pumps. The arguments are compelling: zero emissions, no need for complicated seal support systems, no need to replace expensive seals periodically. All manufacturers of sealless centrifugal pumps agree on these basic advantages. But their agreement ends there.

As the two major camps in the sealless centrifugal debate — canned motor or magnetic drive — try to position their chosen technology as the most reasonable choice, they let loose a flurry of claims and counterclaims. It can get confusing.

To help you prepare for the barrage, we present advantages for both types of sealless centrifugal pumps as commonly stated by manufacturers and users. Consider the arguments and decide which are most pertinent to your situation. Then you'll be better prepared to discuss your specific concerns with manufacturers.

THICKNESS OF CONTAINMENT SHELL

Magnetic drive pumps can use thicker containment shells since their inner and outer magnetic rings do not have to be as close together as the rotor and stator in a canned motor pump. Manufacturers of mag-drive pumps claim that the thicker shell — up to five times thicker than that of canned motor pumps — vastly reduces the chances of breaching the shell, especially as a result of bearing wear.

Manufacturers of canned motor pumps counter that claim with two arguments. First, the thicker shell of a magnetic drive pump reduces operating efficiency. Second, the shell must be thicker (and the internal clearances wider) because magnetic drive pumps do not contain bearing monitors. Unmonitored bearing wear can cause the inner magnetic ring to contact the shell. A thin shell would be too prone to such damage. Canned motor pumps can use a thinner shell because bearings are closely monitored and bearing wear can be projected from the data.

HIGH TEMPERATURE SERVICE

Permanent magnets can tolerate heat better than motor windings can. Thus, magnetic drive pumps can pump hot liquids — up to 750° F with just air cooling. Canned motor pumps can also be used in hot service but need water cooling jackets. Manufacturers of canned motor pumps agree that canned pumps should be water cooled for high temperatures. But, they reply, so should magnetic drive pumps since even rare earth permanent magnets cannot tolerate extremely high temperatures.

SOLIDS HANDLING

With greater internal clearances and thicker containment shells, magnetic drive pumps can handle solids more easily, their manufacturers say. Canned motor manufacturers counter that internal clearances are designed to accommodate bearing wear, not to accommodate solid particles. The crucial dimension, they say, is bearing clearance and that is the same for both types of sealless pumps. Further, canned motor pumps can effectively handle solids if they are outfitted with external flush or filters to remove particulates from the pumpage before they circulate around the bearings.

HIGH PRESSURE APPLICATIONS

One point on which all parties agree is that magnetic drive pumps cannot tolerate as high pressures as canned motors pumps can. A canned motor pump is a pressure vessel

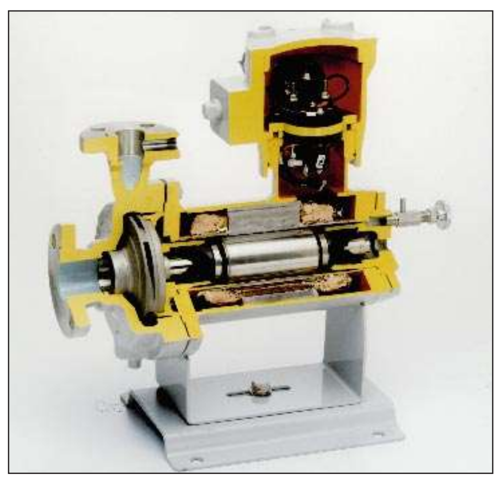


PHOTO COURTESY OF TEIKOKU USA

Sealless canned motor pump designed for hazardous liquids.

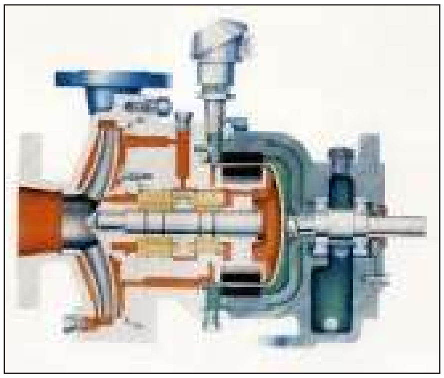


PHOTO COURTESY OF GOULDS PUMPS, INC.

Magnetic drive process pump designed for zero leakage services.

since the stator windings lend additional mechanical strength.

DIFFICULT-TO-HANDLE FLUIDS

According to one manufacturer of magnetic drive pumps, the biggest advantage magnetic drives offer is the ability to use non-metallics. These pumps are thus able to pump highly corrosive materials, solvents, and other difficult fluids. That may be true for some fluids, counter manufacturers of canned motor pumps, but other issues are involved. Hazardous materials require failsafe containment. Canned motor pumps, they point out, offer sealless double containment. If the stator lining blows, a backup shell will contain the materials. Doubly contained magnetic drive pumps rely on a mechanical seal — the very thing we're trying to avoid, say the manufacturers of canned motor pumps.

Canned motor proponents point to another benefit of their technology in hazardous environments: UL listing for the entire unit. Because a canned motor pump integrates the electrical and mechanical portions, the entire pump must be UL listed for use in, say, explosive atmospheres. Sundstrand canned motor pumps, for example, are tested under a procedure in which UL fills the pump with oxygen and ethylene and ignites the gas. The explosion must be contained in the pump with no propagation of flames up or down the discharge piping. Magnetic drive pumps are not UL listed — only the motors need to be. If a magnetic drive decouples or loses a bearing, the skin temperature on the drive unit can exceed the autoignition temperature of the explosive compound.

COMPACT DESIGN

Canned motor pump manufacturers cite compact design as an added advantage. Canned motor pumps not only save space but also require no foundation work. Magnetic drive manufacturers counter that they can make compact pumps by using a close coupled design. Besides, they add, the absolute dimensions aren't as important as meeting ANSI standard dimensions. ANSI standard dimensions make magnetic drive pumps easier to retrofit, according to their proponents.

ALIGNMENT

Canned motor pumps have an integrated single shaft and thus come perfectly aligned from the factory. Alignment of the motor and magnetic coupling can be tricky in a magnetic drive pump.

A USER'S PERSPECTIVE

For one user, an engineer at a chemical processing plant in the midwest, UL area classification is the most important reason he prefers canned motor over magnetic drive pumps — in situations where canned motor pumps are optimal. This user also relies on magnetic drive pumps for high temperature applications (e.g., heat transfer fluids), high horsepower requirements and for aqueous hydrochloric acid service (which requires nonmetallic pumps).

Another advantage this user states for canned motor pumps is he can predict bearing wear and thus schedule maintenance more easily.

FURTHER ADVICE ABOUT GOING SEALLESS

"We've been using canned motor pumps for some time and are comfortable with the technology," the user says. "Our electricians are adept at repairing the pumps — both mechanically and electrically. We are bringing in more magnetic drive pumps for certain applications and we're getting comfortable with that technology, too," he said.

This user and another engineer at a major chemical plant report high reliability of both types of pumps. Both report that reliability increased as they gained more experience with sealless pumps. In each facility the major cause of damage to sealless pumps is operator and specification error. And as they learned to size the pumps correctly — to operate at the best efficiency point of the pump — and to avoid operating the pumps off design, mean time between failure increased substantially. "We've gotten seven years without failure from some of our sealless pumps," said one user, "but we've also had cases where we replace the pump nearly every month because of dead head operation, running dry or cavitation."

All the above manufacturers agree that pumps must be specified correctly for the application and that operators must be trained adequately. "Users need to make sure we know everything about the application," says one manufacturer. "We especially need to know temperatures and vapor pressures at startup and shutdown, not just normal operating conditions."

Another key point: don't simply substitute pump problems for seal problems. In other words, if you're faced with recurring seal failures, be sure to root out the cause of the failure before you simply bring in sealless pumps. Maybe the fault isn't the seal. If the problem lies elsewhere in the system, you'll be left wondering why your sealless pumps failed just like the seals did. ■

National Electric Code Impact on Sealless Centrifugal Pumps

BY: ROBERT MARTELLI

What users need to know about the National Electric Code and how monitoring options can help.

Pump users are no different from other users of industrial processing equipment who must comply with several codes and government regulations. It can be a formidable task to keep up to date and appropriately apply rules to specific situations. A greater effort is required to get code and regulations updated and clarified to keep pace with changing technology. Nonetheless, users need to understand the impact of the National Electrical Code on sealless centrifugal pumps and to know what monitoring options are available.

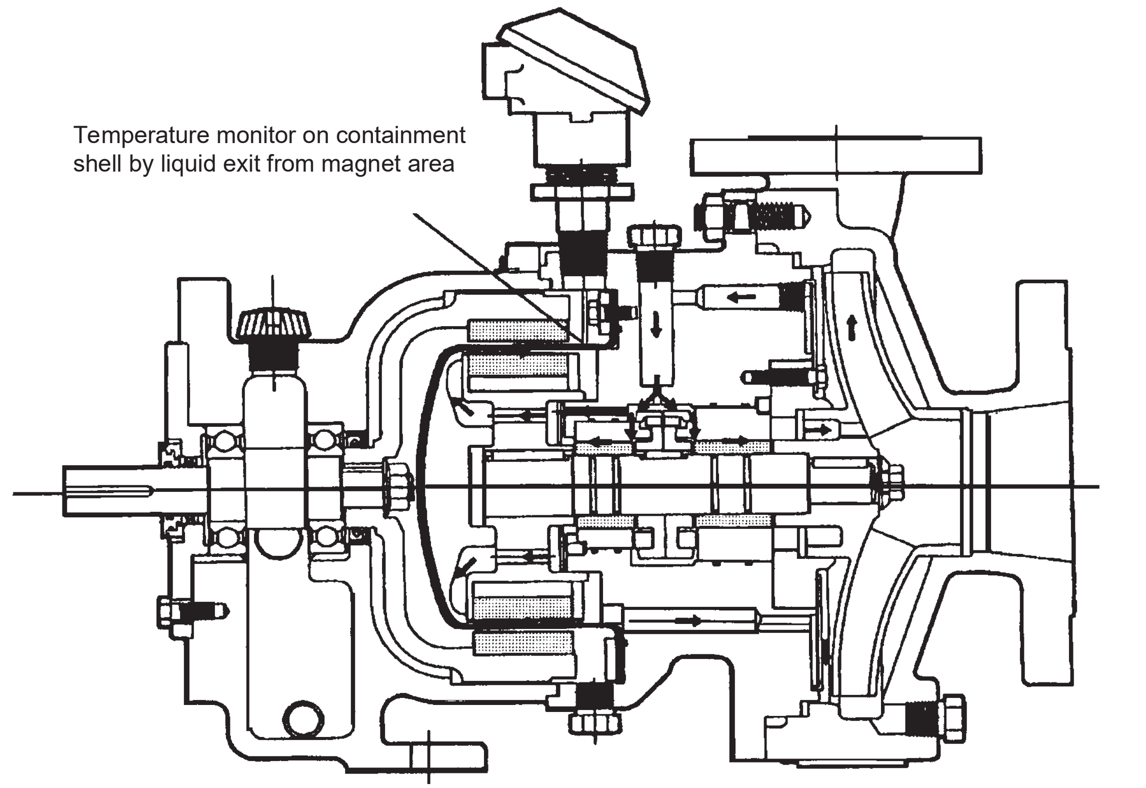
ELECTRICAL DEVICES IN HAZARDOUS LOCATIONS

The National Fire Protection Association (NFPA) has produced several codes for reducing the risk of and damage from fires. One of these is the National Electric Code (NEC). Section 500 of this document applies to electrical devices operating in hazardous environments—where flammable/explosive materials are either routinely or may be present in the atmosphere. These materials can be gasses/vapors, liquids, a solid dust or liquid mist. Since electrical devices are present in these areas, the NEC imposes requirements to reduce the risk of fires and explosions.

A Division 1 area is where explosive materials are routinely present in the atmosphere (such as the bottom of a spill containment, or a below-grade installation where vapors could collect) and requires U.L.-approved electrical devices. Most sealless pumps, however, are operated in Division 2 areas where explosive materials may occasionally be present in the atmosphere. (In the chemical industry, the vast majority of materials are handled in closed systems.) The two areas are covered in the NEC where they have the potential to form an explosive cloud in the atmosphere. A growing cloud that comes in contact with a source of ignition, such as a hot electrical device, can cause a large explosion and fire.

To reduce this risk, the NEC requires that the Auto-Ignition Temperature (AIT) be determined for each stream in the process area as well as its geographical area. Electrical devices intended for operation in hazardous areas are also

FIGURE 1



Mag drive pump cooling circuit flow temperature measurement is made after the fluid has picked up eddy current heat and partial bearing heat.

required by the code to have "T ratings." If users follow this section of the code, these electrical devices will not constitute a potential source of ignition, vastly reducing the chance of an explosion should an explosive cloud ever develop.

AITs are a concern for light hydrocarbons including n-butane and acetylene, which have AITs below 600°F; pentane and hexane, which have AITs below 500°F; and diethyl ether and heptane, which have AITs below 400°F.

NONELECTRICAL SOURCES OF IGNITION

Ignition by nonelectrical sources—for example, steam, heat transfer lines and reactor vessel walls—are also possible in process areas. The NEC does not address these sources. Another NFPA code covers nonelectrical sources of ignition in section 30, the Flammable and Combustible Liquids Code. Specifically, Chapter 5 requires that you take precautions to prevent the ignition of flammable vapors from nonelectrical sources. Preventive measures are to be determined by an engineer and/or "the authority having jurisdiction." Since "hot surfaces" are normally present in chemical processing environments, one precaution typically taken is to handle materials in closed systems.

EDDY CURRENT HEAT GENERATION

Canned motor and magnetic drive pumps with metallic liner/containment shells generate heat due to eddy current loss. Eddy currents are created by changes in magnetic field strength during pump operation in a given area of a stator liner or containment shell. In most applications, pumps will operate at temperatures well below 400°F (which is below most AITs) because of the cooling effects of the pumpage. In general, the eddy current heat source is not regarded as heat produced by an electrical device and, therefore, not clearly addressed in the NEC.

CANNED MOTOR PUMPS IN HAZARDOUS AREAS (DIVISION 2)

The "skin" or outside temperature of the canned motor pump is an issue in hazardous areas. These pumps contain a thermal cut-out switch, which is located in the stator winding hotspot and shuts down the motor if its setpoint is exceeded. The user is required to wire this switch into the motor control circuit. If the motor cooling is lost due to some upset or misoperation, the pump will heat up and eventually open the switch and shut off the power, preventing an excessive "skin" temperature on the can. If the pump is in a volatile liquid service, it's usually destroyed. In most cases, the switch will not protect the pump from dry running—it is there only to meet NEC requirements.

For canned motor pumps, the NEC currently covers only conduit seals. This is to prevent hazardous pumpage from traveling through the conduit system to the motor starter room in case of a stator liner and primary seal failures.

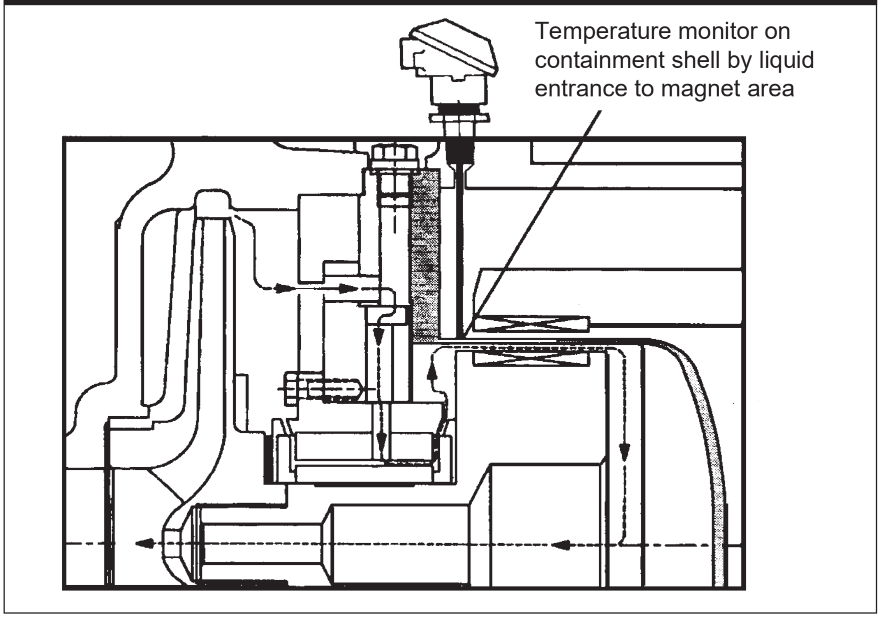
MAGNETIC DRIVE PUMPS (DIVISION 2 AREAS)

Mag drive pumps with metallic containment shells are not typically regarded as electrical devices, despite eddy current generation. (Mag drives with nonmetallic containment shells have insignificant eddy current generation and associated heat-up potential.) These losses are about 17% of the maximum rated horsepower of the drive, which works out to between 2 and 3 KW of power loss for a drive rated for 20 hp. If an upset or misoperation results in dry running, recent tests have shown that the containment shell temperature can reach 800°F to 1200°F in one to two minutes of continued operation. Furthermore, mag drive pumps that operate for several minutes with no cooling provided to the magnet area have straw-blue rings in the areas of the strongest magnetic flux, indicating temperatures of at least 900°F. Even more disturbing, with this type of failure there is a good chance of a spill or release occurring!

Currently, there is no requirement to monitor mag drive pumps for an abnormal condition and subsequently shut down the pump. In most cases, containment shell temperature, motor power, or pump flow monitoring with alarm and shutdown capabilities can greatly reduce the possibility of ever reaching unacceptable temperatures. Today these monitoring options are routinely available. Most mag drive manufacturers provide the option of containment shell temperature monitoring. However, there is no widely accepted agreement on the best monitoring method for mag drive pumps. Each method has strong and weak points.

Several mag drive pump manufacturers have recently taken steps to isolate the containment shell from the outside atmosphere, eliminating the air cooling used in some designs. This will not necessarily prevent the migration of flammable vapors or gasses to come in contact with the containment shell. Depending on maintenance procedures, there is also a small possibility that the pumpage will leak undetected past an improperly installed gasket and collect near the bottom of the containment shell, next to the outer magnet assembly. Therefore make sure the containment shell temperatures do not rise above the AIT of the materials present. Some mag drive manufacturers offer a leak monitor option for this section of the pump, partially addressing this concern.

FIGURE 2



Mag drive pump temperature measurement is made on the cooling circuit inlet. Temperature variations will be much smaller here.

HEAT TRANSFER FLUID PUMPS

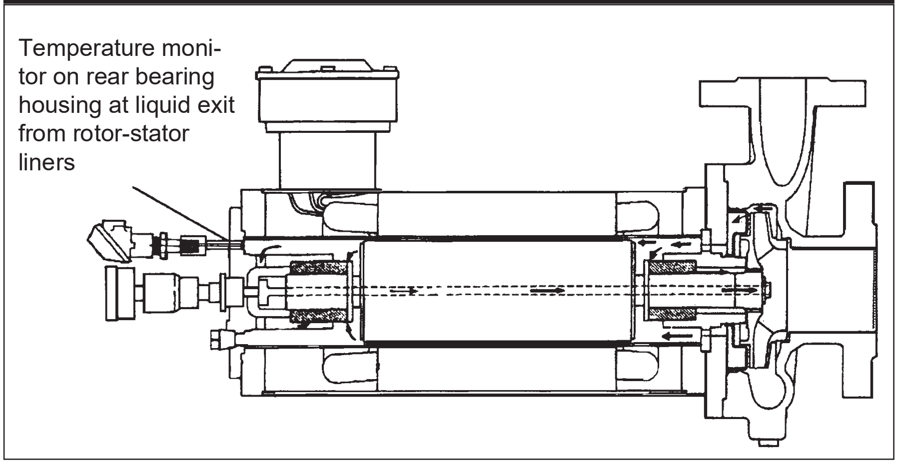
Mag drive and canned motor pumps with ceramic insulation on the stator windings in heat transfer service may present a problem since in some situations the suction fluid temperature can be in the 600°F range. If this temperature exceeds an AIT for other nearby materials, there is no increase in safety by applying this portion of the NEC to either pump. In this case, the "skin" temperature already exceeds limitations set by the NEC for electrical devices! Current NEC interpretations by several users preclude the use of a canned motor pump with ceramic insulation in these instances. The only option for a canned motor pump is a unit with a cooling jacket, the necessary service water lines, and a conventional stator with the appropriate thermal cut-out switch. The current mag drive claim, again, is that there is nothing in the pump that meets the definition of an electrical device; therefore, no special monitoring or shutdown devices are required by code. If the magnets can operate at the required service temperature, no cooling water is required.

Although difficult, it may be prudent to revise pump and piping layouts so that no low AIT materials are near the pump.

CODE APPLICATION AND CLARIFICATION

A word of caution: The NEC has been adopted by OSHA as a reference standard and you are required to follow it as a minimum. Be careful when interpreting the code and remember that common sense does not always apply! When making interpretations or determinations regarding legal regulations, a team approach is advisable. More informed determinations are made, and mistakes are less likely.

FIGURE 3



Canned motor pump cooling circuit temperature measurement made at its hottest point.

If the NEC panel would clarify the application of the code to mag drive pumps with metallic containment shells, it would help pump users considerably. Specifically, does eddy current generation fall within the code definition of an "electrical device"? And for electrical devices of which the "skin" temperature already exceeds an applicable AIT by non-electrical sources, does the NEC prevent its use?

A National Electric Code change can occur no sooner than 1999, when it's scheduled for update. Until then, users will have to operate under the current code, taking precautions as they deem appropriate.

CONTINUOUS MONITORING: SAVING MORE THAN THE PUMP

Although monitoring adds cost, users can take advantage of automatic shutdowns for other abnormal conditions (such as dry running) before the pump is destroyed and provide better assurance that AITs are not exceeded. Monitoring can also improve pump reliability in handling heat sensitive materials.

Monitoring is especially important with silicon/tungsten carbide bearings. (Most sealless pump manufacturers offer carbide bearings at least as an option.) Monitoring these bearings requires a different approach than for carbon bearings in order to extend life. Carbide bearings can be more prone to sudden breakage and failure due to misoperation; hence, these conditions must be identified and the pump automatically shut down if encountered.

Pump monitoring is a relatively new concept for most operations people and not well understood. Yet these workers play a key role in implementing monitoring methods. Everyone involved should have patience in finalizing the alarm and shutdown setpoints for successful implementation of the methods used. Nonetheless most users go through several "false shutdowns" or even a pump failure before determining the proper setpoints.

TEMPERATURE RISE MONITORING

For a single method, temperature rise monitoring offers the best overall protection against most pump failures, including dead-head/very low flow, dry run operation, and restricted cooling circuit flow in the magnet area. Moderate cavitation and gas entrainment in the pumpage are also involved when they reach the point of upsetting the cooling circuit flow.

Two temperature points are required to implement this monitoring. One is on the containment shell of the pump. In all current designs, this point must be located between the magnet assembly and the containment shell flange limiting what temperature can be monitored. Note the direction of the cooling circuit flow next to the temperature measurement point. A more sensitive measurement results by monitoring at the exit point for the cooling circuit flow after it has picked up heat from the magnet area (Figure 1). This flow configuration can find this exit point near the rotating magnets in pumps that use a discharge-to-discharge pressure circulation with a pumping vane near the rotating magnets. On pumps that use a discharge-to-suction pressure configuration to drive cooling circuit flow (that is, where the cooling circuit inlet flows past the containment shell at the measurement point, before the temperature rise takes place) temperature rise monitoring will not be as effective (Figure 2). Canned motor pumps may also have temperature monitoring installed. More sensitive readings can be taken when the monitor point is located after the cooling fluid passes between the rotor and stator liners (Figure 3). With the temperature probe in this location, dry run protection will not be as effective as what can be provided by power monitoring.

The second temperature monitoring point is on the suction line or supply vessel, providing suction temperature compensation and takes into account temperature changes from day to night, and seasonal variations. This greatly eliminates false shutdowns and failures. Keep enough distance between this point and the pump to ensure that suction recirculation will not conduct heat from the pump and up the suction line to the measurement point during dead-head operation. Locating this point upstream of a suction basket strainer may provide enough isolation to be effective. If you go to the tank for this temperature, keep in mind that the sun can warm up the suction line and pump unit much faster than it can warm the tank during nonoperation. If this happens and you get an inaccurate measurement, you may shut down the pump on start-up when there is nothing wrong.

The temperature rise is determined by the difference between the containment shell and suction temperatures (Figure 4). Pump suppliers can provide an expected "normal" temperature rise. Typical alarm and shutdown points may be 10°C and 20°C above this value. Field experience will be required to finalize these setpoints for each application, since this is a relatively new concept for pump users. A good approach is to find a temperature rise that is sufficiently far away from the normal operating range (with its usual variations), and that still results in liquid in the containment shell, with a few degrees of boiling point margin left in the magnet area. Pump suppliers can help by supplying pump cooling circuit pressure. Knowing the pressure, you can calculate the liquid boiling point in the cooling circuit. The maximum cooling circuit temperature needs to stay below this value.

Recent discussion about containment shell temperature rises downplays the effectiveness of containment shell temperature monitoring. Some say the temperature measurement is not sensitive enough for the rapid rise found in dry running. However, temperature rise monitoring does not need a large change in containment shell temperature to be effective.

Also, temperature rise monitoring does not protect against motor or outer magnet bearing failure. Periodic vibration monitoring or additional bearing temperature monitoring are two proven ways to protect against these types of failure.

Wear of the inner sleeve bearings may be detected by temperature rise monitoring if there is enough wear to alter the cooling circuit flow or the eddy current heat generation. This will depend on the pump used since temperature rise is not always a direct result of bearing wear.

FLOW AND LEVEL MONITORING

For processes where the supply tank level or pump flow are already measured and sent into a process control computer, adding these monitoring devices can be relatively inexpensive. The only other hardware required is an output relay in the pump motor control circuit. Then the software is programmed to implement a low level shutdown or a high flow/low flow shutdown for the pump. The low level method is effective against running the tank dry but does not cover other common pump failures.

Flow monitoring provides a little better protection because it protects against closed suction and discharge valves in addition to dry running. It can also protect against excessive flow. The narrower the range between the shutdown setpoints and normal operation, the better the protection; however, false shutdowns must be avoided. Neither of these methods protects against mild cavitation, a plugged cooling circuit flow path, or worn inner bearings. In services where these other modes of failure are unlikely, this method can be quite effective.

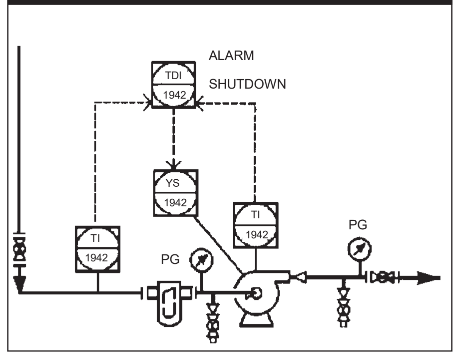
POWER MONITORING

In this case, motor power (kilowatt draw) is monitored, usually in the pump starter room. It has the advantage of not requiring any process connections to install, making it one of the easiest to incorporate into existing processes. This can eliminate corrosion/erosion concerns in slurry or acid service where exotic materials of construction are required. It is easier to establish shutdown setpoints if the pump is operating in the range of 60% to 90% of its BEP.

As with any electronic device in an operating plant, it must not be affected by radio frequency interference (e.g., portable radios). Power monitoring has the same advantages and disadvantages as flow monitoring in protecting against previously described failures.

Current monitoring of the motor amp draw can also be effective as long as the horsepower draw is near the motor's nameplate rating. Otherwise, the amp draw versus pump curve becomes flatter, and it's more difficult to determine realistic shutdown setpoints.

FIGURE 4



Temperature rise is calculated in a process control computer and compared to alarm and shut-down setpoints for appropriate action.

CONCLUSIONS

Until the National Electric Code is either revised or clarified, users will need to make their own determinations of the potential hazards of pump operations and choose suitable means of reducing the resulting risks. Efforts are underway at several pump manufacturers to improve continuous pump monitoring. A more universally accepted method should result. ■

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Pumping Hydrofluoric Acid

Consider proper metallurgies, compatible bearing materials, and hydraulic and pump configurations when pumping this acid.

BY: JOHN V. HERONEMA

Hydrofluoric acid has touched all of our lives because so many industries use it in their manufacturing processes. For example, a beryllium-shafted golf club and a coffee mug with an etched design have been manufactured using hydrofluoric acid. It has also been used as a catalyst in the manufacture of ozone-friendly refrigerants. Yet hydrofluoric acid is a potentially dangerous chemical. Acid leaks can yield devastating effects, ranging from toxic fume inhalation to severe chemical burns, injuring people and damaging equipment.

Many plants pump hydrofluoric acid using traditional seal technology. Of course, mechanical seals can leak. Because of the potential danger involved, hydrofluoric acid leaks are not tolerated. One way to reduce the threat of leakage is to use a sealless technology. Consider several key factors when selecting a sealless pump for hydrofluoric acid applications, including proper metallurgies, compatible bearing materials, and hydraulic and pump configurations.

METALLURGIES

Proper pump metallurgy is critical for pumping hydrofluoric acid. Two primary variables dictate what metallurgies are necessary under pump operating conditions. The first is temperature. Hydrofluoric acid is similar to many other acids in that as temperature increases, so does the aggressive nature of the fluid. The second variable is the percent concentration. Table 1 and Figure 1 define the most suitable metallurgies for given applications. When chemical process industries use hydrofluoric acid, its nature is generally aggressive. Consequently, worst case scenarios have more significance in decision-making choices.

Temperature and concentration are not the only variables that impact corrosion rates. Factors such as velocity, aeration and other contaminates play equally important roles in metallurgical corrosion.

Five metallurgies (Table 1 and Figure 1) are suitable for any given condition. Silver, gold and platinum are among the metals most resistant to hydrofluoric acid corrosion. Two other metallurgies are more affordable and provide excellent results, maintaining corrosion at less than 20 mils per year (mpy) during adverse conditions. One is 66Ni 32Cu (Monel 400), and the other is 54Ni 15Cr 16Mo (Hastelloy C-276®). There are some pitfalls in the composition of 54Ni 15Cr 16Mo. This alloy is less resistant to corrosion than 66Ni 32Cu, especially if oxygen is present; whereas 66Ni 32Cu is generally corrosive resistant, even to temperatures up to 300°F. (With either alloy, stress-corrosion cracking may be inevitable if water or oxygen are present, and in that case, corrosion and cracking would be widespread and not localized.) Both of these metallurgies are excellent choices for handling hydrofluoric acid.

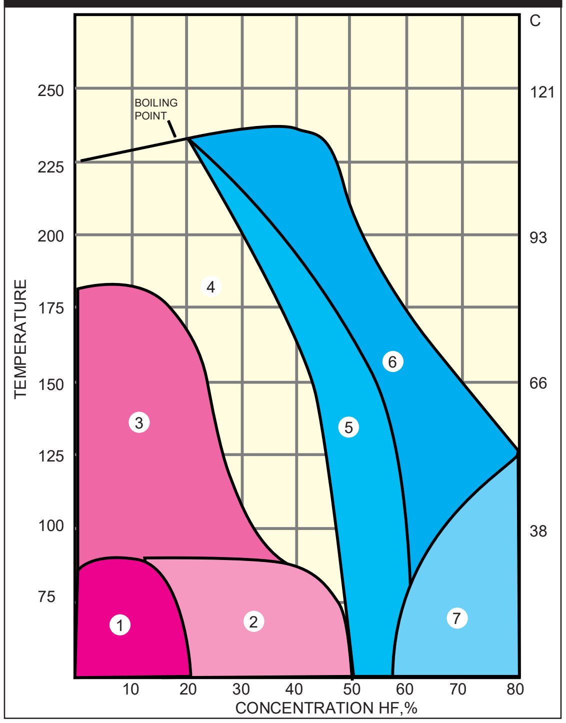
Temperature and percent concentration dictate what metallurgies are necessary under operating conditions.

BEARINGS

Bearing material is every bit as crucial to a pump's mechanical stability as its overall metallurgical composition because the bearings are exposed to the acid. This is particularly important in canned motor technology because the pumping process is responsible for the cooling and lubrication of the bearings.

What is the proper bearing material? What will hold up under the unforgiving corrosiveness of hydrofluoric acid? The answer is 100% alpha grade silicon carbide, which is a pressureless sintered silicon carbide. Bearings made of this material can withstand high temperatures and maintain dominating resistance to strong acids. Alpha grade has better resistance to wear and abrasion than the beta version of silicon carbide. However, both are pressureless sintered, or self-sintered, silicon carbide products.

FIGURE 1



Zone definition for common metallurgies

Do not use reaction-bound silicon carbides for hydrofluoric acid processes. These forms of silicon carbide contain free silicon or graphite because reaction-bound silicon carbides require silicon as a sintering aid. Free silicon is subject to the attack of corrosive acids, resulting in bearing breakdown. In alpha and beta grades of silicon carbide, no sintering aids are used, giving both grades almost complete chemical inertness. The bottom line is that there is little difference between the alpha and beta grades of silicon carbide. Most of the difference lies within the processing of the final products. Nonetheless, alpha grade silicon carbide is the preferred material for chemical processes that use hydrofluoric acid. Both alpha or beta grades of silicon carbide should exceed bearing expectations.

HYDRAULIC CONFIGURATIONS

For hydrofluoric acid applications, the same challenges arise again and again: low NPSHA, low flow and high head. In centrifugal pumps, low flow and high head yield low specific speed, meaning poor hydraulic efficiency.

Specific speed is a dimensionless number that relates the hydraulic performance of centrifugal pumps to the shape and physical properties of its impeller. The equation to calculate specific speed is shown in Figure 2. Where low flow and high head are requirements, use a partial emission pump with an open or closed radial vane impeller. A standard guideline for pumping a fluid as volatile as hydrofluoric acid is to keep the specific speed above 200. This ensures the pump will maintain a reasonable hydraulic efficiency (25% to 30%).

Specific speed can be easily manipulated by increasing, gallon by gallon, the flow of a pump until the desired N_S value is achieved. Another way to impact specific speed includes increasing the rotative speed. This technique is sometimes difficult because many motors have fixed rotating speeds. To manipulate speed, a variable frequency drive must be used. A variable frequency drive can increase the speed at which a motor runs while maintaining a constant voltage. However, these devices can be expensive. Regardless, the results are the same—increased hydraulic efficiency.

Hydraulic efficiency is important when pumping hydrofluoric acid because it has a steep vapor pressure curve. Unproductive energy, which is a direct byproduct of inefficiency, is lost in the form of heat. This added heat must not be allowed to localize in the suction zone of the pump case. If it does, and suction pressure is not great enough to suppress vaporization, the pump may fail. The ability to carry the heat away is directly related to the specific heat of the fluid. Specific heat is the ratio of a fluid's thermal capacity to that of water at 15°C; in other words, a fluid's ability to carry away energy in the form of heat. Unfortunately, this thermodynamic property of hydrofluoric acid is fairly poor. If adequate NPSHA existed for most hydrofluoric acid applications, the ability of the process to dissipate heat would not be crucial. However, NPSHA is often lacking.

NPSHA is the net pressure of a process fluid at the suction of a pump. Having adequate NPSHA is important when pumping hydrofluoric acid because of the volatility of the process. The graph in Figure 3 demonstrates the relationship of tem-

TABLE 1. CODE FOR HYDROFLUORIC ACID GRAPH

Materials in shaded zone have repeated corrosion rate of <20 mpy

Zone 1
20Cr 30Ni
25Cr 20Ni Steel
70Cu 30Ni[1]
66Ni 32Cu[1]
54Ni 15Cr 16Mo
Copper[1]
Gold
Lead[1]
Nickel[1]
Nickel Cast Iron
Platinum
Silver

Zone 2
20Cr 30Ni
70Cu 30Ni[1]
54Ni 15Cr 16Mo
66Ni 32Cu[1]
Copper[1]
Gold
Lead[1]
Nickel[1]
Platinum
Silver

Zone 3
20Cr 30Ni
70Cu 30Ni
54Ni 15Cr 16Mo
66Ni 32Cu[1]
Copper[1]
Gold
Lead[1]
Platinum
Silver

Zone 4
70Cu 30Ni
66Ni 32Cu[1]
54Ni 15Cr 16Mo
Copper[1]
Gold
Lead[1]
Platinum
Silver

Zone 5
70Cu 30Ni[1]
66Ni 32Cu[1]
54Ni 15Cr 16Mo
Gold
Lead[1]
Platinum
Silver

Zone 6
66Ni 32Cu[1]
54Ni 15Cr 16Mo
Gold
Platinum
Silver

Zone 7
66Ni 32Cu[1]
54Ni 15Cr 16Mo
Carbon Steel
Gold
Platinum
Silver

[1] = No air

FIGURE 2

Example A

$$N_s = \frac{NQ^{1/2}}{H^{3/4}} \qquad \left(N_s = \frac{3550\,(15^{1/2})}{900^{3/4}} \qquad N_s = 83.7 \right)$$

N_s = Specific speed
N = Revolutions per minute
Q = Capacity, at best efficiency, in gpm
H = Total head developed by maximum diameter impeller at best efficiency, in feet

Equation to calculate specific speed

perature to pressure. As the temperature rises, the required pressure to maintain the acid in a liquid phase increases, and the vapor pressure curve becomes dramatically steeper at higher temperatures. Any point left of the curve means the process is liquid; conversely, any point right of the curve means the process is vapor. If hydrofluoric acid is being pumped at 100°F, the NPSH must be equal to or greater than 27 psi. If not, the process will flash, resulting in a heavily cavitated or dry running pump.

To ensure adequate NPSHA, the heat input from the pump must be considered. Hydraulic temperature rise can be calculated. The equation (Figure 4) considers three variables: hydraulic efficiency, head, and the specific heat value of a process. By using this equation and considering the vapor pressure versus temperature rise curve, you can predict if adequate NPSHA is provided. The following is an example.

PUMPING SPECIFICS

Fluid Pumped = HF acid
Head (H) = 790 feet
Flow (Q) = 20 gpm
NPSHA = 7 feet, Mechanical NPSHR = 6 feet
Temperature (P) = 95°F
Vapor Pressure (P.T. PSIA) = 25
Specific Heat (BTU/lb°F) = 0.78
Pump Hydraulic Efficiency (n) = 15%
Specific Gravity (sp gr) = 0.92

1. Solve for the slope of the vapor pressure curve. Pick one temperature/PSIA point below the design temperature and another 20°F above the operating point. Convert data into PSIA per °F.

$$\frac{17\text{ PSIA} - 36\text{ PSIA}}{75°F - 115°F} = \frac{19\text{ PSIA}}{40°F} = 0.475\text{ PSIA/°F}$$

2. Solve for the maximum allowable temperature rise that can occur before the HF flashes.

$$\frac{(\text{NPSHA} - \text{NPSHR})\text{ sp gr}}{2.31} = \frac{\text{PSI}}{0.475\text{ PSI per °F}}$$

= maximum allowable temperature rise °F

(Actual)

$$\frac{(7 - 6)\,0.92}{2.31} = \frac{0.39\text{ PSI}}{0.475\text{ PSI per °F}}$$

= 0.83°F allowable temperature rise °F

3. Calculate hydraulic temperature rise due to inefficiencies.

$$\frac{H\,(1 - n)}{778 \times n \times Cp} = \text{Temperature Rise}$$

Where
n = hydraulic efficiency
Cp = Specific heat

(Actual)

$$\frac{790\,(1 - 0.15)}{778 \times 0.15 \times 0.78} = 7.37°F$$

Conclusion:
Slope of curve = 0.475 PSIA/°F
Max allowable temperature rise allowed = 0.83°F
Total hydraulic temperature rise = 7.37°F

In this example adequate NPSHA is not being supplied. If this data is plotted on a vapor pressure versus temperature curve, the end result is obvious—the HF is vapor (Figure 5).

To calculate how much NPSHA is necessary to keep the HF from vaporizing:

1. Total hydraulic temperature rise = 7.37°F
2. Convert 7.37°F to PSIA using calculated vapor pressure curve slope and consider allowable temperature rise (0.83°F)

   7.37°F – 0.83°F (Allowable Temperature Rise) = 6.54°F
   6.54°F x 0.475 = 3.1 PSI

   Convert 3.1 PSIA to feet

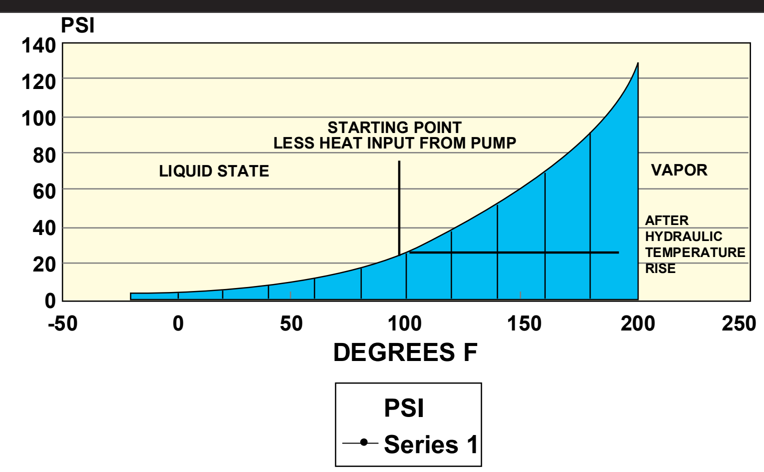
   $$\frac{2.31 \times 3.1\text{ PSIA}}{\text{sp gr }(0.92)} = 7.78\text{ feet}$$

3. Thus, 7.78 feet in addition to current NPSHA must be provided.

Current = 7 feet + 7.78 feet Newly Calculated = 14.78 feet Total NPSHR

These calculations are conservative because they assume that the total temperature rise will take place at the suction of the pump. This tends to be valid at minimum flows, but is conservative at design flow. These examples do not encompass every possible scenario that could be experienced, but they are effective

FIGURE 3



Graph demonstrating the relationship of temperature to pressure for hydrofluoric acid

guidelines in the determination of adequate NPSHA. In addition to these calculations, always multiply your final calculated NPSHR by a 1.3 safety factor to ensure a successful pump application. Clearly, increasing NPSHA can be an expensive proposition. However, it may be lower in cost than reinvesting money into a problem pump caused by borderline NPSHA versus NPSHR margins. NPSHA, flow and head play equally important roles when selecting a pump configuration.

PUMP CONFIGURATIONS

Several effective pump configurations exist for handling hydrofluoric acid. Before selecting a configuration, accurately evaluate the NPSHA versus NPSHR, flow, head, efficiencies, and temperature rise.

Canned motor pumps offer two designs that are extremely effective for pumping hydrofluoric acid. These designs can pressurize the fluid in the motor to increase vapor pressure margins or to reverse the motor flow (internal circulation) direction, routing the heated process to the suction tank rather than the pump. These design capabilities are important due to the temperature gained from viscous drag, eddy current losses, and motor inefficiencies. Although different, they fundamentally achieve the same end result, keeping the process from flashing in the motor. The basic premise of both designs is to increase the pressure in the motor so that, even though the process temperature is rising, there is still adequate pressure in the motor to keep the acid liquid. These configurations are often referred to as pressurized or reverse circulation designs.

Magnetically coupled pumps are also ideally suited to the handling of hydrofluoric acids. Mag-drive pumps with metallic containment shrouds (sometimes called cans) also produce eddy current losses that transmit heat to the pumpage. Some manufacturers offer different internal circulation paths, including rear-mounted impellers to compensate for pressure drop and temperature rise.

Several mag-drives are available with nonmetallic shrouds. In these no heat is produced due to eddy current losses, which increases overall pump efficiency and decreases motor requirements in most cases. Shroud materials include ceramics, silicon carbide, PEEK and reinforced fluoroplastics. Remember, factors such as inlet temperature, NPSH, contaminates and system requirements must be taken into account with either canned motor or magnetically coupled pumps.

Regardless of the pump style, several auxiliary items can smooth the path toward safe and effective operation. If low NPSHA is a factor, an inducer can lower a pump's NPSHR. This can sometimes alleviate costly system cha-nges. Over and under current measuring relays effectively protect against dry running. These simple devices enable the pump operator to set a minimum amperage draw based on the specific functional curve amperage draw. If the current drops below the set point, the pump will automatically shut down.

Thermowells and temperature switches are also effective in detecting overheating of the process within the pump. Often when pumps or systems are experiencing disturbances, process temperatures increase. These devices signal a possible problem and allow for review of the pump and system before extreme damage occurs. However, they are not independent of process temperature fluctuations and may be effective only in constant temperature applications. Bearing monitors are also important because they can detect problems, such as fracturing with silicon carbide bearing systems. Some of the most effective bearing wear monitors detect axial and radial wear. These monitors are important in scheduling proactive maintenance versus reactive maintenance, which is critical when unscheduled downtime can mean lost revenue.

If pump metallurgies, bearing materials, and hydraulic and pump configurations are approached properly, pumping hydrofluoric acid can become as routine as brushing your teeth. ■

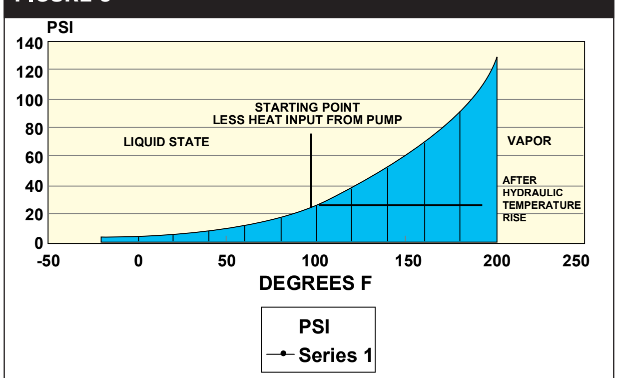
FIGURE 4

$$\frac{H\,(1-n)}{778nCp} = \text{Hydraulic temperature rise due to inefficiencies of pump performance}$$

H = Rated head in feet at design flow
n = Rated efficiency at design point
Cp = Specific heat of a process fluid defined as BTU/lb°F

Equation for hydraulic temperature rise

FIGURE 5



Vapor pressure versus temperature curve

ACKNOWLEDGMENTS

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User Perspective: When to Apply Mag Drive Pumps

Making the move for the right reasons.

BY: MAURICE G. JACKSON

Magnetic drive centrifugal pumps offer an advantage over normal single mechanical seal centrifugal pumps by preventing fugitive emissions from leaking to the atmosphere. Given proper application and operating procedures, these pumps can perform for years without failure. Rather than discuss the design of these pumps - a subject that has already been thoroughly addressed in articles, papers and presentations - let's review the justification for installing mag drives and provide installation keys to insure reliability of the investment.

KNOW SECONDARY CONTAINMENT OR CONTROL REQUIREMENTS

Secondary containment and secondary control are important terms to understand when selecting your mag drive pump. **Secondary containment** insures the fluid will be contained if the primary can fails. Some mag drive suppliers accomplish containment by installing a secondary can around the primary unit. If the primary can develops a leak **secondary control** insures the leakage will be controlled to a defined amount, but not contained. In selecting a mag drive pump be sure to know whether secondary containment or secondary control is specified for your application.

CONSIDER LIFE CYCLE COSTS

Magnetic drive pumps are often the only alternative to meet government hazardous materials and safety regulations, such as OSHA 1910, requiring stringent levels of containment or control. In addition, many companies now have policies, odor-free imperatives for example, requiring strict control of emissions. However, for some zero emissions applications tandem seal pumps offer a viable alternative to mag drives and life cycle cost must be considered in the selection criteria. Table 1 shows calculations of life cycle costs of tandem seal versus mag drive pumps for two specified applications. In the first example, the mag drive pump has a significantly lower initial cost and operating costs only slightly higher than the tandem seal. In this application, because mag drives offer more reliable containment most users would select the mag drive. In the second example, however, the mag drive proves to be more costly in terms of both investment and operating cost, and use of the mag drive can not be justified in terms of cost alone.

DON'T INSTALL MAG DRIVES TO OVERCOME SYSTEM PROBLEMS

Magnetic drive pumps should not be installed to solve a maintenance problem, such as a troublesome mechanical seal, without first determining the real reason for the problem. Once the problem has been identified, insure that installation of mag drive pumps will not create a ripple effect. Typical pump and system problems to watch for are:

- cavitation
- operating too far from best efficiency point (BEP)
- Net positive suction head available (NPSHA) too low
- slurries
- pump operating without liquid in the unit

COMMUNICATE WITH YOUR VENDOR

A user of magnetic drive pumps should also be aware of potential problems and communicate with the vendor to insure they are avoided. For example, the drive motor should always be sized smaller than the magnets to prevent decoupling if the impeller is overpowered. Decoupling of the magnets will create excessive heat build-up in the fluid. Decoupling may also result in a locked rotor due to failure of the pump bearings and can. In addition, because magnetic drive pumps are often used to pump low – less than 1.0 – specific gravity fluids and are generally sized for such applications, an operator should be aware that employing the pump for water or higher specific gravity applications may overpower the motor or magnets. Failure to communicate fluid properties may lead to additional problems. A vendor will need to know more than fluid viscosity and specific gravity to size your magnetic drive pump. Users should also specify vapor pressure vs. temperature data and specific heat, as well as size and percentage of solids for the fluid being pumped.

USE PROTECTIVE INSTRUMENTATION TO INSURE RELIABILITY

To insure the reliability of mag drive pumps, protective instrumentation is recommended. Listed below are some typical instrumentation available and their features.

- Power meter – monitors power to the motor driving the mag drive pump. The meter can be used to prevent dry and dead headed operation. The power meter is probably the best choice if you are limited to the selection of one type of monitoring instrumentation.
- Can thermocouple – mounted on the can, it senses dry running and bearing problems.
- Bearing wear detector – is used to sense the position of the shaft or rotor. It can provide an indication of the condition of the pump's bearings.
- Level detector – is placed in the suction or discharge piping to insure liquid flow to the pump.

TABLE 1

Based on one year[1] operation

Application One: 400 gpm; 120 ft head		
Pump Type	Tandem Seal	Magnetic Drive
Initial Cost	$13800	$11350[2]
Basic hp Required	17.5	20.4
Seal hp Required	1.5	NA
Total hp Required	19	20.4
kW•h	129400	138900
Power Cost	$6470	$6945[3]
Product Loss	$300	$0[4]
Maintenance Cost	$600	$600[5]
Total operating cost per year	$7370	$7545[6]

Application Two: 100 gpm; 240 ft head		
Initial Cost	$7000	$8000[2]
Basic hp Required	11.4	14
Seal hp Required	1.5	NA
Total hp Required	12.9	14
kW•h	84426	95320
Power Cost	$4220	$4776[3]
Product Loss	$300	$0[4]
Maintenance Cost	$600	$600[5]
Total Operating Cost per year	$5120	$5376[6]

[1] Table data based on operation 350 days per year, 24 hours per day.
[2] Material of construction is 316 stainless steel.
[3] Electric power calculated at $50 per 1000 kW. Electric motor efficiency of 92% assumed in calculation of kW usage.
[4] Product loss calculated at $50 per pound.
[5] Maintenance costs assumed a failure once every three years. The failure modes are assumed to be seal failure for the tandem seal and bearing failure for the mag drive.
[6] Figures do not include initial cost.

Life cycle cost calculations for Tandem Seal Vs. Magnetic Drive Pumps

CONCLUSION

Recent developments in magnetic drive pumps, harboring many functional and maintenance advantages for pump users, are testimony to an exciting future for magnetic drive centrifugal pumps. ■

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Interpreting Sealless Pump Failures

The causes of part failures in sealless centrifugals may determine system and operational problems.

Sealless pump failures can highlight system or operational problems once taken for granted or blamed on mechanical seals. Once a pump has failed, it should be taken apart to identify the broken part or problem area. Frequently, a broken part can indicate the cause of failure. By establishing and remedying the origin of the failure, pump service life can be extended and future failures minimized.

The following describes part failures and their causes that indicate system and operational problems.

DAMAGED THRUST SURFACES (FRONT OR REAR)

Cause - operation below the acceptable minimum flow rate

Many pump users think of minimum flow relative to temperature rise and bearing wear problems. However, extreme low-flow operation in a centrifugal pump can also create hydraulic imbalance of the impeller, generating thrusting and vibration. Because sealless pumps do not have the shaft overhang typical of sealed pumps, an imbalance can cause extreme axial shuttling of the rotating assembly which may break thrust surfaces.

When a pump's desired operating point is at a very low flow rate, check with the pump manufacturer for the minimum rate. If the desired flow rate is below the recommended minimum, add a recirculation loop to increase throughput and prevent hydraulic imbalance.

Cause - insufficient net positive suction head (NPSH) available

A sealless pump may require more NPSH to insure that the hydraulic balance is maintained and the bearing system contains enough fluid. NPSH problems can result in shuttling of the rotating element, making it bang against thrust surfaces, and this can lead to rupture of the containment shell or liner. Repipe the system to reduce suction piping friction losses. Removing unnecessary valving or changing the pump elevation will solve the problem.

Cause - water hammer

With sealless pumps, water hammer can manifest itself by causing failure of the thrust surfaces as the rotating element is slammed against them. To solve the problem, review valve operating sequences and piping arrangements. Slow down valve closing speeds or change valve types to reduce water hammer.

INTERNAL SLEEVE BEARING FAILURE

Cause - operating the pump dry

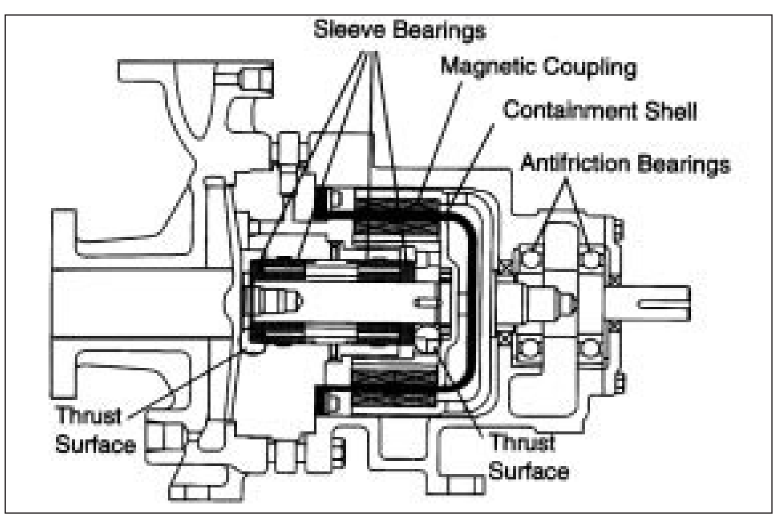
Sealless pumps require fluid to cool the bearings. Lack of fluid passing through the bearings causes thermal expansion of the bearing or journal, depending on the particular design. This expansion constricts passages, increasing friction and heat, and thereby causing the pump rotating element to lock up and cease operating. Alternatively, if the dry running operation is short, the bearings may heat up enough to fracture due to thermal shock when fresh fluid is introduced.

The first solution is to avoid running sealless pumps dry. If running dry is possible due to the desired method of operation, install a recirculation line around the pump to insure that fluid will always be running through it.

Cause - low fluid vapor pressure

Pumping fluids at temperatures close to their vapor pressure can create problems. In a sealless pump, fluids close to vapor pressure can flash as the fluid, in passing, picks up heat from the containment shell or bearings. This additional temperature rise brings liquids closer to their vapor pressure, and only a small amount of additional heat from the bearings may increase the liquid temperature above the vapor pressure limits, causing the liquid to flash and preventing it from cooling the bearings. Bearing failure in a sealless pump requires prompt attention to minimize the cost of the repair and prevent external leakage of the fluid.

If the fluid is close to the vapor pressure before it enters the pump, or if it has characteristics that suggest that a small temperature change will produce a large change in the vapor pressure, ask your sealless pump manufacturer to predict the expected temperature rise in order to verify that flashing will not occur in the bearings. To prevent flashing, some pump designs incorporate sec-



Magnetic Drive Pump

ondary pumping devices to increase pressure as fluid moves into the bearings. Other designs offer secondary cooling to solve the problem.

CONTAINMENT SHELL FAILURE

Cause – drive magnet contacting the outer surface

When the antifriction bearings supporting the drive magnet in a magnetic drive pump fail, the magnet may contact and rupture the containment shell. Such contact is indicated by grooves and rub marks on the external surface of the containment shell. If the pump design has safety rub rings installed, check clearances to insure they are correct. Replace the rings when repairing the antifriction bearings.

Cause – internal pressure higher than containment shell design limits

Water hammer can burst the containment shell. The sudden increase in pressure can drive rotating elements against thrust surfaces and put increased shock into the containment shell. Alternatively, the increased pressure alone can distort and burst the containment shell. In this case, the containment shell will show signs of expansion or distortion from the inside out. Repipe the system to reduce suction piping friction loss. Removing unnecessary valving or changing the pump elevation will solve the problem.

Cause – pump hydrotest pressure was above design limits

Caution should be used when hydrotesting assembled pumps that have nonmetallic containment shells. Shells using a fiber fill for strength may not rupture the first time they are exposed to pressure, but the fibers inside the material may be broken. If so, the next time pressure hits them, the shells may burst due to ineffective fiber reinforcement. Review instruction manuals and technical data and do not hydrotest nonmetallic shells above recommended pressures.

Note - Hydraulic thrust balance

Magnetic drive pumps and canned motor pumps frequently have specific clearances that hydraulically control the amount of thrust which the rotating elements experience. When thrust surfaces or bearings fail, the subsequent internal rubbing that takes place can increase wear on balancing surfaces and reduce the effectiveness of the hydraulic thrust control. If only the failed part is replaced, the next failure may then result from lack of hydraulic thrust control. When in doubt, always replace a hydraulic thrust surface part that is worn.

SUMMARY

The best solution to pump failures is always prevention. Pump products should be properly applied at all times. Don't hesitate to contact the manufacturer or his representative to ask for help, and be sure to describe the application, installation and operating conditions for the pump thoroughly. Also, save all parts. An examination of them may provide invaluable clues to the origins of pump failure, offering keys to overcoming systems or operational defects. ■

ABOUT THE AUTHOR:

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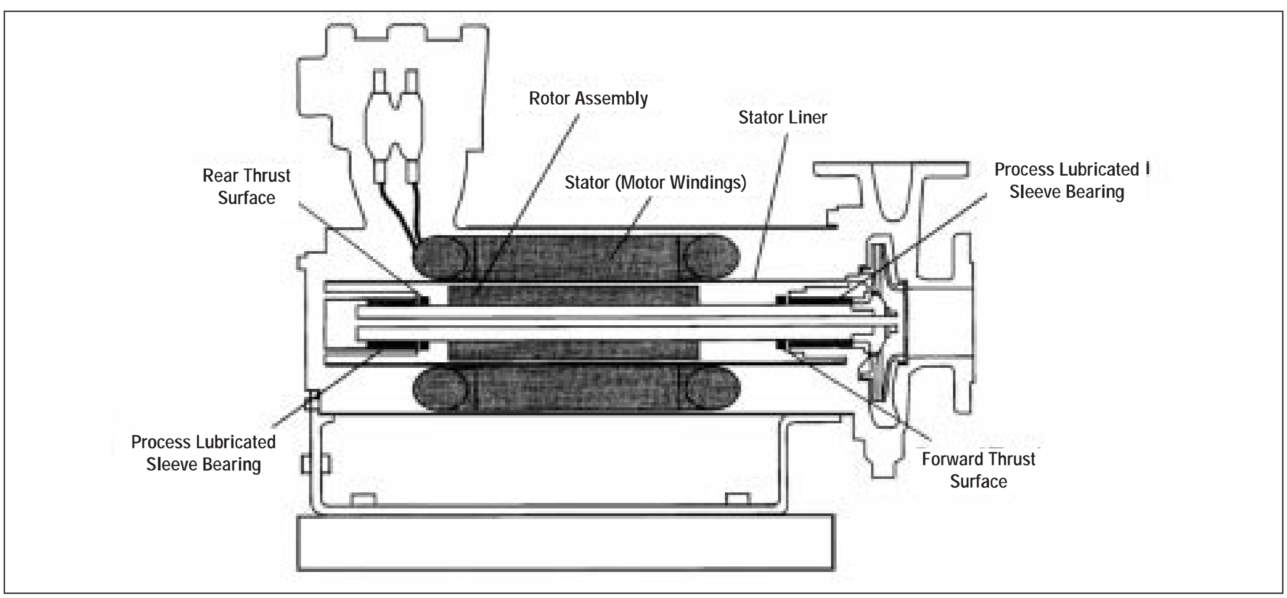


Canned Motor Pump

Magnetic Couplings for Sealless Pumps

Elimination of seals ends leakage concerns.

BY RONALD P. SMITH

BACKGROUND

Electric-motor-driven pumps have been around for about 75 years, and so has the nagging problem of the shaft packing or seal. Because water was the common fluid pumped, it rarely became a dangerous problem. However, as the chemical industry developed, leakage became a major concern, and better seals were needed and developed.

Industries are now under scrutiny for hazardous emissions of all types, and must comply with clean air and water regulations dictated by Congress and implemented by the EPA, OSHA, and other government agencies.

Currently, any leakage of liquid or gas is a problem and must be minimized or eliminated. The state of the art for mechanical seals is in the range of 500 ppm leakage, with some releasing as little as 100 ppm. By using secondary seals with drainage and control instrumentation, levels closer to zero can be accomplished at increased cost to the user.

THE BASIC PROBLEM

If we can accept that contacting surfaces with relative motion between them will eventually wear, then we can conclude that in the case of mechanical pump seals, leakage will ultimately occur. So it is desirable to do away with any shaft seal. By not penetrating the pump housing with a shaft, the seal is eliminated, along with any potential leakage. This can be done by totally enclosing the impeller/pump assembly and isolating it from the prime mover.

The question is, how do you drive the pump with no direct connection to a prime mover (motor)?

A SOLUTION

Fortunately, we have a natural force, magnetism, that can be used to our benefit. As children, we experienced the magnetic force of two magnets operating through a table top or pane of glass. One magnet would follow the other until the gap between them became too large and reduced the force. That basic idea is used in synchronous magnetic couplings.

There are two basic styles of magnetic couplings in use. Figure 1 shows a face-face coupling and Figure 2 illustrates a co-axial design.

Magnetic couplings can be made to develop almost unlimited forces, based on choice of material and scale. Coupling designs for hundreds of foot-pounds of torque are available.

One of the most fascinating aspects of permanent magnet couplings is that although they exhibit powerful forces of attraction and repulsion, they require no outside sources of power. If properly used, they last indefinitely.

COUPLING CHARACTERISTICS

In any synchronous coupling, torque is developed in the same way. The maximum attractive force between the poles occurs when the poles are aligned in opposite polarity. Maximum repulsive force occurs when the same polarity is aligned. In both instances, the transverse force (torque) is at a null (zero). The latter position is the least stable. The maximum transverse force (torque) occurs between the two positions where the normal force is zero. Stable positions occur only once per pole pair, so in the case of a 10-pole coupling, there would be five stable positions.

The proper application of a permanent magnet coupling requires knowledge of the maximum torque produced by the motor. This is typically twice the amount produced at the rated horsepower.

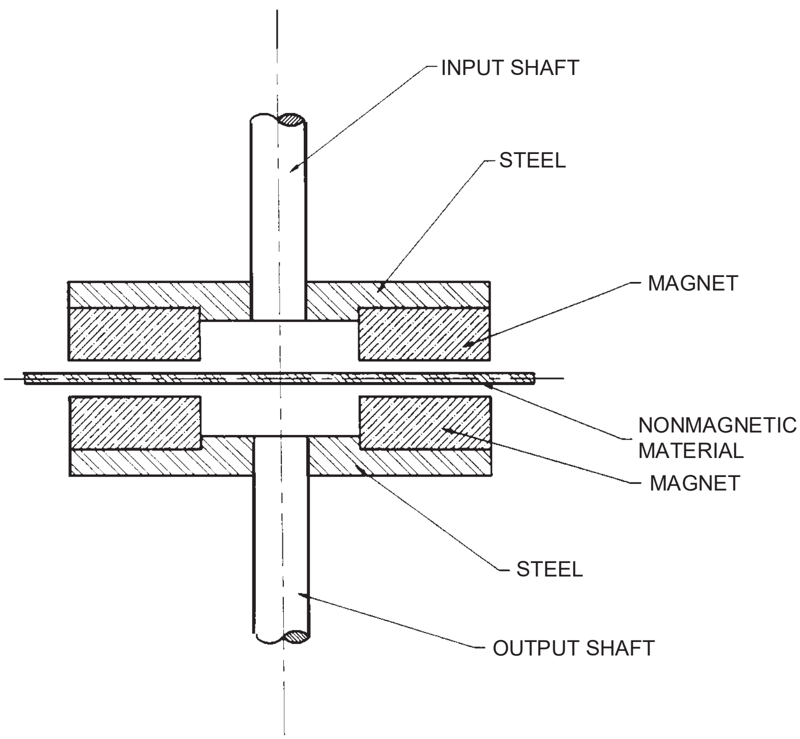
Running torque =
(Rated Horsepower x 5,250) /rpm (ft-lbs)

In the case of a 5 Hp motor at 1,800 rpm with no load, the running speed with about 3% slip is 1,750 rpm and running torque is:

(5 Hp x 5,250) /1,750 = 15 ft-lbs.

However, the motor will develop about 30 ft-lbs peak torque during line start, and a magnetic coupling must have a peak torque rating at least that high to prevent loss of coupling. Figure 3 displays the relationship of peak to running torque. The amount of safety factor for the appli-

FIGURE 1



A face-face magnetic coupling

cation will determine the exact design point on the curve. Slow start conditions can reduce the amount of peak torque required in the coupling and provide overload protection for impellers in case of a mechanical jam.

In a coaxial coupling the radial forces are balanced if all of the magnet segments are of equal strength. Concentricity of the inner and outer assemblies is also required for equal air gap distance. These factors develop "magnetic balance," which is as significant as physical balance in reducing noise and bearing wear.

Face–face couplings develop significant axial forces. When in the aligned, attractive mode, the force is at a maximum. At peak torque, the axial force approaches zero. If slippage occurs, it goes through a maximum in the opposite direction. This coupling design requires proper bidirectional thrust bearings on each member to handle the variable forces. Inadequate bearings will allow air gap variations that cause mechanical noise and can be self destructive. For this reason, the face–face coupling is generally restricted to special applications.

The torque of face–face couplings is limited by the allowable maximum diameter of the assembly. A coaxial coupling can be made longer for increased torque once a maximum diameter is reached. The torque is essentially linear with axial length. This benefit makes the coaxial design the one of choice for most applications.

STIFFNESS

If rapidly fluctuating loads cause mechanical resonance with the pump couplings, changing the number of poles will modify the stiffness. Relationships of pole spacing to gap length must be taken into account to maintain design efficiency.

UNCOUPLING, SPECIAL CASES

The uncoupling phenomenon limits torque and is very useful. In pumps, it might protect the impeller from damage or detect unacceptable thermal conditions. Obviously, a coupling can be made to exceed the torque of the motor, as in a mechanical coupling, and use motor thermal or electrical overload protection to shut down the system.

Most pumps are designed with a coupling that will not slip or uncouple within rated performance and proper motor application.

In the unique case where the peak coupling torque is exceeded, slippage or uncoupling results. The impeller then stops, and no fluid is pumped. The seriousness of this situation will depend on the application and coupling design. The system should detect lack of flow and shut down the pump before any major damage occurs. A low-level audible warning may be heard from the coupling.

Inertia of the system will not allow "pick up" of the impeller magnet until the motor is stopped. Before restarting the pump, the cause of the uncouple should be determined.

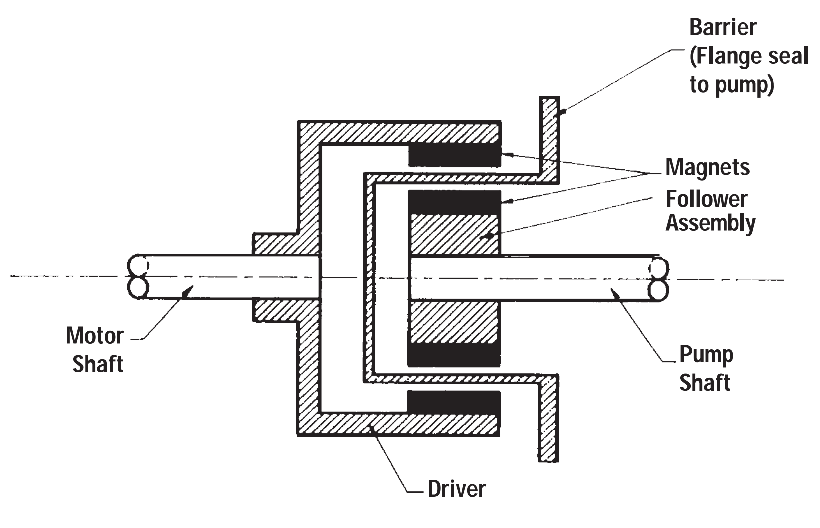
Running uncoupled for long periods should be avoided. Because the impeller is not rotating and no fluid is being pumped, no fluid is being circulated through the containment can and no cooling of the coupling occurs.

When one magnet element is rotating past the other, a significant amount of energy is converted to heat, and because the inner unit usually has a poor thermal escape path, it will get hot. If the temperature rises past the design point, demagnetization can occur. This is either temporary (recovered when the coupling cools down) or permanent (recovered only by remagnetizing).

If a coupling has a nonmetallic barrier such as ceramic or plastic, there will be no additional uncoupling effect.

Metallic barriers of stainless steel, Hastelloy, etc., will heat up rapidly due to eddy currents and, depending on the fluid contained, could represent a dangerous condition. If the additional heat raises the

FIGURE 2



A co-axial magnetic coupling

temperature of the magnets, a further reduction of force by demagnetization is possible. Either of these cases could affect restart and necessitate a "cool down" before restart. This might not be a concern, because some time should be spent identifying the cause of the high torque requirement.

CONTAINMENT BARRIERS

The containment barrier is a key element to sealless pump success. It provides the primary fluid containment and the "window" to couple torque in the system. Like other elements in the system, it usually is connected to the pump with a flange and an O-ring seal.

The containment barrier is also a critical part of any permanent magnet coupling design. Its magnetic and electrical characteristics affect the heating and power losses of the system. The wall thickness and associated mechanical gaps determine the magnetic air gap and the amount of magnetic material required for a given torque, and therefore significantly impacts the cost of the coupling.

Table 1 displays some of the common barrier materials, along with their benefits and related costs.

Permanent magnet couplings can easily handle larger air gaps than allowed in canned motors. This is a major benefit when handling high viscosity fluids or when suspended particles are in the fluid. Magnetic particles are to be avoided because they may collect between the magnet and barrier. Air gap clearance on either side of the barrier should be as small as possible, but their size depends on allowable bearing wear. If either rotating magnet assembly is allowed to contact the barrier, the pressure vessel may be compromised and failure can occur. Mechanical "rub rings" or proximity detectors can be used to indicate bearing failure.

It is usually not practical to make the coupling barrier shell an integral part of the pump housing. This shell should have the thinnest wall possible that satisfies the design pressure requirements. The material must be nonmagnetic and preferably nonmetallic to reduce eddy current heating and associated power losses.

There are many barrier designs, the most common being plastic shells for small pumps and stainless steel for large pumps, with pressure requirements up to thousands of pounds per square inch. Chemicals being pumped dictate the choice of material, and frequently the shell is made of the same material as the pump housing.

Ceramic shells, coated metal, and laminated metal are used for special cases.

In the case of solid metallic barriers, eddy current heating is developed. This is torque transfer loss and can amount to 5–10% of the input power. Generally ignored in small systems, it may be significant with motors over 100 Hp. Most cooling can be through the fluid if a generous flow within the barrier is established. Additional heat dissipation through the pump housing is possible if the barrier shell has a metal–metal contact to the housing. Eddy current heating can be reduced by lowering the speed of the motor, as losses are proportional to speed.

Because the containment barrier becomes a pressure vessel, fabrication techniques are important. Designs are guided by ASME standard and manufacturing processes dictated by quantity. Small- to medium-size barriers are usually machined from solid bar stock in small quantities. Spun, hydroformed, or deep-drawn shells may be more economical in large quantities. Welded units, which are feasible in all sizes, require close process control to avoid stress corrosion problems. Pressure testing may be part of part certification.

TABLE 1- BARRIER MATERIAL COMPARISON

Material	Wall Thickness	Pressure Capability	Chemical Resistance	Eddy Current Heating	Relative Cost
Plastic	Medium	Medium	High	No	Low
Ceramic	Thick	Low/Medium	Medium	No	High
Stainless Steel	Thin	High	High	Yes	Medium
Hastelloy	Thin	High	High	Yes	High
Titanium	Thin	High	High	Yes	High

COUPLING DESIGN

In concentric couplings, the driving element connected to the motor is usually the outer magnet assembly. This part of the magnetic elements has the highest mass and inertia. It can be made of magnetic iron and painted, plated, or coated as necessary. It is usually not subjected to high corrosion environments and does not require special sheathing.

Salient magnet poles have gaps between them. These may be filled with epoxy or other potting compounds to improve cleaning and minimize magnet damage during assembly.

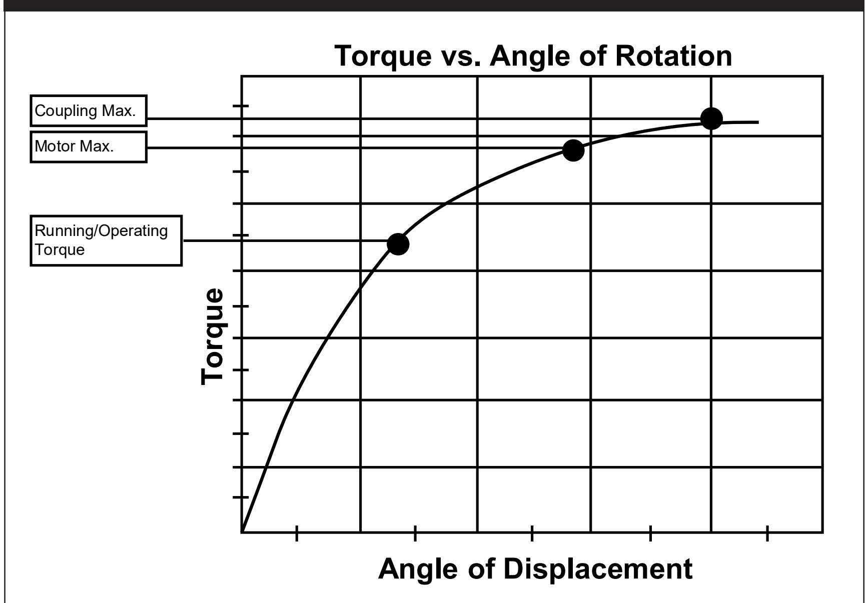
The inner magnet assembly is the "follower." Because it is in the process fluid, special care must be taken to prevent corrosion or contamination of the pumpage. This assembly typically has rare earth magnets mounted on an iron ring. If these parts will be corroded by the fluid, they are sheathed or coated with appropriate materials. Frequently, a stainless hub is used with a stainless sheath welded to it, totally encapsulating the magnet assembly. Welding techniques are critical to long-term life, and leak or pressure testing is advised.

Bathed in the process fluid, the follower assembly may be affected by temperature extremes. For temperatures over 100°C, efficiencies decrease and designs become more material specific.

Factors affecting design are:

- magnetic gap length (barrier + clearances)
- peak torque required
- space available for coupling
- form factor desired for complement to pump (diameter x length)
- stiffness required
- fluid and corrosion concern
- maximum operating temperature
- running speed
- shaft sizes
- barrier material and allowable eddy current heating

FIGURE 3



The relationship of peak to running torque

INDUSTRY STANDARDS

Permanent magnet couplings are an integral part of a pump. Due primarily to the need for critical alignment, they are sold as a unit with the motor. Pumps are manufactured to meet industry standards such as those published by ANSI, API, and the Hydraulic Institute. These organizations have recently included standards for magnetic couplings.

Standards are based on voluntary compliance and in most cases insure interchangeability of parts among manufacturers. To meet customer needs for quality in specific applications, pump manufacturers have their own rigorous standards.

Permanent magnet materials are also produced to industry standards that allow sizeable variation in magnetic properties within material grades. Critical applications require greater control of properties and/or selection of parts for uniformity.

MAINTENANCE

Because there is no mechanical wear in a magnetic coupling, there should be no need for maintenance. However, bearings do wear, and occasionally a pump will require disassembly or a motor will need to be replaced. The manufacturer will have recommendations for handling this operation. Disassembly of the magnetic coupling should be done only by trained personnel using the proper fixtures.

Permanent magnet couplings contain some of the most powerful magnet materials ever made, and they are always energized. In systems of 5 Hp and over, the forces are greater than a person can control by hands alone. Large pump couplings have axial forces in the hundreds of pounds. Fixtures or mechanical means to guide the parts and prevent damage to the containment barrier are required during disassembly and re-assembly. Training is also required to avoid personal injury. Magnets can be very unforgiving of mistakes. ■

Ronald P. Smith is Manager of Engineering for the Magnetic Materials Division of Dexter Corporation.

Suction Side Problems - Gas Entrainment

BY: JAMES H. INGRAM

Multiple symptoms associated with noncondensable suction side gas entrainment, such as loss of pump head, noisy operation, and erratic performance, often mislead the pump operator. As a result, entrained gas is generally diagnosed by eliminating other possible sources of performance problems. To adequately control gas entrainment a user should first be aware of systems most likely to produce gas, and then employ methods or designs to eliminate entrainment into these pumping systems.

ENTRAINMENT VERSUS CAVITATION

The audible pump noise from noncondensable entrained gas will produce a crackling similar to cavitation or impeller recirculation. However, cavitation is produced by a vapor phase of the liquid which is condensable, while noncondensable entrained gas must enter and exit the pump with the liquid stream.

To test for gas entrainment over mild cavitation, run the pump back upon the curve by slowly closing the discharge valve. The noise will diminish if it originated from cavitation and the pump is not prone to suction recirculation. In contrast, with entrained gas, continued performance at this portion of the curve will choke off or gas-bind the pump, causing unusually quiet operation or low flow.

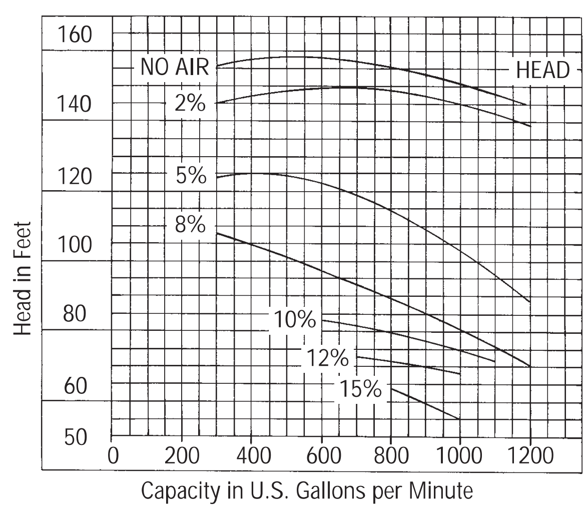
GAS BOUND IMPELLERS

As a process stream containing entrained gas nears the impeller, the liquid pre-rotating from the impeller tends to centrifuge the gas from the process stream. Gas not passing into the impeller accumulates near the impeller eye. As entrained gas flow continues to increase, the accumulating groups of bubbles are pulled through the impeller into the discharge vane area where they initiate a fall in flow performance. The bubble choking effect at the impeller eye produces a further reduction of Net Positive Suction Head Available (NPSHA). At this stage long term damage to the pump from handling entrained gases is generally negligible when compared with the damage due to cavitation. If the process stream gas volume increases, however, further bubble build-up will occur, blocking off the impeller eye and stopping flow (Ref. 1).

A pump in this gas bound state, will not re-prime itself, and the gas, with some portion of the liquid, must be vented for a restart against a discharge head. The effort to restart a gas bound impeller depends on

FIGURE 1. ENCLOSED IMPELLER-ENTRAINED GAS HANDLING PERFORMANCE

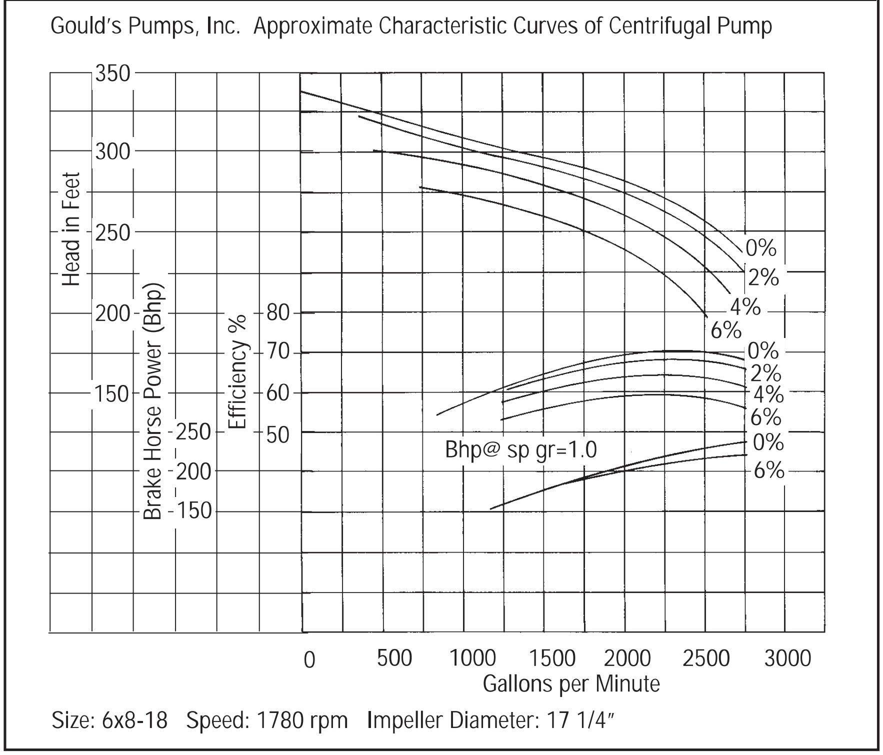
The LaBour Company, Inc. Effect on head and capacity of varying quantities of air with water being pumped.



Size: no. 55; Type: SQ. Speed: 1750
Impeller Diameter: 11″

Air quantities given are in terms of free air at atmospheric pressure referred to % of total volume of fluid being handled.

FIGURE 2. OPEN IMPELLER-ENTRAINED GAS HANDLING PERFORMANCE



impeller position, type and valving arrangement, among other variables. Degassing is easier to accomplish with a variable speed driver, such as a steam turbine, than with a constant speed electric motor drive. In addition, a recycle line to the suction vessel vapor space is often an effective method for degassing an impeller, since with this arrangement the pump is not required to work against a discharge head. (Ref. 1 describes methods for venting gas on modified pumps that are gas bound.)

As a rule, if the probability of entrained gas exists from a chemical reaction, the inlet piping design should incorporate a means to vent the vapor back to the suction vessel's vapor space or to some other source.

EFFECTS OF ENTRAINED GAS ON PUMP PERFORMANCE

Figures 1 and 2 illustrate the effect of entrained gas on a LaBour enclosed impeller and a Gould's paper stock open impeller. As illustrated by the figures, 2% entrained gas does not produce a significant head curve drop. Note that while the LaBour impeller experiences a 22% head loss at 5% gas volume, the Gould's open impeller experiences a 12% head loss at this volume. Some open impeller paper stock designs can actually handle up to 10% entrained gas because clearance between the case and impeller vanes allows more turbulence in the process fluid, which tends to break up gas accumulation more efficiently than an enclosed impeller with wear rings. In addition, other designs, such as a recessed impeller pump, may handle up to 18% entrained gas. In fact, most standard centrifugal pumps handle up to 3% entrained gas volume at suction conditions without difficulty. (Ref. 2 discusses open impeller pump modifications.)

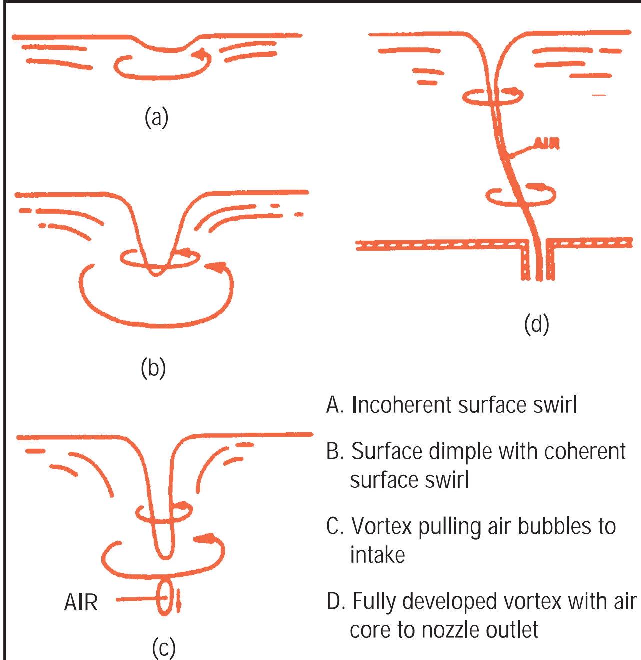
SYSTEMS PRODUCING ENTRAINED GAS

The most common conditions or mechanisms for introducing gas into the suction line are:

1. Vortexing
2. Previously flashed process liquid conveying flashed gas into the suction piping.
3. Injection of gas, which does not go into solution, into the pumpage.
4. Vacuum systems, valves, seals, flanges, or other equipment in a suction lift application allowing air to leak into the pumpage stream.
5. Gas evolution from an incomplete or gas producing chemical reaction.

If a particular application produces entrained gas or has the potential to do so, the best solution is to eliminate as much entrainment as possible by applying corrective pump system design and/or a gas handling pump. If liquid gas mixing is desired, employ a static mixer on the dis-

FIGURE 3. DEVELOPMENT OF A VORTEX



A. Incoherent surface swirl

B. Surface dimple with coherent surface swirl

C. Vortex pulling air bubbles to intake

D. Fully developed vortex with air core to nozzle outlet

FIGURE 4. "HAT" TYPE VORTEX BREAKER

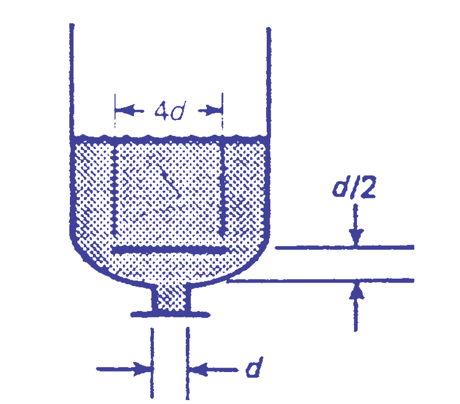
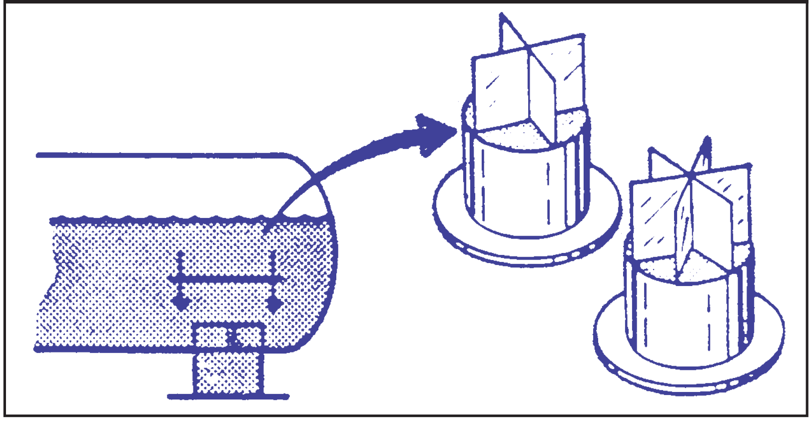


FIGURE 5. "CROSS" TYPE VORTEX BREAKER



charge of the pump. In addition, an anticipated drop in pump head due to an entrained gas situation may be offset by oversizing the impeller.

Of the five aforementioned mechanisms, vortexing is the most common source of entrained gas. Therefore, a user should be especially cautious employing mechanical equipment, such as tangential flash gas separators and column bottoms re-boilers, likely to produce a strong vortex.

VORTEX BREAKER DESIGN

The extent of gas entrainment in the pumped fluid as the result of vortex formation depends on the strength of the vortex, the submergence to pump suction outlet, and the liquid velocity in the pump suction nozzle outlet. Vortices form not only through gravity draining vessel applications, but also in steady state draining vessels, and in vessels under pressure or with submerged pump suction inlets. Vortex formation follows conservation of angular momentum. As fluid moves toward the vessel outlet, the tangential velocity component in the fluid increases as the radius from the outlet decreases. Figure 3 shows various stages of vortex development. The first phase is a surface dimple. This dimple must sense a high enough exit velocity to extend from the surface and form a vortex. (For experimental observations regarding vortex formation see Refs. 3, 4.)

The most effective method to eliminate entrained gas in pump suction piping is to prevent vortex formation either by avoiding vortex introducing mechanisms or by employing an appropriate vortex breaker at the vessel outlet. A "hat" type vortex breaker, illustrated in Figure 4, covers the vessel outlet nozzle to reduce the effective outlet velocity. This design doesn't allow a vortex to stabilize because the fluid surface senses only the annular velocity at the hat outside diameter (OD). In addition, the vanes supporting the hat introduce a shear in the vicinity of the outlet to further inhibit vortex formation. An annular velocity of 1/2 ft/sec at the hat OD produces a viable solution. Variations in hat diameters from 4d to 5d and hat annular openings of d/2 to d/3 are acceptable when annular velocity criteria are met. Annular design velocities of more than 1 ft/sec are not recommended.

"Cross" type breakers, installed above or inserted in vessel nozzle outlets as shown in Figure 5, work for some applications by providing additional shear to inhibit a mild vortex from feeding gas into a nozzle outlet (providing enough submergence is available). However, this design will not stop a strong vortex and will decrease NPSHA. A user should be aware of these limitations.

COLUMN VORTEXING

If a column draw-off pump is erratic and/or nearly uncontrollable, a vortex may be feeding gas into the draw-off nozzle of the pump as illustrated by Figure 6a.

It may be difficult to understand how a pump with 60 ft of vertical suction could be affected by entrained gas, but in this real case example Murphy's law applied twice. First, since the pump system in question has a NPSHA greater than 50 ft, the piping designer employed a smaller suction pipe with a liquid velocity of 10 ft/sec. Second, the column draw-off nozzle was sized according to normal fluid velocity practice. As a result, the tray liquid had an exit velocity of 5 ft/sec with a liquid level 6-in. above the top of the draw-off nozzle and a vortex formed, feeding gas into the draw-off nozzle.

As in the above example, due to a lack of proper submergence, gas is carried into the pump suction piping as a high liquid downward velocity exceeds the upward velocity of a gas bubble.

Many draw-off vortexing problems may be eliminated by proper pump system design or by one of two vortex breaker designs illustrated by Figures 6b and c. The selection of the breaker design may depend on the downcomer arrangement and space limitations. The most effective vortex breaker is the slotted pipe design shown in Figure 6c.

Application of these corrective pump systems designs or installation of an appropriate gas handling pump can solve suction side gas entrainment problems, resulting in a smoother process operation. ■

FIGURE 6. DESIGN MODIFICATIONS FOR A SYSTEM EXHIBITING A LACK OF ADEQUATE SUBMERGENCE AND PROHIBITIVELY HIGH EXIT VELOCITY

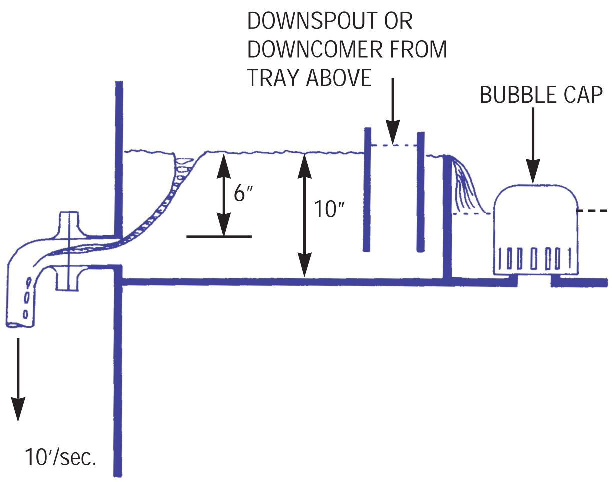


Figure 6a. Tray take off nozzle with vortex from lack of correct submergence and too high exit velocity.

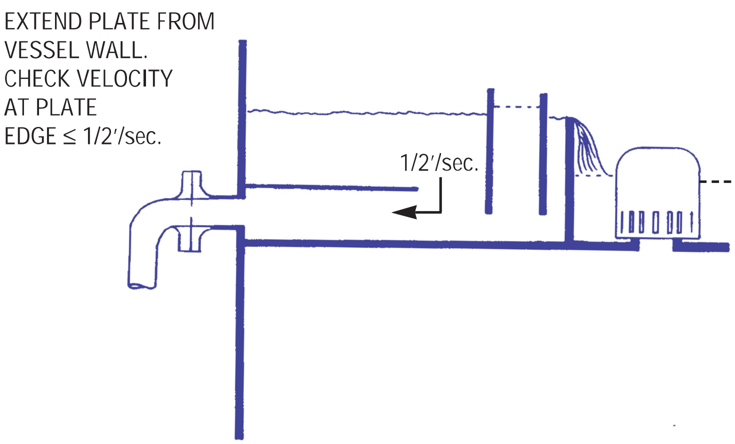


Figure 6b. Plate extension over outlet nozzle lowers high outlet velocity.

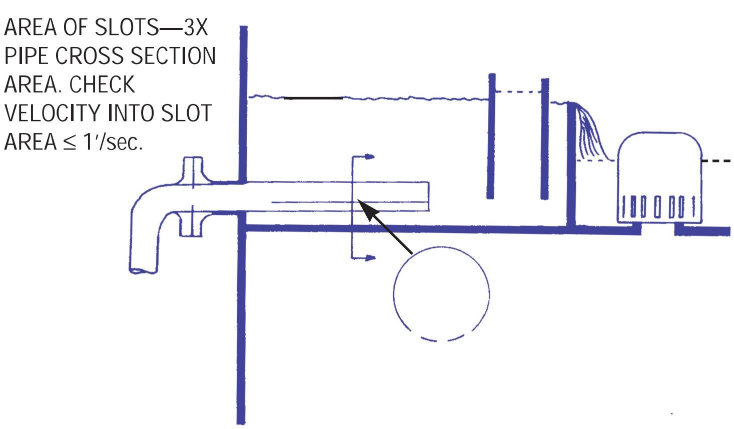


Figure 6c. Slotted pipe vortex breaker.

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Nozzle Loading – Who Sets the Standards?

Or, to what extent should the pump be used as a piping anchor?

BY: KIMBERLY FORTIER, ASSISTANT EDITOR

This past year's Texas A&M International Pump Users Symposium at the George R. Brown convention center in Houston, TX included a discussion group entitled *Nozzle Loading and Pump Operability* co-coordinated by John Joseph of Amoco Oil and Willie Eickmann of Houston Lighting and Power. According to Gary Glidden, also a discussion leader for this group, the two day discussion was a "standing room only" affair. Clearly, nozzle loading is a subject of concern to pump users.

ESTABLISHED LOADING STANDARDS QUESTIONED

Much of the discussion focused on the difficulty of establishing standards for allowable nozzle loads. Although the current API 610 7th edition standard for centrifugal pumps in general refinery service provides values for maximum loads (Table 1, Figure 1), many pump users believe the API allowable loads are too high—especially for use as specifications for installation designs which fail to recognize the possibility of "unplanned" stresses on the piping, such as those produced by foundation settling. However, as noted by James E. Steiger in his paper, *API 610 Baseplate and Nozzle Loading Criteria*, "Before the 6th Edition of API 610 was published, there were no industry accepted standards for allowable piping loads acting on centrifugal pumps." Moreover, when these piping load standards are absent or not universally accepted by pump users, manufacturers and piping engineers, these groups tend to set independent, often contrary standards, further complicating the design process.

In an attempt to overcome these

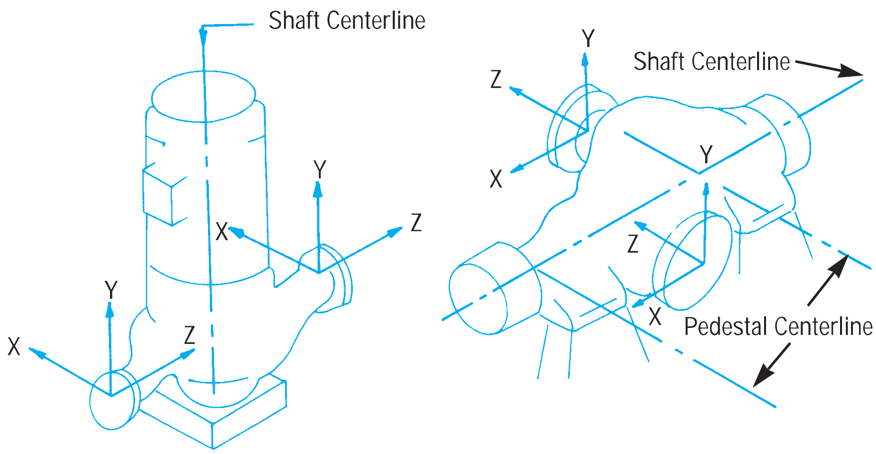
TABLE 1. API ALLOWABLE NOZZLE LOADS

Note: Each value shown below indicates a range from minus that value to plus that value; for example, 160 indicates a range from -160 to +160.

	Nominal Size of Nozzle Flange (inches)								
Force/Moment*	2	3	4	6	8	10	12	14	16
Each top nozzle									
FX	160	240	320	560	850	1200	1500	1600	1900
FY	200	300	400	700	1100	1500	1800	2000	2300
FZ	130	200	260	460	700	1000	1200	1300	1500
FR	290	430	570	1010	1560	2200	2600	2900	3300
Each side nozzle									
FX	160	240	320	560	850	1200	1500	1600	1900
FY	130	200	260	460	700	1000	1200	1300	1500
FZ	200	300	400	700	1100	1500	1800	2000	2300
FR	290	430	570	1010	1560	2200	2600	2900	3300
Each end nozzle									
FX	200	300	400	700	1100	1500	1800	2000	2300
FY	130	200	260	460	700	1000	1200	1300	1500
FZ	160	240	320	560	850	1200	1500	1600	1900
FR	290	430	570	1010	1560	2200	2600	2900	3300
Each nozzle									
MX	340	700	980	1700	2600	3700	4500	4700	5400
MY	260	530	740	1300	1900	2800	3400	3500	4000
MZ	170	350	500	870	1300	1800	2200	2300	2700
MR	460	950	1330	2310	3500	5000	6100	6300	7200

* *F* = force, in pounds; *M* = moment, in foot-pounds; *R* = resultant

FIGURE 1. COORDINATE SYSTEM FOR THE API FORCES AND MOMENTS



Vertical In-Line Pumps

Horizontal Pumps with Side Suction and Side Discharge Nozzles

complications, one user developed a standard operating procedure based on measured changes in pump alignment to be applied universally throughout their plant. Changes in alignment subsequent to connection of suction and discharge lines indicate shaft deflection. This user set the maximum shaft deflection at 0.002" regardless of pump size or configuration. However, because this procedure relies on establishing a baseline alignment before the lines are connected, this standard cannot be applied to all pumps. For example, the feedwater pumps Glidden operates at Houston Lighting and Power employ welded nozzles, which don't allow the pump to be isolated in order to determine the "zero-load" alignment, as opposed to flanged nozzles. And, since this procedure depends on measuring alignment rather than forces and moments as for the API specifications, making a correlation between the two standards is "almost apples and oranges," says Glidden.

Figure 2 illustrates a common consequence of nozzle loading on a pump. While the case bows in one direction due to piping loads, the shaft sags in the opposite direction as a result of thermal deformation. Pump operators witness the end results of overloading a pump nozzle in misalignment, vibration, bearing or coupling failure, and shortened seal life; but, according to some, the established standard fails to address the correlation between loading levels and these failure modes. And, these users are concerned that relatively slight levels of nozzle loading, even those within API specifications, may have costly ramifications, in terms of downtime and pump life, in the long run. In fact, according to Joseph, the discussion at the Pump Users Symposium quickly progressed beyond the question, "How much (loading) is too much," to whether the pump should "even be considered an anchor for the piping." Joseph concludes, "The piping (should) exert as little force as practically possible during operation."

CAREFUL PIPING DESIGN CAN REDUCE STRESSES

But how much is "practically possible"? Joseph recommends "shooting for 10% or less of API (allowable nozzle loading specifications) during running conditions." He also points out, however, that piping stresses can be reduced to zero, "My personal preference under hot conditions is that the piping exert nearly zero forces and moments." In most cases to obtain zero stress under hot conditions requires exerting some stress on the piping in the cold condition. These stresses will then relax with the thermal growth that occurs during hot operation. Joseph favors careful calculations in the design phase to insure that "the spring hanger forces and the deflection of the beam they're supported from matches the weight and growth of the piping when it's full of liquid at temperature." For example, a spring hanger supporting a 20' straight vertical piping section might relax a full 1/4-1/2" due entirely to thermal growth in the vertical direction. Add this growth to the pull of the process-liquid weight and the result is, the piping stress and strains during hot running differ drastically from those prior to start-up. One operator actually measured a 0.150" horizontal movement of the pump. "You've got to think, what is it (the pump) going to look like with a hot flow of liquid and then back calculate to the cold, empty position you want that pipe at," says Joseph. To obtain minimum loading during the running condition, the pipe should be supported in a position requiring it to be pulled down to the pump. During hot operation, the thermal growth of the piping and the weight of the liquid will then depress the piping into the relaxed position.

ECONOMICS

Piping engineers counter these arguments for low to zero piping loads, claiming, as Steiger notes, that "the pump manufacturers and rotating equipment engineers are too conservative and the higher piping loads do not usually lead to significant operability problems." The larger piping loads are desirable because they result in simpler and significantly less expensive piping configurations. Yet, the 1985 Pressure Vessel Research Committee (PVRC) Pump-Piping Interaction Experience Survey indicates that there is "a significant pump-piping interaction problem and that it has an annual impact on the order of one half-billion dollars" (Ref. 1). "Economics plays a big role in these decisions," adds Glidden, "However, if you do a bad job up front, this will compound, resulting in a terrible-running pump."

CONFRONTING THE ROOT CAUSE

Understanding how the base plate and piping design relate provides one key to maintaining shaft alignment and thereby pump reliability. However, as Steiger maintains, "The pump-baseplate assembly represents a complex structure whose response to piping loads is difficult to predict with a high degree of certainty." Joseph agrees, "While some pump cases and base plate foundation designs can take very high loads, there are others for which simply tightening a nut one flat at a time changes alignment significantly." If the pump case and base plate construction is reasonably rigid, higher forces may be applied with little deflection at the coupling. "However," Joseph warns, "if users depend on the pump case and the base plate to provide the rigidity, and the piping is significantly high in stress application, then they're just covering one problem with another solution. They're not getting at the root cause—the piping strain."

Joseph recommends proper warm-up procedures and piping supports, in addition to good piping design, as the primary remedy for piping strain. His recommendations are outlined as follows:

- **Warm-up procedures**

Large, hot (and, as some users have pointed out, expensive) pumps are especially affected by nozzle loading, and for these pumps proper warm-up is essential. API recommends that the warm-up procedure be very well thought out. Without proper warm-up the pump may suffer from uneven thermal growth in the piping, case, shaft, stuffing boxes, and bearing housings. As the pump comes to equilibrium, it will experience transient thermal growth which may put it under considerable stress. The design of the piping is also crucial to adequate warm-up and should allow hot product to flow from the discharge line to the bottom of the pump case.

Many operators bring hot product to the pump by employing a by-pass to the discharge check valve. This is a small piece of pipe with a block valve which enables the discharge pressure product to by-pass the discharge check valve and follow the discharge nozzle, then cross the top of the pump and exit at the suction nozzle. This method is inadequate because it heats only the top of the pump, resulting in a 150-300°F differential within the pump. This temperature differential creates a very large humping tendency since the pump expands much more at the top than the bottom. As a result, the bearing brackets might shift down at both ends of the pump. The rotor bows in the same direction. In fact, the rotor will actually roll over by hand about 70° until the wear rings rub. A few minutes later, the heat in the top of the rotor will allow it to roll about 70° further.

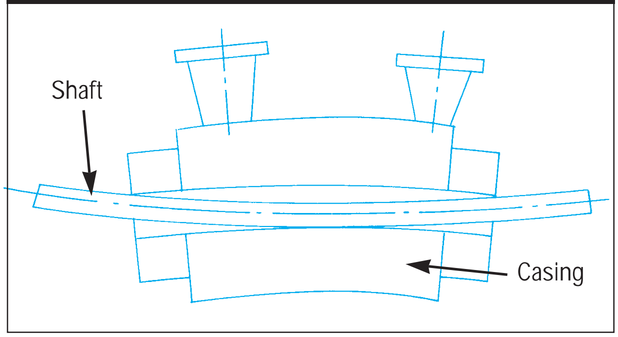
To assure that hot product is dispensed to the bottom of the pump, the product should be evenly distributed in all suction and discharge cavities at the drain connections. Even distribution provides the best opportunity for thorough heat delivery to the pump case, rotor, and discharge/suction piping, prior to pushing the start button. Slow rolling the pump will also aid warm-up. Establish the slow roll at 50-100 rpm and then initiate the warm-up flow to the pump with hot product.

- **Piping supports**

Bad installation, deteriorating hangers and foundation settlement are some of the most common causes of piping strain. Piping should be well supported by spring hangers, anchors, expansion loops or compression spring cans. In addition, all piping supports should have adjustment capability to enable repositioning in response to deterioration and settlement. Glidden envisions a monitoring system, which would examine alignment once a year or so, to test for changes due to deterioration over extended periods.

The hangers and other supports, including the beams that support these hangers, should be designed to support the weight of the piping and the process-liquid. If the support system deflects, the pump may become the anchor for the piping.

FIGURE 2. RESULTS OF NOZZLE LOADING, COMPOUNDED BY HOT OPERATION



Placing a firm anchor on a knee brace right at the pump base may provide a good solution. The knee brace, which should also be supported by spring hangers, forces the piping up and away from the pump, so that when the load changes as the process liquid flows into the piping, it will relax down to the pump, and the spring hangers will bear the full load, allowing the pump to operate with very low stresses.

CONCLUSION

The response to the discussion group at the Pump Users Symposium indicates a real need for these kinds of practical solutions to nozzle loading. The pump operators present at the Symposium recognized that an inexpensive piping design can be costly down the road. And, many are hopeful that the 8th edition of the API 610 standard, currently in progress, will advance one step further toward an agreeable solution for more uniform nozzle loading practices. ■

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Low Flow Options

This service range demands an innovative approach.

BY PUMPS AND SYSTEMS STAFF

Process requirements often demand capacities below those achievable with a conventional centrifugal pump. Figure 1 illustrates the range of service conditions considered to be low flow. The minimum continuous stable flow of a typical 1"x2"x7" overhung pump at 1800 rpm is approximately 7 gpm, while at 3550 rpm the minimum continuous stable flow is about 13 gpm. A pump of this size will produce about 240 ft. of head. As the head requirement increases to 5000 ft., the minimum continuous stable flow will increase to about 190 gpm.

Conventional centrifugal pumps will not handle these low capacities very well for two main reasons:

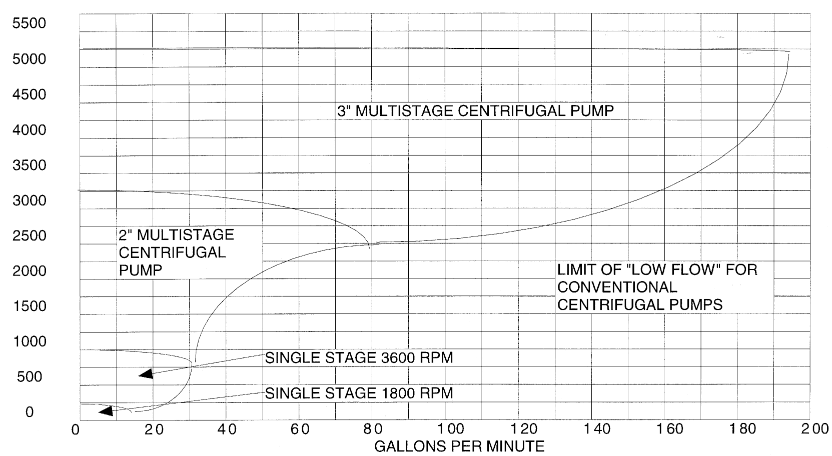
Suction recirculation

The minimum continuous stable flow is usually set by the pump manufacturer to avoid suction recirculation. Suction recirculation results in increased vibration and imparts continuous axial movement to the shaft, decreasing the life of bearings and mechanical seals. The point at which suction recirculation begins may be calculated as described by Dr. S. Gopalakrishnan in his presentation at the 5th International Pump Users Symposium in 1988. The pump manufacturer should perform these calculations and set the pump minimum continuous stable flow at a capacity greater than the calculated capacity.

Temperature rise

The ultimate limitation on low capacity is minimum continuous thermal flow. Temperature rise through a pump determines the minimum flow rate. The maximum safe temperature rise through a pump should be limited to 10°F. The formula for determining thermal rise through a pump is:

FIGURE 1



RANGE CONSIDERED LOW FLOW

$$\delta T = \frac{H}{778C_p} \times \frac{1}{(Eff - 1)}$$

H = total head in feet

C_p = specific heat of the liquid in $\frac{Btu}{lb}$ x °F

788 ft-lbs = the energy to raise the temperature of one pound of water by 1°F

PARTIAL EMISSION PUMPS

The type of pump most frequently applied to fulfill low flow requirements is a single port diffuser pump with a "Barske" straight vane impeller close coupled to an electric motor, also known as a partial emission pump (Figure 2). Theoretically, in this kind of pump, the only liquid discharged as each chamber passes the diffuser port is the liquid between the impeller vanes. In reality, however, due to the clearance between the case and impeller, some additional liquid also gets swept out the diffuser port. Unfortunately, this pump has a head capacity that droops at shutoff which inhibits the ability to control the pump capacity by increasing pressure with decreasing flow (Figure 3). As a result, installation of a flowmeter is necessary to effectively control this type of pump.

FIGURE 2. BARSKE STRAIGHT VANE IMPELLER WITH SINGLE PORT DIFFUSER PARTIAL EMISSION PUMP

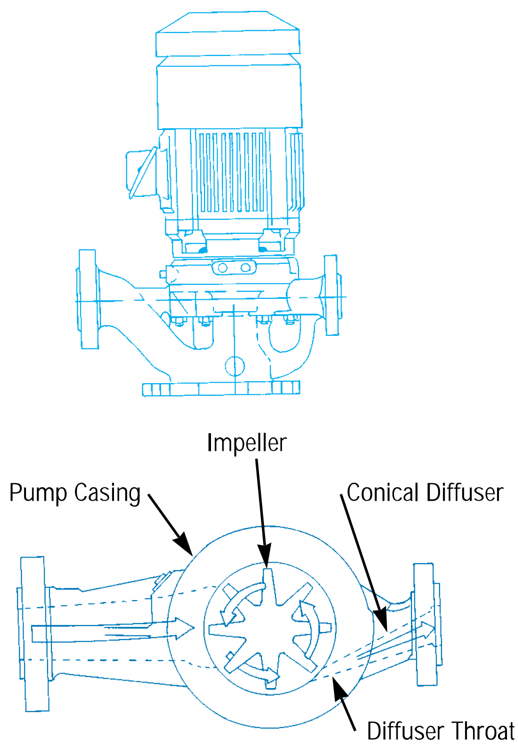
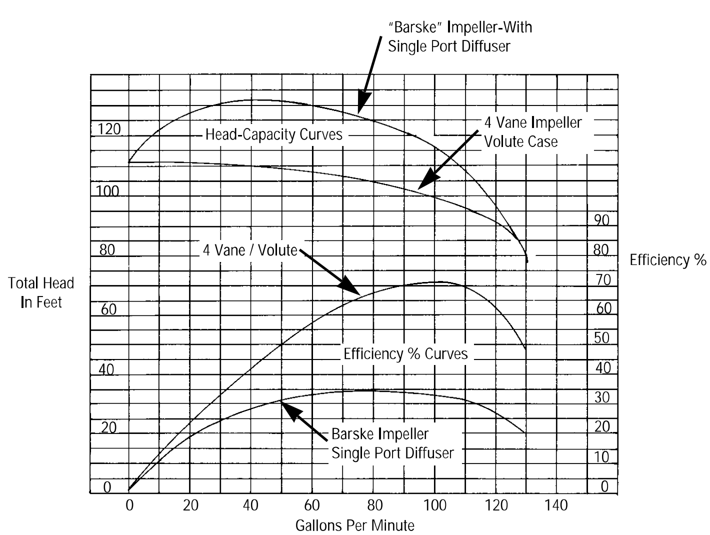


FIGURE 3. TYPICAL CURVE SHAPES



Because the characteristic curve for the "Barske" impeller, also referred to as a high solidity impeller, is exactly the opposite of a centrifugal pump (where increasing the number of impeller vanes will flatten the curve and eventually cause a droop), the droop of the head capacity curve towards shutoff can be minimized in a single port diffuser pump by increasing the number of impeller vanes.

Another method of eliminating head capacity droop is to install a discharge orifice. Since the friction across an orifice increases as the flow increases, the pressure of a discharge orifice will increase the pump curve slope so that the pump can be pressure controlled. Unfortunately, a discharge orifice decreases the pump efficiency.

The partial emission pump is also available with an integral gear (either single or double increaser) to produce a higher pressure head than a single stage pump. Since high head application with the integral gear may call for speeds up to 20,000 rpm, an axial flow inducer is often employed in conjunction with this gear to lower the net positive suction head (NPSH).

Another means to achieve low flow combined with high head requirements is to drive the pump with a special motor capable of high speed. Application of a variable frequency drive will produce speeds nearing 7200 rpm. With this type of construction a partial emission pump may also be coupled to a canned motor for sealless pump construction.

One manufacturer builds the partial emission type pump with an in-line configuration, giving the pump its own bearing frame. In this configuration the pump is flexibly coupled to a standard vertical solid shaft motor. This same manufacturer also builds this pump in a horizontal centerline-mounted configuration.

FLOW RESTRICTION DEVICES

Conventional centrifugal pumps can handle low flow conditions with the incorporation of a restriction device on the discharge to shift the best efficiency point (BEP) capacity back toward shutoff and increase the pump curve slope. Unlike the partial emission pumps which employ a construction requiring removal of a motor with a special shaft extension for mounting the impeller, or in the case of high speed applications removal of the motor and gear, the application of flow restriction devices on conventional API or ANSI pumps provides the benefit of an easily maintained single stage pump.

Even though a restriction device reduces the efficiency by a considerable amount, low flow pumps are generally low horsepower machines, so consuming a little more horsepower to obtain a steep curve rise previous to shutoff is a small price to pay for the more desirable performance. Moreover, the required motor horsepower for the restricted pump is less than that for a non-restricted pump, as the restriction will not allow the pump to run to the extended portion of the curve.

The use of an orifice to restrict flow will produce the desired performance. However, if the orifice diameter is considerably smaller than the pump discharge and discharge piping, cavitation and noise may occur on the downstream side of the orifice.

For this reason one pump manufacturer incorporates a venturi to modify pump performance. The advantage of the venturi is that the gradual taper down to the required hole size then back up to the discharge pipe size effectively eliminates the cavitation, noise and vibration. Pumps equipped with venturi have been observed to run smoother and quieter as they approach shutoff.

OTHER PUMP OPTIONS

- Another type of centrifugal pump that will operate effectively in the low flow range is a vertical can pump. A 5-6 in. diameter bowl assembly will experience its BEP capacity in the 60-120 gpm range at 3600 rpm. At these operating conditions the pump will produce heads up to about 1000 ft. A primary advantage of the vertical pump is the ability to stack many stages so that a low capacity impeller of fairly good efficiency will produce a high head. These pumps usually incorporate two, sometimes three or four, impeller designs of various capacities. Mixing impellers will result in a rated point capacity very near BEP. There is a limit, of course, to how many stages a vertical can pump may have. The limiting factors are shaft diameter size required to transmit the horsepower and torque and the availability of shafting in long sections (usually 20 feet). Another limitation is dependent on the machining tolerances of the register fittings of the bowl assembly. Since the tolerances are additive as the bowl is assembled, they may cause shaft binding if they are not tight enough.
- Regenerative turbine pumps will also fulfill low flow requirements. These pumps, available in single and multistage construction, have a very steep head characteristic and will operate on pressure control. The regenerative turbine does not demonstrate any apparent problems with minimum continuous stable flow, so the only limiting factor to set minimum flow is temperature rise. The formula for temperature rise through these pumps is identical to that for centrifugal pumps. The disadvantage of a regenerative turbine is the close internal clearances required to produce the pumping action. To accommodate this close clearance, the pumpage must be very clean.
- Gear or other rotary positive displacement pumps also will operate in the low flow range without difficulty. These pumps do not, however, operate well in low viscosity services.
- Controlled volume metering pumps can be applied for low flow services and are one of the few types of pumps that will operate at flow rates below 1 gpm. The disadvantages of using a metering pump are the inherent pulsations which may damage downstream piping and instruments. Pulsation dampeners help to smooth out pulsations but never entirely eliminate them.

CASTING LIMITATIONS

Development of a truly efficient low capacity centrifugal pump requires prohibitively small liquid passages. These small passages are troublesome to produce in the casting process because the sand mold is prone to collapse at such small sizes and small interior passages are difficult to clean to the degree required for good efficiency in operation.

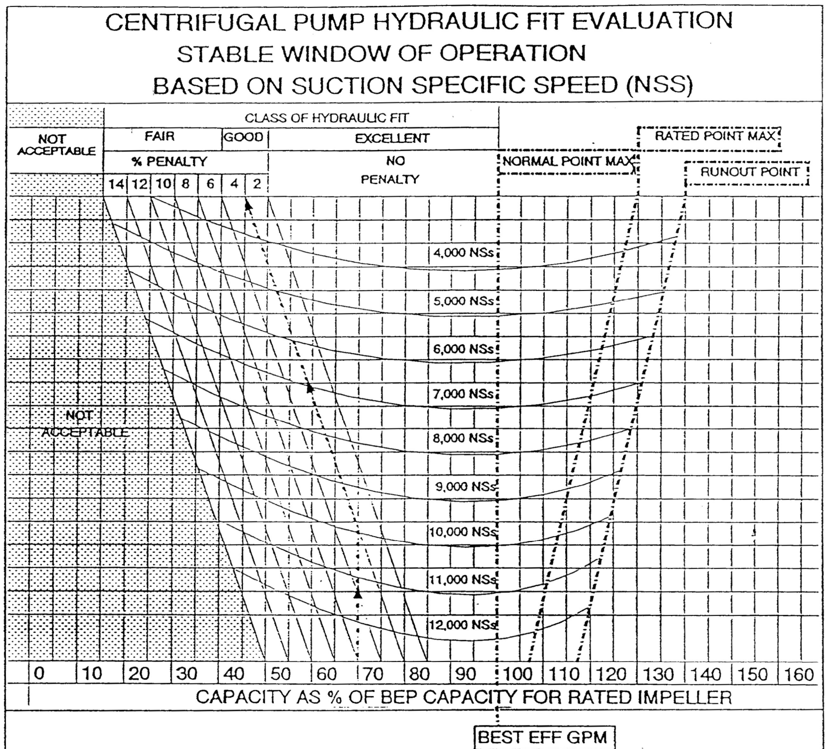
A semi-open impeller is easier to cast and clean. This design is, however, in violation of API 610, which calls for an enclosed impeller cast in one piece. If sufficient advantages of the semi-open configuration are demonstrated, this standard might be changed. Very small impellers might even be machined from billet stock (similar to some centrifugal compressor impellers), thus eliminating all of the casting problems.

Similarly, the casing of a low flow pump is difficult to cast and clean, requiring very small passageways which must have a smooth surface in order to produce good efficiency. This obstacle to producing a low flow pump case might be overcome by eliminating the need for a case casting in favor of a machined and fabricated construction.

EVALUATING HYDRAULIC FIT

The fact of the matter is most manufacturers usually make little profit on their small model pumps. To convince manufacturers that quality low flow pumps are actually in demand, users must let them know that their quotations for pumps in the low flow area are being

FIGURE 4



NOTES:

1. NSS FORMULA $\dfrac{RPM\ (GPM)^{0.5}}{(NPSHR)^{0.75}}$
2. CALCULATE NSS AT BEP CAPACITY AT MAXIMUM DIAMETER IMPELLER AND NPSHr AT THAT POINT.
3. FOR DOUBLE SUCTION IMPELLERS USE GPM/2
4. APPLY PENALTY TO PUMP PRICE (EXCLUDING DRIVER).
5. EXAMPLE:
   NORMAL CAPACITY IS 700 GPM.
   BEP AT RATED IMPELLER DIAMETER IS 1000 GPM.
   SUCTION SPECIFIC SPEED (NSS) IS 10,000.
   PENALTY IS 3% OF PUMP PRICE.
   (SEE EXAMPLE ARROWS ON CHART)

evaluated for hydraulic fit. One method of evaluating hydraulic fit is shown in Figure 4. This evaluating tool adds a penalty, as a percentage multiplier, to the pump price for rated capacity to the left of BEP capacity. This tool is based on the fact that a higher suction specific speed correlates with a smaller stable window of operation.

Applying this tool consistently and sending it along with your request for quotations will convince pump manufacturers that low flow performance is an area of hydraulic design that needs to be addressed. ■

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ACKNOWLEDGMENTS

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Pump Design Changes Improve Lubrication

Quantifying the benefits of modifications.

BY LEV NELIK

Proper lubrication is a key to long, trouble-free life of centrifugal pump bearings. In recent years the issue of lubrication has received renewed attention from pump users in chemical plants, pulp and paper mills, refineries, and other industries.

Budgetary pressures have forced many plants to reduce maintenance capital. Many knowledgeable maintenance workers have been laid off. Not surprisingly, the ability to maintain pumping equipment properly is reduced, resulting in increased outages, lost production, and rising maintenance costs.

Users have started to look to pump manufacturers to pick up the slack and help solve pump reliability problems, extend component life, and increase mean time between between failure (MTBF) and mean time between scheduled maintenance (MTBSM).

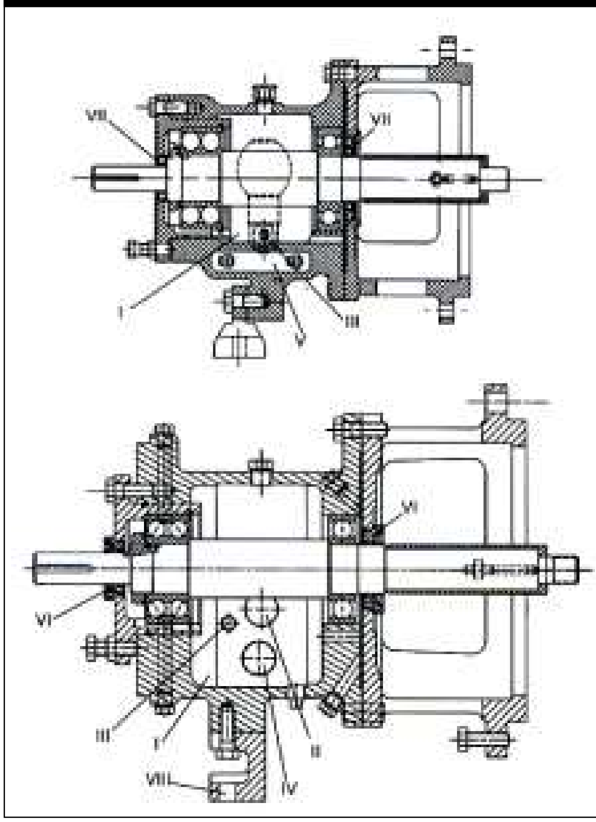
Statistics show (Ref. 1) that most pump failures are related to bearings and seals. In this article we will look at bearings, analyzing how design changes affect bearing life in a quantifiable way.

The need for improved pump reliability and increased MTBF led to a new design, introduced by Goulds in 1990/1991. Figure 1 shows cross sections of two single-stage, end-suction ANSI pumps. Both have identical wet ends (impeller and casing), but the power end and the seal chambers are different. Improvements in the seal chamber are significant. The new design has a larger chamber to ensure better heat transfer and cooler operation of the mechanical seals. The previous design incorporated a tight stuffing box.

The new power end design in Figure 1 features approximately three times the volume of the oil sump (I), an oil level sight glass (II) to assure the proper oil level versus the constant level oiler (III), improved cooling via a finned cooler insert (IV) versus bottom cooling pockets (V), labyrinth oilframe seals (VI) versus lip seals (VII), and stiffer footing (VIII) for reduced vibrations.

A testing program has been conducted to compare the two designs under extremely adverse operating conditions, such as running endurance testing at overspeed and below minimum flow. This program was conducted at the R&D lab of the Technology Center at Goulds, resulting in quantifiable correlations between changes in pump design and their effect on life extension.

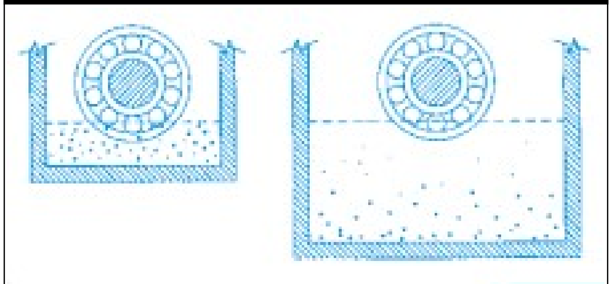
FIGURE 1



Cross sectional views of old and new power end designs.

Feedback from users comparing two designs was also obtained , specifically in relation to the operating temperature of the bearing frame surfaces.

FIGURE 2



Larger sump results in reduced concentration of contaminants, which settle to the bottom.

ANALYSIS OF THE POWER END

With regard to the power end (Figure 1), the belief that "the bigger the better" is not uncommon in the pumping community. This idea has some merit, but manufacturers often overlook the importance of *quantifying* the benefits of a particular design or modification. Frequently, little information is given as to *how much* life extension can be obtained by, say, having a deeper sump, or how much added value and savings can be realized from the increased bearing frame heat transfer surface.

It is clear that a systematic approach to identify, measure, and improve pump component design is impossible without a proper balance of theory, experimentation, user feedback, and data from real world installations. Theory and experimentation should be balanced by clear communication between manufacturers and users.

INCREASED FRAME OUTSIDE HEAT TRANSFER SURFACE

Heat is transferred from the pump bearings to the oil and through the housing frame walls to the outside air. Some of the heat is also conducted through the casing to and from the pumpage, depending on the temperature of each. Typically, the difference in temperatures is small for the pumping conditions of chemical plants, and the effects are omitted for simplicity.

Our investigation has shown (Ref. 3) that the larger surface area can result in a nearly 40°F reduction in bearing operating temperature. The cooler bearings, in this case, result in approximately 13% longer life.

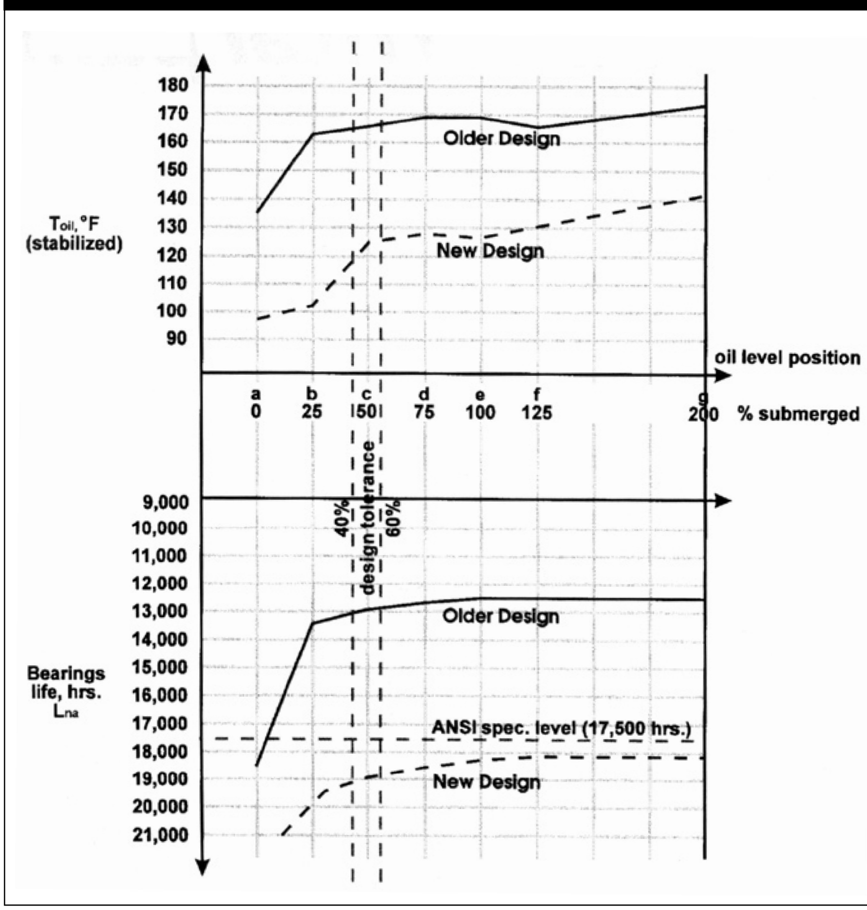
INCREASED OIL SUMP DEPTH

A deeper sump allows contaminants to settle farther from moving parts, resulting in a cleaner layer of oil near the ball bearings (Figure 2). Contamination of the bearing races and the balls is the cause of microscopic deterioration of load surfaces, leading to failure. Statistics show (Ref. 2, 4) that a cleaner oil operation can increase bearing life by nearly 2.1 times (Ref. 3). Similarly, due to decreased air concentration, the oil oxidation rate by air is reduced for the larger sump. Again, for the type of pump studied in this work, this results in a 2% extension in bearing life (Ref. 3).

LABYRINTH OIL VERSUS LIP SEALS

The effects of oil contamination are further reduced by improved oil seals. A proprietary labyrinth seal design was tested against the lip seal. Both pumps were sprayed with water from a hose, simulating plant washdown. The spray was directed at various angles to the frame at the oil and seals area. The oil was then analyzed for water content. It was found that the previous design equipped with lip seals contained 3% water after 30 minutes of spraying, while the new design, with labyrinth seals, showed no water at all. Also, lip seals may

FIGURE 3



Comparison between bearing submergence in oil, operating temperature, and bearing life for the old and new designs.

cause wear and leakage after approximately only 2,000 operating hours.

OIL LEVEL SIGHT GLASS VERSUS CONSTANT LEVEL OILER

A large sight glass allows direct visual observation to ensure proper oil level. It is standard in the new design, although a constant level oiler option is available. A constant level oiler is preferred by many users. When properly installed and maintained it can result in satisfactory operation.

However, because operation of the oiler is "blind," depending solely on strict conformance to correct (and nontrivial) oil filling and maintenance procedures, it may lead to an incorrect oil level inside the frame. This can lead to hot operation and premature failure. Another problem is known as the oiler "burping" effect, resulting in a higher actual oil level than perceived (Ref. 5). Obviously, the new pump design can be equipped with both the sight glass and oiler if they are desired by the user.

Such improvements in design can be combined because they benefit pump reliability independently. Based on this research, when all are added together, an improvement in pump life of up to 125% may be obtained. Even longer life may be realized due to other design upgrades, such as providing the pump with a more rigid foot, reducing vibration, improving (finned) cooling, and creating larger chambers for mechanical seals. For brevity, these effects are not included in this analysis, but they can be accounted for in the references (Ref. 3, 5).

TEST PROGRAM

To support and validate the theoretical derivations and assumptions as outlined below, a testing program was conducted, including lab testing and field data analysis. Figure 3 shows a comparison between operating temperatures and bearing life for the old and new designs. Tests were conducted with oil covering different levels of the lower ball of the bearings. The proper design level (marked 50% on Figure 3) corresponds to oil at the middle of the lower ball of the bearing. At design setting, the new frame ran 40°F cooler, with a corresponding predicted life extension of approximately 6,000 hours.

ECONOMIC BENEFITS

Having measured the life increases resulting from these improvements, it is not difficult to assess the economic benefits of the new design.

Assuming the average value for MTBF of two (2) years for the old design, a 125% improvement results in a four and a half (4.5) year bearing life for the new design. The reciprocals of these numbers (1/2 years = 0.5; 1/4.5 years = 0.22) give an approximate number of failures or scheduled maintenance per year. The difference, 0.28, when multiplied by the average cost of repair of, say, $260 in parts and labor and 3,000 pumps per plant results in a yearly plant savings of:

0.28 x $260 x 3,000 = $218,400

In addition, savings resulting from increased uptime and a reduction of lost production at approximately $500 per off-line hour, assuming an average four hours per repair for off-line time, would be:

0.28 x ($500 x 4) x 3,000 = $1,680,000

The total, $1.9 million, is annual plant maintenance savings. Obviously, these numbers are approximate and can best be determined by individual maintenance departments using their operating specifics, but the savings potential due to improved design is clear.

CONCLUSIONS AND RECOMMENDATIONS

Our study demonstrated that substantial savings can be realized through improvements in pump design. To gauge such improvements systematically, it is imperative to quantify the benefits of each pump enhancement.

It is also important to maintain a proper balance between the solid theoretical foundations used for the analysis and the laboratory work, field testing, and data supporting such theory. Users should seek quantitative data demonstrating improvements from pump manufacturers, including improvements in MTBF and MTBSM, enabling them to determine added value and other economic benefits.

This approach will improve communication between manufacturers and users, and lay the ground work for the next step: further improvements in pump reliability.

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For more information on these references please call (315) 568-2811. ■

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CPI Pumping

Increase reliability and reduce emissions through pump selection.

BY RICHARD BLONG AND BOB MANION

Today chemical manufacturers and users are faced with global competition and pending environmental restrictions that threaten to reduce profitability. The need to reduce overall operating costs has driven pump users at chemical plants to focus on improving reliability and eliminating or reducing fugitive emissions.

SEALED PUMPS

The mechanically sealed chemical process pump, which meets ASME/ANSI B73.1M standards, is the workhorse of chemical processing industries. It will continue to be used on a wide range of process applications—such as liquids containing significant amounts of solids (sodium chlorate, alum, sodium carbonate, chemical wastewater), light slurries (silver nitrate and acetone slurries), viscous liquids (above 150 cP, including black liquor and titanium dioxide), and stringy materials where sealless pumps may not be economical to use. In addition to its ability to handle tough services, the flexibility of the design—along with improved low-emission mechanical seals—continues to make ANSI pumps the standard in this field.

To elaborate on why sealless pumps are not economical to handle the above materials, we must note that they use enclosed impellers to reduce the axial thrust and increase reliability. (Although several manufacturers have tried using open or semi-open impellers in sealless designs, many of these have not been reliable at two-pole speeds.) Also, standard sealless pumps have small internal passageways to circulate liquid for bearing lubrication and drive-end cooling, and mechanical seal manufacturers are rapidly improving the reliability of their products while reducing emissions to well below 500 ppm. With this in mind, consider the following:

1. Enclosed impellers are prone to plugging and premature wear in the above services due to small wear surface area. (Performance and efficiency cannot be renewed without replacing wear rings.)
2. Open or semi-open impellers are reliable in these services and are standard for ANSI pumps. (Simple external impeller adjustments allow easy maintenance of performance and efficiency, and there are no wear rings to replace, yielding long-term energy savings.)
3. The small internal passageways in sealless pumps are subject to plugging while handling liquids with only small amounts (5%) of solids. Viscosity handling is also limited.
4. Design solutions separate the pump end from the drive end to allow sealless pumps to handle these services, but these modifications can be expensive and may not be cost effective.

Considering all the facts, it's understandable that mechanically sealed ANSI pumps are the more economical choice to handle these types of liquids.

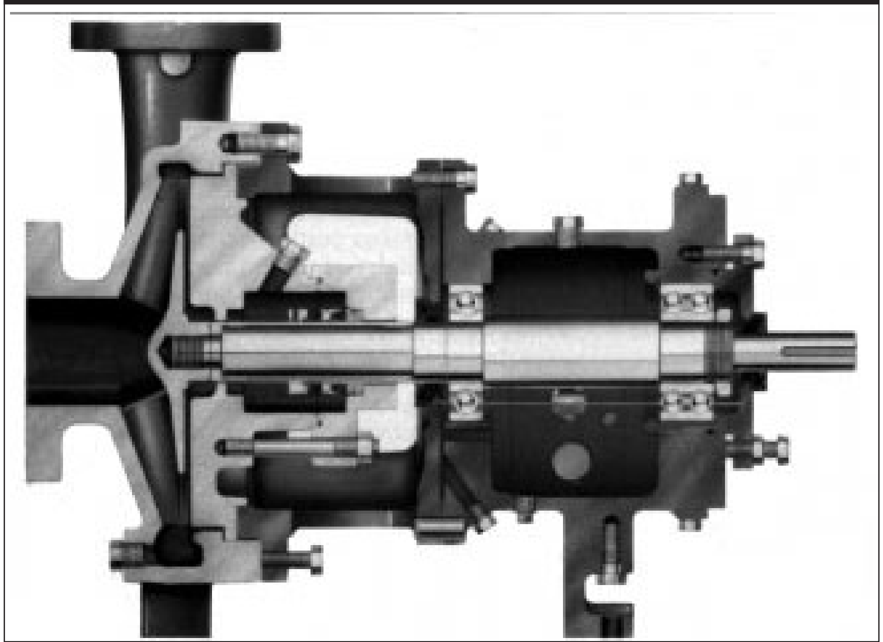
ANSI RELIABILITY IMPROVEMENTS

To meet emissions regulations and improve reliability, process industries have pushed ANSI pump manufacturers to improve performance. Some manufacturers have formed alliances with users to share technology and improve standard designs. By working together, the theoretical has been combined with the realities of applying pumps on a multitude of services and making them last.

These efforts have produced the features listed below that many major ANSI pump manufacturers have incorporated (Figure 1). At a minimum, users should purchase ANSI pumps with features that best meet their application needs. However, most new designs incorporate features systematically to provide reliable products. Compromising designs to save money or add standard plant features—substituting a vendor's standard labyrinth seal with the plant's standard oil seal, for example—may not be advisable. New ANSI pump features include the following:

1. Labyrinth oil seals are designed to prevent premature bearing failure from lubricant contamination or oil loss. These non-contacting seals have replaced Buna-rubber lip seals, whose useful life was three to six months under normal conditions. Materials of construction include carbon-filled Teflon, bronze, or stainless steel.
2. Increased oil sump capacity provides better heat transfer for more effective oil cooling. Bearings operating at lower temperatures contribute to longer life.
3. A rigid frame foot reduces the effect of pipe loads on shaft alignment. Misalignment won't exceed 0.002 in. under load, and pump and driver alignment is better maintained.
4. Bull's eye sight glasses insure proper oil level, which is critical to bearing life. Level oilers have often been misused, leading to

FIGURE 1



ANSI pump improvements.

over- or under-filling sumps, both of which contribute to bearing failure. Sight glasses are also convenient for checking the oil condition visually to determine if a change is necessary. Constant-level oiler manufacturers are just now introducing oilers that eliminate the potential for improper oil level settings while providing a sight glass, combining the best features of both methods.

5. Mounting flanges accommodate an optional adapter that simplifies pump/motor shaft alignment, saving the user time and money during installation.
6. Condition monitoring bosses on power ends provide consistent measurement points for temperature and vibration sensors. Many users report increased pump life from using predictive maintenance to identify and correct problems early. Taking measurements at the same point aids in proper interpretation of readings and allows personnel to move through the plant more quickly on inspections.
7. Engineered large seal chambers, specifically designed for today's mechanical seals, increase seal life through improved lubrication, cooling, air venting, and solids handling. The chambers allow seal manufacturers to engineer and apply more reliable designs, including cartridge seals.

These developments extend pump and seal life and reduce emissions at the same time. Experience shows that one cannot be accomplished without the other. For example, a mechanical seal with emissions in excess of regulations has already failed in its application.

Another benefit of these features is that several manufacturers and seal suppliers are extending unconditional warranties to as long as three years, helping to further lower operating costs.

SPECIALTY PUMPS FOR IMPROVED RELIABILITY

Many diversified chemical producers are moving production of commodity chemicals to the Asia-Pacific and Latin American regions to take advantage of lower labor and production costs. As a result, chemical production in the United States is being driven toward manufacturing specialty chemicals typically produced in small runs or batches. Examples include methylisobutyl- ketone (MIBK) and paratertiarybutyl- phenol (PTBP). Pump applications for these batch-type processes are usually low flow, in the range of 0 to 100 gpm.

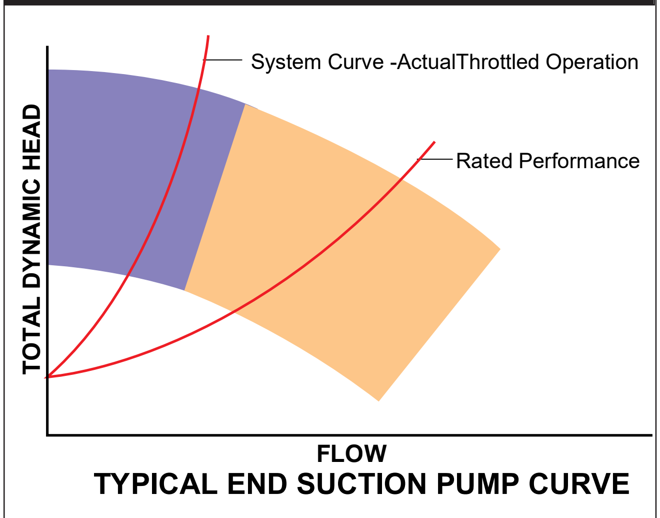
Traditionally, users install standard process pumps and throttle the discharge valves to obtain low-flow performance. However, these pumps are not designed to operate continuously in this range (Figure 2). Higher radial loads and increased shaft deflection lead to premature bearing and seal failure. Costly downtime and maintenance expenses result.

For low-flow operation, users should specify a pump designed to meet specific service conditions (Figure 3). ANSI pumps designed for low-flow operation are available to increase pump and plant reliability.

Improvements come from a casing and impeller designed for low-flow operation. Low-flow designs use concentric volutes and radial vane impellers to reduce radial loads, eliminating hydraulic and mechanical problems from throttled low flows (Figure 4). Some designs reduce radial loads as much as 85% compared to end-suction expanding volute pumps in this service (Figure 5). Shaft deflections from high radial loads are minimized, optimizing bearing, mechanical seal, and overall pump life. A disadvantage of low-flow ANSI pumps is that they sacrifice some efficiency to reliably handle viscous and solids-containing liquids.

Another approach to low flow–high head applications is the regenerative turbine pump. This design directs liquid by a passageway so that it circulates in and out of the impeller many times on its way from pump inlet to outlet. Both centrifugal and shearing action work together to efficiently develop relatively high heads at low flows. Regenerative turbines also use concentric volutes and radial vaned impellers to obtain the reliability benefits discussed above. One drawback is that this type of pump utilizes close running clear-

FIGURE 2



Off-design (throttled) operation range (darker gray) and recommended operation range (gray).

ances to keep efficiency high and it is therefore normally used on clean liquid applications.

SEALLESS PUMPS

With the implementation of the Clean Air Act, sealless pumps offer a dynamic solution to controlling emissions. Not only should sealless pumps be strongly considered to control emissions of the 149 volatile organic compounds identified by the Environmental Protection Agency, but they should be viewed as solutions to many difficult applications encountered in CPI plants today. For example, if users are experiencing sealing problems because of the pumped product's poor lubricity (typical of acidic products in the range of 0–3 pH, such as sulfuric or hydrochloric acids), difficulty with product crystallization at seal faces (usually with caustic products in the range of 10–14 pH, such as sodium hydroxide and potassium hydroxide) or are frustrated with sophisticated auxiliary piping plans to provide clean, cool flush liquid to mechanical seal faces, sealless pumps may be the answer.

IMPROVED RELIABILITY WITH MAG DRIVES

It is well recognized that mechanical seals and bearings are the two items that fail most often in pumps. These failures are often directly related to improper application and installation, poor operating practices or lack of maintenance, pipe strain, or misalignment. All of these again lead to high bearing loads, shaft deflection, and bearing and seal failure. Magnetic drive pumps have neither a mechanical seal that can fail nor a driven shaft that can be subjected to pipe strain or misalignment. The driven shaft is separated from the drive shaft by a magnetic coupling, eliminating the two major causes of pump failure.

CRITICAL MAG DRIVE FEATURES

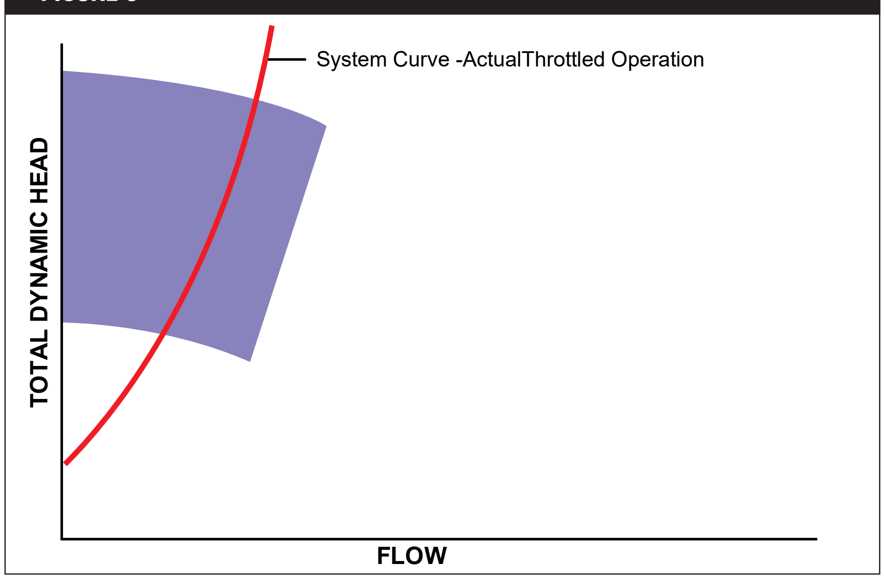
Reliable magnetic drive pumps must address two critical concerns:

1. proper lubrication of the journal bearings
2. removal of heat generated by eddy currents in the recirculation circuit

The design must deliver liquid to lubricate the bearings—it should not be flashing or have risen in temperature, which decreases lubricity, prevents proper cooling, and leads to bearing failure. Proper journal bearing lubrication directs cooling liquid to the bearings, then to the magnets. Dual path designs provide lubrication to these areas separately. Both approaches prevent flashing at the bearings, a leading cause of failure.

Another typical mag drive pump failure is liquid flashing at the impeller eye after being circulated through the drive end to remove eddy current heat. The result is a vapor-bound pump. New mag drive designs have virtually eliminated this problem by creating a constant pressurized circulation circuit that prevents flashing of cooling liquid and the associated failures (Figure 6). Not all new designs use pressurized circulation, and because most regulated liquids are volatile, this feature is necessary to achieve extended life in these services.

FIGURE 3

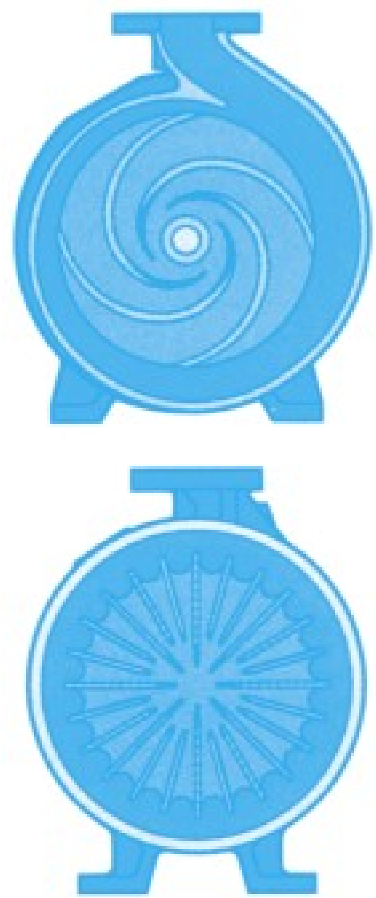


Pump curve for a low-flow ANSI pump.

Regardless of the design features and modifications available from manufacturers, users are responsible for providing suppliers with as much data as possible on fluid and operating conditions. To apply sealless pumps properly, many factors must be considered:

- Is the flow continuous or intermittent?
- Upon shut-down, what reaction (if any) will the process fluid have to residual heat? Chemicals like butadiene and formaldehyde may polymerize, leaving deposits inside the drive section and on the bearings.
- Can the process shut down automatically, resulting in the pump operating at shut-off condition?
- Conversely, can the system allow the pump to operate at the extreme right of the pump curve, which can adversely affect $NPSH_R$ and cause motor overload or excessive thrust?
- What are the fluid characteristics, including vapor pressure curves, specific heat, viscosity over the process temperature range, and the effects of heating and cooling on the process fluid? Benzene freezes at 42°F (depending on the installation location, address the possibility of exposure to low temperatures), and toluene diisocyanate freezes at 72°F and begins to polymerize at 127°F (again, protect the installation or use jacketing if necessary). Maleic anhydride freezes at 130°F (use heating jackets or temperature control).
- What about the customer's practical knowledge of the corrosive nature of the chemical? Sometimes the standard corrosion charts don't give the whole story.

FIGURE 4



An expanding volute pump (top) and a circular volute pump with a radial vane impeller (bottom).

ABRASIVES

When pumping fluids containing particles, the traditional solution is to use very hard bearings (silicon carbide) operating against a hard or coated journal. The application of sealless pumps should go beyond this seemingly easy approach. Consideration must be given to:

- the abrasiveness of the solids
- the size of the particles
- the quantity of particles
- whether they can agglomerate
- what creates the particles (reaction, catalyst, temperature)

The size of the particle that can be handled is usually determined by the impeller design and the clearances in the fluid passages. The effects of the quantity of particles are usually predicted from previous experience and test data. Solids may be formed by reactions to moisture (titanium tetrachloride), temperature (butadiene or formaldehyde), or a catalyst (any process that uses a catalyst that may vary in quantity or is subject to upsets).

When the fluid is understood, it may be best to use modifications, including: backflushing to keep particles out of the drive section, heating or cooling jackets, heat exchangers in flush lines, filters or specially designed units that utilize isolation chambers, built-in seals, and precision back-flushing to reduce process stream dilution, if economical. Otherwise a mechanically sealed ANSI pump may be the best solution.

MAG DRIVE CONDITION MONITORING

Magnetic drive pump reliability is also affected by operating practices. Condition monitoring devices can be applied to shut pumps down before a critical failure. Maintenance can then be performed, or operator errors corrected, before the pump is put back into service.

Temperature detection and power monitoring together provide the best basic protection. Temperature detection indicates internal pump problems such as plugged recirculation paths, while power monitoring prevents dry-run failure. Other devices available include low amp relays, leak detection indicators, and package control systems.

INSTALLATION

The effort involved in selecting the right pump for a given CPI application can be nullified by poor installation. As much effort, if not more, should be put into installation design to insure expected performance is achieved. (To understand how proper procedures improve equipment reliability see "Installation and Start-Up Troubleshooting," *Pumps and Systems*, November 1993.) Important steps include:

1. Lay out suction piping to provide NPSH available to the pump in excess of NPSH required. A common recommendation: $NPSH_A > NPSH_R$ + 2–5 ft. See "Pump Suction Conditions," *Pumps and*

ers' reps and rely on their expertise, but be informed, as well, and together you can apply pumps properly in your facilities. ■

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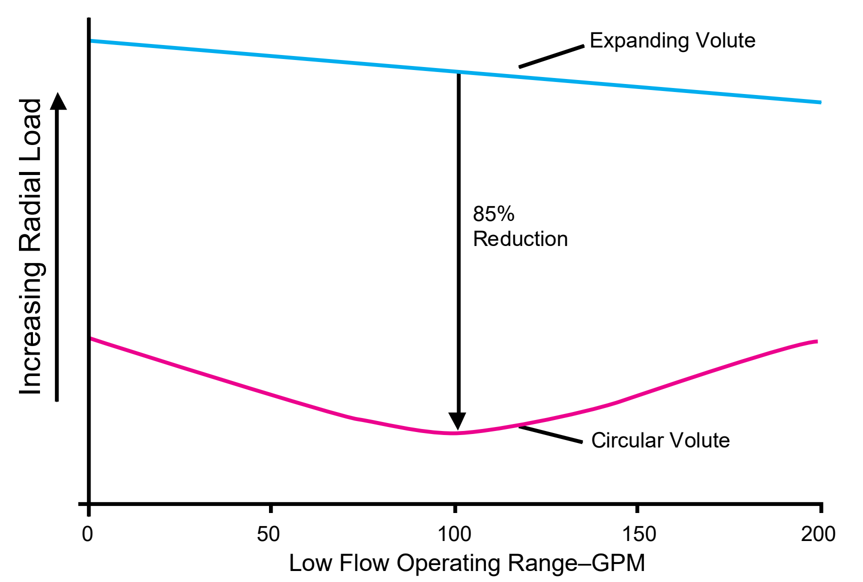
Bob Manion is product manager for magnetic drive and non-metallic pumps for Goulds Pumps Inc. He has held marketing and sales management positions related to developing, selling, applying, and servicing centrifugal pumps for 13 years. Mr. Manion holds a bachelor's degree in marketing from the Rochester Institute of Technology.

FIGURE 5



Radial load curves.

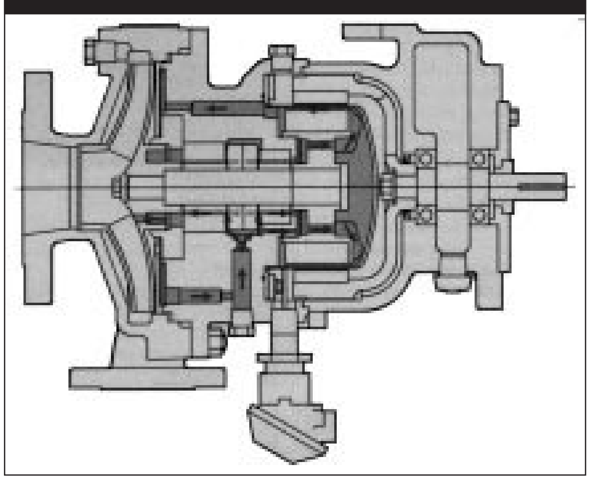
Systems, May 1993 and "How Much NPSH Is Enough?" September 1993.

2. Provide a straight run twice the length of the pipe diameter (2D) to the pump suction flange to prevent added turbulence at the impeller eye, which could lead to premature (incipient) cavitation.
3. Install conventional or cartridge mechanical seals according to manufacturer recommendations.
4. Meet seal flush requirements by providing an external flush at the necessary pressure and temperature, or add auxiliary piping for flushing on the pump.
5. Prepare the foundation before grouting the baseplate.
6. Select grout that will meet installation requirements.
7. Select a baseplate to maximize pump, seal, and motor reliability. Many vendors offer baseplates with enhancements such as .002 in./ft flatness, leveling screws, motor alignment screws, continuous drip rims, and other features designed to ease installation and alignment and increase pump life.
8. Align equipment according to manufacturer specifications.
9. Select and install condition monitoring devices for sealless pumps.

CONCLUSION

Selecting a pump to improve reliability will reduce emissions and operating costs at the same time. Neither a mechanically sealed ANSI pump nor a sealless pump can be universally applied on every process application. Make an informed decision based on specific service conditions and total cost (initial + maintenance + operating costs). To insure a return on investment, as much time and effort must be expended on the design of equipment installation as on pump selection. Although selecting equipment for increased reliability and reduced emissions may seem expensive in the short term, it saves money in the long run. Work with manufactur-

FIGURE 6



Recirculation circuit.

Canned Motor Pumps

When the canned-motor pump is the choice to solve a specific pumping problem and control costs, the following points must be considered to achieve satisfactory results:

CHEMICAL CHARACTERISTICS

There is seemingly never enough information available on the chemicals to be handled. The supplier must depend on the customer to provide this information, but it is also very important that the supplier and the customer exhaust their resources in an attempt to anticipate what a chemical will do inside the sealless pump. Will it cause corrosion, boil, decompose, freeze, or polymerize? Any of these properties can result in rapid failure unless anticipated.

APPLICATION AND METHOD OF OPERATION

Will the pump be used for transfer, condensate return, reboiler, or batch operation? Will it be running continuously or intermittently? Will the location be remote, exposed to the elements, or in a hazardous location? How will the pump be operated and what will the process demand? Can the flow range over the complete curve? Is it close to shut-off, which may require a by-pass orifice? Or, conversely, will it occasionally pump at the extreme right of the curve? This can result in cavitation and subsequent failure if allowed to continue. All of these factors, combined with the knowledge of the fluid pumped, will determine the proper selection and modifications necessary for successful pump operation.

DIAGNOSTICS AND CONTROL

Once the above factors have been determined, the user and supplier should agree on the type of diagnostic devices and process control that will assure a successful installation. Diagnostics available include:

- bearing wear monitors
- rotation indicators
- motor diagnostic devices
- bearing temperature sensors
- leak sensors
- flow sensors

Any of the above may be recommended. The pump and motor can also be fitted with a control device such as:

- a water or steam jacket
- a water-cooled heat exchanger
- a heat exchanger in the circulation line
- complete jacketing of the pump and motor

(Consider if the insulation will create motor heat problems.)

MAINTENANCE

The final consideration must be maintenance. Does the user have a planned maintenance program? Does the user's and supplier's experience indicate more frequent maintenance intervals than normal with the chemical product in this particular mode of operation?

Proper maintenance and replacement of less expensive bearings and gaskets can prevent a major failure and yield increased savings.

CONCLUSION

Using a sealless pump can be easily justified due to the elimination of leakage and emissions because the value of the chemical lost using a sealed pump can be calculated. But there are many other factors that are more difficult to quantify, including housekeeping costs, safety, odor, and public and employee relations. The major elements leading to long-term savings using sealless pumps is the upfront analysis of the application and the supplier's knowledge of his product. ■

Joe Cleary is the Vice President (retired) of Sales for Crane Co., Chempump Division.

Pump Buying Strategies

BY: J.T. MCGUIRE

To get the right pump, all you have to do is decide what you want, state those requirements clearly and place your order with a capable manufacturer. Sounds easy, doesn't it?

At first glance, pump requirements don't really seem that complicated. After all, a pump only needs to:

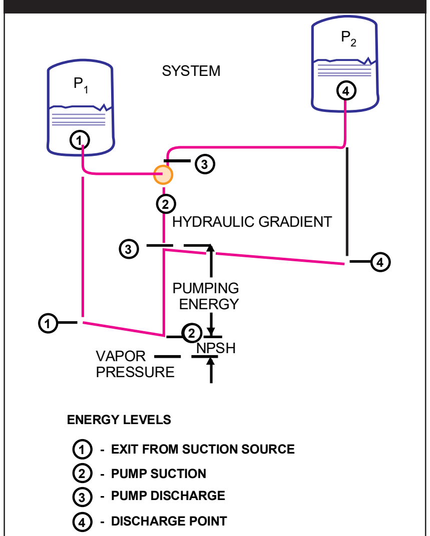
- move a specified volume of liquid through a given system
- be energy efficient
- comply with any applicable laws regarding leakage
- achieve certain mean time between overhauls and replacement
- be delivered on time with complete documentation
- and all at minimal cost

It may sound simple at first, but it's not. For example, some pump purchasers may not know the volume of liquid their system handles. Reliability data is hard to come by, too. With so many factors affecting pump life, mean time between overhauls and replacement may not be known. And the goals are conflicting. Increased service life may also increase energy consumption and purchase price. How do you sort through these factors? How do you determine what you need from a pump, develop a meaningful specification for those requirements and finally buy the right pump? The answer is to take it one step at a time and follow a disciplined approach to pump specification and purchasing.

DETERMINING PUMP REQUIREMENTS

To write requirements for a pump, you should review the basics of pumps. Pumps are designed to move liquid against a hydraulic gradient; in other words, to move liquid from the suction reservoir to the discharge reservoir, which differ in elevation and/or pressure (Fig. 1).

FIGURE 1



You can see immediately from the figure that the pump must supply adequate energy to overcome the difference in elevation and pressure along with the friction losses in the conduits on both sides of the pump. It's obvious that the pump, as the sole source of energy in the system, must supply all the needed energy. Thus it's no wonder that severe consequences await those who overlook that simple fact!

Another, perhaps not so obvious, fact is that the energy available at the suction side must provide a certain net margin over the liquid's vapor pressure at the pump suction. This net margin, called Net Positive Suction Head Available (NPSHA), is necessary to prevent cavitation — the boiling of liquid in the system. Cavitation impairs pump performance and shortens the service life of the pump. An excessive amount of boiled-off vapor impairs the machine's hydraulic performance. In addition, the subsequent collapse of the vapor bubbles as they move to regions of higher pressure can cause cavitation erosion.

To prevent these problems, you must specify total system head accurately. In most applications, you can determine the normal pump flow and the static components of the total head associated with ideal operation of the plant or process at its design output. Add estimated piping friction losses and control valve pressure drop (if applicable) to find the total system head for that capacity. (Remember friction head varies as the square of the flow ratio.)

Normal pump flow and static components aren't the whole story of operating conditions. You must also factor in the range of operating conditions your pump will be called on to perform under. Changes in operating conditions can be caused by:

- process unit downturn
- flow swing to cover upset or transient
- change in static head as vessel levels or pressures in both change with time
- change in friction head as system fouls or scales or as discharge vessel fills
- pump wear

You use these data to compute the rated flow — the flow under which your pump will need to operate. You then match the performance data (power, NPSHA, speed) quoted by the manufacturer against the rated flow.

You can set the rated flow to the maximum rate at which your pump will be called on to operate. But if the flow range is very wide (and you plan to use a centrifugal pump), you might set the rated flow to the most frequent or efficient flow rate. In either case, once you set the rated flow and required operating flow range, you will need to look at the NPSHA for the pump at these flows.

In addition to the rated flow, you must consider the range of flow. Pumps cannot operate across the entire range of flow from maximum flow to zero flow. With the exception of direct acting steam pumps, no pump has an infinite range. Centrifugal pumps, the most common pump in use today, can operate under a wide range of flows if they are designed appropriately. Thus, set the flow margin to allow for process transients and pump wear, but don't set it larger than necessary. And be sure the stated NPSHA reflects a value normally available, not some possible minimum value with a hefty hidden margin. Otherwise, you'll find yourself with an unsuitable pump — an oversized pump or one designed for abnormally low NPSHA.

Stated NPSHA should reflect a value normally available, not some possible minimum value with a hefty hidden margin.

DEVELOPING A MEANINGFUL SPECIFICATION

Once you determine the specifications for your pump, you need to communicate those specs to the manufacturer in concise terms.

For the manufacturer to understand what you really want, your specification must state, in an orderly manner, all the requirements you have for the pump. But that doesn't mean you should strive for a thick specification document. The value of the specification is not proportional to volume or weight. If anything, the inverse is true. Overly long specification documents often fail to state what the purchaser really wants.

Formats for the specifications range from a very simple functional spec to a very elaborate functional design and manufacture. The simple functional spec states only what the pump will be called on to do. You give complete freedom to the manufacturer for designing the pump. An advantage to such a specification is that you can get very interesting designs for unique pumping problems. But, such specs can be difficult to write and you'll need to evaluate the engineering behind the bids carefully.

To avoid the back-end expenses of a simple functional specification (and since most pumping requirements are relatively straightforward), most pump buyers write detailed functional requirements and manufacture specifications.

As a start, these specifications must address:

- operating environment
- liquid to be pumped
- pump performance and life
- materials of construction
- extent of supply

Many other items related to function, design and manufacture can be addressed (Table 1). The number of requirements you choose to include will depend on the pumping application and your confidence in the potential pump manufacturers.

Items you may want to pay special attention to include:

- Degree of redundancy (Item 4). This item refers to the proportion of spare capacity the pumping arrangement has to provide in the event one pump is lost. Typical values are 100, 50, 25 or 0 percent. Most purchasers have design standards relating degree of redundancy to the type of service involved.
- Type of pump (Item 7). This issue is complex, determined by the hydraulic duty, the degree of flow regulation required, and the nature of the pumped liquid.
- Number of pumps (Item 7). This item incorporates the required degree of redundancy (Item 4) along with the total flow to be handled, the total head to be developed, or the total power absorbed. It can also incorporate the physical size of the pump (often related to type of pump) and the required turndown in flow when rated flow is high.
- Service lives of the pump and various components (Items 12 through 15). These requirements are usually expressed as mean time between failure. As noted earlier, data on the life of various pump components are meager. Thus, these requirements are often not specified. Generally, antifriction bearings and the first stage impeller of high energy centrifugal pumps are the only components for which minimum service lives are commonly specified.
- Materials of construction (Item 16). You'll have to handle this item since the pump manufacturer does not control the pumped liquid. If you have little or no experience pumping the particular liquid, manufacturers will suggest possible materials. But the only guarantee a pump manufacturer makes for materials is that they will conform to their producing specification.
- Extent of supply is an essential issue. (Items 7, 8, 22, 23, 25 and 26). When faced with increasing complexity and extent of specifications, many pump purchasers find it beneficial to summarize extent of supply (also known as terminal points or battery limits) in a list or diagram. That's a good idea. It helps you state more clearly what you need.

To simplify the specification, you should note all technical elements on a basic data sheet. And remember that the basic data sheet should be just that — a sheet. Multi-page data sheets are unwieldy. If your system is complicated, cite and add supplementary sheets rather than cluttering the basic data sheet.

Instead of building custom specs, some purchasers in particular industries use general specifications issued by that industry (for example, ANSI B73.1M-1991, which addresses horizontal end suction pumps for chemical process and API-610, 7th edition, which addresses centrifugal pumps for petroleum refining). Some buyers use these general specifications verbatim, others use them as a base and add supplement covering changes they wish to incorporate.

A cardinal rule for any meaningful specification, whether homegrown or based on an industry standard, is to avoid multiple tiered references to other specifications. With more than one tier of references, such specifications become too complicated to be meaningful. For example, when addressing government regulations, be sure to identify and specify the exact rules and regulations the equipment has to meet. The old catchall, "comply with applicable local, state, and federal rules and regulations," doesn't add anything to the specification.

Beyond technical requirements, the specification also must address the proposed terms of purchase or commercial terms and conditions. Although these items are generally the province of the purchasing department, you, as a specifying engineer, should be aware of what is involved. The major items covered in the terms and conditions are:

- delivery period or date
- point of delivery
- liquidated damages
- terms of payment
- warranty
- default
- cancellation
- bankruptcy

Delivery period and terms of payment should be of special interest to you. For example, if the delivery period is too short, there arises the risk that somewhere in the manufacturing of your pump, a shortcut or two will be taken resulting in a pump that will not function adequately in the field.

Also of interest to an engineer is method of payment. By tying payment to achieved manufacturing milestones, you can expedite the manufacturing of your pump, and thereby help to ensure on-time delivery.

TABLE 1. ELEMENTS OF A PUMP SPECIFICATION TECHNICAL

Item		Function	Design	Manufacture
1.	Location and environment	X		
2.	Liquid pumped and properties	X		
3.	Hydraulic duty	X		
4.	Redundancy in pump arrangement	X		
5.	Future performance margin	X		
6.	Application margins		X	
7.	Type and number of pumps		X	
8.	Driver and arrangement		X	
9.	Minimum tolerable piping loads	X		
10.	Allowable seal leakage	X		
11.	Allowable noise	X		
12.	Minimum pump life	X		
13.	Mean seal life	X		
14.	Bearing life and basis	X		
15.	Mean period between overhauls	X		
16.	Materials of construction		X	
17.	Rotor design requirements		X	
18.	Hydraulic design requirements		X	
19.	Allowable stress		X	
20.	Type of shaft seal		X	
21.	Type of bearings and lubrication		X	
22.	Type of coupling		X	
23.	Type of base		X	
24.	Piping: systems required and construction		X	
25.	Auxiliary systems: specification		X	
26.	Instrumentation		X	
27.	Material tests			X
28.	Welding procedures approval			X
29.	Inspection during manufacture			X
30.	Component and equipment tests			X
31.	Painting and inhibiting			X
32.	Documentation			X

BUYING THE PUMP

Once you've clearly specified the pump, it's time to place the order. This process can go smoothly, if you:

- double check that the pump you're ordering is really the pump you want
- order the pump in time to allow for orderly manufacture
- have a post-award meeting, within one month of ordering, to ensure the order is clear and started
- don't change the order unless safety or a major performance problem is involved

The final three steps are self-explanatory, but the first two deserve some explanation.

Double checking your order is especially important for complex units with an extensive specification. As a check, hold a pre-award meeting with the manufacturer to clarify the bid. If your unit includes major auxiliary equipment or systems, review and settle the basic unit plot plan at this meeting.

Selecting a manufacturer can be done in one of two ways: specify-and-evaluate or partnership-purchasing. Under the specify-and-evaluate method, you prepare a very detailed specification and issue inquiries with extensive data requirements, then thoroughly evaluate the data in the resulting bids and purchase based on the numerical results of the evaluation. The evaluation generally takes the form of a weighted matrix which includes:

- energy consumption
- maintenance cost
- risk of lost production
- purchase price
- delivery

Manufacturers are free to bid whichever pump they feel meets your specifications. That leaves you, the purchaser, to make the final determination of whether a pump meets your requirements. Thus, you'll need to build a rigorous inspection regime into your selection process. Over time, you can build a list of acceptable bidders to help narrow the field.

Partnership-purchasing avoids the cost of preparing an elaborate specification, issuing inquiries, and evaluating bids. Under this method, you select one manufacturer to work with and provide just a minimal specification. The manufacturer then chooses the best pump for your needs.

To select the best manufacturer to work with you, you need to assess the caliber of the various manufacturers that make the class of pump you've chosen. Your assessment should cover each manufacturer's:

- order engineering and manufacturing processes
- emphasis on quality as an inherent facet of all processes
- product design philosophy
- detail designs for and experience with the class of pump required

After choosing the best manufacturer to work with, you can negotiate prices for equipment according to some fixed relationship to published price lists. While you will incur some costs in assessing manufacturers, this process is likely to be less expensive for a major product or a period of two or three years between assessments than open bidding would be.

Which approach is better? For innovation in design and reliability, I've found that partnership-purchasing yields distinctly better results than specify-and-evaluate has. Engineers are not surprised by this; they know that technical endeavors proceed best in a cooperative arrangement.

The specify-and-evaluate method might help you find a company that will furnish equipment nominally capable of the same function for less money (even when factoring in the cost of writing the specs and evaluating the bids). But the issue isn't just cost. The real value of partnership-purchasing is innovation in design and reliability of products. Partnership-purchasing is actually the way the pump industry used to operate before competitive bidding became so popular. As an industry, we, the suppliers and purchasers, would do well to resurrect it. ■

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A Common Sense Approach to Combating Corrosion and Abrasion

BY JOHN RINARD

Photo 1. New and corroded centrifugal pump impellers.

A single look at Photo 1, above, should be enough to convince anyone of the destructive nature of corrosion and abrasion on pumps, and lead to the question of how to prevent this from happening.

This article should serve as either a primer or a reminder of factors involved in properly selecting or troubleshooting a pump in corrosive and/or abrasive service. Historically, pump selection has consisted of finding a pump that will, "pump stuff from here to there," or that will "deliver so many gpm at such-and-such a head." A greater degree of sophistication leads to "and that will hold up in acid," or "and that will pump solids." Obviously, the more that is known about the solution being pumped, the more appropriate the pump selection will be. An interrelationship exists where the chemical and physical properties of the pumpage determines the materials of construction, which dictates pump design, which affects pump performance, which in turn determines the proper pump selection. The more difficult selections involve services pumping corrosives and/or abrasives, and these are the major factors governing which pump is chosen.

You don't need to be a rocket scientist to select a pump, and you don't need a PhD in metallurgy to make some basic materials selections and understand the reasoning behind them. We all know that water will "rust" iron, acids "corrode" certain materials that come into contact with them, and solids "wear" when rubbed together; conversely, we know that "stainless" steel is corrosion resistant and that either a hard or soft "rubber" material will resist abrasion or wear. These simple facts lead us to a closer examination of the mechanisms of corrosive and abrasive attack.

Corrosion is the wearing away or deterioration of a material by chemical or electrolytic action or attack.

Abrasion is the wearing away of a material caused by a solid rubbing or impinging on another. Abrasion caused by the velocity of a liquid or gas is commonly called erosion.

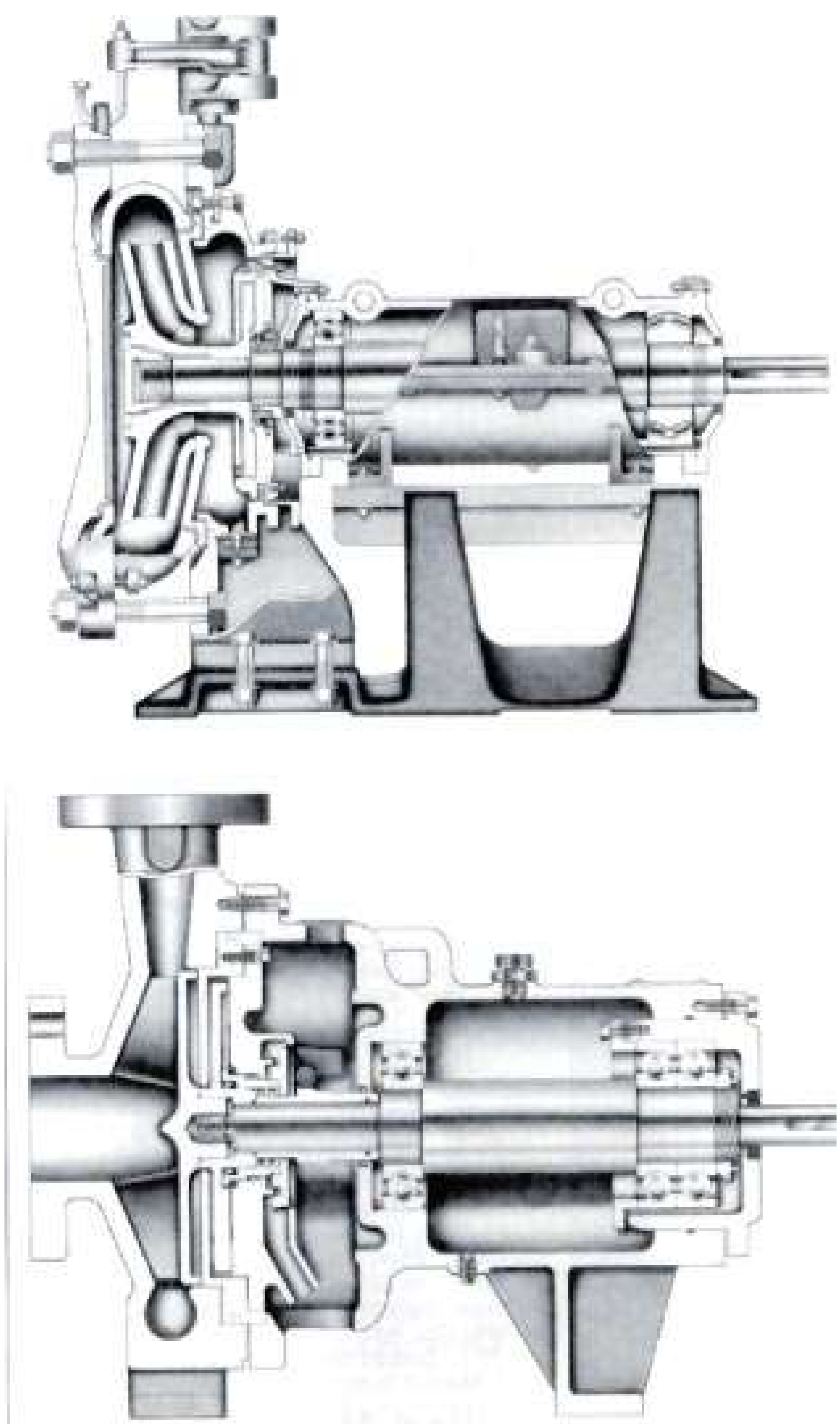
Corrosion-abrasion is a combination of both corrosion and abrasion that results in an accelerated attack on material. It is generally more severe than either corrosion or abrasion alone, due to the severe wear caused by the continuous abrasive destruction of the passive protective film built up by corrosion.

Table 1 shows the basic types of corrosion. Corrosion and abrasion take many forms, and numerous combinations of these forms exist. Detailed analysis of these combinations can be quite complex and goes beyond the scope of this discussion.

MATERIALS

There is no material that will withstand attack from *all* combinations of liquids and solids found in pumped solutions. However, a basic knowledge of material categories will give us a general idea of what materials will and will not work in certain environments, and then we can zero in on the right pump for a given job.

FIGURE 1



Pump design comparison. On the top is a hard iron slurry pump with side suction. The bottom is a stainless ANSI B73.1 chemical pump with an expeller-type seal.

It becomes obvious with examination of Table 2 that the mechanical properties of a material determine the design of a pump. Pumps constructed of hard materials are more difficult to design (flanges, stack tolerances, and clearances), cast (sharp angles and complex shapes), and machine (drill, tap, and finish surfaces); non-metallics may need to be reinforced, supported, or protected with metal armor; and thin or highly stressed components must be made of strong materials.

Figure 1 shows a typical configuration of both a chemical (corrosion resistant) pump and a slurry (abrasion resistant) pump. One can readily see that the slurry pump's hard metal materials of construction dictate the use of through-bolt construction rather than drilled and tapped holes. Less apparent are the facts that slurry pumps are generally more massive than chemical pumps; are designed with open clearances, blunt edges, and looser tolerances due to "as cast" hard metal surfaces and the need to handle solids; and are commonly designed with metallic or nonmetallic liners. As a result of these design constraints, slurry pump efficiencies suffer, and in most cases are lower than chemical process pump efficiencies.

Identification of materials that can handle the liquid to be pumped does not necessarily complete the material selection process; quite often this step leads to other considerations. Options and compromises almost always present themselves with either chemical or slurry pumps when comparing service life and wear with cost and availability.

CHEMICAL PUMPS

Wear. The chemical process industry generally considers that any corrosion rate equal to or less than 20 mils per year is acceptable wear. This, however, may be considered excessive depending on either pump design (pump impellers with relatively thin vanes and shrouds effectively see double this wear rate because they are totally immersed in the liquid and therefore exposed to attack from both sides) or a need for extended service life for pumps in critical services and inaccessible or remote locations.

Cost. Some material costs may be prohibitively high and therefore lead to selection of less corrosion resistant alternatives or a lined rather than a solid material pump.

Availability. While materials such as 316 stainless steel, CD4 MCu, and Alloy 20 are commonly stocked and available for chemical pumps, alloys such as monel and Hastelloy are more likely to be special orders. "Standard" materials of construction vary from manufacturer to manufacturer. Depending on the pump type and the manufacturer, material availability can vary anywhere from being in stock to needing up to several months lead time.

TABLE 1. TYPES OF CORROSION

Type	Characteristics	Remarks
General	Uniform attack over entire exposed surface	Most common type of corrosion
Erosion-Corrosion	Corrosion accelerated by erosive action of fluid or slurry vortex	
Crevice	Localized attack at crevices or stagnant areas	Commonly found at gasketed or flanged surfaces
Galvanic	Occurs when two dissimilar metals are immersed in a corrosive or conductive solution	
Intergranular	Grain boundary attack	Weld decay is a type of intergranular corrosion occurring in areas adjacent to a weld
Cavitiation	Pitting on high pressure areas such as impeller vane tips and/or low pressure areas such as eye of impeller vanes or trailing edge of impeller vanes	
Pitting	Localized accelerating attack by chlorides; associated with stagnant conditions	Common in 304 and 316 SS, and A20
Selective Leaching	Dissolves one component of an alloy	Zinc removed from brass or bronze is called dezincification; when grey cast iron is attacked, graphite is left undisturbed

Lined or coated pumps and nonmetallics offer possible solutions to the high cost and long lead times of nonstock special metallic materials. Lining pumps is where nonmetallics really shine, and they are less expensive and more readily available than special metallics. However, just as there is no single metallic that is good for handling every solution pumped, there is also no single nonmetallic for all services. Each must be carefully selected to fit the service.

SLURRY PUMPS

The selection of abrasion-resistant materials for slurry pumps, much the same as corrosion-resistant chemical pump material selection, also involves consideration of service life (wear), cost, and availability. Abrasion, unlike corrosion, is generally combated by the use of either very hard materials or soft, resilient elastomeric materials. Hard materials are generally used for slurries with large or sharp solids. Soft, resilient elastomeric materials are used for small or blunt solids. Once again, here we find that non-metallic elastomers lend themselves for use as pump liners.

CHEMICAL-SLURRY PUMPS

It was mentioned earlier that a pumped solution that is both corrosive and abrasive, an acid sludge, for example, presents the greatest challenge in pump selection. Many materials are essentially suitable to either corrosion or abrasion, but not to both; titanium, for example, is a very strong, corrosion-resistant material, but it is unsuitable for slurries because of its softness, and white iron is a very hard, abrasion-resistant material that is not practical for corrosive conditions. Nonmetallic elastomers, on the other hand, may be used in a service that is both corrosive and abrasive. When selecting elastomers, consideration must be given to solids size and configuration, temperature, and a pump design that must generally preclude liquid contact with any metallic armor or reinforcing.

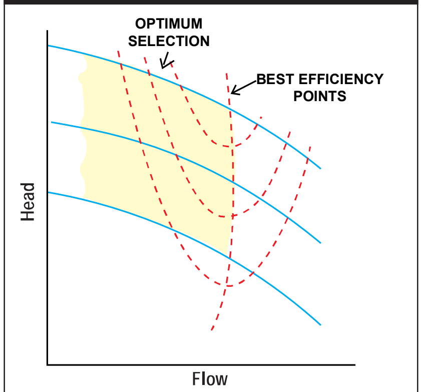
PUMP PERFORMANCE

The efficiency of a pump as well as the location of the operating point on the pump performance curve is often overlooked or ignored during pump selection. The location of the operating point is overlooked more often than the pump's efficiency. This alone will contribute as much as any other factor to pump failure when abrasion or corrosion-abrasion are present. Efficiency is a measurement of smooth flow—and therefore reduced turbulence and recirculation—within the pump. Turbulence and recirculation result in increased liquid and solids contact with wetted pump surfaces, as well as unpredictable angles of impingement. Proper pump selection, therefore, dictates selection at or near the best efficiency point of the pump. Selection just to the left of best efficiency is considered good practice, as illustrated by Figure 2.

Analysis can often trace the cause of pump problems to operating the pump at or too close to shut off (to the far left of best efficiency) because the pump is oversized. Intentional oversizing may occur through the use of system design safety factors, selection for future increases in performance, and using an existing pump without consideration of size. Unintentional oversizing may occur because of miscalculations or changes over time in the process or the piping system. The end result is the same; the pump is operating too far away from the best efficiency point.

When analyzing pump performance, we must think in terms of a pumping system rather than just the pump. A system consists of the pump and all the related piping, valves, and process equipment on both the suction and discharge sides of the pump. All of these items directly affect the pump performance in that the "system curve" (which can be analytically derived from the pressure drop/resis-

FIGURE 2



Optimum pump selection results in a pump operating just to the left of its best efficiency point.

tance to flow across various in-line hardware) dictates where the pump will operate on its curve, the pump point of rating. Less sophisticated considerations include rules of thumb such as:

- Keep the suction piping as short and straight as possible.
- Slope the suction piping toward the pump suction when handling slurries.

Centrifugal pumps tend to become unstable the closer they approach either shut off (zero flow) or maximum flow. This instability may be manifested in cavitation, recirculation, and turbulence. Recirculation and turbulence can result in a liquid temperature rise in the pump that can cause accelerated corrosion as well as erosion-corrosion.

PUMP SELECTION

The final selection decision is made by the pump user. This decision may be more subjective than analytical, but should include such factors as:

- Availability (of both pump and parts)
- Maintainability
- Reliability
- Service life
- Standardization
- Cost

There are always trade offs. The user ultimately makes a selection based on the priorities that best meet the process needs. This paper has presented an overview of corrosion and abrasion factors that should be a part of that selection process.

ACKNOWLEDGMENT

My thanks to Dr. George Calboreanu, Chief Metallurgist, Western Foundry, a division of A.R. Wilfley and Sons, for his materials expertise and assistance in preparing this paper. ■

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TABLE 2. MATERIALS COMPARISON

Category	Subcategory	Material	Typical Mechanical Properties		
			Typical Hardness*	Tensile Strength (Min, psi)	Elongation (Min % in 2")
Metallics	Ferrous	Steel	150 Brinell	70,000	22
		Ductile Cast Iron	160 Brinell	60,000	18
		27% Chrome	600 Brinell	80,000	Nil
	Stainless	304 SS	150 Brinell	70,000	35
		316 SS	150 Brinell	70,000	30
		CD4MCu	225 Brinell	100,000	16
		A20	125 Brinell	60,000	35
		Hastelloy B/C	225 Brinell	75,000	20-25
	Copper base	Brass	60 Brinell	37,000	30
		Bronze	65 Brinell	35,000	18
	Miscellaneous	Aluminum	130 Brinell	65,000	8
		Titanium (pure)	200 Brinell	80,000	18
		Zirconium (pure)	210 Brinell	55,000	12
Non-Metallics	Elastomers	Rubber (gum)	35 Durometer A	3,500	500
		Neoprene	55 Durometer A	3,000	650-850
		Urethane	75-95 Durometer A	4,500-7,500	250-900
	Plastics	Teflon (PTFE)	50-65 Shore D	3,000-4,000	200-400
		Epoxy (cast)	M75-110 Rockwell	2,000-12,000	Nil
		Polypropylene	R85-95 Rockwell	5,000	500-700
	Ceramics	Silicon Carbide	2,500 Knoop	44,500	Nil
		Aluminum Oxide	1,000-1,500 Knoop	22,000-45,000	Nil

There are numerous alloys, formulations, and compounds of metallics and non-metallics; those shown are typical and are not to be considered all-inclusive.

*Conversion relationships of hardness scales/numbers are discussed in ASTM E140 and Metalcaster's Reference and Guide, 2nd Edition, 1989, The American Foundrymen's Society, Inc., (for metals), and ASTM D2000 (for rubber).

Recommendations For Vertical Pump Intakes

BY: HERMAN GREUTINK

LOCATION

A vertical turbine, mixed flow or axial flow pump's location in a sump is critical to good performance. Figures 1 and 2 provide good design criteria for sump layout. These criteria are based on a maximum bell entrance velocity of 6 ft/s. However, because bell diameters vary from manufacturer to manufacturer, these ratios must be adjusted to accommodate the differences.

According to the U.S. Army Corps of Engineer's design guide, "For satisfactory pump performance based on research and prototype experience, recommended submergence, S, should be 1.25 D or greater, and the dimensionless flow ratio through the individual pump should not exceed a value of 0.40 for:

$Q/\sqrt{gD^5}$

where

Q= discharge, cfs

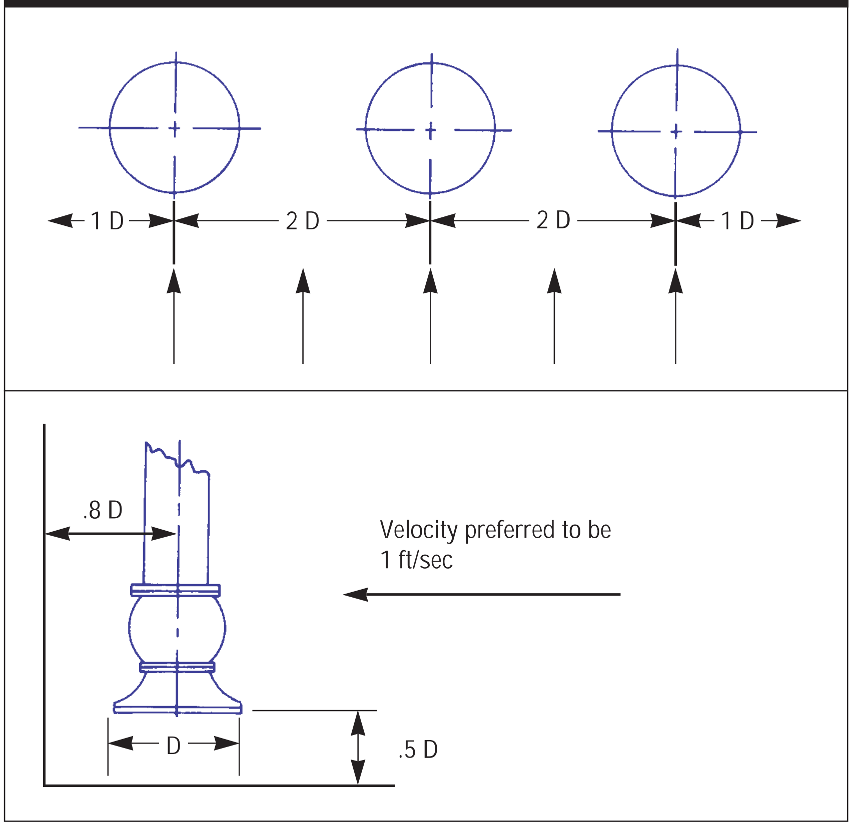
D= pump bell diameter, ft

g= acceleration due to gravity, 32.2 ft/s^2

Submergences that are less than, and flow rates that exceed the above limits were investigated, and more complex designs were required for satisfactory hydraulic performance."

The recommendation of 1.25 D minimum submergence is suitable for storm water and flood control pumps (provided a vortex supressor beam is used as illustrated by Figure 2); however, for continuous service pumps a submergence of 1.75 D is recommended. If the submergence is less than these values, the bell diameter must be enlarged. For instance, to meet a 1.25 D submergence value, the bell diameter should be enlarged to produce an average entrance velocity of 3.3 ft/sec. This velocity may be a bit conservative, but the cost of enlarging the diameter is low and the benefits are tangible.

FIGURE 1



VORTICES

If a vortex still occurs after you have followed the above guidelines, it is not generally difficult to alleviate. It takes very little energy to form a vortex; therefore, it takes very little energy to get rid of it!

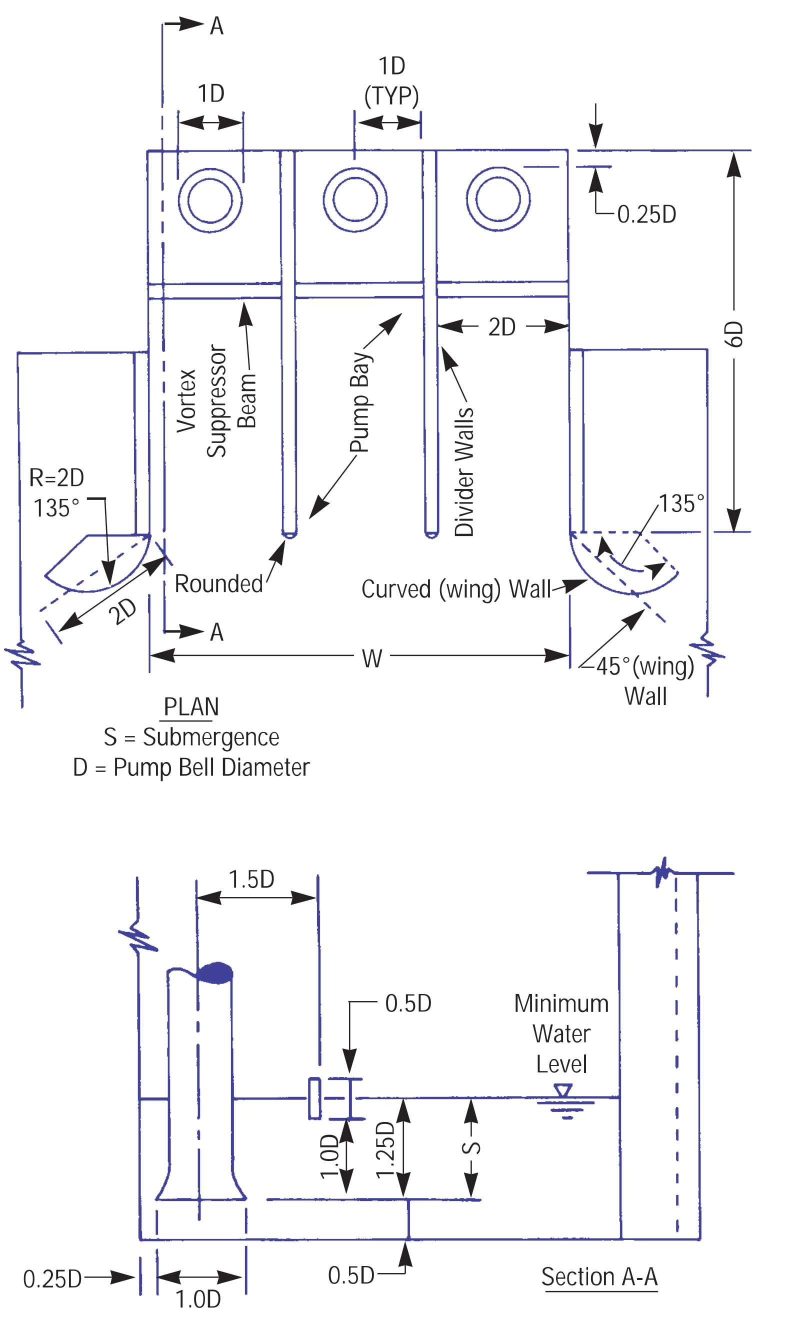
Submerged vortices, however, can be troublesome. These vortices will touch the floor and/or wall of an intake. They are the result of swirling masses of water next to or under the pump and are not continuous. Although submerged vortices sound like cavitation due to the lack of net positive suction head (NPSH), the noise created by a vortex comes and goes as the vortex comes and goes. To mitigate submerged vortex formation, apply the following strategies:

- Place a cone under the bell.
- Employ splitters.
- Fill-in intake corners.
- Use diffuser screens.

HIGH VELOCITY

As a rule, high velocity to a pump in the intake and/or at the bell leads to reduced life of the pump. For a given head and capacity, today's pumps operate at approximately double the speed of the pumps in use before the

FIGURE 2



1960's. The net result of these higher speeds is a drastically increased frequency of pump repairs. Slowing down continuous service pump speeds may be more expensive initially but the long-run savings on maintenance will more than compensate for the increased pump costs.

A high velocity stream aimed at or near the pump could also be a source of premature failure. A fluid force of this nature should be diffused by piling, screens or walls in front of the conduit outlet. Figure 3 provides a simplified depiction of distances required to diffuse a high velocity flow out of a conduit.

The breakup of jet streams can be achieved by baffles as shown in Figure 4. This configuration also promotes better distribution to multiple pumps.

DIVIDING WALLS

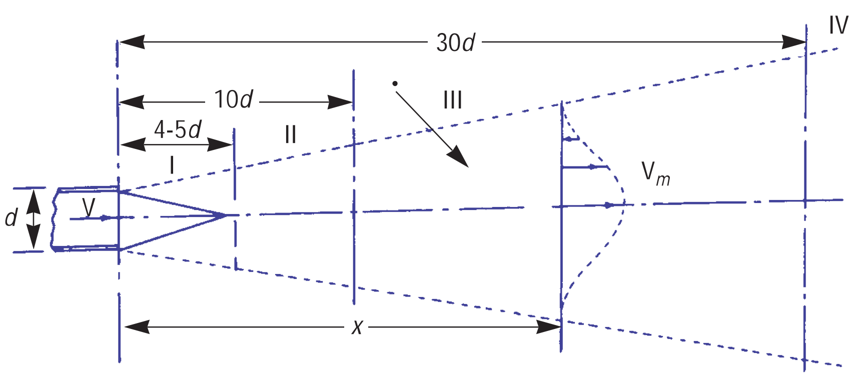
Because short dividing walls are not recommended, they are not pictured in any of the figures. (Figure 1 shows no dividing wall while Figures 2 and 4 show long dividing walls.) With multiple pump stations, the front of the short walls can propagate vortices when one or more pumps are out of service. So it is better to have no walls than short walls. Long walls provide easy support for the pumps, as well as drainage for individual pump sumps when stop logs are used.

INTAKE TESTS

When guidelines such as those published by the Hydraulic Institute and the British Hydro-mechanics Research Association (BHRA) cannot be followed, model intake tests should be performed, especially for pumps larger than 50,000 gpm. ■

FIGURE 3

Area I - Potential Core
Area II - Transition
Area III - Similar Velocity Profiles
Area IV - Jet Center Line Wanders

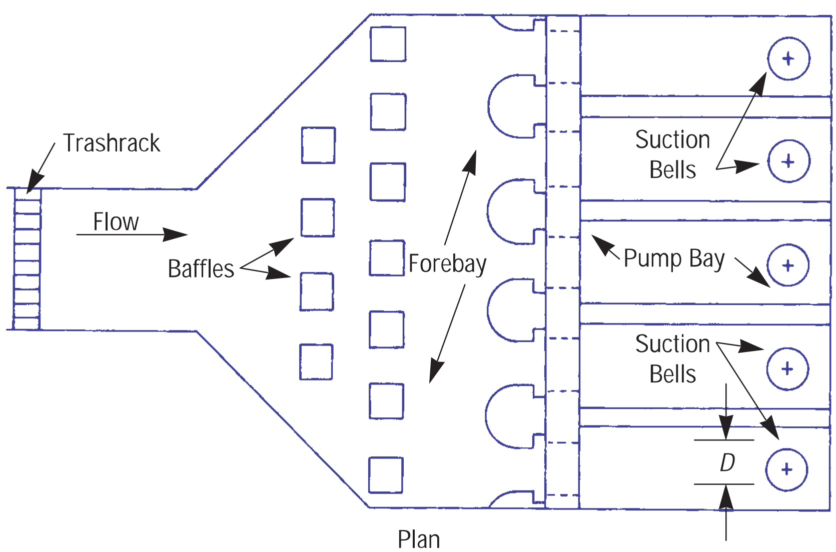


Up to about 30 times diameter d, the formula

$$\frac{V_m}{V} = 6.5 \times \frac{d}{x}$$ is used to determine V_m

V and V_m (ft/sec.), d and x (ft.)

FIGURE 4



Plan

D = Suction bell diameter, ft.

U.S. Army Corps of Engineer's Design Instructions for Flood Control Pumps

REFERENCES

1. U.S. Army Corps of Engineers, Engineering Technical Letter No. 1110-2-313 "Hydraulic Design Guidance for Rectangular Sumps of Small Pumping Stations with Vertical Pumps and Ponded Approaches."
2. Prosser, M. J. "The Hydraulic Design of Pump Sumps and Intakes." British Hydromechanics Research Association.
3. Hydraulic Institute Standards, 14th Edition

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Hydraulic Instabilities and Cavitation

Causes, Effects and Solutions

Hydraulic excitation forces and pressure pulsations created by excessive flow deceleration at partial load have a profound impact on the possible failure of a variety of pump components. These forces and pulsations are the result of flow recirculation in the impeller inlet, diffuser or volute sections of the pump. Some degree of recirculation is present in every centrifugal pump below a specific flow rate representing the "onset of recirculation." In fact, recirculation is of minor concern for the majority of pump designs. On the other hand, excessive recirculation can be extremely harmful and destructive. Consequently, the onset of damaging recirculation is of greater concern to pump operators than the onset of recirculation itself.

IMPELLER INLET RECIRCULATION

Three physical mechanisms trigger flow recirculation during partial load at the impeller inlet:

1. deceleration of the velocity upstream to the impeller relative to the velocity in the impeller throat
2. pressure gradients perpendicular to the direction of main through-flow
3. excessive incidence (i.e., difference between blade angle and flow angle at the impeller vane leading edges).

The primary geometrical parameters impacting the above phenomena are:

- impeller throat area
- angle of approaching flow
- impeller vane angles
- ratio of impeller eye diameter to hub diameter
- ratio of vane tip diameter to hub diameter
- impeller shroud curvature
- impeller leading edge position (in planar view and meridional section).

However, no simple general relationship exists between the onset or amount of recirculation and the geometry of the impeller. Relationships have been derived that are valid only for particular families of impellers (Ref. 1). Applying these relationships to impellers developed according to design rules differing from those underlying the correlation would be very misleading.

IMPELLER OUTLET RECIRCULATION

Downstream to the impeller the flow may be decelerated in a stationary component of the casing. This deceleration can occur in a diffuser, a volute, an annular casing or a combination thereof. The physical mechanisms of downstream deceleration are quite similar to those previously mentioned for upstream recirculation. They are:

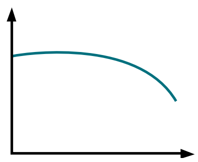
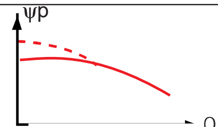
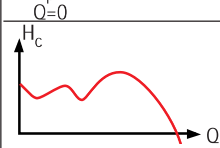
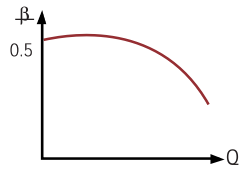
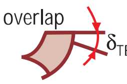
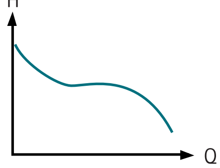
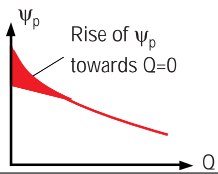
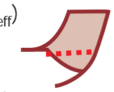
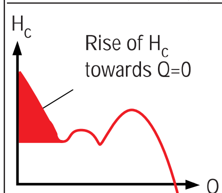
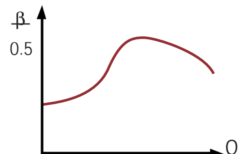
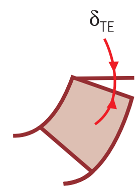
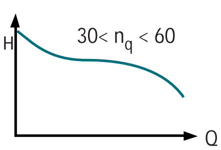
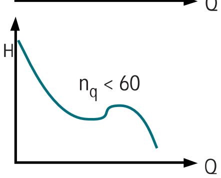
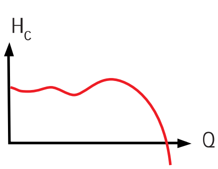
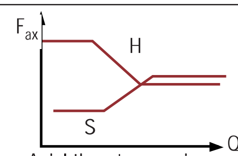
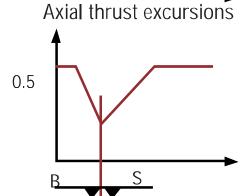
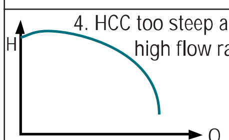
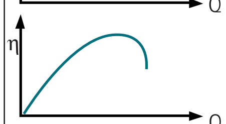
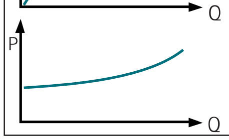
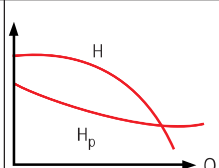
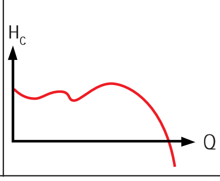
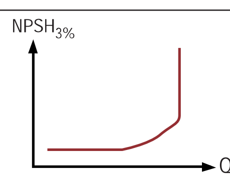
1. deceleration of the absolute velocity from the impeller outlet to the throat of the casing
2. incidence at the diffuser vanes or volute cutwater
3. pressure gradients perpendicular to the direction of the main flow (particularly for semi-axial or axial pumps).

The main geometrical parameters impacting flow recirculation at the outlet are:

- velocity distribution at the impeller outlet as determined by the geometry of the impeller

BY: J.F. GUELICH AND T.H. MCCLOSKEY

TABLE 1. MEANS TO UNDERSTAND AND TO MODIFY THE SHAPE OF THE HEAD/CAPACITY CHARACTERISTIC (HCC)

Symptoms: Q-H-curve	Symptoms: Internal pressures	Symptoms: Axial thrust, $\beta/\omega = f(Q)$	Possible causes or mechanisms	Possible remedies (NOTE: any modifications may have side-effects which should be carefully assessed)
1. HCC drooping towards shut-off  Typical for $n_q < 30$	 H_p, ψp flat towards Q=0		• Insufficient recirculation at impeller inlet (insufficient centrifugal head increase at low flow)	• advance impeller vane leading edge (reduce $d_{1\ eff}$) • advance return vane trailing edge (reduce c_{ou}) • increase d_{1a} (increase $\Delta\ d_{1\ eff}$) • reduce hub dia. (increase $\Delta\ d_{1\ eff}$) 
	 H_c, flat or drooping towards Q=0		• Insufficient recirculation at impeller outlet (insufficient exchange of momentum between impeller and diffuser at low flow)	• reduce a_3 or s_8 • increase b_3 / b_2 • reduce gap A, increase overlap • increase δ_{TE} • increase b_2 • reduce gap B (with small n_q) but beware of increased pressure pulsations 
2. Excessive shut-off head and/or excessive shut-off power  Typical for $n_q > 35$			• Excessive recirculation at impeller inlet	• reduce d_{1i} (increase $d_{1\ eff}$) • cut back return vane (increase c_{ou}) • reduce d_{1a} • increase hub dia. • inlet ring } reduce $d_{1\ eff}$ 
H = total dynamic head Q = net flow through pump	 ψ_p = static head rise of impeller / $(u_2^2/2g)$ H_c = head rise in casing	 Sudden decrease of β due to outlet recirculation F_{ax} = axial thrust towards suction $\frac{\beta}{\omega}$ = ratio fluid/shroud rotation	• Excessive recirculation at impeller outlet	• increase a_3 or s_8 • reduce b_3 / b_2 • reduce b_2 • increase δ_{TE} 
3. HCC with saddle or flat position  		 Axial thrust excursions  S = outlet recirculation on shroud B = outlet recirculation on hub	• Flow separation in diffuser (volute) but not yet fully developed recirculation • Shifting of flow patterns (zones of recirculation / flow separation) • Very sensitive to manufacturing tolerances of diffuser and impeller • Very sensitive to impeller inlet flow conditions and impeller inlet geometric parameters • Sensitive to axial rotor position: HCC and F_{ax}	• Axial thrust fluctuations and sensitivity to rotor position / axial stage stacking tolerances can be eliminated by reducing gap A and introducing proper overlap • To remove saddle type instability detailed flow analysis and / or testing often required • Differences in stage geometry to get onset of flow separation in different stages at different flow rates
4. HCC too steep at high flow rates   	 	 Steep rise of NPSH little or not at all affected by impeller inlet	• Excessive flow acceleration in diffuser or volute throat • Cavitation in diffuser or volute	• Increase diffuser or volute throat area

itself as well as the velocity distribution at the impeller inlet

- diffuser or volute throat area
- diffuser vane or cutwater angles
- ratio of impeller outer diameter at tip to hub (oblique cut of radial, semi-axial or axial impellers).

INCREASING SHUT-OFF HEAD

Ample evidence suggests that increasing the recirculation at the impeller inlet and/or outlet increases the head. A number of geometrical parameters can be altered to increase the head in this manner. If the flow versus pressure head (Q-H) curve is drooping towards shut-off, invoking recirculation may be appropriate, especially since the shut-off head is particularly affected by an increase in recirculation.

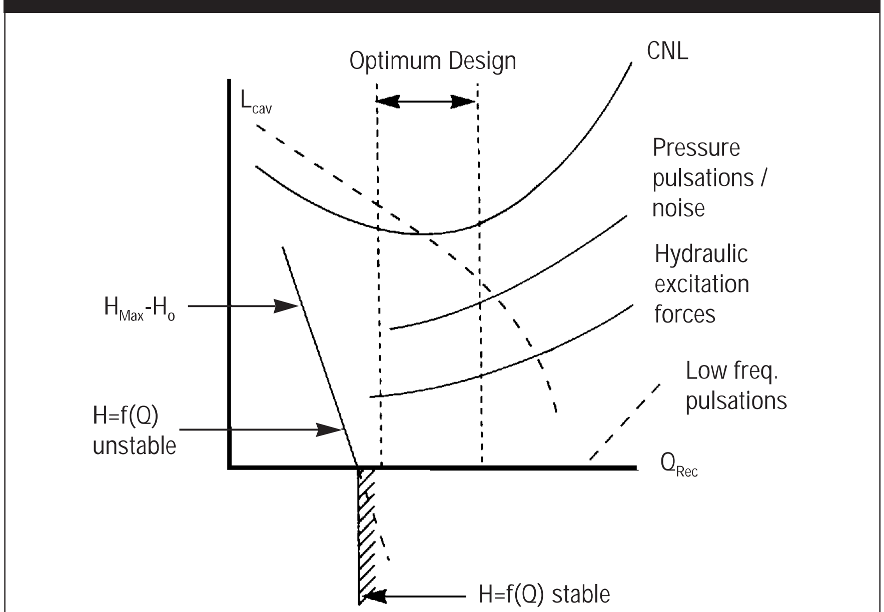
Table 1 presents typical shapes of Q-H curves, explains the physical mechanisms responsible for these curve shapes and suggests possible remedies and geometric parameters by which the shape of the Q-H curve can be corrected. However, the reader should be aware that these remedies may produce undesirable side effects (e.g., reduction in head or efficiency at best efficiency point (BEP)).

DAMAGING RECIRCULATION

For every type of pump there exists a range of optimum recirculating flow, and operating a pump within it avoids the risk of unstable Q-H-curves on one side and the risk of damaging levels of recirculation on the other. This range is illustrated qualitatively by the Figure 1 graph. Unfortunately, there is no established method to predict exactly the onset of damaging recirculation. An unacceptable level of recirculation can be determined indirectly, however, in individual cases by applying the following strategies:

- Measure cavitation noise to assess the risk of cavitation erosion.
- Test for vibration.
- Monitor shut-off head or shut-off power. If excessive, this may indicate inordinate recirculation.
- Measure the radial or axial hydraulic excitation forces. A sudden rise in these forces as the flow rate is reduced and/or consistently excessive levels of these forces could indicate an unacceptable degree of recirculation.
- Measure pressure pulsations. A sudden rise in pressure pulsations as the flow rate is reduced and/or consistently excessive pulsation levels could indicate damaging flow recirculation. However, a sudden rise of pressure pulsations might also result from standing wave resonance. For this reason, to avoid the possibility of misinterpretation of high loads of pressure pulsation, careful testing and data analysis is imperative when diagnosing the true nature of pulsation.

FIGURE 1. OPTIMUM AMOUNT OF CIRCULATION



Hydraulic excitation forces and pressure pulsations may be responsible for a number of possible component failures. Table 2 details the root causes and mechanisms of such failures along with possible remedies.

Partial load flow phenomena also strongly influence pump vibration. Observed vibration phenomena causes and mechanisms, along with possible remedies, are given in Table 3.

CAVITATION EROSION

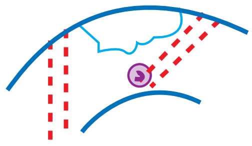
If, as a result of recirculation, the local pressure at the impeller inlet drops below the saturation pressure of the pumped liquid, vapor bubbles are generated and are then swept by the flow into zones of higher pressure, where they implode and may cause erosion of the impeller.

To eliminate or reduce cavitation damage, the following remedies are available:

- Change operation procedures if damage occured at partial load or overload.
- Increase net positive suction head available (NPSHA).
- Reduce speed or use a varible speed drive if partial load is required.
- Increase cavitation resistance of material.
- Modify geometry of impeller (profiling of blades, impeller redesign).
- Improve inlet flow conditions by geometric modifications.
- Increase gas contents.

In addition, Table 4 outlines cavitation damage mechanisms and offers correlating remedies. As illustrated

TABLE 2. EFFECT OF HYDRAULIC EXCITATION ON COMPONENT FAILURE

Failure / incident	Possible hydraulic causes or mechanisms	Possible remedies	Possible non-hydraulic causes / remarks
1. Fracture of impeller blades at outlet, diffuser vanes at inlet, tie bolts, instrument piping, or other components	• High dynamic stresses induced by pressure pulsations (impingement of wake flow from impeller blade trailing edge on diffuser vanes or volute cutwater)	• Increase gap B by cutting back (1) diffuser vanes if diffuser throat does not increase by more than 3% (2) impeller blade trailing edge (head of pump will be reduced unless speed cannot be adapted) • Reduce excitation at part load by modifying hydraulic components (careful analysis and redesign)	• There are a number of other failure mechanisms related to design, material selection and quality Remark Pressure pulsations and dynamic stresses are expected to decrease with a power of -0.77 of gap B. For example to achieve half of the original level gap B must be increased by a factor of about 2.5
2. Side plate breakage 	• High dynamic stresses induced by pressure pulsations • Impeller side plate resonance if $\lvert z_3 - z_2 \rvert = 2$ and z_3 n/60 close to impeller side plate natural frequency	• Increase gap B (see previous item) • Change z_3 / z_2 combination • Modify natural frequency • Reduce exciation at part load by modifying hydraulic components	• Insufficient quality of impeller casting and / or finish (notch effect) • Insufficient thickness of impeller side plates
3. Mechanical seals	• High pressure pulsations caused by wake flow or recirculation / separation • High frequency pressure pulsation due to cavitation • Shaft vibrations	• (see above) • Reduce cavity volume by redesign of impeller and / or inlet • see table 3	• There are a number of other failure mechanisms related to design, material selection and quality
4. Excessive labyrinth wear	• Excessive radial thrust • Excessive vibration	• Reduce flow asymmetries around impeller by - double volute in case of single volutes - analyzing / eliminating cause of asymmetry (casting tolerances, differences in resistance in channels of double volutes, discharge and suction nozzle,...) • see table 3	• Thermal deformations of casing and rotor
5. Failure of radial bearings	• Excessive radial thrust • Excessive vibration	• see above item • see table 3	• Mechanical / design
6. Failure of axial bearings	• Axial thrust excursions • Excessive labyrinth wear (high leakage increases rotation on shroud; reduces rotation on hub with multistage-pumps)	• see table 1, item 3 • Replace wear rings	• Mechanical / design • Transients

by the following case study, geometric modification of the impeller is frequently the only feasible solution.

CASE HISTORY

After a boiler feed pump had operated for more than 50,000 hours with no trace of cavitation on the suction impellers, the load demand of the process changed, requiring prolonged partial load operation. The pump operated about 1000 hours at 60% load and 1100 hours at 80% load before cavitation damage was discovered on the pressure side of the impeller blades. The attack varied between 2-4mm from blade to blade. Since the cavitation damage occurred at partial load on the pressure side of the blades, flow recirculation was identified as the most probable cause.

To improve the partial load range and thereby increase the impeller life, an inlet ring was designed and installed in the pump. Figures 2 and 3 show the fluid-borne and solid-borne noise prior to and after this modification. Prior to modification the noise recorded at 100% and 80% flow is virtually equal. Since no erosion occurred during more than 50,000 hours of operation at full load, this evidence suggests that the operation at 60% load was entirely responsible for the damage. As illustrated by the figures, the modification of the pump decreased the noise at 60% flow to the unmodified 100% flow noise level, and the erosion problems were solved. ■

ACKNOWLEDGEMENTS

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2. J.F. Guelich et. al. ”Feed Pump Operation and Design Guidelines.“ EPRI Final Report TR-102102. June, 1993.

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NOMENCLATURE

A	amplitute
a_3	diffuser throat width
b_2	impeller exit width
b_3	diffuser inlet width
C_{om}	absolute velocity at meridonal inlet point
CNL	cavitation noise level
c_{ou}	absolute velocity upstream of impeller
D_1	impeller eye diameter
D_2	impeller outer diameter
d_1	D1/D2
d_{1eff}	impeller vane inlet diameter where flow enters the impeller
F_{ax}	axial thrust towards suction
f	frequency
f_n	rotational frequency
H	head per stage of pump
H_c	head rise in casing
H_p	static head rise of impeller
L_{cav}	cavity length
n_q	pump specific speed (metric convention)
Q	flow rate
Q_{SL}	flow at shockless entry
S	outlet recirculation on shroud
S_8	volute throat area
u_1	circumferential velocity
z_2	number of impeller vanes
z_3	number of diffuser vanes
ß	angular velocity of liquid
δ_{TE}	trailing edge angle
σ	slip factor
σ_{u1}	cavitation coefficient $(2gNPSH/(u_1)^2)$
ψ_p	static pressure rise of impeller
ω	angular velocity of impeller

Subscripts:

av	available
BEP	best efficiency point
rec	recirculating flow

FIGURE 2. FLUID-BORNE NOISE IN A FEED PUMP (1 TO 180 kHz)

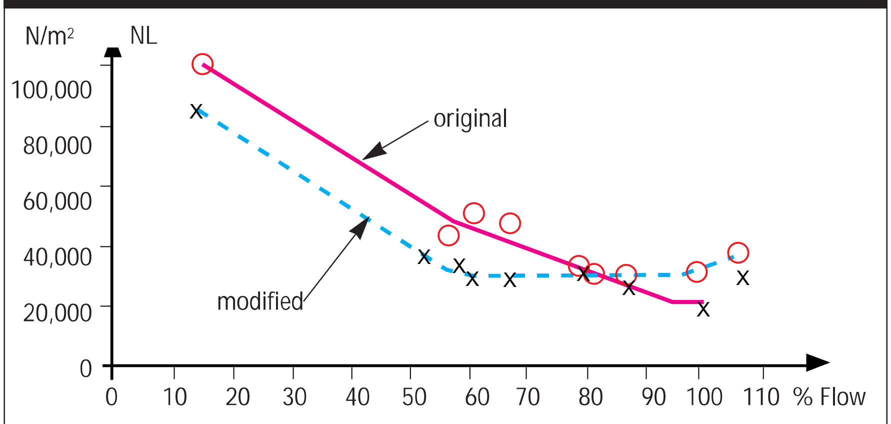


FIGURE 3. SOLID-BORNE NOISE AT A FEED PUMP (10 TO 180 kHz)

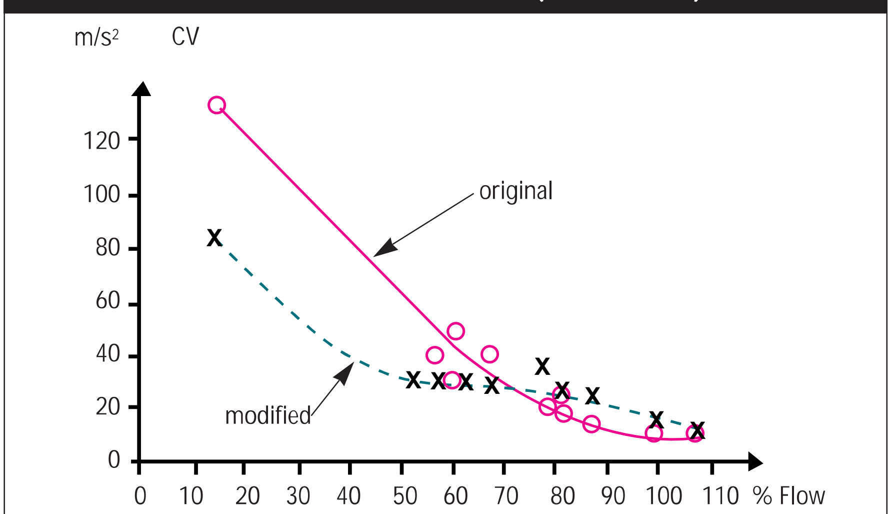


TABLE 3. INTERACTION BETWEEN FLOW PHENOMENA AND VIBRATION

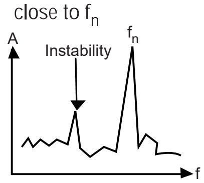
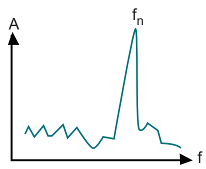
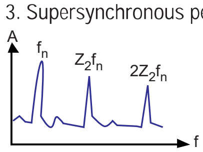
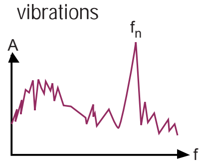
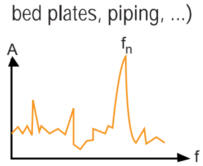
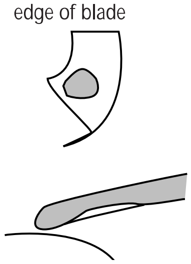
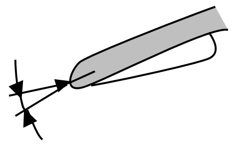
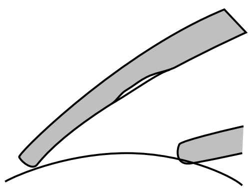
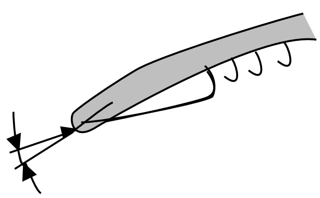
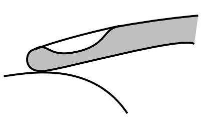
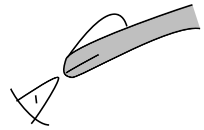
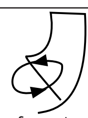
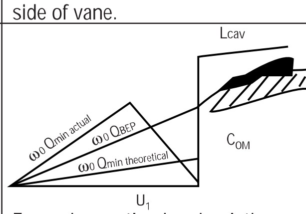
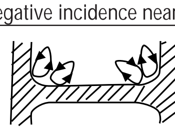
Observed vibration: Spectral component	Q/Q_{BEP}	Possible hydraulic causes and mechanisms	Possible remedies	Major non-hydraulic causes / remarks
1. Subsynchronous peak close to f_n  Subsynchronous vibrations increase with time	< 1.0 > 1.0 all	• Increased labyrinth preswirl ⇒ reduced rotor damping • Unloading of bearings due to change in radial thrust with flow • Rotating stall • Labyrinth wear ⇒increased leakage ⇒increased preswirl ⇒reduced rotor damping	• Increase rotor stiffness and damping by introduction of plain labyrinths or shallow serrations only • Reduce preswirl to labyrinths by swirl brake • Impeller or diffuser redesign	• Labyrinth design • Thrust balancing device does not provide sufficient rotor damping • Bearing design / bearing unloaded Remark: Instability may be recognized by very steep increase in amplitude with increasing speed
2. Synchronous vibration 	all	• Hydraulic unbalance due to various impeller tolerances	• Reduce casting / manufacturing tolerances of impeller (precision casting, ceramic core procedures, manufacturing) and implement more stringent question/answer procedures	
3. Supersynchronous peaks  A = amplitude f = frequency f_n = rotational frequency z_2 = number of impeller blades	all	• Pressure pulsations caused by wakes from the impeller blade trailing edge • Harmonics other than blade passing frequency are due to impeller casting tolerances (pitch of the blades) • $\lvert z_3 - z_2 \rvert$ = +/- 1 resulting in non-zero radial blade force component at $z_2\ f_n$	• Peaks nearly always present. If excessive: - Increase gap B (see table 2, item 1) • Harmonics other than blade passing frequency: reduce impeller casting tolerances • Change number of impeller or diffuser vanes • Reduce excitation force by proper staggering of impellers on shaft	
4. Broad band shaft vibrations 	typically below 50% of BEP flow	• Flow recirculation at impeller inlet and outlet • Some broad band vibrations are unavoidable. If excitation is excessive this can be due to oversized throat areas of diffuser or volute, oversized impeller eye or excessive incidence • Fluctuating cavities	• If excessive: reduce diffuser or volute throat area; reduce impeller eye (careful review of hydraulic design required) • As a cure of the symptoms the rotor damping might be increased (swirl brakes, labyrinth redesign, see above) • Reduce cavitation extension (higher $NPSH_{av}$, redesign of impeller or inlet)	
5. Structural resonances below frequency of shaft rotation excited by broad band hydraulic forces (e.g. bearing housing, bed plates, piping, ...) 				Remark: It is typical that structural resonances excited by broad band forces do not depend on the speed of the shaft
6. Rotating stall	typically below 90% of BEP flow	• Stall cells in diffuser or impeller rotating with a frequency below the frequency of rotation	• Analysis and redesign of hydraulic components • Increase rotor damping (swirl brakes, labyrinth redesign)	Remark: The peaks are expected to be proportional to the rotor speed
7. Surge-like strong pulsations	low	• Vapour core forming in the suction pipe at low $NPSH_{av}$ due to strong part load recirculation	• Structures upstream of impeller to avoid formation of core (flow straightener, cross, inlet ring, hub diameter, "back-flow catcher") • Impeller / inducer redesign • Air admission (if possible)	

TABLE 4. CAVITATION DAMAGE MECHANISMS AND REMEDIES

Type of cavitation / damage pattern	Flow mechanisms likely to induce damage	Possible causes	Possible remedies
1. Suction side of blade, starting close to leading edge of blade 	 Sheet cavitation on suction side of blade at $Q < Q_{SL}$ • Damage near shroud • Damage near hub	• High incidence • Unfavorable leading edge profile • Outer blade angle β_{1a} too large • Inner blade angle β_{1i} too large	• Increase flow rate • Reduce blade inlet angles • Improve leading edge profile • Reduce incidence by inlet ring • Reduce impeller eye diameter if above optimum range • Increase pre-rotation
2. Suction side of blade, within channel  Sometimes erosion is also observed on the shroud and / or the hub or on pressure side of the blade	 Vortex cavitation on suction side of blade at $Q < Q_{SL}$ at low σ. Vortexes developing downstream of a long, thick cavity attached to the blades. Bubbles created in the vortexes are swept away by pulsating flow and can implode anywhere in the channel. Q_{SL} = flow at shockless entry	• High incidence at low σ (typically $\sigma_{u1\,av}$ = 0.15 to 0.3) • Insufficient $NPSH_{av}$ • Insufficient cavitation resistance of material	• Reduce impeller eye diameter if above optimum • Reduce blade inlet angle • Increase NPSH available • Increase cavitation resistance of material
3. Pressure side of blade, any location, starting close to leading edge of blade 	 Sheet cavitation on pressure side at $Q > Q_{SL}$	• Negative incidence due to excessive flow • Unfavorable leading edge profile • Excessive (run-out) flow	• Reduce flow rate • Increase blade inlet angle (but beware of partload cavitation) • Improve leading edge profile if damage close to leading edge
4. Pressure side of blade, damage at outer half of impeller width starting close to leading edge	 Bubbles in free stream generated by shear flow due to partload recirculation. Bubbles impinge on pressure side of vane.	• Excessive flow deceleration (excessive impeller eye diameter, excessive impeller throat area, excessive blade angles)	• Increase flow rate • Impeller redesign (decrease eye diameter / throat area / blade angles) • Inlet ring at the impeller entrance (reduce deceleration, reduce shear flow effects)
5. Pressure side of blade, damage near hub. Difficult to distinguish from item ③ unless it can be determined whether pump has operated at partload or above BEP	 Excessive partload recirculation creates a negative incidence near hub	• Negative incidence near hub due to partload recirculation	• Increase blade angle at hub • Reduce recirculation • Improve leading edge profile to reduce flow separation near hub under recirculation • Reduce preswirl (vanes, ribs, backflow catching elements) of recirculating flow
6. Damage on hub or shroud or in fillet radii	 Comer vortex cavitation often combined with high incidence	• Blade angles not properly matched to the flow • Fillet radii too large	• Impeller redesign (adaption of blade angles) • Required fillet radii

High Speed/Low Flow Pumps: Top 10 Issues

The authors answer your top ten questions about high speed/low flow pumps that perhaps you were afraid to ask.

The September 1994 issue of *Pumps and Systems* featured an article entitled "Low Flow Options." Among other things, it discussed the single port diffuser pump design credited to Dr. U. M. Barske. A high-speed version of this design was popularized in 1959 with its application as a water injection pump for the Boeing 707 airliner. An industrial version of this concept was first marketed in 1962 and tens of thousands have now been placed worldwide. The single port, or partial emission, pump has been developed commercially by a number of manufacturers. The designs accommodate both two and four pole electric motor drives and are most often applied for low specific speed services. Figure 1 shows the basic geometry of an open impeller, "partial emission" diffuser, design. The high-speed version of this design has garnered its share of "wives' tales" over the 30-plus years of commercial manufacture. Following are the "Top 10" notions and real issues that generate continuous discussion among users of these products. Although many of the topics are worthy of individual papers, we are limited here to an overview along with some helpful hints.

PHOTO COURTESY OF KOP-FLEX, INC.

Photo 1. A centrifugal pump performing high speed/low flow duties in a Gulf Coast plant.

10. EFFICIENCY: HERE TODAY, GONE TOMORROW?

Partial emission pumps use open impellers and therefore do not rely on wear ring clearances to maintain hydraulic efficiencies. Although this design allows more recirculation back to the suction than an enclosed impeller design, efficiency definitely depends on the disk friction component. Disk friction is the drag loss between the body of rotating fluid being carried by the impeller and the stationary walls of the chamber (Figure 2). In clean, non-corrosive streams, the finish on the chamber walls remains in "as new" condition resulting in steady, long term hydraulic efficiency.

Typical finishes are machined to a 62 (micro inch) RMS value. Thus, a surface finish of 250 can reduce the hydraulic efficiency by as much as 10 points. In extreme cases, total power consumption has been known to double that of the "clean" rating. Of the

BY: DAVE CARR AND ROBERT LINDEN

FIGURE 1. OPEN IMPELLER

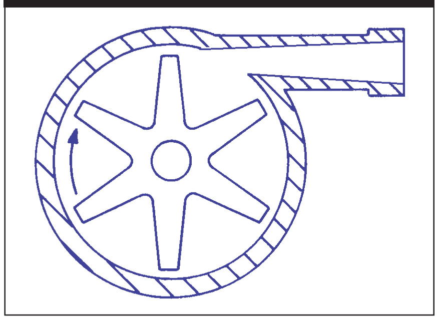
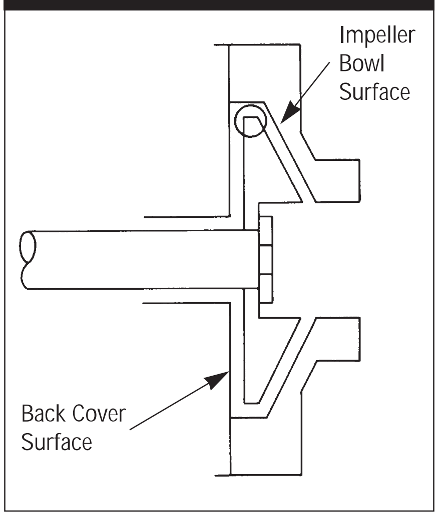


FIGURE 2. PARTIAL EMISSION CHAMBER SURFACES



two chamber surfaces, the finish of the backing plate is actually of greater significance than the impeller bowl in its effect on efficiency. Machined skin cuts at depths of 0.005 to 0.010 inch have been shown to efficiently restore the pump's original performance by reconditioning the surfaces to the material's original finish.

In addition to wear of the bowl material, there is also potential for foreign material build up on the surfaces. The corresponding skin, or film, similarly changes the effective finish and may also degrade the efficiency. The pump's sensitivity to this condition increases with tip speed and operation with lighter liquids. For this reason, wetted materials may be selected to shed any build up or to resist the potential for accumulations to occur. For example, salts have an affinity for carbon steel materials. Build-up of the type commonly experienced in contaminated butane streams has been negated by upgrading to 316 stainless steel construction.

Either of these effects can be verified by measuring the motor amperage or the liquid's temperature rise. The latter indicator increases proportionately to the decrease in hydraulic efficiency, i.e. by the equation:

$\Delta T=\{H(1-\eta)/778C_p\eta\}$

Units: T is in °F, H is head in feet, Cp is specific heat in BTU/LB-°F and η is efficiency expressed as a decimal.

Comment: Practically speaking, temperature rise analysis can be difficult because it typically ranges between 2 and 10°F.

9. THE SUCTION SPECIFIC SPEED IS WHAT?

Quoting from Lobanoff & Ross, "The inducer is basically a high specific speed, axial flow pumping device...that is series mounted preceding a radial stage to provide overall system suction advantage." This relationship is paramount to evaluating suction specific speed with high-speed pumps. Figure 3 shows a portion of a well-rounded inducer family.

An industry rule of thumb is to limit suction specific speed to a value of 11,000 (English units) in heavy duty process pumps. It is based on the premise that pumps operating at greater suction specific speeds have demonstrated a shorter service life, most likely due to rougher off design operation.. As conventional impellers are capable of suction specific speed values greater than 11,000, via oversized impeller eye configurations, this may be a good guide. To provide a useful net positive suction head (NPSH) performance guide for high-speed pumps, however, will require a change in the reference point!

High-speed pumps inherently require inducers to achieve competitive performance. Inducer designs have advanced to the point of providing reliable, cavitation-free operation with suction specific values of approximately 24,000. Efforts to desensitize inducer operating ranges have been addressed by optimizing blade number, angle and passage areas as well as using various inlet bypass designs. One design that resists cavitation surge, referred to as a "backflow recirculator," is reported to improve inducer turndown to a near shut off condition. Suction vortex breakers are also used but are less effective than flow stabilizers. Nonetheless, they can improve low flow stability, with high-speed/inducer style pumps, by as much as 25 to 35%.

In general, inducer use is discouraged with conventional API pumps, but high-speed pumps clearly require them. Several leaders in the Hydrocarbon Processing Industry

FIGURE 3

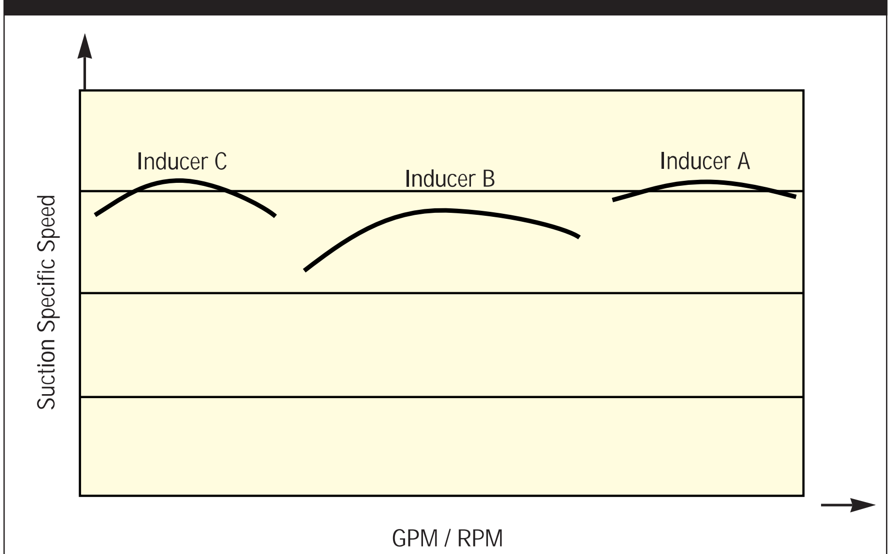
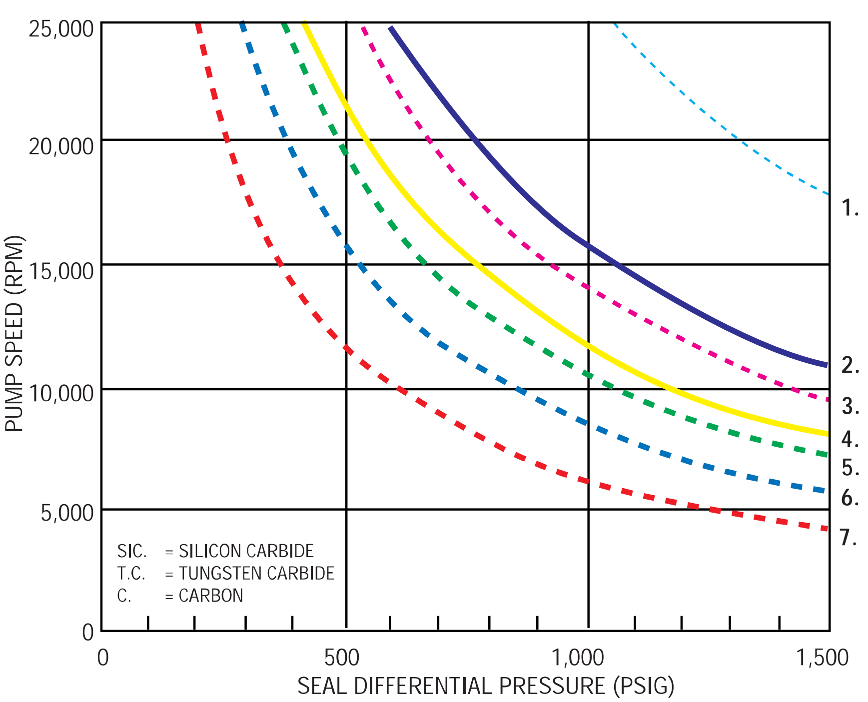


FIGURE 4. PRESSURE-SPEED LIMITS AS A FUNCTION OF SEAL FACE MATERIALS



1. 1 1/4", 1 1/2", & 2" SIC. vs. C.

2. 1 1/4 IN. T.C. vs. C.

3. 1 1/2 IN. T.C. vs. C.

4. 1 1/4 IN. T.C. vs. SIC.

5. 1 1/2 IN. T.C. vs. SIC.

6. 2 IN. T.C. vs. C.

7. 2 IN. T.C. vs. SIC.

(HPI) have recognized the unique position that this equipment occupies in the marketplace and have explicitly exempted such designs from the 11,000 suction specific speed limit. Care still must be exercised, however, in matching particular inducer configurations with the pump's operating flow range, and interaction with manufacturers that recognize cavitation erosion limits within their application guidelines.

8. HOW CAN MECHANICAL SEALS HANDLE THAT SPEED?

Mechanical sealing of high-speed pumps presents numerous obstacles including the potential for vibration, high sliding speeds, heat generation and high sealing pressures. Much of this is the direct result of rotational speeds that reach a maximum of 25,000 rpm, which is more common with compressors than with pumps. These conditions place high-speed pump manufacturers in the unique position to accept "seal success ownership." The bulk of the issues are addressed by reversing the conventional seal configuration. Vibration control dictates the spring-loaded component as the stationary part with the hard face as the rotating member. Small seals, typically with either 1-1/4" or 1-1/2" diameters, are used to minimize sliding speeds. A 1-1/4" seal at 15,000 rpm has a face sliding speed of approximately 82 feet per second. This equates to a 5-1/2" seal operating at a traditional 3,550 rpm (Figure 4.). When combined with the fact that open wheel impellers present seal pressures nearer to suction than to discharge (i.e., approximately 10% rise over the suction pressure) manageable pV values are seen at the seal faces. This allows most applications to be handled by conventional carbon vs. tungsten carbide material combinations. Silicon carbide also may be used to raise the operational limits.

Most high-speed pump seal problems are not caused by high speed, but by a lack of understanding or information regarding the fluid's properties. The most common problems are rust and scale, inadequate vapor pressure margin and a build up of solute at the atmospheric interface. Proper application of typical API seal flush plans, e.g., -31 (flush through a separator), -13 (reverse flush) and -54 (quench), normally negate these concerns and promote good seal reliability with high speed pumps.

7. PREPARING THE PUMP FOR STARTUP.

High-speed pumps are typically accorded extra care during the startup process due to respect for the technology. The lubricant level, oil cooler venting, fresh oil filter installation (if applicable) and driver rotation are inspection points which are consistently addressed. Seal piping, however, can be quite another story. API Standard 610 dictates, and industry practice provides, that all seal ports be plugged prior to factory shipment. In the field, the permanent appearance of some of these plugs frequently leads to failure to remove them. The pump case tags, engineering drawings and the instruction manual all provide a critical definition of the ports' functions and the corresponding auxiliary piping requirements, and must be followed.

Unfortunately, failure to properly configure the pump often will not cause any immediate problems. A common occurrence is an improperly vented port that can direct accumulated process seal leakage toward the back side of the gearbox mechanical seal, or bearing seal. The potential for lubricant contamination exists when an atmospheric drain is connected to a flare header that experiences significant upsets. Care should be exercised to ensure that these actions do not occur since the livelihood of the gearbox depends upon limiting this pressure to an absolute maximum of 10 psig.

6. SUBTLE TRUTHS ABOUT NPSH.

Net positive suction head (NPSH) is a subject worthy of its own full fledged article, and has been covered previously in *Pumps and Systems*. It should be emphasized, however, that a centrifugal pump's NPSH performance is established based upon the breakdown of the standard head versus capacity curve. When inducers are used, the total pump NPSH performance is measured, not just the inducer's performance. The most commonly accepted parameters are based on the Hydraulic Institute's 3% head suppression criterion. By definition, however, the pump will cavitate when conditions at the suction flange meet those test, or predicted curve, conditions.

High-speed inducers (in essence axial pumps) can develop as much as 25 to 100 feet of head. This energy, however, cannot be included within the overall pump requirements due to corresponding inlet eye losses at the centrifugal impeller. The net result is that the centrifugal portion of the pump still must be sized for the full design head rating.

The subtle aspect of NPSH revolves around the pumped fluid's properties and its potential to flash. Cavitation damage is a function of a liquid's propensity to release vapor. Water pumps typically will produce rated head and flow, albeit with the potential for some material damage, despite close proximity to the 3% head suppression value. This may be attributed to the high surface tension characteristic of water. The same trait makes it aggressive toward cavitation damage but the pump generally works!

Difficulty occurs at the other end of the spectrum where high speed pumps are used with low specific gravity services. Using the 3% suppression value for light fluids, plus the industry rule of thumb for an additional 2 to 3 feet safety factor, the NPSH margin may not be adequate with low specific gravity fluids. One example of this situation is realized when a slight heating of the fluid occurs on the suction side of the pump, particularly with above-ground supply pipes from storage vessels to transfer pumps. This can result in off-gassing and failure of the pump to hold prime. Ironically, this situation contradicts the API hydrocarbon offset factors that are typically prohibited by company specifications. In general, increasing the NPSH margin by an additional 4 to 6 feet is appropriate with light gravity liquids, i.e., less than 0.7 specific gravity.

Also of concern is the fact that the NPSHR value typically increases beyond the best efficiency point of the machine. Pump startups are commonly uncontrolled and result in operation at the end of the curve due to a lack of sufficient back pressure. The practical result of this situation is the fact that the pump may then run at too high an NPSH requirement, and could promote a disconnect between the impeller/inducer and the liquid stream. Therefore, the control valve's initial trim position and manual venting of the pump system should be anticipated in preparation for startup.

5. CENTRIFUGAL PUMPS IN A POSITIVE DISPLACEMENT WORLD.

High-speed centrifugal pumps are often installed in applications designed for positive displacement (PD) pumps. This is due to the inherent ability of both pumps to deliver a high differential pressure. Unfortunately, the two designs must operate under significantly different control schemes. This fact must be recognized when retrofitting from one configuration to another.

Figure 5 shows the theoretical characteristics of PDs and centrifugals with regard to flow and head capabilities. It is evident that the positive displacement design is limited by head (pressure) and the centrifugal by flow. Consequently, the PD uses a pressure relief valve to prevent over pressurization and to bleed off excess capacity. In practice, centrifugal pumps exhibit only a moderate head rise across their operating region. The radial vaned centrifugal, in particular, demonstrates a 5-10% head rise from the best efficiency point (B.E.P.) to the peak of the head versus capacity curve. This margin does not facilitate the use of a pressure relief system for control purposes. Further, the relief valve scheme can result in wasted power when the pump is allowed to run out to the extreme right of the B.E.P., under "low load" conditions. The recommended high-speed pump control system is with the use of flow control. At first glance, this concept can be intimidating but is essentially synonymous with level and mass control schemes that are typical within process systems. Some processes do demand strict pressure control. When that is the case, a pressure controlled throttle or bypass source may be required.

4. WHAT ABOUT UNCONTROLLED FLOW OPERATION?

The effect of low flow operation on centrifugal pumps is commonly discussed, but rarely is the opposite end of the performance curve considered. Regardless of the hydraulic hardware, NPSHR generally increases with a pump's operation at greater than its rated flow. This topic has been mentioned within issue #6 and therefore will not be discussed further. Single volute and diffuser style pumps experience increased radial loads when applied at greater than

FIGURE 5. THEORETICAL HEAD RISE

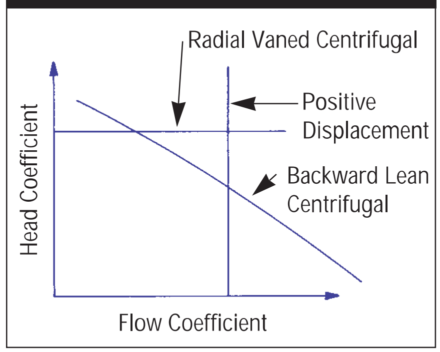


FIGURE 6. CENTRIFUGAL PUMP RADIAL LOAD CHARACTERISTICS

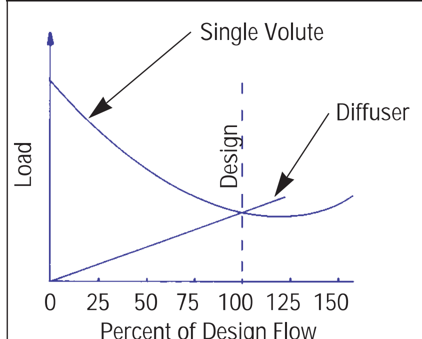


FIGURE 7. POINT EMISSION PUMP PERFORMANCE CHARACTERISTICS

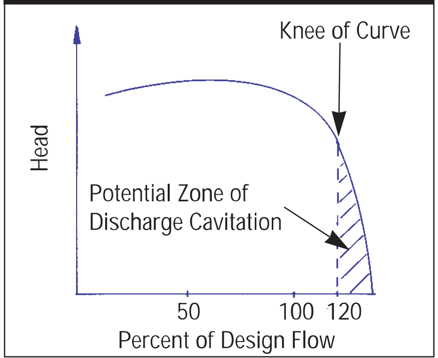
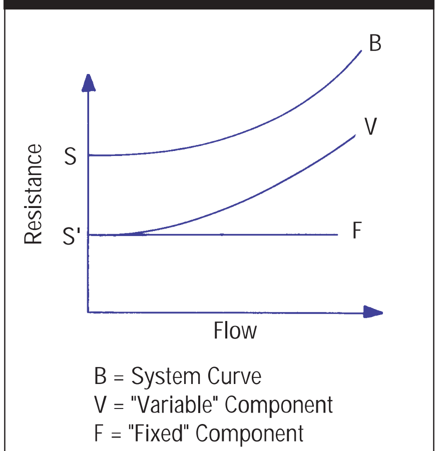


FIGURE 8



design flow rate (Figure 6). It is generally understood that sufficient bearing over capacity, and/or control limits, must be applied to the pump to account for these events. High-speed centrifugal pumps share these same basic design needs, but also must address a phenomenon often referred to as discharge cavitation.

Discharge cavitation occurs within single divergent conical (point emission) diffusers when the pump is operated at rates to the right of knee of the performance curve (Figure 7). Under such conditions, a low pressure zone forms on the trailing edge of the impeller's blades. Vapor bubbles are formed and subsequently collapse in a manner consistent with the standard definition of cavitation. The result is impeller blade pitting, increased pump vibration and the classic "rock pumping" noise associated with suction performance problems.

Such an occurrence is common with systems where the pump has been oversized on head, inadequate control is afforded to maintain operation within a maximum flow limit, or simply during a process startup where the system is being filled. A good rule of thumb is that this type of pump should be controlled to a maximum flow of 120% of the pump's rating. System requirements exceeding this value require a conversion to a larger diffuser throat to accommodate the actual process demand. Use of the design's conversion capabilities is always preferable to grossly oversizing the machine for future requirements.

3. A CONFLICT BETWEEN CURVES!

Some, but not all, high-speed pumps produce flat or "drooping" curves. This characteristic, which is common to low specific speed pumps, has elicited much discussion regarding their inherent ability to be controlled. Such pumps, however, have been successfully applied in tens of thousands of applications in spite of these concerns. Unfortunately, however, many times this has been accomplished at the expense of energy by oversizing the pump's rated head and then employing a discharge orifice to artificially steepen the curve as it runs back toward its minimum flow point.

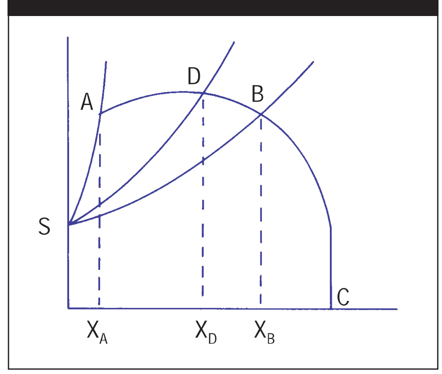
The application key is to understand the pump and the system curves. It is a common belief that drooping curves are difficult to control because the pump has two flow points associated with a single head (pressure) point. The result is a tendency for the pump to "hunt" between the two flows. What is often overlooked, however, is that the pump merely reacts to what the system presents it with (i.e., it operates at the exact point where the system and pump curves intersect).

A process system's characteristic resistance curve typically is made up of two components. The first is referred to as the "fixed" element and is associated with the system's static component, e.g., the operating pressure of a process tower. This element is considered to be constant with respect to the flow rate of the system. Conversely, the "variable" component may be simply thought of as the frictional element that is related to the pumped flow rate. These two factors are shown in Figure 8. When combined, they become the basic system curve.

Figure 9 shows a typical head versus capacity curve (ABC) with a drooping characteristic. Point B signifies the best efficiency point (B.E.P.), point A the cutoff flow and point C is the stonewall condition. Superimposed on this curve is the basic system curve (SB) which was derived from the previous discussion. Without supplemental pump control, the system will demand a flow rate equal to XB. The system head curve can be modified with changes to the piping system or by regulating the pressure drop across a control valve. The latter approach is a typical means of controlling centrifugal pumps and yields new system curves as indicated by (SD) and (SA). It is seen that the pump's head capability is equal at the XA and XB flow points yet successful pump operation is accomplished as a result of the modified system curve.

We would be remiss in not pointing out that this type of curve does have its shortcomings. First, pressure control normally is not practical due to the relatively small head rise that occurs between the rated and maximum head points, typically on the order of 5-10%. This fact favors the use of flow control methods. Second, systems that are comprised predominantly of the "fixed" component, i.e., those that exhibit little influence as a result of the system's demand flow,

FIGURE 9



may result in an undersized pump if an active appraisal of the pump/system interaction is not performed.

Regardless of the centrifugal pump type, a discrete intersection between the pump and system curves will always complement pump stability and controllability. Conscientious attention to the interaction between pumps and systems can tame both so that they work in harmony.

2. WHY WON'T THIS PUMP WORK ON WATER?

Pump manufacturers typically use water as a performance test medium for safety and convenience reasons. Many pumps, however, are sold for process liquids that vary between a 0.4 and 0.8 specific gravity. Factory tests often use one-half speed motors, or other speed changes to compensate for the increased power that results from water's density difference to the contract liquid. Field operation, however, commonly uses an initial run-in on water to flush and prove the system. Two typical repercussions of this action are an expected overload of the driver or an unexpected overload of a high-speed gear or bearings. The high-speed pump is particularly vulnerable to this off-design operation as a result of its common use in light hydrocarbon processing applications and as a result of small, custom matched, components which capitalize on the specific application's needs.

Throttling the pump may or may not meet the pump's needs because bearing loading may be violated in extreme cases. The moral of the story is to check with the pump vendor before proceeding with off-spec tests. The optimal approach is to advise him of your complete run-in conditions prior to placing a pump order to ensure that it will meet all of the intended uses.

1. WHERE IS THAT CONTROL VALVE?

Centrifugal pump designers expect throttle valves to be very near the pump discharge, while system designers prefer a location near the demand point. This issue becomes more than one of aesthetics when it involves high energy pumps.

A surge phenomenon may occur with pumps with either continuously rising or drooping head versus flow curve shape attributes. It is distinguished by fluctuations in head capacity at the pump's low flow conditions. The combination of high-speed technology with relatively low design flows introduces unique challenges to the pump designer and user.

Empirical testing shows that this pump design's low flow stability is directly influenced by the system within which it operates. The single point emission diffuser and a Barske impeller may be simplistically characterized as discharging flow each time a blade passes by the throat. It is reasonable to envision a void between the time in which one blade passes and the next arrives to distribute its supply. This interim period represents the opportunity for the discharge control valve's location to influence the pump's low flow stability.

The valve's interactive process may be visualized in the context of a simple spring mass system. The liquid in the system acts as the mass and all storage devices within the system, e.g., piping, vessels, etc., provide the spring medium. Excitation of this system may be initiated from a number of sources but often may be related to the blade passing frequency. The greater the energy that is accumulated within this system, i.e., the spring, the greater the propensity for it to disrupt the pump's stability. The momentary flow reversals cause surge circulation between the blades. This type of system is sometimes referred to as "soft" or "spongy" since it reinforces the amplitude of the theoretical spring force. The destructive energy of this situation is greatest with an increasing mass of liquid, when the throttle valve is remotely located, and with increasing pump power.

This phenomenon is minimized when the system can be described as "hard." This is accomplished by placing a control valve, or orifice, near the pump's discharge flange. In effect, this change reduces the liquid mass, thereby minimizing the amplitude of the spring's movement, and the flow oscillations. The valve/orifice also disrupts the frequency of the excitation force and further improves the pump's low flow stability.

The "hard" system should always be the goal since it discourages the formation of a potentially dangerous energy source that can damage piping and induce mechanical vibration into the high-speed pump. Good rules of thumb are that transmitted power levels of less than 25 horsepower are minimally affected by this phenomenon and the optimum control valve location is within 5 feet of the pump's discharge flange. Failure to address this situation can reduce a 200 horsepower pump's minimum recommended continuous flow rate from 40 to 65% of the B.E.P., based upon the valve's placement 25 feet, rather than 5 feet, from the pump's discharge flange. ■

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Donald P. Sloteman, Paul Cooper and Jules L. Dussord, Control of Backflow at the Inlets of Centrifugal Pumps and Inducers, presented at the First International Pump Users Symposium (1984).

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Dave Carr is a senior marketing specialist with Sundstrand Fluid Handling.

Pump Ratings Vital When Pressure's On

Though a relatively simple subject, pump ratings can generate much disagreement. Using objective evaluation procedures, however, we can shed light on the topic without generating heat and pressure.

All pumps are pressure rated. The rating is the maximum pressure a pump casing can safely contain, often termed maximum allowable working pressure or MAWP, at a given temperature (Figure 1). Temperature affects the rating because the strength and stiffness of the materials used for casings – mostly metals – vary with temperature. From that simple definition, the subject becomes more complicated as we add definitions for the pump's maximum discharge pressure, the means of correcting for different temperatures, and the question of what to do with the casings of pumps that develop high differential pressures.

The maximum allowable working pressure (MAWP) of a pump's casing is a function of its geometry, the material from which it is fabricated and the intended service temperature. Before delving into numbers, let's address a fundamental aspect of casing geometry, namely the casing joint.

THE CASE FOR CASING JOINTS

Pump casings must have a joint — either at right angles to the shaft axis (radially split) or parallel to it (axially split) — to allow the pump to be assembled and dismantled. Most pump casings are radially split: in small overhung pumps, either horizontal or vertical, because of their inherently lower cost; in medium size and large vertical pumps for both lower cost and ease of maintenance; and in high pressure pumps, either horizontal (Photo 1) or vertical, because it's the more cost effective solution. In between these extremes, axially split casings are preferred for horizontal single stage double suction and multistage pumps (Photo 2) for lower cost and ease of maintenance. They are, however, limited in the pressure rating they can economically achieve.

API-610, 7th Edition[1], recommends using radially split casings for hydrocarbon service when the maximum discharge pressure is above 1,000 psig (70 bar), the pumping temperature above 450°F (235°C), or the liquid specific gravity below 0.7. These conservative limits reflect the difficulty various refiners have had maintaining pressure tightness in axially split casings. There are, however, many examples of axially split casings being used successfully in hydrocarbon service beyond these limits, and the forthcoming 8th Edition of API-610 will recognize this by raising the recommended pressure limit to 1,450 psig (100 bar). In water injection and boiler feed applications, axially split casings are regularly used to pressures of 2,500-2,750 psig (175-190 bar). Higher pressures are possible, but the cost of the casing can become prohibitive, and maintenance of the split joint gasket a major concern.

The radially split casing in Photo 1 is one piece with a cover or head at the outboard end. This design has one high pressure seal, and the pump can be dismantled without breaking its suction or discharge connections or moving its driver. There is an alternative form of radially split casing, known variously as "ring section" or "segmental ring," composed of many pieces clamped together with tie bolts (Photo 3). This design achieves low cost at the expense of maintenance. It has many casing joints, does not comply with API-610 and was wisely dropped from use in the US in the mid 1930s for all but small industrial

By J. T. ("Terry") McGuire

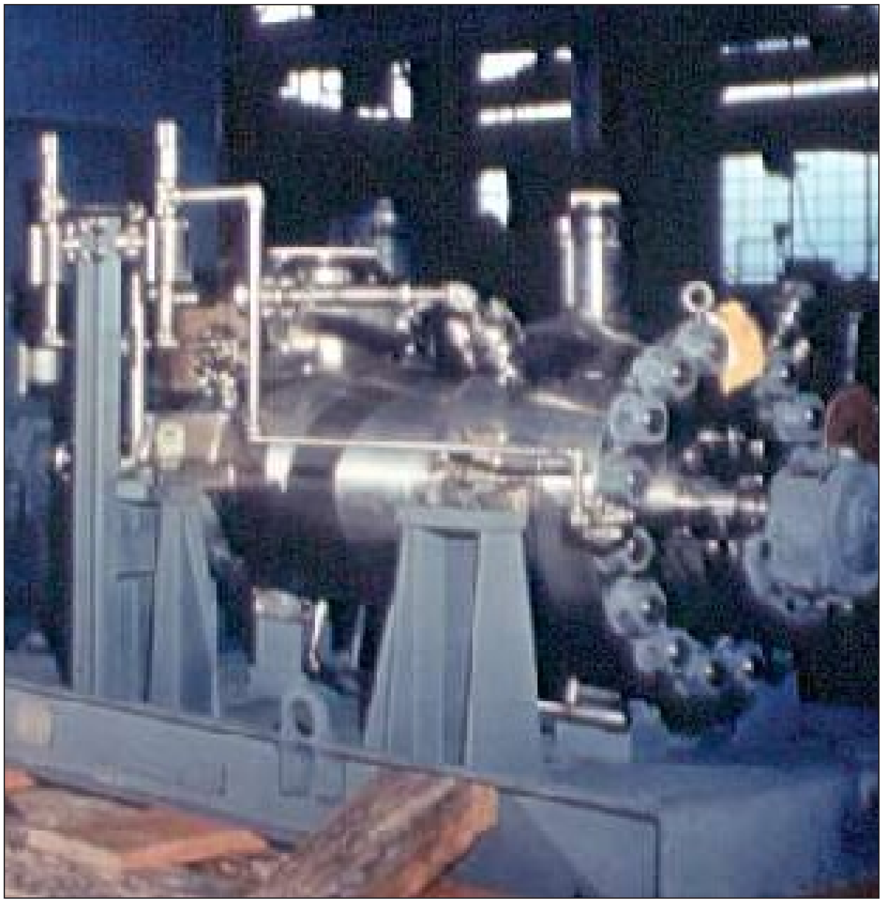


Photo 1: High pressure radially split casing

Photo 2: Axially split multistage pump casing

Photo 3: Segmental ring casing

boiler feed pumps.

DETERMINING MAWP

For a given casing geometry, a pump's MAWP is determined by allowable stress unless strain (deflection) at critical sealing surfaces dictates a lower stress. The allowable stress may be that from ASME Section VIII, Division 1[2], as required by API-610, or some other similar limit. Designs using the ASME stress limits also include a casting integrity factor, which is 0.8 unless volumetric NDE of the castings allows a higher factor. Note that the allowable stress from ASME Section VIII, Division 1, includes a large design or "ignorance" factor to account for design using simple means of estimating the stress. When more sophisticated means of estimating the stress are employed, such as finite element analysis (FEA), using a higher allowable stress is justified because the local stress values are now known with quite high accuracy. In such cases, the allowable stress from ASME Section VIII, Division 2, can be used provided that the FEA model has been verified for accuracy and that appropriate material quality is used.

CALCULATING MAXIMUM ALLOWABLE DISCHARGE PRESSURE

In operation, the maximum discharge pressure developed by a centrifugal pump is equal to the sum of the maximum suction pressure it can be exposed to plus the maximum differential pressure it can develop. In a simple world, that definition would be sufficient, but the world's not so simple. Pumps are purchased with various reserves, margins and tolerances in head and rotative speed that complicate the definition of maximum discharge pressure. Following the requirements of API-610, 7th Edition, produces two definitions, one for fixed speed pumps, the other for variable speed. With Figure 2 as a reference, the definitions are:

Fixed speed pump

$$P_{d,\,max} = P_{s,\,max} + 1.05(\Delta P_{max})f_H \qquad (1)$$

Variable speed pump

$$P_{d,\,max} = P_{s,\,max} + (\Delta P_{max})f_H(f_N)^2 \qquad (2)$$

Where:

$P_{d,\,max}$ is the maximum discharge pressure

$P_{s,\,max}$ is the maximum suction pressure

ΔP_{max} is the maximum differential pressure at rated specific gravity

f_H is a factor to account for the allowable tolerance on shut-off head

f_N is a factor to account for the allowable overspeed to trip.

API-610's requirement for variable speed pumps covers the head reserve in the allowable positive head tolerance, and it assumes the pump will be operated as if fixed speed, i.e. its flow will be controlled by throttling. Boiler feed pumps for central stations are often variable speed drive to avoid the losses associated with control by throttling. As such, their head always corresponds to the system resistance at any given capacity, and it's therefore appropriate to use a different definition for the maximum discharge pressure. Referring to Figure 3, it is:

$$P_{d,\,max} = P_{s,\,max} + P_{CMR} \qquad (3)$$

Where:

P_{CMR} is the differential pressure at the pump's continuous maximum rating (CMR).

With this definition, the 10-15% higher discharge pressure that could be developed in the event the driver went to overspeed while the pump was blocked in is classified as a momentary excursion into the margin provided by the casing's hydrotest pressure.

Positive displacement pumps, unlike centrifugal pumps, will develop pressure equal to the resistance they encounter, up to the mechanical limit of the pump or its drive. This is obviously an extremely dangerous possibility, and so the cardinal rule in the application of positive displacement pumps is the provision of a full capacity relief valve at their discharge, upstream of any possible obstruction. The accumulation pressure of the relief valve, the additional pressure drop across the valve to discharge its rated capacity, should be no more than say 20%. With this provision, the pump's maximum discharge pressure is (Figure 4):

$$P_{d,\,max} = P_{set}f_A \qquad (4)$$

Where:

P_{set} is the relief valve set pressure

f_A is a factor to account for the accumulation pressure.

PUMP RATING VERSUS REQUIRED

For a pump's pressure rating to be acceptable, the MAWP of its casing, at the intended service temperature (Figure 1), must be at least equal to the pump's maximum discharge pressure, as calculated by the applicable equation above. Depending upon the design status of the pump being considered, there are two approaches to achieving this. For existing designs, which is the usual case, the MAWP of the pump's casing at the design temperature, generally 100°F (40°C), is corrected to that for the intended service temperature, T, by the equation:

$$MWAP_T = MWAP_D(\sigma_T / \sigma_D) \qquad (5)$$

Where:

σ_T is the allowable stress at the intended service temperature

σ_D is the allowable stress at the design temperature

Unless specifically stated otherwise in the engineering specification, the intended service temperature for casings is taken as the pump's normal operating temperature. The rationale for this is that the maximum temperature generally represents a possible short term transient condition.

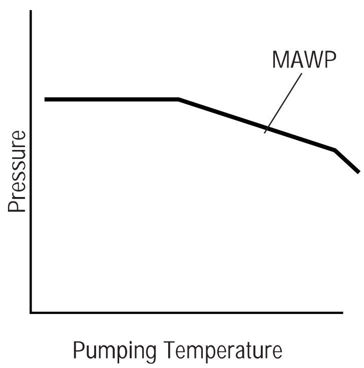
When the casing is being designed specifically for the application, a common practice with fabricated casings, the pump's maximum discharge pressure is used to calculate a minimum design pressure, MDP, by the equation:

$$MDP = P_{d,\,max}(\sigma_D / \sigma_T) \qquad (6)$$

The MDP is generally rounded up to the nearest common increment, 25 psi being the ASME practice, or to a minimum pressure required by the engineering specification or a connected flange.

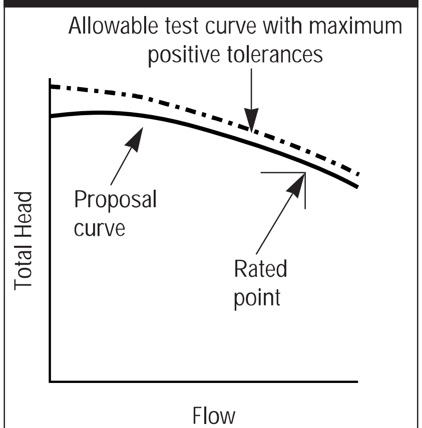
MINIMUM CASING DESIGN PRESSURE

FIGURE 1



Variation of MAWP with pumping temperature

FIGURE 2



Allowable actual pump head curve, API-610, 7th Edition

FOR SHOP TESTING

When designing a pump to handle a liquid of low specific gravity, it may be necessary to raise the MDP calculated from equation (6) to accommodate the pressure that will be developed during shop testing with cold water. The equation for the maximum discharge pressure on test is:

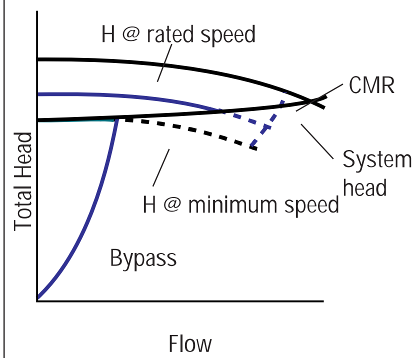
$$P_{d,\ test} = P_{s,\ test} + (H_{max}/2.31)f_H \qquad (7)$$

Where:

$P_{s,\ test}$ is the maximum suction pressure during the test

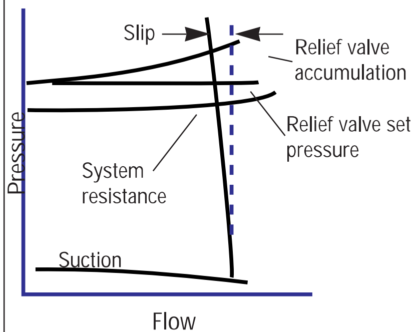
H_{max} is the highest head the pump will develop during the test, in ft.

FIGURE 3



Variable speed boiler feed pump

FIGURE 4



Pressures in positive displacement pump applications

The maximum pump head developed during the shop test is usually at shutoff, except in the case of a pump with a drooping head characteristic, or high energy multistage pumps, which are tested only down to their minimum continuous flow. The head tolerance factor, f_H, is that for shutoff.

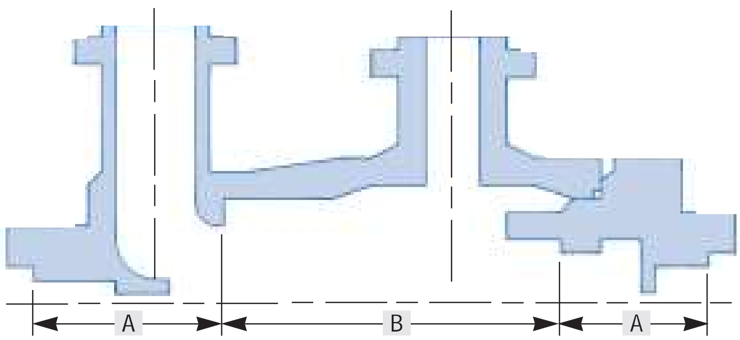
CONTROVERSY OVER DUAL PRESSURE CASINGS

Multistage centrifugal pumps develop large pressure differentials. This means various regions of their casing normally are subjected to distinctly different pressures. Once the pump exceeds a certain size and pressure rating (3-inch discharge and ANSI 1,500 # flanges are a good starting point) it becomes more economical to design the casing for two pressures, a convenient low pressure for the regions subjected to suction pressure, maximum discharge pressure for the remainder of the casing (Figure 5). This practice is normal in the utility industry and is recognized by API-610, the standard for the process industry. API-610 also recognizes this is a controversial topic within the process industry, so it includes the option of specifying that the entire casing be designed for maximum discharge pressure.

Designing the normal low pressure regions of a barrel pump casing for discharge pressure has two negative effects that need to be taken into account. The first is that the shaft seal design is often compromised. Hot charge pumps, for example, are best equipped with metal bellows type shaft seals. These cannot withstand static pressures of more than 350-400 psig (24.0-27.5 bar). Therefore, pusher type seals have to be used. Pusher seals rely on some form of cooling to preserve their elastomer dynamic gasket so the design has become more complicated and potentially less reliable. The second drawback is that the flange of cartridge mounted seals becomes so large and heavy that installing and removing it presents a serious handling problem.

The controversy over dual pressure casings appears to have its origin in operating practices and the static pressure tightness of mechanical seals. In the utility industry and about half of the process industry, standard isolation practice is to block in the discharge, then open a drain before blocking in the suction. When this sequence is followed, the entire casing and suction piping back to the

FIGURE 5



A = suction pressure + 75 psi
B = maximum discharge pressure

Regions of barrel pump casing subjected to different pressures.

suction block valve cannot be accidentally subjected to discharge pressure by a small leak past the discharge valve. The alternative sequence, blocking in the suction before opening a drain, when carried out on pumps with mechanical seals, does risk subjecting the entire pump casing and suction piping downstream of the suction block valve to discharge pressure. There have been instances where doing this has ruptured the suction piping or the pump casing. It does seem that a judiciously placed burst disc or relief valve could practically eliminate the risk of misadventure without compromising the design of the shaft seals.

HYDROSTATIC TEST PRESSURE

A casing, or the various regions of a dual pressure casing, is hydrostatically tested at

$$P_{hydro} = 1.5\ P_{design} \qquad (8)$$

P_{design} is either $MAWP_D$ for pre-engineered casings (see Equation 5), or the greater of MDP from Equations 6 or 7 for engineered casings.

REVIEWING THE OPTIONS

Casings with the highest pressure ratings are radially split. There are actual examples of these designed for 14,000 psig. Axially split casings are preferred for large axis single stage double suction pumps and horizontal multistage pumps because of ease of maintenance. They cannot always be used, however, because their pressure rating is currently limited to 1,000 psig by API-610, 7th Edition, for hydrocarbon service. This is due to be increased to 1,450 psig effective with the 8th Edition of API-610, although there are examples of their operation at 2,500 to 2,700 psig for boiler feed and water injection services.

Higher pressures are possible, but the cost of the casing can become prohibitive, and the maintenance of the split joint gasket a major concern. Single stage, single suction pumps are generally radially split – for economics in the smaller sizes, both horizontal and vertical, and for ease of maintenance in the very large vertical axis designs.

When a methodical approach is taken, the issue of pump pressure ratings is not too difficult. This evaluation and selection process can be further simplified by following the evaluation procedure outlined in the box accompanying this article. ■

REFERENCES:

[1] API-610, 7th Edition, Standard for Centrifugal Pumps for General Refinery Service, American Petroleum Institute, Washington, DC, 1989.

[2] ASME Boiler and Pressure Vessel Code, Section VIII, Divisions 1

Evaluation Procedure

Answering the following questions in sequence will help avoid errors:

Is the preferred form of casing joint suitable for the intended pressure, temperature and liquid specific gravity?

Is the maximum discharge pressure, calculated as required by the applicable industry standard, less than the casing's MAWP at the intended service temperature?

Is there a minimum industry or code pressure rating for this class of pump?

Do the casing flanges or nozzles have a pressure rating at least equal to that of the casing region to which they are connected?

Is the minimum design pressure of the casing determined by the pressure developed during shop testing with water?

Does the purchaser's specification require all regions of multistage pump casings to have a MAWP equal to or greater than the maximum discharge pressure?

Communicating Your Pump Needs

Purchasing and installing a new pump requires a team effort between customer and supplier.

By John Bertucci

Many industry pump problems are not caused by improper operation or faulty maintenance, although these are the primary focus of most reliability improvement efforts. In reality, most problems are traceable to improper initial application or changed operating conditions. The pump is just not right for the job. Or it once was, but is no more. Misapplication can be avoided, however, by observing three important principles: communication, communication and communication.

In the past, and sometimes even today, pump users assume an adversarial relationship with their suppliers. Mutual suspicion is the order of the day, with each side trying to "win" or gain an advantage over the other. On the other hand, wise pump users are treating suppliers as valuable resources. They seek win-win outcomes where everyone benefits. Pump manufacturers, too, have found that making today's sale is not as important as building a long term relationship with customers. Out of this industry-wide shift in attitude has come a new opportunity for suppliers and users to work together to put the best pump in a given application.

With all of today's corporate downsizing and re-engineering, pump users cannot afford to ignore the wealth of help available from pump suppliers. After all, suppliers can draw from a broad spectrum of industry experience to help a user solve a particular problem. This information exchange benefits users by taking solutions developed in one industry segment and introducing it to other sectors. Pump suppliers benefit by getting valuable feedback on their designs so they can improve their products and broaden their applications. And as this article explains, communication is the key to making it happen.

There are two primary areas of pump user/supplier interaction. The first is when the user initially purchases a new pump. The second is when a user attempts to improve the performance of an existing pump. These two activities are different yet have much in common. The most important of these is the need for clear, open and honest communication between users and suppliers.

NEW PUMP PURCHASES

Purchasing a new pump requires a team effort between customer and supplier. Much effort has been expended over the years to develop specifications and industry standards such as API and ANSI. These standards form the foundation of a new pump purchase. Following are some additional ideas that will help you select and purchase the best pump for a given application.

PROCESS DATA

Accurate process data is needed to achieve a successful pump application. Without good information, the pump supplier is already fighting with one hand tied behind his back, and the battle hasn't even started. A whole book could be written on the subject of properly sizing pumps, but the calculation and engineering aspects are beyond the scope of this article.

Good communication is the key to getting accurate process data onto the pump data sheet. Typically, the flow rate is established by the process for which the pump is being selected. A calculation then determines the discharge pressure required to move this amount of flow through the piping system. This process is fairly straightforward, but pitfalls exist even at this elementary stage. Many process design engineers do not understand that a typical centrifugal pump has an operating range of only 40–120% of Best Efficiency Point (BEP) flow rate. This may lead them to set the rated flow rate of the pump higher than required, possibly to allow for future expansion of the unit. Unfortunately, this future expansion either is many years away or never happens. As a consequence, the process unit is left with a pump that is operating at, or at much less than, the lower limit of its operating range. This low flow operation is the root cause of many pump reliability problems.

Good communication during the sizing process can help avoid this and other sizing errors. The plant's rotating machinery group should be brought in early to work with the process designers. Working together, they can explore options other than pump oversizing. The rotating equipment group also can bring pump manufacturer expertise to the sizing exercise. Pump manufacturers frequently offer other choices such as an upgradable pump, variable speed drive or other means to improve operational flexibility.

SITE/INSTALLATION DATA

It is important to communicate the location and type of installation both internally and to the pump sup-

INFORMATION ITEMS	POTENTIAL USES
Suction Specific Speed	1. Determining Stable Flow Range. 2. Re-rates, especially to lower rates.
Number of Impeller Vanes	Vibration Analysis, especially vibrations caused by low flow.
Seal Flush Flow Rate Calculations and Stuffing Box Pressure	1. Determine the cause of seal failures. 2. Change seal design.
Stable Flow Range	1. Determine the minimum or maximum allowable flow. 2. Determine if pump is in a low or high flow condition.
Thermal Growth	1. Determine if pump is distorting due to thermal forces. 2. Aid in getting good alignment.

Table 1

INFORMATION ITEMS	POTENTIAL USES
Wear Ring Clearances	1. Checking existing clearances to determine if wear ring replacement is necessary. 2. Determine if performance problem was due to excessive wear ring clearance.
Bearing Number (Rolling Element Bearings)	1. Vibration Analysis - determine ball pass frequencies. 2. Analyze potential bearing upgrades.
Bearing Clearance (Hydrodynamic Bearings)	1. Analyze vibration problems. 2. Set alarm limits on probe type vibration monitors. 3. Determine bearing replacement needs.
Stuffing Box Dimensions	Mechanical seal upgrades and changes.
Materials of Construction	1. Determine repair methods. 2. Emergency fabrication of replacement parts.

Table 2

DOCUMENT	REVIEW BEFORE PURCHASE	COPY AFTER PURCHASE
Installation, Operation & Maintenance Manual	NO	YES
Cross Sectional Drawing with Parts Identified	YES	YES
Dimensional Outline Drawing	YES	YES
Spare Parts List	YES	YES
Performance Curve	YES	YES
Curve Family	YES	NO
Completed Data Sheet	YES	YES
Test Data (if applicable)	NO	YES
Driver Data	YES	YES

Table 3

plier. Teamwork in this area is especially vital in vertical pit pump installations such as cooling tower pumps. Errors in designing the pit and suction approach to the pumps are easy to correct when the pit consists of lines on paper. But they are extremely expensive to correct once concrete is poured.

USER RESTRICTIONS AND PREFERENCES

Every plant has certain things such as seal type, maximum suction specific speed and bearing type that they like and dislike in their pumps. These preferences and restrictions are usually based on years of experience in solving that plant's particular pump problems. They must be explained well internally so that the pump data sheet accurately conveys them to the suppliers. Unfortunately, these preferences usually are in the heads of the plant rotating equipment group while a lot of the pump selection is done by a project engineering group or outside engineering design firm. One way to assure that these preferences are given due consideration is to require review of the pump data sheet by the rotating equipment group. Another good way is to place a member of the rotating equipment group on the project design team where his/her knowledge can be tapped by the project design engineers. A third way of accomplishing this is to develop a local specification that contains the various preferences and restrictions from the rotating equipment group.

COMMUNICATION WITH THE PUMP SUPPLIER

Communication with the pump supplier should be a two-way street. The information should flow freely back and forth between the user and supplier. This should happen even in situations where competitive bidding will determine the ultimate supplier of a new pump. The only difference in the competitive bid situation is that all of the data sent from the user to the supplier should go to all suppliers equally, with no favoritism shown. Of course, all commercial information (prices, delivery details, etc.) should be kept confidential.

INFORMATION OBTAINED FROM PUMP SUPPLIER

Much information can be obtained from the pump supplier that will aid in future pump maintenance and troubleshooting. Some of the less obvious or often forgotten items are listed here.

TROUBLESHOOTING AND MAINTENANCE

Table 1 shows information that can be obtained from the pump's manufacturer for future use in troubleshooting. This is information that the manufacturer generally has and will provide upon request. Table 2 lists pump supplier information that can be very useful in future pump maintenance.

OTHER DOCUMENTATION

Table 3 shows what can be considered a good minimum requirement for a documentation package.

PERFORMANCE IMPROVEMENTS

Pump performance improvements generally come in two flavors: upgrade of an existing pump that is performing well, and correction of a problem pump. In either case, the pump's original supplier can be an extremely valuable resource.

Be Open to New Ideas. When your friendly pump supplier calls and asks for an appointment, make time for him (even if it's not lunch time). Suppliers are constantly coming up with new and better ways to do things. Examples include A and B gap modifications, new overlay materials for severe service and new impeller designs. You can't consider these and other potential solutions to your problems if you don't take time to learn about them from the experts.

Invite Your Supplier to Participate. The manufacturer's local representative should be a regular part of your maintenance resources. You may have a problem that your competition solved years ago. You'll never find this out from the competition, but the pump supplier may already know the solution. However, he can't help if he doesn't know about your problem. In addition, the pump supplier will be intimately familiar with the design issues of a specific pump.

Visit Supplier's Repair Facility or Factory. A visit to the supplier's repair shop can give you much valuable information: It enables you to evaluate the shop's capabilities, in case you ever need them.

It allows you to develop face-to-face contacts with the people who repair these pumps every day. A good relationship with these folks may help you in the future when you need information. Also, the information you get there is not filtered through a salesman.

A visit to the factory can be even better. It allows you to meet with the people who know the most about the design of your pump. Factory contacts can get you information in a hurry. They can also help expedite shipment of desperately needed parts.

Go to Conferences. There are many good conferences and symposia that have pumps as a primary theme. One highly recommended conference is the International Pump Users Symposium held in early March each year in Houston. Sponsored by Texas A&M University's Turbomachinery Laboratory, it includes short courses, tutorials, discussion groups and hundreds of pump manufacturer and related industry displays. These conferences are like having many pump factories under one roof. You can go from booth to booth and meet representatives of many companies. The discussion groups are also a good way to interface with other users as well as suppliers.

Share Knowledge/Experience with the Pump Supplier. Users accrue valuable practical experience with pumps over their years of operation. This experience should be shared with the pump supplier. It will help the supplier improve his products and eventually benefit all users. Also, it is only fair that knowledge go both ways in any relationship.

In summary, your pump supplier will never know or understand your needs if you don't take the time and effort to develop a mutually beneficial relationship. Most pump suppliers place a high priority on meeting their customers' needs, but they need all the help that they can get. ■

John Bertucci is a Mechanical Equipment Engineer with the Norco Manufacturing complex of Shell Oil Products Co.

Impellers and Volutes: Power with Control

A review of centrifugal pump impeller and volute design applications can help you optimize both power and control.

By Robert R. Ross

All centrifugal pumps have two major components: the rotating element and the pump case. Together they establish how much head will be generated, best efficiency point (BEP) capacity, the slope of the head capacity curve and net positive suction head required (NPSHR), Figure 1. The rotating element consists of a shaft and one or more impellers whose function is to convert mechanical energy into high velocity kinetic energy. The pump case directs liquid to the impeller from the suction nozzle, collects the liquid discharging from the impeller, and then converts the kinetic energy into pressure by controlled deceleration in the diffusion chamber immediately following the volute throat.

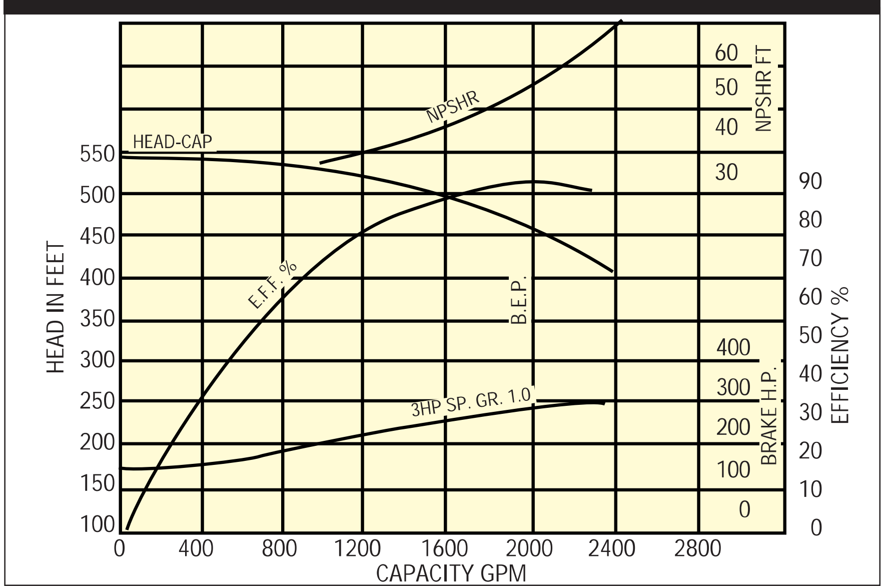
TYPES OF VOLUTES

Single volute pumps have only one volute throat and one diffusion chamber, and because of their simple casting geometry, they can be produced at lower cost than the more complex double volute designs. Figure 2 shows a single stage version with the diffusion leading directly to pump discharge. Figure 3 is a multistage version in which the liquid, following diffusion, is directed through the crossover to the next stage impeller. Pressure distribution around the impeller is non-uniform and produces a radial load on the impeller which, depending on the developed head, may deflect the pump shaft and cause wear at impeller wear rings and seal faces.

Single volutes are used routinely in slurry and sewage pumps to minimize plugging at the throat, and on low head pumps where radial loads are nominal. They are also used on low specific speed pumps where the throat area and hydraulic passages are too small to cast as a double volute.

Double volute casings were introduced to minimize the radial thrust problems of single volute pumps. They are actually two single volute designs 180° apart with a total throat area equivalent to a comparable single volute design. The non-uniform pressure distributions are opposed, thereby greatly reducing radial loads (Figure 4). Double volute pumps are the preferred choice on higher head pumps.

FIGURE 1



Pump performance curve

TYPES OF IMPELLERS

Pumps can be built with a single impeller or, in the case of high pressure applications, with two or more impellers. They will be either single entry or double entry type, more commonly known as single or double suction (Figure 5). Because double suction impellers have a greater total eye area, velocity of the liquid entering the eye is reduced, producing a lower NPSHR. The shape of the impeller depends on specific speed (N_s), which should only be calculated at BEP with maximum diameter impeller. In U.S. units this is:

$$N_s = \frac{RPM \times GPM^{.5}}{(\text{Head/Stage feet})^{.75}}$$

Figure 6 shows the change in shape from the low N_s radial flow impellers to the high N_s axial flow types.

Whereas the suction geometry is selected to reduce inlet losses for low

FIGURE 2

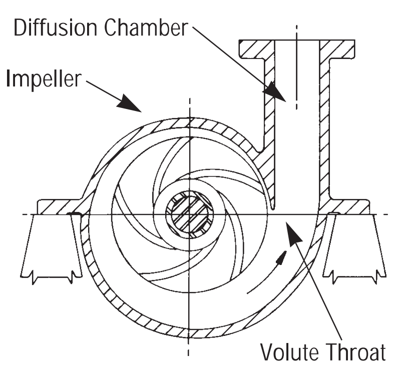


FIGURE 3

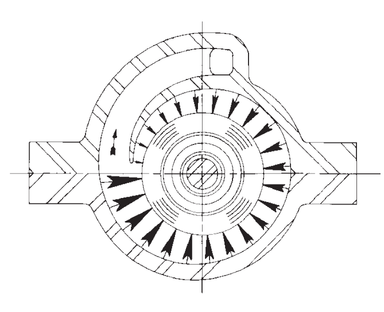


FIGURE 4

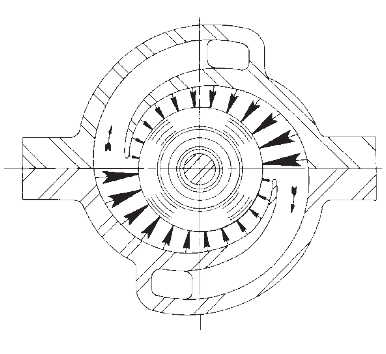
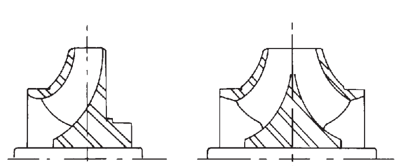


FIGURE 5



Centrifugal impellers

NPSHR, the discharge geometry is selected to satisfy the required head, slope of the head capacity curve and BEP capacity.

HEAD CAPACITY CURVE SLOPE ANALYSIS

The desired slope is determined during system analysis with the percentage rise to shutoff (zero flow) from rated head often determined by system limits. When the pump is started against a closed discharge valve, the pressure up to the valve will be the pump differential head at shutoff plus suction pressure. Because this should not exceed the safe working pressure of the system, the rise to shutoff can be critical and is controlled by impeller discharge geometry.

An evaluation of the system head curve is needed to determine if the pump should have a flat or a steep curve. This is a graphical plot of the total static head and friction losses for various flow rates. For any desired flow rate, the head to be generated by the pump is at the intersection of the head capacity curve and system head curve. In a simple pump application in which system head is due entirely to friction loss, a flat head capacity curve with 10 to 20% rise to shutoff from the head at rated capacity would satisfy the application and minimize shutoff pressure on the system (Figure 7). Where system head consists of both friction and static head — that is, where there is a change in elevation — a flat curve also would be appropriate if little or no change was anticipated in the static head (Figure 8).

Figure 9 represents an application in which changes in static head caused by changes in the suction tank level result in a range of system head curves. For illustration, two pump head capacity curves have been superimposed — one flat, the other steep. The advantages and disadvantages of both must be considered in deciding which pump curve is more suitable for the application.

Advantages of the flat curve are low shutoff pressure and relatively small differences in operating pressure as the system head moves. The disadvantage is a larger variation in flow rates.

Advantages of the steep curve are smaller variations in flow rates and additional head margin to accommodate potential increases in static head. Disadvantages are high shutoff pressure and larger variations in head.

IMPELLER DISCHARGE GEOMETRY

Various methods are used to modify the impeller so the percentage rise to shutoff will match the slope resulting from system head analysis. Among these are changes in the number of vanes, changes in the vane discharge angle and changes in the exit width b_2. The curve can be changed with variations in vane number and discharge angle, while BEP capacity is held constant by changing b_2. Another method is to use a constant discharge angle and b_2 with changes in the number of vanes only.

FIGURE 6

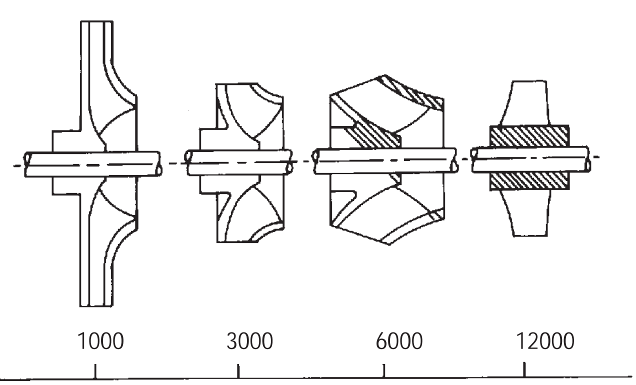


FIGURE 7

SYSTEM HEAD CURVE, ALL FRICTION

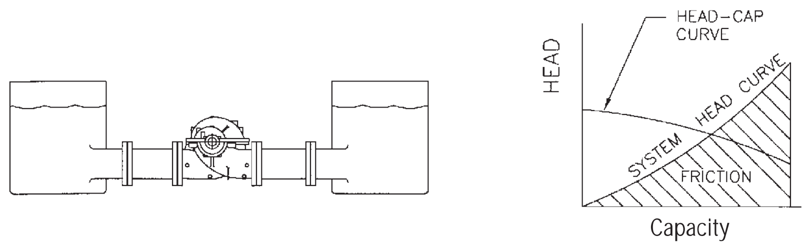


FIGURE 8

SYSTEM HEAD CURVE, STATIC HEAD PLUS FRICTION

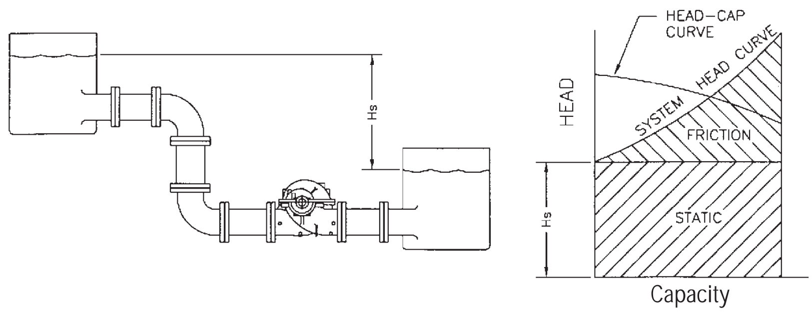
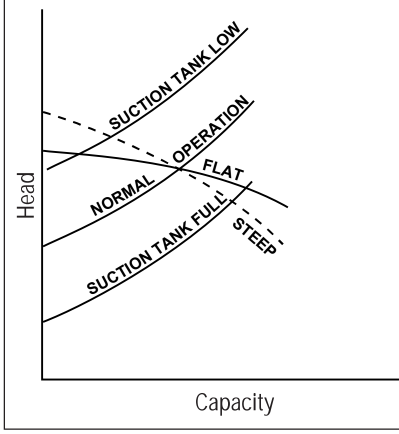


FIGURE 9

EVALUATING PUMP CURVES AGAINST VARIABLE SYSTEM HEAD CURVES



BEP CAPACITY

The impeller discharge geometry and volute throat area establish BEP capacity. By adjusting the ratio of liquid velocity leaving the impeller to liquid velocity entering the volute throat, BEP can be increased or decreased. Modifications of this type are used to upsize or downsize existing pumps hydraulically, moving BEP to the normal operating capacity for optimum efficiency and generating significant savings in the cost of power. ■

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Editor's Note: Some text and figures for this article have been excerpted with publisher's permission from Lobanoff, V. S. and Ross, R.R., Centrifugal Pumps: Design and Application, 2nd Edition, Gulf Publishing Company, Houston, TX 1992.

Fully Lined Slurry Pump for FCCU Bottoms Use

By replacing conventional API pumps with fully lined slurry pumps in FCCU bottoms applications, refineries are improving production and profitability.

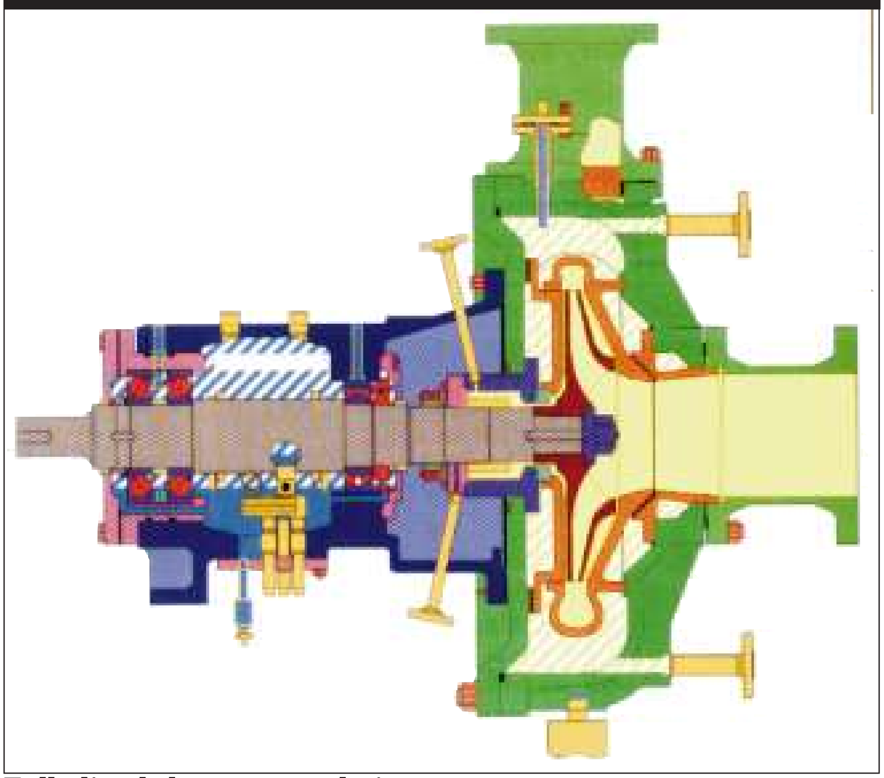
By Dan Clark and Julio R. Cayro

The Fluid Catalytic Cracking Unit (FCCU) is one of several processes critical to a refinery's productivity. Its long term, safe operation translates into increased production and profitability. Yet prior to introducing fully lined pump technology for use in FCC (Fluid Catalytic Cracking) main column bottoms applications, refineries replaced or repaired conventional double volute API process pumps several times a year. This caused serious safety risks and a process shutdown frequency unacceptable in today's production environment.

The fully lined slurry pump has thus emerged as the pump technology of choice for providing 3 – 5 years of maintenance-free operation for FCCU refinery bottoms applications. This particular application involves pumping a highly erosive high temperature (350 – 800°F) slurry at flows to 12,000 gpm, pressures to 600 psig and heads from 90 – 900 feet. Catalysts used in FCC processes are also extremely erosive, and they are applied in varying concentrations depending on the process unit configuration and/or upset operating conditions.

The fully lined slurry pump design (Figure 1) is an engineered approach to providing long term, reliable pump performance in this severe-duty application. To understand how this new pump technology meets such demanding requirements, we will take a closer look at FCC processes and the design of fully lined slurry pumps.

FIGURE 1



Fully lined slurry pump design

FCCU PROCESSES

Conventional Fluid Catalytic Cracking. A typical fluid catalytic cracking unit (Figure 2) consists of a reactor, catalyst regenerator and fractionator column. This process converts straight run heavy gas oil from the crude distilling unit, and flasher tops from the vacuum flasher unit, into high octane gasoline, light fuel oils and olefin-rich light gases.

In the vertical reactor vessel, vaporized oil contacts fluidized catalyst particles, causing a reaction that yields lighter hydrocarbon and coke. During the reaction, carbon (coke) is deposited on the catalyst, rendering it inactive. This inactive catalyst is recirculated from the cyclones at the top of the reactor back to the regenerator where the coke is combusted, rejuvenating the catalyst. Vaporized cracked products flow through the cyclones at the top of the reactor into the vapor line that feeds the bottom of the main fractionator column. The cyclones operate at less than 100% efficiency so that some coke and catalyst particles continuously reach the fractionator.

In this type of FCC process, the main column bottoms pumps must pump the bottom oil at a high rate through the heat exchanger and over a vapor contact section within the fractionator tower. This desuperheats and scrubs the fine particles of catalyst from the reactor vapors entering the fractionator without causing oil coking. A small concentration of alumina-based catalyst particles that have been scrubbed out of the vapors is continuously circulated with the main bottoms product, causing erosion of the pump internals.

New FCC Processes. Unlike conventional FCC processes that recycle part of the main column bottoms directly back to the reactor vessel, newer FCC processes such as the UOP process continually circulate the main column bottoms in a closed loop through heat exchangers and back to the fractionator tower. With the closed loop design, catalyst concentrations run as high as 2% by weight, compared to conventional systems where the catalyst concentration is 0.25 – 0.5%.

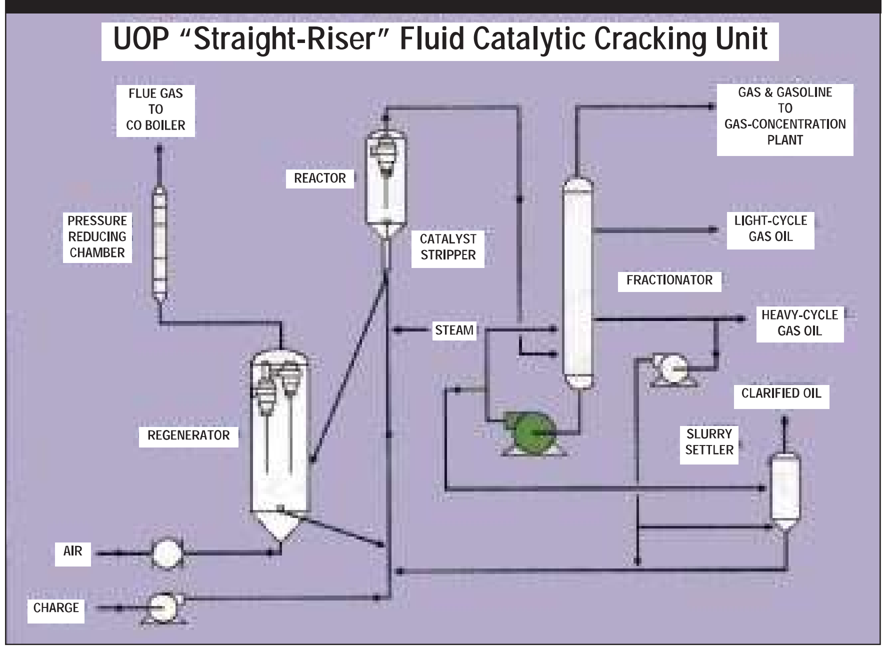
With conventional FCC processes, catalyst levels reached 1 – 1.5% only during prolonged upset operating conditions, destroying conventional API process pump internals in a matter of days or weeks (Photo 1). Today, a 2% concentration of catalyst continuously circulating with the main column bottoms is considered a normal operating condition.

PHOTO 1

PHOTO COURTESY OF LAWRENCE PUMPS

Conventional API process pump internals can be destroyed in a matter of days during prolonged upset operating conditions.

FIGURE 2



FULLY LINED SLURRY PUMP TECHNOLOGY

Unlike conventional API process pumps designed for maximum efficiency with clean liquids, fully lined slurry pumps are engineered to provide maximum reliability when handling abrasive hydrocarbon slurries. All pumps are selected to operate in the optimum hydraulic fit (80 – 110% BEP) for specified flow, head and erosive characteristics of the slurry. Design considerations such as using larger diameter impellers at lower speeds (870 – 1,770 rpm), selecting proper construction materials and maximizing appropriate mechanical seal designs all help optimize life cycle cost.

Liners. The fully lined slurry pump uses replaceable, abrasion-resistant 28% chrome iron liners to protect the pressure casing, providing 5 – 6 times the life of diffusion coated CA6nm components (Figure 3).

Abrasion-resistant liners are machined, toleranced components that form the hydraulic wet end of the pump. They are easily replaced individually or as a set. Flow stream turbulence is reduced by using a 125 rms machine finish on the liner surfaces.

Rotating Elements. Like the liners, impellers are constructed of abrasion resistant 28% chrome iron. These high efficiency enclosed impellers (Photo 2) use front and back repelling vanes to reduce slurry recirculation. Repelling vanes eliminate the clean oil flush required by conventional wear ring impeller designs. Large open passages reduce frictional losses and allow maximum solids-passing capability to avoid clogging due to coke buildup in the fractionator column.

Several impeller mounting configurations are available depending on horsepower and catalyst slurry properties. Impellers can be fastened to the shaft using a tapered polygon or straight bore. Each is locked in place with an enclosed impeller nut and then secured with a locking bolt.

As shown in Figure 1, fully lined slurry pumps use large diameter shafts that meet the stiffness criteria set by API-610. Each shaft is engineered to provide space for single, double or tandem seal arrangements and a conservative L3/D4 deflection index. Stiff shaft designs limit deflection, maximize mechanical seal and bearing life, and minimize vibration.

A heavy duty bearing assembly employs 7300 Series bearings with slight preloads to support the shaft. The anti-friction bearings provide a minimum L-10 rated life of 100,000 hours at the rated pump condition. Thrust bearings are duplex, angular contact type mounted back to back. The radial bearing is either the anti-friction ball or spherical roller arrangement, depending on radial loads and rotative speeds.

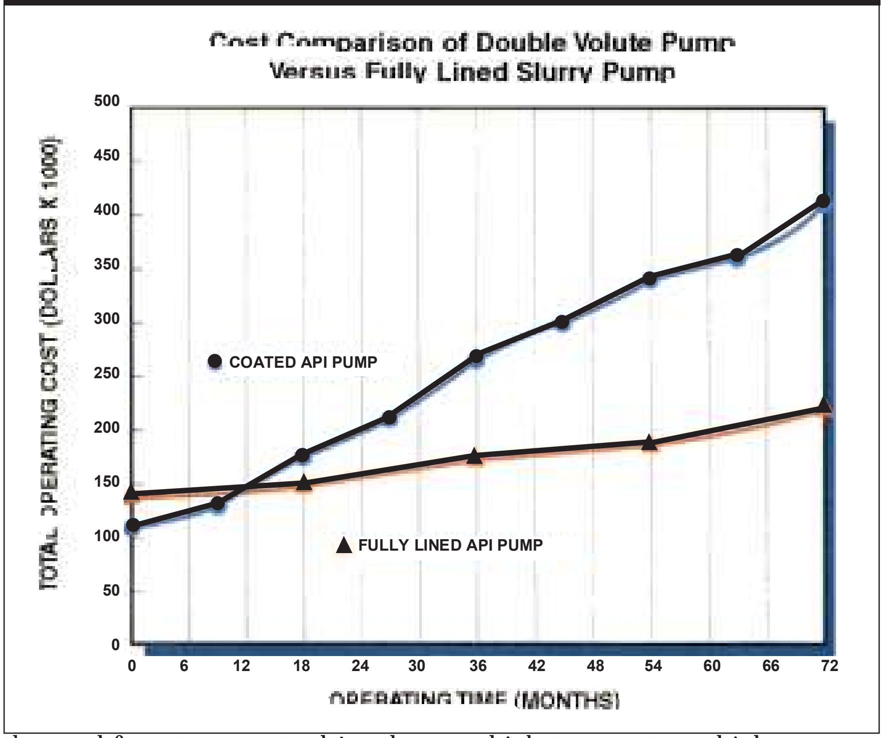
Bearings are lubricated using ring oil, flood oil, oil mist and forced feed arrangements. Oil mist is used to minimize friction, but it is not acceptable for cooling bearings that are subject to heat transferred from an external source such as the shaft. Thrust and radial bearing covers are equipped with isolators that have a deep grooved labyrinth which prevents oil from escaping from the bearing frame.

MECHANICAL SEAL CONFIGURATIONS AND OPTIONS

All fully lined slurry pumps operating in FCCU bottoms applications feature a removable seal chamber that allows pressure testing of the seal before installation. This design gives the user the ability to switch seal types easily to accommodate catalyst slurry changes and/or to comply with environmental regulations.

Seal chambers are designed to accommodate single, tandem and double mechanical seal arrangements. A replaceable throat bushing mechanically fastened to the seal chamber rides on the impeller hub. The shaft sleeve is 316 stainless steel, and its straight design allows easy adjustment of impeller clearances without disturbing the seal setting. A Ringfeder locking collar replaces the conventional locking collar with set screws. The Ringfeder eliminates the need for set screws to drive the shaft sleeve. It also seals the cavity between the shaft and sleeve by compressing the sleeve onto the shaft. This clamping arrangement prevents slippage of the sleeve on the shaft under severe slurry upset conditions, and it eliminates the need for grafoil packing by providing a high temperature, high pressure metal-to-metal seal between the shaft and sleeve.

FIGURE 3



STARTUP AND OPERATION ISSUES

The startup procedure for fully lined slurry pumps operating in the FCCU bottoms application is critical to the pump's long term performance.

PHOTO 2

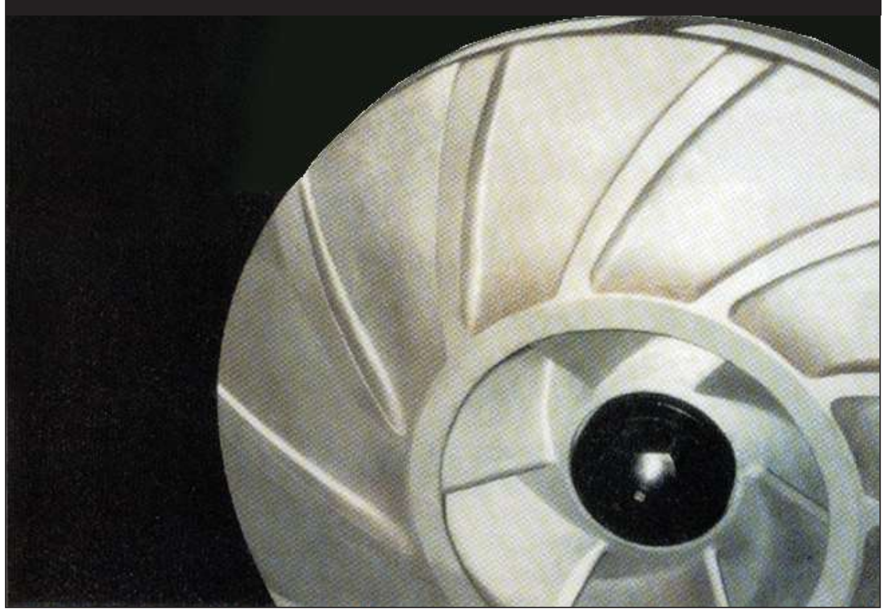


PHOTO COURTESY OF LAWRENCE PUMPS

Enclosed impeller with front and back repelling vanes to reduce circulation of the slurry.

To avoid thermally shocking the hard metal liners and impeller, hot oil is injected into the pump casing to preheat these internal components gradually — at a rate not exceeding 150°F per hour.

Initially, the oil steam is introduced at less than 250°F through either the casing drain or seal flush connection at a pressure higher than the downstream discharge pressure.. The steam is then allowed to flow through the casing. Operators are asked to maintain a constant preheat rate until the pump is heated to within 150°F of its actual operating temperature and allowed to soak for one hour at the maximum preheat temperature before startup.

Providing the proper cooling to the bearing frames and pedestals, and flush oil to the mechanical seals, becomes very important once the pump reaches its startup temperature.

To ensure long term successful operation of a fully lined slurry pump in the FCCU bottoms application, the following steps are recommended.

- Provide dual strainers in front of the pump to allow cleaning the strainer without shutting down the pump.
- Use slurry impeller designs to allow for larger mesh openings in the suction strainers and increase the cycle time between cleaning by 300%.
- Equip the FCCU system with a minimum flow bypass line so the pump can be operated continuously at its BEP.
- Use mechanical seals in the cartridge canister arrangement to allow performance testing during seal design stages and hydrotesting before installing the seal.
- Consider using double seal configurations for added safety and provide for inboard seal faces to run continually on a clean liquid.

MAINTENANCE ISSUES

Fully lined slurry pumps operating in the FCCU bottoms application require maintenance, repair or replacement of the casing liners and impeller every 3 – 5 years. This maintenance is simplified by a back pull-out arrangement allowing access to the liners and impeller without disturbing suction and discharge piping.

After assembly, impeller clearances can be adjusted easily by moving the thrust bearing cartridge relative to the bearing frame. Shims are then placed between the cartridge and bearing frame to lock the shaft and impeller into position. The large oil reservoir for the bearings ensures a continuous source of clean oil to lubricate the bearings. Oil level is monitored using a 2" bulls eye in the side of the bearing frame.

The entire pump can be rebuilt by mechanical craftsmen using standard shop tools. All wear components are replaceable, and all fits and clearances are standard.

LONG TERM PERFORMANCE

Several hundred fully lined slurry pumps are operating in FCCU bottoms applications in refineries around the world. More specifically, a fully lined slurry pump installed in a UOP FCCU unit has been running continuously since 1991, requiring only scheduled maintenance.

Fully lined slurry pump technologies continue to evolve to meet the increasingly demanding performance criteria of the FCCU bottoms application. Refineries that have installed fully lined slurry pumps in their cracking units have eliminated the expense of replacing conventional API process pumps several times a year. More important, they've eliminated the safety risk of catastrophic pump failure and the prohibitive cost of shutting down the FCCU for several days. ■

Dan Clark has been with Lawrence Pumps Inc. since 1973 and has overseen the development of the FCCU bottoms fully lined slurry pump.

Julio R. Cayro is a Mechanical Equipment Consultant and owner of Cayro Engineering Company in Houston, TX.

Options for Sealless Centrifugals

In recent years, the case for using sealless centrifugal pumps has centered mainly on zero emissions—and the fact that they do not require seal support systems and periodic mechanical seal replacement.

But this is only part of the story. Design modifications and accessories are expanding the performance ranges of both canned motor and magnetic drive pumps. Here is a generic comparison of what these pumps offer when it comes to hydraulic application features and options:

1. DOUBLE CONTAINMENT

This is an important consideration in services that pose extreme health and/or safety concerns. Several manufacturers of canned motor pumps (CMP's) offer double-containment (i.e. welded primary containment, hermetically sealed secondary containment). Because of efficiency issues, few magnetic drive pump (MDP's) manufacturers have approached secondary containment.

Most MDP's can provide secondary "control" by utilizing mechanical seals on the OMR shaft penetration. Additionally, an MDP provides a thicker containment shell, and thus more resistance to penetration by corrosive or mechanical failure. Typical CMP primary liner thicknesses are 0.022–0.35″, while an MDP containment shell is 0.029–0.060″. Bearing monitoring plays a role, however, because bearing and internal rotor positions are easier to monitor in CMP's, by design, than in MDP's. This enables the CMP to detect extreme bearing wear prior to containment shell contact or violation.

2. SOLIDS/SLURRY HANDLING

Both the canned motor and magnetic drive designs will handle moderate amounts of solids, and optional designs for both will handle higher concentrations of slurries. For CMP's, if low concentrations of solids are present (about 2%) and their size is relatively small (25µm maximum), hard bearings running against a hardened journal work well. CMP's can handle increased solids if they are outfitted with external flush or filters to remove particulates from the pumpage before they circulate around the bearings. In fact, CMP's with a slurry modification are able to handle slurries in the concentration ranges handled by a standard centrifugal pump. An MDP cannot be isolated as easily as the CMP, and therefore would require a completely different internal bearing-mag coupling flow path than is conventionally offered.

3. HEAT INPUT

Both sealless centrifugal pump designs add heat by hydraulic and drive inefficiencies. For a CMP design a "high efficiency" motor will be 80–85% efficient. But because of the ease in isolating the motor area, CMP's can offer optional configurations to control the fluid temperature, pressure, or both, to prevent product vaporization. However, "heat soak" can occur. In this situation the process fluid will be heated to higher temperatures at potentially lower pressures than during normal operation. This is because a CMP motor tends to be a large insulated mass once the unit is shut down. This may result in flashing of the contained fluid with a potential of vapor locking if restarted.

4. COOLING REQUIREMENTS

Both the CMP and MDP can be configured for operation at high temperatures. However, permanent magnets can tolerate heat better than motor windings can, so MDP's are able to pump hot liquids—up to 750°F—with just air cooling. Canned motor pumps require water cooling jackets for high temperature service. For example, some CMP designs can operate up to 1,000°F with water cooling. Various designs require temperature limitations by component and must be evaluated on a case-by-case basis.

Several things must be considered on MDP designs. A synchronous drive is more efficient than an eddy current drive. A non-metallic containment shell is more efficient than a metallic shell. The MDP lacks the large insulated mass around the containment shell and is less susceptible to "heat soak."

5. JACKETING

Either design can accept steam or hot oil jackets added to the pump to maintain the proper temperature of the product with a high freezing point. This insures that the pumpage remains a liquid during pump operation and shut-down.

CMP designs allow jacketing of the pump case, stator and rear bearing housing. Users must be careful not to exceed the thermal limits of the stator insulation with the heating media. Most MDP's can jacket the casing and add some heat in the area of the containment shell without fully encapsulating it.

6. HIGH SUCTION PRESSURE

Typically, the CMP is more efficient than the MDP in high pressure applications because of the increased primary containment thickness. A canned motor pump is a pressure vessel since the stator windings lend additional mechanical strength.

For applications in which the suction pressure and maximum allowable working pressure require-

PHOTO COURTESY SUNSTRAND FLUID HANDLING

Cutaway of a Kontro A-range ANSI sealless magnet drive pump used in chemical processing services.

ments exceed the standard pressure design capability of either a CMP or MDP, optional designs are available for both. In fact, some CMP's are available that can have as high as 5,000 psi system pressure design.

Modifications in the CMP design for high pressure applications include the use of primary containment shell backing rings, thicker secondary containment shells, additional pressure-containing bolting and high pressure terminal plates. For MDP's high pressure application modifications include usage of a thicker containment shell and additional bolting.

In addition to application features, users should be aware of certain hydraulic and design differences between canned motor and magnetic drive pumps. Here again, both offer advantages.

7. ANSI

While a few manufacturers of canned motor pumps offer units with ANSI dimensions and/or hydraulics, most do not. On the other hand, the majority of mag drive pump suppliers do offer ANSI dimensions and hydraulics. In addition, most manufacturers of sealed and sealless ANSI pumps offer interchangeability between their pumps' wet ends and bearing frames.

8. EFFICIENCY

CMP wire-to-water efficiency is defined as hydraulic times motor efficiency. As mentioned in the section on heat input, a typical CMP motor will be 80–85% efficient. MDP wire-to-water efficiency is defined as hydraulic efficiency times motor efficiency times coupling efficiency. Again, the magnetic coupling is 80–85% efficient. However, containment shell metallurgy (or lack thereof) and magnetic coupling type play a big role in coupling efficiency.

Further complicating the efficiency discussion is wet end hardware. Impeller and casing geometry play a vital role in hydraulic efficiency.

For example, a Barske design (open radial blade impeller and diffuser discharge) is usually more efficient in low flow/high head hydraulics. A Francis design impeller (enclosed with backswept vanes and an increasing-radius volute) is more efficient at moderate to high flows with low to medium heads.

You need to evaluate "wire-to-water" efficiency to get a truly accurate picture of efficiency.

9. INTERNAL CLEARANCES

While clearances between bearing ID and mating surfaces are typically the same (0.003–0.007″), differences occur between other rotating parts.

Typical CMP clearances between the rotor and stator liners vary by manufacturer between 0.018–0.044″ radially. Typical MDP clearances between the inner magnetic ring (MR) and containment shell range from 0.030–0.045″ radially. This larger clearance gives MDP's the advantage of allowing more bearing wear to occur prior to containment shell contact.

10. BEARING MONITORING

While different monitoring methods are available for both MDP's and CMP's, the latter design lends itself more to real bearing monitoring. A CMP bearing monitor can provide axial, radial and liner corrosive wear indications. However, bearing monitoring features vary by manufacturer and care must be exercised when selecting a sealless pump vendor to insure that the desired monitoring features are provided.

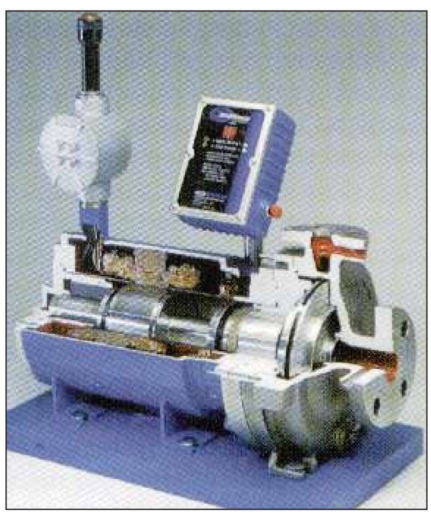


PHOTO COURTESY CRANE CO., CHEMPUMP DIVISION

Cutaway view of a canned motor pump. These pumps are used on a wide variety of fluids at temperatures from cryogenic service to 1000° F and at system pressures up to 5,000 PSI.

11. SPACE CONSIDERATIONS

In general, a CMP (integral pump and motor) occupies less of a foot print than a comparable MDP (pump, coupling and motor). However, close-coupled MDP's are available, and these may require the same or less space than a CMP.

12. COUPLING ALIGNMENT AND VENTING

Coupling alignment is not required for CMP installations because the pump impeller is directly mounted on the motor shaft inside of the containment area, so no coupling exists.

Typical MDP installations utilize frame-mounted motors which require coupling alignment. Some MDP suppliers, however, offer close-coupled designs which eliminate coupling alignment. Both MDP and CMP designs are usually self-venting back into the process piping and do not require additional external lines.

In addition to the issues already discussed, some manufacturers offer canned motor and magnetic drive centrifugal pumps with options such as the following: two-phase flow designs; tachometers; open impellers (non-shroud) with isolated motor sections for solids handling; diagnostics that indicate rotor position, stator liner rupture, temperature and pressure; vibration pads, and redundant systems to indicate breach and contain fluid.

The best thing you can do if you're considering options for your sealless centrifugal pumps is to make sure your supplier(s) know everything about your application, particularly temperatures and vapor pressure at startup and shutdown, not just normal operating conditions. ■

This article was developed with the assistance of Steven A. Jaskiewicz, of Crane Chempump (Warrington, PA) and David Carr, of Sundstrand Fluid Handling (Arvada, CO).

Tips for Selecting ANSI Process Pumps

Versatility is the key to ANSI process pump applications.

By Charles Cappellino and Richard Blong

A revolution swept through the chemical process pump industry more than 30 years ago. It was the beginning of chemical pump design and dimension standardization. Before the 1960s, chemical process pump manufacturers offered a proliferation of designs. Each manufacturer had its own design and dimensional envelope. Industrial users faced significant piping, baseplate design and potential foundation changes if existing pumps had to be replaced. This very expensive possibility became the driving force behind the development of what industry today refers to as an "ANSI" pump. During the 1960s and 1970s, the American Voluntary Standard (AVS pump) served as the chemical process pump standard. In 1974, the American National Standards Institute (ANSI) used the AVS standard as the foundation for its B73.1 specification covering chemical process pumps.

After several revisions today's ANSI/ASME B73.1M - 1991 Specification for Horizontal End Suction Centrifugal Pumps for Chemical Process serves as the industry standard. It covers dimensional interchangeability requirements for 20 pump sizes. This includes mounting dimensions, suction and discharge flange size and location, input shaft size, baseplates and foundation bolt holes. It also addresses many mechanical design features such as pressure limits, temperature limits, drain and gauge connections and seal chamber dimensions. This enables today's pump user to replace a pump with one from a different manufacturer easily.

Chemical process pump specifications developed through the B73 committee include:

- ANSI/ASME B73.1M - 1991, Specification for Horizontal End Suction Chemical Pumps for Chemical Process (Photo 1)
- ANSI/ASME B73.2M - 1991, Specification for Vertical In-Line Centrifugal Pumps for Chemical Process (Photo 2)
- ANSI/ASME B73.3M - (in process), Specification for Sealless Horizontal End Suction Centrifugal Pumps for Chemical Process
- ANSI/ASME B73.5M - (in process), Specification for Thermoplastic and Thermoset Polymer Material Horizontal End Suction Centrifugal Pumps for Chemical Process

PRINCIPLES OF OPERATION

The pumps covered by ANSI/ASME B73.1M are classified as end suction centrifugal pumps. Centrifugal pumps use an impeller and angled vanes to impart velocity to the liquid entering the pump. The liquid leaving the impeller is collected by the pump casing, an action that converts a portion of the fluid velocity into pressure. Either mechanical packing in a stuffing box or a mechanical seal in a seal chamber is used to seal the rotating shaft. Figure 1 shows the basic components of an ANSI pump.

Impellers. Centrifugal impeller designs are of two basic types, the open style and the closed style, as shown in Figure 2. Most ANSI pumps employ some type of open impeller and axial adjustment feature. This allows critical operating clearances within the pump to be maintained, which is important for maximizing hydraulic performance. The open impeller is also better at handling solids and pumping liquids with entrained gases. Open impellers – particularly those running at 3600 rpm – must be carefully engineered to control axial thrust, seal chamber pressure and mechanical integrity. Closed impellers are typically employed in less corrosive environ-

PHOTO COURTESY OF GOULDS PUMPS, INC.

Photo 1. Example of a horizontal metal ANSI process pump

PHOTO COURTESY OF GOULDS PUMPS, INC.

Photo 2. Example of vertical in-line ANSI centrifugal pump for chemical process service

ments such as light duty chemical, petrochemical and utility applications. This is because closed impellers utilize renewable wear rings to maintain performance-sensitive running clearances. Renewable wear rings are subject to crevice corrosion and are generally undesirable for corrosive services.

Casings. Three basic types of centrifugal pump casing designs are used in chemical process pumps: the circular volute, single volute and double volute (Figure 3). The different casing designs are used to reduce hydraulic radial loads. Pumps designed to operate at extremely low flows normally use circular volutes to minimize hydraulic radial loads. Single volute casings are simple to manufacture and are the most commonly used for ANSI pumps. Larger pump sizes require a double volute to reduce hydraulic radial loads. The casings utilize ANSI/ASME B16.5 Class 150 or Class 300 flanges for suction and discharge connections. Casings must withstand a hydrostatic pressure test of 1.5 times the maximum design pressure for the material used, and they must employ an 0.125" corrosion allowance by design.

Seal Chambers. ANSI/ASME B73.1M provides dimensional guidelines for shaft sealing. The guidelines cover a stuffing box design used for packing, a large diameter cylindrical seal chamber and a self-venting seal chamber used for mechanical face seals. Large radial clearance between the shaft and the inside of the seal chamber is specified due to its importance to mechanical seal face temperatures. The large radial clearance also allows mechanical seal manufacturers to build more robust and reliable designs. Because mechanical seals are one of the most significant causes of pump downtime, most ANSI pump manufacturers have developed new seal chamber designs that enhance the operating environment for mechanical seals. Typical seal chambers offered by manufacturers are shown in Figure 4. The most recent self-venting designs incorporate some type of flow modifying ribs or superior performance vanes to control solids and entrained gas. Most chemical pump manufacturers offer several seal chamber designs, as well as selection guidance for various services.

Bearing Housings. ANSI/ASME B73.1M requires a bearing selection that provides 17,500 hours of life for the radial and axial thrust bearings, calculated according to ANSI/AFBMA-9&11. This typically results in a double row thrust bearing being used at the coupling end of the shaft and a deep groove ball bearing at the impeller end. (Photo 3)

The pump industry is steadily improving the reliability of pump components. To increase mean time between planned maintenance, many manufacturers have improved bearing housing designs and added features to improve reliability. Some ANSI pump manufacturers offer heavy-duty housings that use angular contact bearing pairs to handle higher hydraulic thrust loads. Most bearing housings are sealed using either lip

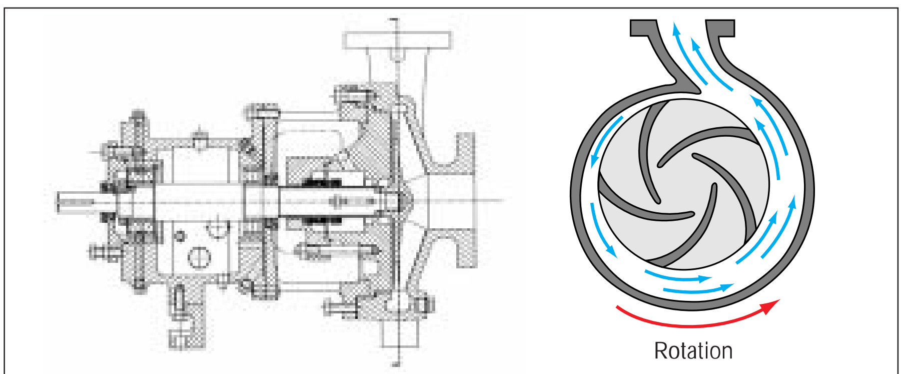


Figure 1. Basic components of an ANSI B73 pump

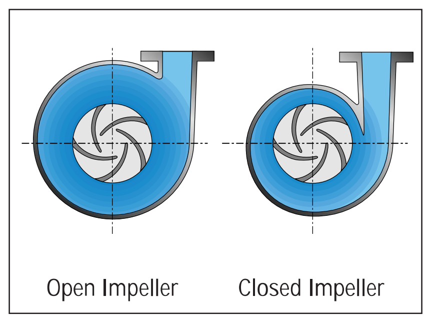


Figure 2. Centrifugal impeller designs

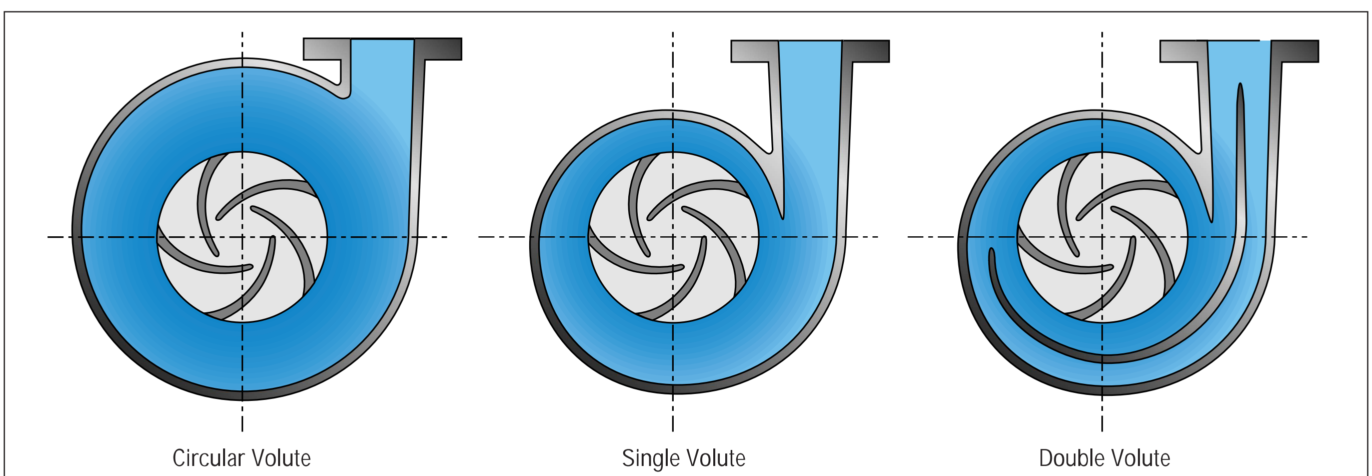


Figure 3. Circular, single and double volute casing designs

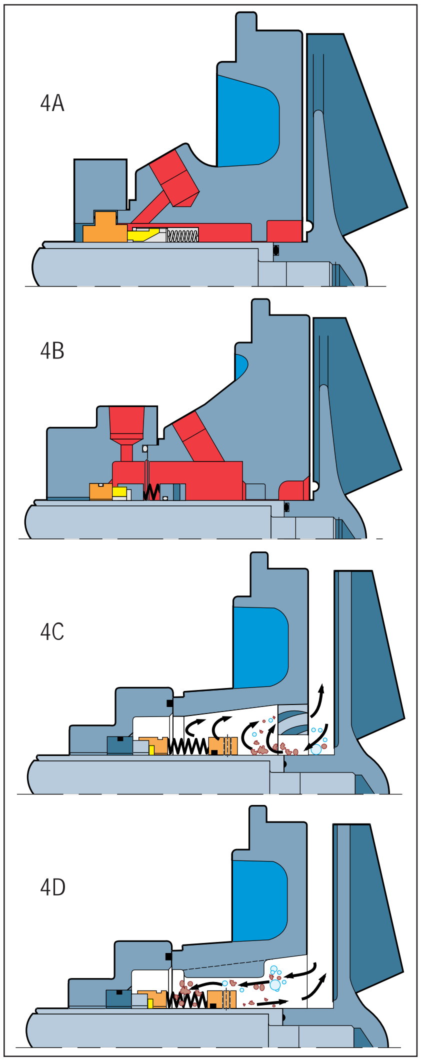


Figure 4. Seal chamber styles

A. Standard bore (packed box) is characterized by long, narrow cross section. Originally designed for soft packing, mechanical seals were forced into cavity envelope. Requires an API/CPI flush plant for optimal performance.

B. Enlarged bore features increased radial clearances over the standard bore. This chamber design enables optimal seal design. Restriction at bottom of the seal chamber limits fluid interchange. Requires an API/CPI flush plan for best performance.

C-D. Tapered bore features increased radial clearances similar to the enlarged bore, except there is no restriction at the bottom of the cavity and is open to the impeller backside. Current designs include vanes or ribs to provide solids and entrained gas handling. Flushing is often not required as design promotes cooler running seals by providing increased circulation over faces.

seals or higher performance labyrinth seals to prevent lubrication contamination–the number one cause of premature bearing failure. Many manufacturers also have increased the capacity of oil sumps to provide superior heat transfer and cooler running bearings. Most housings have some type of finned cooler that is submerged in the oil sump. This controls oil temperature in hot services. And large diameter (1") sight glasses are incorporated into many designs to provide a means of viewing oil condition and level.

Baseplate Designs. ANSI/ASME B73.1M specifies a set of baseplate dimensions covering motor sizes typically required for the full range of ANSI pump sizes. Dimensions specified include the pump and motor mounting surfaces, bedplate footprint and foundation bolt hole locations. Proper baseplate design and installation are necessary to maintain accurate pump and driver alignment. This lengthens the life of bearings and seals, which are sensitive to vibration and correct alignment.

Proper baseplate selection is the key to maximizing mean time between planned maintenance. Most pump manufacturers offer a selection of baseplate designs. Camber top cast iron baseplates offer heavy-duty construction with machined pump and motor pads. They also have good vibration damping characteristics. Fabricated steel baseplates provide an economical choice in carbon steel and the option of various metallurgies such as stainless steel. Many ANSI pump manufacturers also offer some type of nonmetallic composite baseplate (such as fiberglass reinforced plastic - FRP) for superior corrosion resistance. FRP bases are used with FRP pumps as well as high alloy pumps. A recent addition to the bedplate options is a heavy-duty fabricated baseplate with integral adjustment features such as adjustment screws and baseplate leveling screws (Photo 4). Finally, most bedplates can be stilt or spring mounted. These supports raise a pump above the floor for improved cleaning access, and they accommodate piping thermal expansion. Stilt and spring-mounted designs must be carefully engineered to ensure proper rigidity. This will maintain alignment and avoid vibration problems.

The ANSI/ASME chemical process pump is the most widely used centrifugal pump in industry. Its wide use is attributed to its adaptability to a wide range of process service pumping conditions. It can be mounted vertically to save installation space, or vertically suspended for use in a sump application. Flexibility in casing and impeller designs enable it to handle extremely low flows, pump solids, move highly corrosive liquids, self-prime or withstand temperatures to 700°F (371°C). Recently, the same basic design has been made sealless by eliminating the need for packing or mechanical seals to seal liquid in the pump. These pumps are magnetic drive ANSI/ ASME chemical process pumps. ANSI/ASME chemical process pumps are perhaps the most versatile pumps in the world.

To help in applying the various types of chemical process pumps, a selection guide is shown in Figure 5. The pump type is shown vertically, and the service parameter is listed horizontally. ANSI chemical pump solutions are available for nearly any pumping service parameter.

Photo 3. Bearing housing configurations

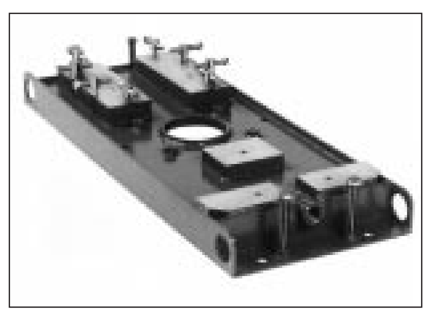


Photo 4. Enhanced fabricated steel baseplate

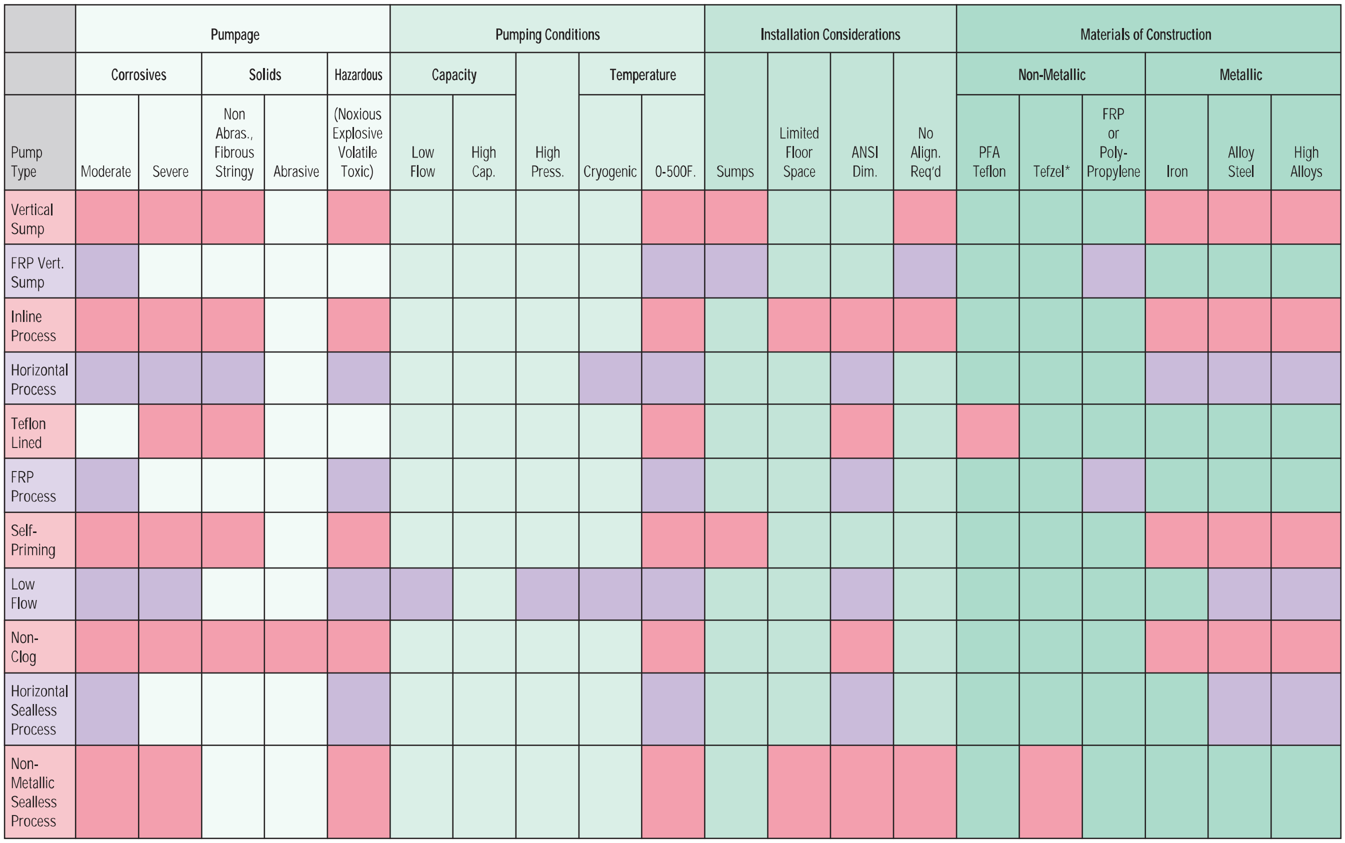


Figure 5. Process pump selection guide

TOMORROW'S CHEMICAL PROCESS PUMP

Having evolved for more than 30 years, the ANSI/ASME chemical process pump now offers users improved reliability, easier installation and broader application flexibility. Its development over the next 30 years will surely produce further improvements in these areas. But instead of 3 years meantime between planned maintenance, the industry will be driving towards 5+ years. Pump emissions will not be acceptable at 1000 ppm. Zero (0) ppm will be the goal. It will not take 2 hours to align a pump and motor to 0.002 TIR. It will take only 15 minutes to align to 0.0005 TIR. And these and other changes will surely take place in less than 30 years.

In fact, manufacturers and users both are demanding change now. The result is that current versions of the ANSI/ASME B73.1M and B73.2M are up for revision this year. The ANSI Pump Committee has also developed two new specifications. In addition, a new group called Process Industry Practices (PIP) has developed two new specifications that supplement ANSI B73 specifications with additional requirements commonly specified in the industry (both horizontal and vertical types).

1. POTENTIAL REVISIONS TO ANSI/ASME B73 SPECIFICATIONS

Changes are under way in the ANSI/ASME B73 Pump Committee. The basic B73.1M Horizontal Process Pump Specification will be revised and probably issued in 1997. Areas being addressed to improve pump reliability are:

a. Nozzle loading
b. Seal cavity dimensions
c. Auxiliary connections to glands and seal cavities
d. Baseplates
e. Additional pump sizes
f. Hydraulic Institute Class A performance criteria
g. Allowable operating range

New Specifications. The ANSI/ASME Pump Committee recently issued a new B73.5M Specification addressing nonmetallic pump designs. In addition, a Canned Motor/Magnetic Drive Specification B73.3M is expected to be approved in 1996.

2. NEW PIP STANDARD

The new specifications covering typical B73 pumps are the PIP (Process Industry Practices) RESP73H and RESP73V. Engineering contractors and pump users have formed a Machinery Function Team whose sole task is to develop a set of standards that will eliminate variations in chemical process pumps manufactured to multiple user and contractor pump specifications. This team and its work will greatly minimize the problems associated with multiple specifications. Lower engineering costs and enhanced pump reliability will be the benefits.

The PIP RESP73H (Horizontal Chemical Process Pumps ANSI B73.1M Type) and RESP73V (Vertical Chemical Process Pump ANSI B73.2M Type) cover the same pump design areas as the original ANSI specification, but they also address:

a. Solid shafts
b. Shaft deflection L^3/D^4
c. Shaft sealing design responsibility
d. Bearing lubrication
e. Preparation for shipment
f. Couplings
g. Baseplate design
h. Hydraulic performance acceptance criteria

The PIP team has the same goals as the ANSI B73 pump team, but it is driving standardization both to the project and local levels. ■

Charles Cappellino is an engineering project manager for the Industrial Products Group of Goulds Pumps Inc. He is a professional engineer with a Bachelor of Science in mechanical engineering from Clarkson University and has been involved in centrifugal pump analysis and design for 15 years.

Richard Blong is a product manager for the Industrial Products Group of Goulds Pumps Inc. He has a Bachelor of Science in chemical engineering from the University of Buffalo and has been involved in centrifugal pump applications for 10 years.

ANSI Upgrades Require More Than Technology

Total program commitment is key to ANSI pump upgrade success.

By Joseph Dolniak

Merely upgrading technology is not enough to increase pump reliability. A technological upgrade is just one of many factors that must be addressed to achieve maximum reliability. Reilly Industries has been increasing the reliability of its large ANSI pump population since 1990. At year end 1995 the company's total pump repairs were at their lowest levels since 1988. Meanwhile, plant production has tripled due to expansions and improvements in efficiency.

We shall examine the procedures Reilly Industries instituted to bring its pumps into compliance with the new ANSI standards. Specific areas covered include new pump installations and orders, inventory, converting old pumps to new standards and future plans. Sealless ANSI pumps are not covered because there are none on site. Testing is currently under way to determine if certain sealless brands will be accepted into the plant.

A specialty chemical manufacturer located in Indianapolis, Reilly Industries is about to celebrate its centennial under the leadership of Tom Reilly, Jr., the founder's grandson. The company has grown over the last century to where it currently manufactures more than 100 intermediate and specialty chemicals for a worldwide market.

Just over six years ago a pump improvement program was initiated to increase the reliability of the company's more than 800 pumps. In implementing pump upgrade projects, certain steps must be followed to obtain maximum benefit. The steps discussed below can be followed to complete any project successfully. They go by many names but most often are referred to as good engineering practices.

GOOD ENGINEERING PRACTICES

Following are some of the quality steps or good engineering practices followed in Reilly's pump reliability upgrade project.

1. Know the current situation
2. Analyze the current situation
3. Formulate a plan
4. Initiate a trial
5. Set up standard and use
6. Train and communicate to work force and others
7. Maintain data
8. Continue making refinements
9. Analyze pump technology
10. Phase in and phase out
11. Stick to the plan
12. Redo poor installations

Steps 1 - 8 are good engineering practices overall. Steps 9 - 12 refer more specifically to the pump upgrading project.

For positive results in upgrading pumps for improved reliability, the pump, mechanical seal, gland, pump base, pump pad and immediate pump piping all must be addressed. The pumpage and flow rate conditions also must be compatible with the type of pump used.

When this project was started in 1990, Reilly Industries had been using a computerized maintenance system for about 5 years. This helped greatly because the maintenance repair and cost history already was on file. The data indicated that pump failures were increasing at an unacceptable rate. Production levels also were increasing, so there was an urgent need to improve pump reliability. The historical data were only a portion of the information that needed to be analyzed to implement a good upgrade program. Visual inspections of the pumps installed at that time revealed other factors that needed to be addressed. The most obvious was a lack of proper grout. Repair inspections also showed that many pump sites had excessive pipe strain. Indicator reverse, or laser alignment, was almost never done on the ANSI pumps. These deficiencies could be handled in-house through better maintenance practices and training. Other difficulties could not. The most important factor that could not be controlled in-house was that mechanical seals were operating in stuffing boxes designed for packing. But a newly approved standard known as ASME B73.1M-1991 changed ANSI pump history.

ASME B73.1M-1991

ASME B73.1M-1991 is the "Specification for Horizontal End Suction Centrifugal Pumps for Chemical Process." One important item addressed in this revision was an increase in pump base sizes to add rigidity. Another was additional motor protection. Perhaps even more important, however, was the new designation of a seal chamber versus the old stuffing box. The introduction of the seal chamber allowed the mechanical seal manufacturers freedom to create new technology and mechanical seal designs. The gas barrier seals that

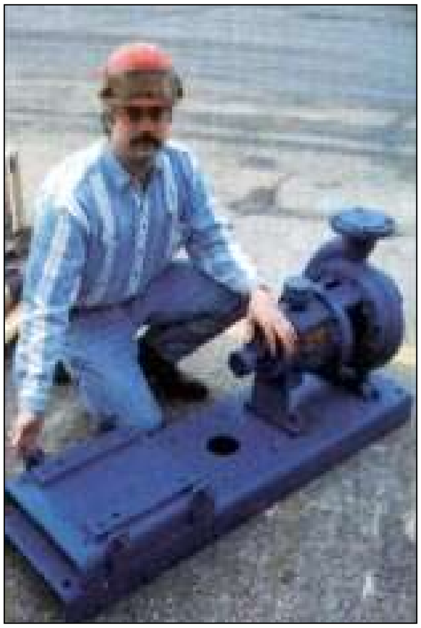


Photo 1. Recently received group two size pump incorporates motor jacking bolts, grout hole, bearing isolators, bull's eye oil indicator and bearing housing expansion chamber as standard features.

are now on the market are a result of this. Specifics not addressed in this revision were the standardization of the gland bolt circle, shaft size at the mechanical seal, and sealing surface diameter at the gland/seal chamber interface and gland piloting area. Standardizing these areas would allow the end user even greater freedom to reduce me-chanical seal and gland inventory while maintaining competitive bidding from various mechanical seal vendors.

With this revision, three basic seal chambers could be used. They included the 4° taper bore, the large bore with a throat restriction and the large bore with no throat restriction. All three designs are beneficial to mechanical seals and can be swapped for the old stuffing box. In each the gland has to be replaced because of the larger static sealing surface and bolt circle diameters.

PUMP RELIABILITY UPGRADE PLAN

With the pump repair history, installation analysis, repair analysis and the new ASME B73.1M-1991 standard in place, a plan could be formulated to address pump reliability. Effort would be directed mainly at ANSI pumps because of their greater population and the fact that many improvements made on them would apply to other types of pumps also. The upgrade plan involved a two-step approach. The first step was to ensure that no more pumps were misinstalled or improperly ordered. This would help reduce maintenance problems from the start. Reilly engineers began by writing standards for proper pump installation procedures. Covered in the standards were such details as pump base dimensions, heights, depths, hold-down bolt designs, pump spacing from one another, pipe strain, grout and alignment. Before the standards were approved, they were tested on four pumps that showed normal repair rates for the plant at that time. After these pumps were reinstalled according to the proposed standard, they ran much quieter and smoother.

Looking at the repair frequency and costs for the three years before the reinstallation and after, total repair costs dropped 94%, from $46,470 to $2,908, and total repairs dropped 69%, from 49 repairs to 15 in the same time period. These pumps were still fitted with the standard stuffing box because the seal chamber was not yet on the market. Shortly after the pumps were reinstalled, however, the standard was accepted as a site engineering standard for ANSI pump installation.

At the same time, new standards were written for future ANSI pump orders. These standards incorporated into all new pump orders many of the improvements specified in ASME B73.1M-1991. Included were requirements that all new pumps have 4° taper seal chambers, drain and discharge taps on the pump casing, labyrinth bearing isolators, bull's-eye oil level indicators and bearing housing expansion chambers. On pump-base assemblies the base would conform to ASME B73.1-1991 di-mensions, have a grout hole centered on the base (4" preferred), and have motor jacking bolts for alignment purposes. Some of these upgrades are shown in Photo 1. This would give us a head start as we would soon begin an alignment and pump installation program. The jacking bolts and grout holes saved maintenance time when these programs were under way.

The accomplishments up to this time took more than a year to achieve. They helped prevent misinstallation of ANSI pumps and eliminate the process of ordering pumps with old technology. This marked the beginning of the next phase of the reliability program, but in actuality it would have little noticeable effect for some time as the new

Photo 2. An upgraded second generation pump receives backup as part of the reliability improvement program.

pumps would be installed only as required through expansion or attrition. The next phase of the upgrade had more immediate impact because the ANSI pumps were upgraded as they were worked on. This step addressed standardizing and consolidating pumps and pump inventory while adding the new-technology parts to stores. This was also a two-step process. Step one was the reduction in numbers of ANSI pump brands and the consolidation of mechanical seals. The second step consisted of phasing in new parts while the old parts were phased out of stores.

PUMP CONSOLIDATION

At least nine specific brands of ANSI pumps had been in use on site. The goal was to reduce that number to three. Some of the factors considered in selecting what brands to retain were past reliability, maintenance shop familiarity, local distributor professionalism, location of the OEM and the size of the current plant population in specific makes. When selection was finished, seven of the nine brands were eliminated, and a new brand was added. The standards list reflecting our preferred vendors was updated, and Reilly is now testing a fourth brand of ANSI pump for possible addition to the list. As for the mechanical seals, we are about 95% committed to one manufacturer. This simplifies the process of consolidating our mechanical seal inventory.

The time-consuming pump brand reduction project is just ending after more than four years of phasing pump parts out of inventory. Caution should be taken when adding additional brands of pumps to a plant site once this point is reached because the amount of work needed to add a brand is almost as time-consuming as it is to eliminate one. Also, the system must be allowed a break-in period to determine how well it is working. Continual changes do not allow this to happen. Also, additional shop training is needed. That is why the preliminary work of determining what pumps will best suit the plant is so important. After finalizing the brands of pumps accepted on the vendor list, work began to phase in the accepted brands and eliminate brands that would no longer be used.

The company was careful not to be wasteful in eliminating brands. Repair parts in inventory that were associated with the pump brands to be deleted were classified as POR (purchase on request). This would cause no parts to be ordered when the reorder point came up, but it also would not delete the part from stores. In so doing, most of the repair parts currently in inventory for the brands of pumps to be deleted could be used. If one or two minor parts were needed to complete a repair and the repair parts were not on hand, giving the stores clerk a work order number would allow the clerk to order the POR part. When most of the parts were used up, the part code would be changed to DELETE, and any remaining parts would be pulled from the shelves. Thus, most of the parts could be used while eliminating unwanted inventory. When no parts or not enough major parts remained, a new pump of an accepted brand was ordered to replace the pump that was being repaired. The new pump incorporated all of the upgrades we required when it arrived in the plant.

Next month in Part II of this article we will assess how Reilly Industries conformed to the new upgrade plan. We will discuss consolidation and parts inventory changes, show how mechanical seals were upgraded, and reveal some of the training procedures that have been instrumental in helping this project succeed. We will end with a look at future plans. ■

CONFORMANCE TO PLAN

It would have been very easy at the conformance stage to deviate from the plan because there were times when the up-front dollar amount involved in purchasing a part being deleted was less than the cost of upgrading the pump technology and purchasing a new pump. Economizing up front, however, would have compromised the plan and been costly in terms of overall life cycle cost. This process is proving profitable for the company as our overall pump repairs have declined as production has steadily risen, as shown in Graphs 1 and 2. Total repair costs for all pumps on site appear to be going up slightly. This was expected because we began requiring our work force to do more thorough repairs, which take more time. Maintenance workers now replace parts that were not normally replaced due to lack of proper inspection. More parts are now being found to be out of specification when checked with dial indicators. And some auxiliary parts now cost more because superior materials are being used. Yet if the pump repair costs were corrected for inflation, the costs would be almost level. Also, the total number of pumps has increased. The in-house maintenance system showed 792 pumps at the end of 1989 and 837 pumps at the end of 1995. Achieving near level repair costs while doing more thorough repairs with superior parts is attainable because we have increased our MTBF (mean time between failure), and thus there are fewer repairs.

Another technique helped maintain costs while phasing out specific pump brands. If a pump being removed for an upgrade or taken out of service was one of the brands that was to be kept, and if it was worth rebuilding, it was put in a specified area. If the pump filled the requirements of a pump brand that was to be eliminated, it sometimes was used in lieu of purchasing a new pump. Several important considerations in doing this were the pumpage material and the repair frequency and the costs of the deleted pump brand being removed. Photo 1 is an example of one of these pumps being brought back into service and adding a back-up pump at this site, which before had no back-up pump. This helped determine if replacing old technology with old was appropriate, or if new should be used, and it improved reliability while maintaining costs. To date, the equivalent of more than $50,000-worth of pump parts has been removed from inventory records.

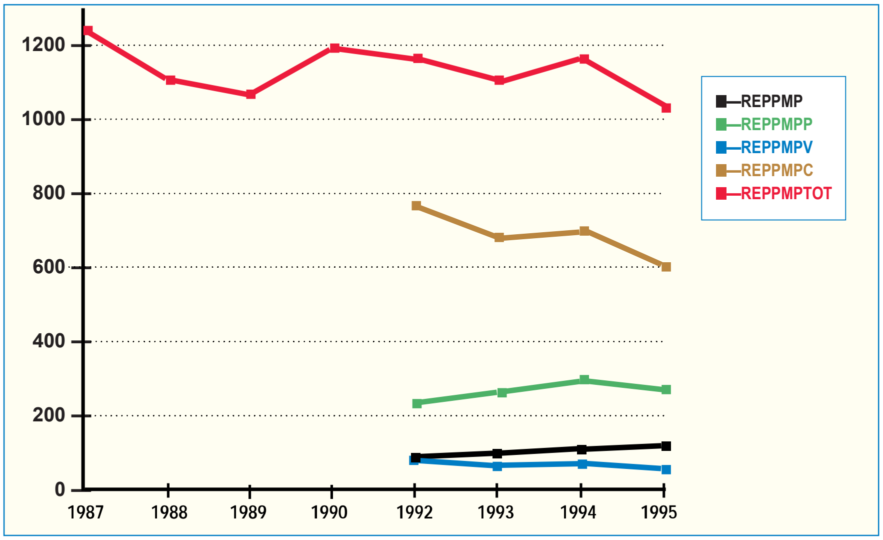
SEAL CONSOLIDATION

Consolidating the mechanical seal parts was more difficult because we were phasing out standard stuffing boxes and phasing in 4° taper bore seal chambers. This required double inventory of glands and gland gaskets for about two years because the gasket and bolt circle on the seal chambers and their glands were larger. There was no need to duplicate any mechanical seal parts. The mechanical seal stationary insert fit both the old glands used with the stuffing boxes and the new glands used with the seal chambers. Parts relating to the stuffing box backheads have recently been eliminated, and now only seal chambers and the new glands remain in inventory. There was another important factor in keeping the additional inventory, now current inventory, to a minimum. In evaluating what pump brands to keep, we also looked at the shaft diameter at the mechanical seal, the seal chamber gland bolt circle, and the gland gasket and pilot diameters. The three ANSI pump brands that were chosen had the same dimensions for the above items. This was important, and it is why, generally speaking, adding these items to ASME B73.1-1991 as part of the standard should be beneficial to the end user. By choosing pumps with the same dimensions on the above items, we were able to use only one gland to fit all three brands of pumps, per pump group size designation.

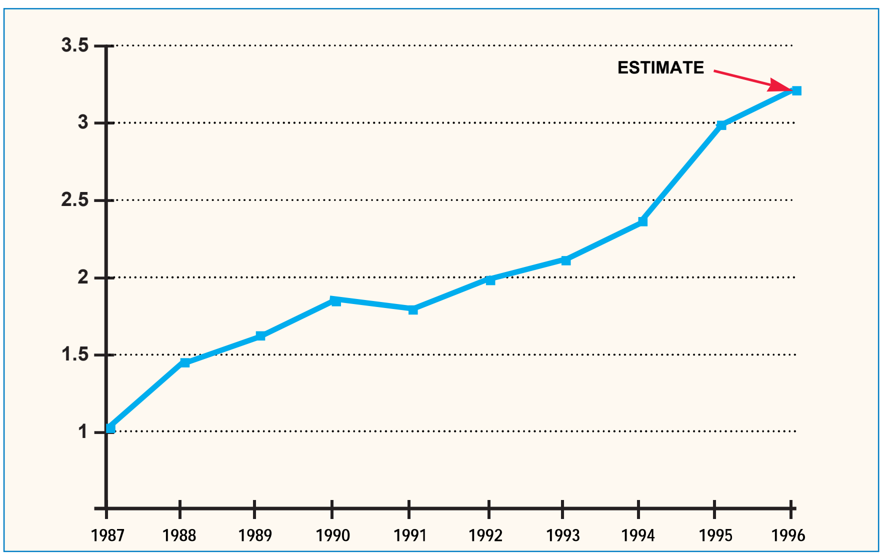
Thus, for all of our group one and group two size ANSI pumps, which are the majority of ANSI pumps in the plant, we have only two glands: one for all group one size pumps, and one for all group two size pumps. The three types of component mechanical seals (one OEM) that we use per pump group size all fit the one gland. The glands come with vent, flush and drain. If they are not needed, they are simply plugged. This has enabled us to eliminate at least nine specific glands from inventory. This strategy also works for consolidating cartridge mechanical seals. The process took much communication between myself, the mechanical seal engineers and the pump engineers. The second major benefit of consolidating the glands besides the inventory reduction is that there is no confusion to the work force on what gland to use on what pump, since there is only one gland. This entire upgrade process is always evolving and being updated as needed. We are currently consolidating three other glands into one.

PARTS INVENTORY CHANGES

Other changes made at this time were the addition of the new technology parts to inventory. These parts included solid shafts (versus the sleeved shafts), bearing isolators (versus oil seals), bull's-eye oil indicator or column sight glasses (versus the constant level), the new glands and the 4° seal chambers (Photo 2). The old parts were earmarked so



Graph 1. Total plant pump repairs: REPPMP = repairs pump (one of a kind pumps), REPPMPP = repair positive displacement pumps, REPPMPV = repair vertical pumps, REPPMPC = repair centrifugal pumps, REPPMPTOT = total pump repairs



Graph 2. Total plant output using 1987 as the base quantity of one unit of output.

that they could be phased out of inventory and deleted at the appropriate time. This created a few minor problems because some of the pump OEMs also were modifying various parts because of the new ASME B73.1-1991 standard. It caused us to add parts that were being changed, and this created minor problems with associated parts. The confusion was minimized, however, by keeping communications open among the end user, vendors and OEMs. As mentioned earlier, this phase of the project is currently drawing to a close.

Because Reilly has a large number of pumps installed that are two generations old, and were designed when mechanical packing was the norm, they have inherent reliability deficiencies when fitted with mechanical seals. These pumps have long thin shafts and stuffing boxes. Due to their age and generation, the pump OEM will not be manufacturing upgraded parts such as seal chambers for these pumps. Because they represent a large part of the plant pump population, however, it was important to improve their reliability also. Two primary points were addressed: seal environment and shaft stability. After conferring with the OEM, it was agreed that the stuffing box could be bored. A new diameter for the stuffing box was determined, and the stuffing box was bored all the way through, giving us what we call a modified large bore with no throat restriction. Our local vendor for this pump brand sees to it that these parts now come into stores already bored through when they are ordered from the OEM. To increase the shaft stiffness, we opted for a solid sleeveless shaft.

This type of part modification is also being applied to some of our larger group two and group three size pumps, which represent a minority of the plant pump population. Because of their limited numbers, we stock only commonly used repair parts (shafts, bearings, mechanical seals). Large-dollar parts such as casings and seal chambers are listed as POR. By boring out the stuffing box and keeping the standard gland, we are able to receive some of the reliability benefits of the taper, or large bore seal chamber, while holding down repair costs. When there is need for more complete repair, these pumps also will be converted to updated technology.

MECHANICAL SEAL UPGRADES

Upgrading the mechanical seals was another important process for us. It went hand in hand with consolidating the mechanical seals. Specific failures were noted on one type of elastomer. Other failures specific to one plant were noticed on one of the seal hard faces. Also, with more stringent regulations approaching, we felt it would be beneficial to upgrade the seal/elastomer combinations used throughout the plant. Because our ANSI pumps generally were not pumping "easy" products, we decided that it would be best to use premium hardfaces and elastomers for all mechanical seals on ANSI pumps. Due to our consolidations, this affects only eight specific seals. The reasoning was that our average pump repair plantwide (ANSI and non-ANSI) was about $1,000 per repair. This covers any action ranging from no repair or minor repair to installing a new pump. Significant failures were noticed on encapsulated o-rings. Converting to an elastomer such as Kelrez added several hundred dollars to the cost of the mechanical seal. Still, if repairs were reduced, costs would decrease in the long run because there would be fewer failures (Graph 1). The new hardface used to replace the hardface experiencing problems in one specific plant would also work throughout the entire Indianapolis site. Therefore, this conversion was completed as well.

Through failure analysis, running dry was determined to be one of the major factors contributing to premature seal failure, even after pump upgrades were completed. To retain the gains discussed here, three op-tions were identified to address this problem. The first is to fit pump motors that can be turned off while the pumps are in service with power monitors. This pertains primarily to transfer pumps. Use of properly calibrated power monitors has all but eliminated seal failures at specific pump sites within our plant. These sites also have been correctly upgraded, which eliminates earlier root causes of pump failures before the power monitors were installed. The second option involves replacing the 4° taper seal chamber with a restricted throat large bore seal chamber. Swapping seal chambers is considered because of the pumpage properties in certain pumps. There are signs of running dry, which could actually be entrained air, or a phase change of the pumpage at the seal faces. The restricted throat large bore seal chamber with various flush plans allows us to obtain a higher pressure in the seal chamber and thus reduce or eliminate this problem. The third option to prevent pumps from running dry pertains to areas where the pump cannot be turned off automatically because of production needs. For some of these areas, we are installing gas barrier seals designed for the pumping requirement. Until recently the gas barrier seals required that a seal chamber be present, as they would not fit in the common stuffing boxes. This situation is changing and gas barrier seals are becoming available for the use in stuffing boxes (see Pumps and Systems, February 1996, pages 8 and 9). As of yet, we have ordered and received only one gas barrier seal, and we are waiting to install it when the current pump comes out for repair.

TRAINING

One aspect not yet mentioned is training for the maintenance work force. Workers were generally concerned about all of the changes. New brands of pumps were being used. Different types of technology were represented. More was expected of workers. All of these were real issues that had to be addressed. Although one-on-one discussions and shop meetings were held, the only way to ensure full communication of all of the changes was through formal training for the maintenance work force. Formal training assures us that all of our workers are up to date on new parts and procedures. Specific training areas have included ANSI pump rebuilding, mechanical seal installation and indicator reverse alignment.

Training in ANSI pump rebuilding covered the specific dial indicator checks that must be performed to ensure that the pump components are in the proper operating specification. Mechanical seal installation training noted the proper techniques to install both component and cartridge mechanical seals by the print, and by the stack method, and testing the seals on the bench before installing the pump in the plant. Indicator reverse alignment training covers how to align pumps properly and overcome common alignment problems such as soft foot, bolt binding and shim pack requirements.

OEM AND VENDOR TRAINING

Training the work force was only part of the project. Training the vendors and OEMs and maintaining good communication among them were critical parts of the training program. This meant keeping them up to date on our new pump requirements, allowing them to read and understand our plant engineering standards relating to the pumps, and ensuring that they held enough pump part inventory to cover our normal use. This was especially important in the beginning stages because being out of a specific new technology part needed for a repair, especially on a critical pump, would add fuel to the fire of the nay-sayers. This was true more so when we implemented the changes because many of the pump OEMs were not yet willing to stock taper and large bore seal chambers because there "was no demand" at the early stages of their introduction to the market. My reply was that we were demanding them, so please stock them. Because these new parts have proved to be important in increasing pump reliability, and there is indeed demand for them, most vendors now stock these parts.

When this program started, one of the major components of upgrading the ANSI pumps was the new seal chambers. Reilly first upgraded the bulk of its ANSI pumps to the new ANSI standard. We felt that this alone would increase the pump reliability. When the general population was updated, more specific problems would become clear and could be addressed. Looking at the three options that were available, it was decided that the taper bore seal chamber would best fit the overall needs of the company. Because of the many other factors that needed to be addressed to get the pumps, stores, and the vendors up to date on the pump upgrades being implemented, only one seal chamber was chosen Adding two seal chambers to the system would be detrimental to the entire process, and it would just add confusion until the project was more mature. Now that this general pump upgrade is accomplished, as mentioned earlier, we are looking at certain specific installations where the large bore seal chamber with the restricted throat might be of more benefit. This is primarily because of the makeup of the product, and not because of gland erosion. We have seen only one instance of gland erosion as a result of the cyclonic effect of solids and entrained air in a seal chamber, and the one instance was in clean acid. We have also had our mechanical seal OEM perform tests on various flushing arrangements using the taper bore seal chamber and our own gland (group one size pumps, 1800 rpm) with respect to particulate and entrained air in the seal chamber, and we have had interesting results that we can directly apply to field use. With conversion to the 4° taper bore seal chambers from stuffing boxes well established, if we now want to change from a taper bore to a large bore seal chamber, the gland, seal and other pump parts will fit. We need only change the type of seal chamber used for the specific pump in question.

FUTURE PLANS

Our ANSI pump upgrade program has been operating for nearly three years now. But along with upgrading the pump technology, we also had to address the root causes of failures. Reilly has begun to eliminate pipe strain proactively. Completely reinstalling problem pump sites has been accomplished with good results. Some of these sites incorporated both epoxy bases and epoxy grout, which give superior chemical resistance and vibration dampening (Photo 3). Steps have been taken to improve our grouting techniques, and we are completely phasing in epoxy grout. Compared to cement types of grout, this gives superior vibration dampening, chemical resistance and adhesion when the pump base and pad components are properly prepared. Also, bearings with tighter tolerances than we normally stock are being phased into stores.

Because of the long term results, there are now more requests from upper management to apply the pump reliability and upgrading techniques further – to specific problem pumps in each plant. This will include improving our vibration analysis program and reiterating the importance of aligning every pump that is worked on, no matter what the size. As pump reliability continues to improve, new "bad" pumps will make the list of pumps to be addressed as old ones are removed from the list because of improved reliability. This is part of the continuing refinement. By analyzing pump failure data, improvements can continue as long as there is a well thought out improvement process. Looking at the repair graph, it may appear that our reliability is improving at a slow rate. But in looking at the entire picture, our plant has been undergoing extensive growth through expansion and efficiency improvement. Our production has steadily increased, our pump population is increasing, new products are being brought to production while production of older core products are being reduced. These changes have created new challenges, and such conditions can have a dramatic effect on the pump and

Photo 3. Upgraded pump site includes epoxy base and grout for improved reliability.

mechanical seal population of any plant. When all these points are considered, however, there is significant satisfaction with the progress achieved in a pump reliability program of this type. ■

Joseph Dolniak has been the maintenance engineer at Reilly Industries for more than 6 years and is involved in reliability improvements of rotating equipment. In addition to producing alignment, seal installation and pump rebuilding training videos for Reilly, Joe has published several articles on ANSI pump reliability and has lectured at the Pump Users Symposium and the Pump Reliability and Maintenance Conference.

Selecting Mag Drive Pumps

Magnetic drive pumps offer irresistible force in sealless pumping.

By Robert C. Waterbury, Senior Editor

Industrial processes involving toxic, hazardous or environment-threatening chemicals often employ the magnetic drive pump (MDP) as a safe, sealless solution. But even though MDPs offer a simple answer to a common need, certain characteristics must be considered to select and apply them cost-effectively.

Kaz Ooka, president and founder of Ansimag, Inc., points out that magnetic drive pumps historically developed along two lines: metallic and non-metallic. The metallic designs traditionally were used in process or heavy-duty applications. But non-metallic pumps, once considered only for light duty applications, have moved up in power and size due to development of improved rare earth materials such as samarium-cobalt and neodymium-iron-boron.

Synchronous MDPs use rare earth magnets. Because they are affected by high temperatures, they often require special cooling provisions for applications in excess of 400°F. Eddy current MDPs employ a torque ring that is normally unaffected by temperatures found in hot oil heat transfer systems. They use a rotating assembly sealed by a containment shell. Power is transmitted by permanent magnets mechanically coupled to the driver rather than through motor windings.

Sealless MDPs prevent liquid leakage and eliminate common environmental concerns. They are also used to move liquids that crystallize upon contact with air, and the seal flush liquids or gases they employ help avoid contamination of process fluids. Yet they don't solve all seal-related problems. Two issues that MDPs still must address are minimum flow conditions and dry-running. Minimum flow rate is greatly affected by radial or thrust load on the bearings or pump shaft and the temperature rise. Dry-running is the most common cause of failure in MDPs and results in thermal damage to the metallic containment shell and/or in mechanical or thermal shock to the bearings and shaft. We shall discuss these in more detail.

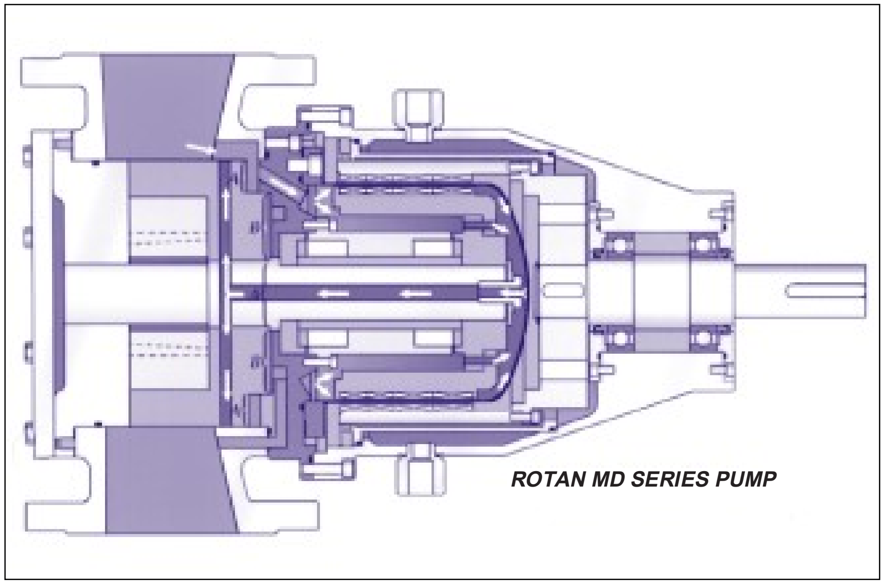


Figure 1. Rotan MD series magnetic drive pump

DESIGN CONSIDERATIONS

The inside of a sealless magnetic drive pump reveals a complex internal flow system that is difficult to model. However, the internal design holds the key to cooling the magnet drive and lubricating the bearings effectively and to the safe transport of solids. Engineers at HMD Seal/Less Pumps in East Sussex, England, which is affiliated with Sundstrand Kontro, identify five critical design elements:

1. the liquid end, comprising pump casing and impeller
2. a magnetic drive including an inner and outer magnet assembly and the containment shroud
3. internal support bearings
4. an internal feed system that circulates among 1, 2 and 3 above and is needed to cool the magnetic drive, lubricate the bearings and transport any solids in suspension
5. a power frame that comprises the external bearings supporting the outer magnet and the interface to the prime mover

INTERNAL FEED SYSTEM

Of these, HMD considers the internal support bearings and internal feed system the most critical, and yet perhaps least discussed. The feed system removes heat generated in the drive assembly by eddy current and viscous friction losses, and it lubricates the process lubricated bearings that support the loads exerted upon the rotor assembly.

Discharge to suction internal flow. To remove heat produced in the drive assembly, liquid is taken from the discharge of the pump and returned at a lower pressure point within the pump. The simplest and perhaps lowest cost system takes process liquid from the pump discharge and recirculates it to the pump suction end. Typically, process liquid leaves the exit of the impeller and returns to the magnetic drive through a hole in the rear casing plate. With this system the liquid always returns to the suction of the pump at a higher temperature than the bulk suction temperature. Thus, it is necessary to ensure that the hotter temperature liquid does not affect the NPSHR of the pump and vaporize as it returns to the impeller eye. This condition is often overlooked, according to HMD, if suppliers test using only water. Because water has a liquid with a high specific heat and gradual vapor pressure curve, test results using water alone may mask this potential problem.

Discharge to discharge internal flow. An alternative is to take the discharge liquid from the pump and return it at a point of pressure higher than suction. The internal feed system is similar to the discharge to suction feed system, except that the liquid is directed to a high pressure area in the casing typically behind the back shroud of the impeller. There are two advantages to this system. First, it eliminates the NPSHR problem. Second, pressure distribution within the drive is related to a high pressure area as opposed to suction and therefore reduces the possibility of vaporization within the drive. The main disadvantage is that the supplier must test to ensure that there is adequate pressure difference under all conditions to force sufficient flow through the drive. If not, low flow and excessive temperature rise will have the same effect as high flow and excessive pressure drop in causing vaporization.

PHOTO COURTESY OF ANSIMAG, INC.

Photo 1. Mag drive pump transfer of 93% sulfuric acid has operated nearly two years without repair

INTERNAL BEARINGS

Michael "Todd" Stevens, senior maintenance engineer at Hoechst Celanese Chemical in Houston, has analyzed bearing failures in magnetic drive pumps and offers some helpful observations. He suggests looking at the lubrication scheme for the roller/ball bearings and the shaft seal that prevents lubricant from entering the drive magnet section of the pump. When using wet sump lubrication, shaft failure will allow an oil level to build in the drive magnet section, causing a heat buildup. Normally under these conditions the pump will begin making noise and vibrate. If the oil level buildup is not detected, the temperature of the drive magnet section will increase and eventually cause a roller/ball bearing failure.

Stevens also suggests selecting silicon carbide rather than carbon as a main bearing material. It has better wear properties and will withstand most thermal shock without failure. Some suppliers even diamond coat silicon carbide journal bearings to withstand brief periods of dry-running without bearing failure.

Thrust bearings should be engineered so that full-face contact is achieved between the bearing and thrust runner. Failure to do so results in point loading, which can damage the bearing. Although Stevens recommends a fixed face bearing, he concedes that a floating-face spherical-seated thrust bearing could be acceptable if there were some way to secure the floating face at all times. Also, open impellers impose higher thrust loads than closed impellers. Thus, thrust loads must be either offset by balancing the impeller or absorbed by the thrust bearings. Lower thrust loads obviously mean increased bearing reliability.

INTERNAL HEAT GENERATION

Bearings are not the only mechanisms that generate heat in a mag drive pump. In fact, Stevens points out that most heat is actually caused by eddy current losses between the driven and drive magnets. With a permanent magnetic coupling, the driven rotor turns at the same speed as the drive rotor. The two magnets are separated by a containment shell. Heat generation is thus a function of pump rotation speed and containment shell construction. Containment shell materials such as 316 stainless steel generate more heat than Hastelloy C, Stevens notes, and Hastelloy C generates

PHOTO COURTESY OF ANSIMAG, INC.

Photo 2. A mag drive pump transferring 50% sodium hydroxide at a specialty chemical manufacturer in the southeastern U.S.

more heat than high performance plastics. The temperature of the containment shell directly between the two rotating magnets can easily rise above 750°F within 30 seconds of the onset of dry-running, and it can eventually reach nearly 1000°F. So internal fluid flow between the driven magnet rotor and the containment shell is needed to remove the heat.

Dry-running heats the shaft. If cool liquid is introduced at this time, however, the shaft and bearings may fail due to thermal shock. Ceramic materials can minimize these effects but offer widely varying thermal shock limits. Ooka says alumina ceramic can withstand only a 200°F thermal shock while sintered carbide offers resistance up to 600°F. Silicon carbide offers such high thermal shock resistance because it has a very low coefficient of thermal expansion combined with a very high thermal conductivity. This allows the material to equalize in temperature very quickly while exhibiting minimal thermal strain.

MDP MONITORING/INSTRUMENTATION

Safe, reliable operation of MDPs clearly depends not only on pump selection and installation, but on monitoring pump operating conditions. Following are some of the more common techniques and instrumentation.

Temperature monitoring. One way to monitor pump condition is to use a thermocouple or RTD (resistance thermal device). It can be positioned to monitor the temperature of the containment shell or placed in a thermowell to indicate the temperature of the fluid leaving the shell. Either way, it monitors the heat produced by the eddy current losses in the magnetic coupling as well as the bearing friction.The temperature can be used as an absolute or a differential measurement. Used as a differential temperature indicator, it is referenced to the pump suction fluid temperature and directly measures heat input to the fluid by the pump.

Flow protection. Minimum flow protection is normally provided by installing a flowmeter in the discharge line of each pump. Stevens points out that the minimum flow to protect against is either the thermal or the stable minimum flow, whichever is greater. It is only reliable, however, if no more than one pump is operating in a two-pump system. Otherwise, each pump must be individually instrumented and protected.

Low suction. Low suction vessel protection ensures that a pump will not run dry. According to Stevens, it is used in tank loading and offloading applications to ensure that the pump will not run dry or suffer from inadequate NPSHR. If a tank must be emptied following unloading, then a mag drive pump should not be used. Otherwise cavitation and subsequent failure of the thrust bearings and possibly the sleeve bearings could result.

Power monitoring. Technically, a power monitor can help guard against low flow, high flow, magnetic decoupling and dry-running. It measures the power consumed by the motor and thus responds quickly to load changes that could lead to mechanical damage. It obviously applies only to pumps driven by electric motors, however, and whether it is sensitive enough to distinguish between low flow and shutoff is questionable. In such cases a flowmeter or other device may be required as backup.

Vibration monitoring. Sleeve bearings typically run so smoothly (less than 0.1 in/sec overall) that periodic vibration monitoring has not proven useful in predicting failure. It has been successful, however, in predicting the failure of the drive magnet support bearings – normally roller or ball bearings. This could help ensure that the drive magnet does not contact the containment shell in case of a support bearing failure.

THE DECISION PROCESS

The Clean Air Act targets 179 liquids in setting limits for allowable chemical leakage into the air. The primary concern is for safety of humans and the environment. And magnetic drive pumps eliminate the dangers normally associated with seal leakage in mechanical pumps. But even though MDPs are purchased initially for safety reasons, many users are now specifying them for reasons of improved reliability, extended service life and longer mean time between maintenance and repair. In the long run they may prove more economical even though the initial cost is higher. Once the decision is made to purchase a magnetic drive pump, however, many questions must still be answered as part of the selection process. The following selection guide developed by Ansimag can help users tailor solutions to meet their specific needs.

Dimensions and design. Does the manufacturer make a pump in the design that you require? There are numerous configurations and standards including ANSI, ISO, API, DIN, etc.

Solution: Determine the design that you need and consider only those manufacturers that build that type of pump.

Vapor pressure of the liquid. Metal magnet drive pumps add heat to the process fluid due to losses in the magnetic coupling. The typical magnetic coupling is anywhere from 70% to 80% efficient. This inefficiency is translated into BTUs that enter the liquid as temperature. As much as 20°F can be added to the liquid that is lubricating the bushings. If the liquid vaporizes, the bushings will be starved for lubrication, and the pump will fail.

Solution: Ask the manufacturer to run a heat balance calculation to determine if the vapor pressure of the liquid will ever exceed the local pressure in the pump. If it does, it is the wrong pump. Also, consider a nonmetallic magnet drive pump with zero losses in the magnetic coupling. This will eliminate the possibility of vaporizing the liquid.

Solids. Because the bushings are lubricated by the process fluid, a "clean" liquid is required. Some pumps will handle more solids than others.

Solution: Ask manufacturers to state maximum limits and give references of applications handling similar solids content. A general rule of thumb is 5% by weight and 150 microns maximum. Flush systems are available from some manufacturers to increase solids handling capacity.

Hydraulic capacity. Some manufacturers have more capability with regard to head and flow than others because they offer more models.

Solution: Look at what models the manufacturer has available and ready to ship (not just planned as future products but currently available). Even if the manufacturer has what you need currently, consider also that you may want to add larger units at a later date. If he does not have them available, you lose commonality of parts and continuity of design.

Temperature. Magnet drive pumps have a variety of temperature capabilities.

Solution: Determine not only what temperature you will be operating at, but also what maximum temperature the pump could see due to excursions or future design revisions. Also, consider the effects of steam cleaning or heat tracing if you plan on either of these. Look at both the magnet capability as well as the material capability of the pump with regard to maximum temperatures.

Simplicity and ruggedness. These two items are critical now that maintenance staffs are slimmer. The less time spent on the pump the better – ruggedness of design is key. When maintenance is required, the simpler the better since the time spent on repair should be minimized and the risk of making an error should be reduced.

Solution: Ask the distributor or manufacturer to demonstrate the pump to determine if it is indeed simple and rugged. Viewing the pump in operation is critical because every manufacturer claims to have a simple and rugged design.

CAUTION: RISK AHEAD

As Stevens says, process engineering people design systems for normal operation and project engineers then use these flow requirements to purchase pumps. This is a normal method of sizing and purchasing a pump, but it is not always successful in purchasing and sizing a mag drive pump. An MDP is somewhat more sensitive to changes in pumping conditions than perhaps an ANSI design pump. Startup procedures, for example, do not always call for operation using the same fluid, pressure, temperature, specific gravity and viscosity indicated on the data sheet. Furthermore, pumps may be used for more than one operation, or they may be required to pump at widely differing flow rates during unit startup, operation and shutdown cycles. The greatest risks are always posed by conditions that fall outside the normal range of operation.

The pump system is designed to operate at the Best Efficiency Point (BEP). However, real world conditions demand more from a pump than a single BEP. A pump may be used to transfer fluid from tower to tower before unit startup to achieve a normal tower operating level. Similarly, it may be used to clear the unit in case of a unit trip, or it may be used as a spare for a completely different service via a jumper line. These scenarios must be postulated and the implications explored before installing a magnetic drive pump. Properly done, this exercise will provide an operating window of minimum/maximum values to be considered in the selection process.

Improper application is perhaps the most frequent cause of failure. Calculations of available versus required NPSH must be extremely accurate and compatible; otherwise, cavitation and pump failure will follow quickly. In addition, obvious practical considerations such as direction of motor rotation must not be overlooked. In a recent pump startup operation, three of nine initial failures were due to incorrect motor rotation.

SELECTED PRODUCTS AND APPLICATIONS

Ansimag. Different versions of the Ansimag K1516 mag drive pump are being used to move an extensive list of hazardous, corrosive and toxic chemicals. Applications noted most frequently in a new Ansimag case history publication involve such chemicals as hydrochloric acid, sulfuric acid, sodium hydroxide and sodium hypochlorite. The main reasons users give for switching to these pumps include zero leakage requirements, safety, elimination of seal problems and increased system uptime. The users include specialty chemical and petrochemical companies, pulp and paper processors, food and pharmaceutical companies, and steel, plastics and electronics manufacturers. In its list of user applications, Ansimag records user chemical concentrations ranging from 1-100%, flows from 1-500 gpm and TDH in feet from 5-250. Process temperatures generally range from -94° to 250°F.

Kontro/HMD. Kontro and HMD Seal/Less pumps are available in a wide range of capabilities designed for specific target applications. The A-Range mag drive pumps feature capacities to 2000 gpm, heads to 700 ft TDH, temperatures to 400°F and system pressures from full vacuum to 275 psig. Applications include toxic or hazardous liquids, high temperature vacuum distillations and liquids that are expensive or require controlled purity. The H-Range pumps offer capacities to 5000 gpm, heads to 700 ft TDH, temperatures to 750°F and system pressures from full vacuum to 300 psig. Applications include heat transfer fluids, molten solids and high temperature vacuum distillations. The API pumps range to 5000 gpm capacity, heads to 700 ft TDH, temperatures to 750°F and pressures to 580 psig. Applications are refinery and petrochemical services. The HSP series fills high system pressure requirements including: capacities to 2000 gpm, heads to 350 ft TDH, temperatures to 750°F and system pressures to 5000 psig. These pumps are used in nuclear, high pressure densitometer and pipeline detection systems. Finally, the self priming SP pumps are designed for truck and tank car offloading of hazardous chemicals. They accommodate capacities to 350 gpm, heads to 300 ft TDH, temperatures to 400°F, pressures from full vacuum to 150 psig and suction lift to 15 ft.

Micropump. Micropump offers precision fluid pumps and systems. Its Integral Series line is used in hemodialysis, chemical dosing, dispensing and filling, water purification, ink jet printing and laser/electronics cooling. Its distinctive features include motor, variable speed control via external signal, low power consumption, pressures to 100 psi and flow rates to 5.8 gpm. The pump head, brushless dc motor and electronic controller are integrated into a single compact unit with no mechanical seals, packing or leakage. The drive system is distinguished from conventional magnetic couplings by sealing the rotor inside the pump and driving it directly by the motor stator. The electronic controller, an integral part of the motor drive, accepts separate 0-5 Vdc or 4-20 mA signals used to adjust speed control.

DESMI/Rotan. Mag drive sealless pumps in the MD series offer capacities to 225 gpm, speed to 1750 rpm, differential pressure to 250 psi, temperature range to 500°F, suction lift to 15" Hg vacuum while priming and 25" Hg while pumping. MD pumps are recognized for their integral pump cooling system, dynamic axial balancing feature that reduces energy consumption and increases MTBM, a thrust control system that maintains correct running clearances and reversible pumping capability through changes in motor rotation. A patented system circulates the pumpage around the magnetic coupling for cooling. A special shaping at the rear of the rotor uses the hydraulic pressure itself to balance the liquid pressure dynamically on the rotor.

Maag Pump. MPS pumps from Maag are known for their use in very high pressure applications. Their operating conditions feature temperature ranges to 300°C, suction pressure to 16 bar vacuum, discharge pressure to 66 bar maximum, differential pressure to 50 bar maximum and viscosity to 1000 m Pa. They are available with either single or double containment shells.

Price Pump. The model CD 100MD from Price pump offers flows to 70 gpm and heads to 95 ft TDH. Incorporating 316 stainless steel or higher alloy materials, its standard configuration withstands pressures to 300 psig and temperatures to 350°F. Factory engineering is available for higher temperature and pressure ratings. Three magnet strengths are available for varied load conditions, and insrumentation options include power meter, temperature probe, and vibration, temperature and pressure switches.

Roth Pump. Magnetic drive pumps from Roth offer the advantages of regenerative turbine pumps in addition to a low NPSH feature. A floating, self-centered impeller is able to produce any level of differential pressure from 50 to 500 ft TDH. Maximum pressure with 316 stainless steel (standard) is 230 psi, or up to 360 psi with optional Hastelloy C material. A power factor sensor indicates both high and low load and reacts to upsets caused by blocked valves or vapor-bound conditions.

Dean Pump. The M300 conforms to the dimensional specifications of ANSI B73.1 and features a one-piece hydroformed containment shell. The seal for this shell is the only o-ring in the pump, and optional flow paths are offered to meet special situations. The RM5000 is a heavy-duty process pump with a centerline mounted refinery type pump wet end. This design offers higher head and capacity ranges along with optional flow paths. The containment shell is gasket sealed with no o-rings. The RMA5000 is a high temperature variation that can be air cooled to pumpage temperatures of 750°F. It features the external-external flush system and uses no o-rings.

Klaus Union. A wide variety of mag drive pumps conforming to ANSI, API, and DIN specifications is available from Klaus Union. They accommodate heads up to 575 ft, temperatures from -300°F to 840°F and design pressures to 5800 psig. Multistage high pressure pumps offer total delivery heads to 3300 ft. A patented double isolation shell is noteworthy among its offerings in addition to a pressure switch for continuos monitoring. Standard products offer designs geared to toxic duty, high temperature, high pressure and slurry applications. Products are available in single and multistage horizontal, vertical and even screw pump configurations. Additional heating and cooling is provided by jackets or coils, and forged casings and special isolation shells are used in high pressure designs.

PUTTING IT ALL TOGETHER

In Michael "Todd" Stevens' experience "...every mag drive pump failure has been the fault of some system design or operation upset. The mag drive pumps, when operated properly, have been very reliable." So how can one ensure the best continued operation of magnetic drive pumps?

The answer is not necessarily simple. First, Rob Plummer of Dean Pumps suggests that all mechanical seal problems really need to be resolved before even considering the purchase of a mag drive pump. His reasoning is that most of the materials used in mag drive pumps are the same used in mechanical seals — just in a different manner. Furthermore, a mechanical seal is tolerant of certain design features such as an elbow on the suction of an end suction pump. A mag drive pump installation, on the other hand, requires a straight pipe with smooth flow. And of course there are the dry-running and eddy current heat problems that we have discussed.

Having considered these issues and still electing to install a mag drive pump, one must closely analyze all possible operating conditions (including startup and shutdown) and pass that information on to the supplier. This involves specifying such characteristics as type of chemical(s), specific gravity, viscosity, specific heat, vapor pressure/temperature rise, percentage of solids and dissolved gases. Describe all possible process operating conditions. Protect the pump by using at least a temperature sensor to indicate the fluid heat as it leaves the containment shell and a power monitor to indicate internal operating conditions. Enlist the assistance of maintenance personnel in system design, pump selection, startup, training and general operation. And be sure to check obvious details such as the direction of motor rotation! ■

Operation Protection for Mag Drive Pumps

Learn how to avoid dry or semi-dry running conditions, which can lead to damage.

By Kaz Ooka and Manfred Klein

Magnetic drive centrifugal pumps are products of an evolving technology utilizing new materials, stronger magnets and new concepts. Beginning shortly after World War II, the mag-drive concept developed along two paths – namely, metallic and non-metallic pumps. Metallic designs have been utilized primarily as process or heavy duty pumps, notably in Europe. In earlier years, non-metallic mag drives were usually considered applicable in light duty situations only – fish tank pumps drawing 100 watts or less, for example. With the development of rare earth magnet materials such as samarium-cobalt and neodymium-iron-boron, however, the size and power of non-metallic designs have been greatly improved over the last ten years.

This rapid increase in magnet strength has allowed for a corresponding reduction in the size and weight of the magnetic coupling. Photo 1 shows an industrial non-metallic mag-drive pump. This machine is clearly a great improvement over the original fish tank pump.

As more magnetic drive pumps have been applied in the process industries to solve increasingly complex problems, some confusion has developed among users accustomed to sealed centrifugal pumps such as those described in the ANSI/ASME B73.1 standard: Specification for Horizontal End Suction Centrifugal Pumps for Chemical Process. Many users have believed that due to its sealless cocnstruction the magnetic drive pump can solve all seal related problems. Sealless pumps are ideal for preventing liquid leakage and mitigating the associated environmental concerns. They also work well in the pumping of liquids that crystallize upon contact with air, and they avoid contamination of process fluids by seal flush liquids or gases. And they are excellent for pumping corrosive liquids. However, several issues need to be addressed, the most important of which are minimum flow conditions and dry-running. While these issues are relevant to traditional sealed pumps as well, this article will concentrate on their effects on mag-drive pumps.

MINIMUM FLOW

Two factors determine minimum flow rate: radial or thrust load on the bearings or shaft of a pump and temperature rise. This discussion applies specifically to single stage, low specific speed (400-800 rpm*$\sqrt{\text{gpm}}/H^{3/4}$) mag-drive pumps in the 1-30 hp range. Low flow operations of higher specific speed pumps can lead to additional problems such as suction recirculation, which will not be discussed in this article.

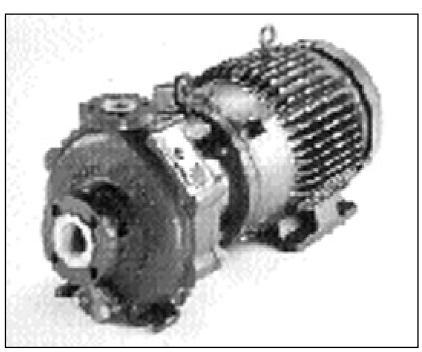


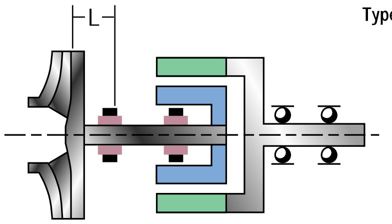
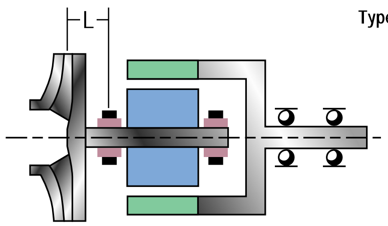
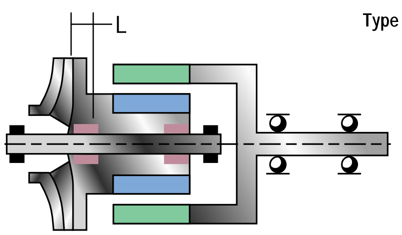
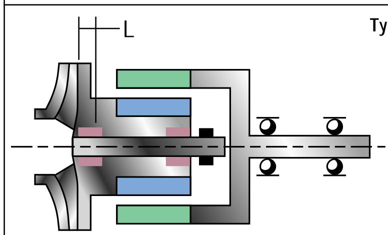
PHOTO COURTESY OF ANSIMAG, INC.

Photo 1. Example of an industrial non-metallic magnetic drive pump

Radial Load on the Bearings and Shaft

When a standard centrifugal pump operates off its best efficiency point, the impeller experiences higher radial loads due to hydraulic unbalance in the casing. The radial load becomes severe when the pump is operated near shut-off. In a standard sealed pump the loads on the bearings are much higher than the load on the impeller. Typically, they are two times higher. This is a consequence of the long overhang distance between the impeller and the first bearing, which is necessary to provide adequate space for the seal.

In contrast, the first bearing of a sealless pump is located very close to the impeller. This results in bearing loads only slightly greater than the impeller loads. Figure 1 shows typical mag-drive pump bearing arrangements. The dimension L denotes the span between the impeller and the first bearing. Each layout has its own strengths and weaknesses, but all provide for a bearing close to the impeller. In Type 3 the bearing rotates with the impeller and consequently is very close to the load. With this design the overhang distance, L, can approach zero. The Type 4 design also has the bearing close to the impeller, but it uses a shaft cantilevered from the containment shell. This design is typically used only in small pumps because of the stresses at the shaft-to-containment shell connection.

Basic Construction	Description
Type 1 	**Impeller and magnet separate, connected via rotating shaft. Bearings stationary and supported in a bushing support.** 1. Front bearing has higher load capability than type 4. 2. High manufacturing cost and more complex than type 3 and 4. 3. Popular in metallic pumps.
Type 2 	**Impeller and magnet separate, connected via rotating shaft. Stationary front bearing behind impeller. Stationary rear bearing in containment shell.** 1. Front bearing has higher load capability than type 1. 2. Longer span between bearings than type 1, 3 and 4. 3. More chance of dry running or vapor lock at rear bearing, since difficult to lubricate.
Type 3 	**Impeller and magnet a single unit. Shaft is stationary and supported at both ends. Bearings rotate with impeller.** 1. Front bearing has highest load capability of all 4 types. 2. Simple design 3. Lower stress concentration on plastic parts than type 4. 4. Non-clogging fluid paths. 5. Popular in non-metallic pumps.
Type 4 	**Impeller and magnet a single unit. Stationary cantilever shaft. Bearings rotate with impeller.** 1. Simple design 2. Non-clog fluid paths. 3. Containment shell (rear casing) supports shaft bending moment. Shaft socket in containment shell must be very strong and resist creep if non-metallic.

• Black squares represent the mounting locations of the stationary bearings (type 1 and 2) or the mounting locations of the stationary shaft (type 3 and 4).
• Red rectangles represent product lubricated bearing locations (type 1-4).

A mag-drive pump, therefore, has a significant advantage in terms of impeller deflection, and it is more resistant to the radial loads encountered during low flow operation. For example, an impeller deflection of 0.005" can be a problem for seal life but is not a concern in most sealless pump designs.

It is important that mag-drive pumps operating at low flows be designed with bearings able to handle consistently higher radial loads. The design must also provide for adequate cooling and lubrication flow to the bearings at these low flow rates. One reason for the increasing acceptance of mag-drive pumps in low flow rate service is that product lubricated bearings can be manufactured from materials such as pure sintered silicon carbide that effectively provide zero wear for the life of the pump.

Temperature Rise

The temperature rise (°F) of the liquid passing through a pump is given by

$$\Delta T = \frac{H}{778 \cdot C_p \cdot \eta}$$

where H is head in feet, Cp is specific heat in Btu/lb °F, and h is efficiency, written as a decimal value. This equation assumes that all losses result in heat that remains in the liquid. To predict the temperature rise in the fluid accurately, the efficiency factor in this equation must include all losses. The three types of efficiency loss in magnetic drive pumps are: (1) hydraulic, leakage and friction losses, (2) radial and thrust bearing friction, and (3) eddy currents in the containment shell (rear casing).

(1) Hydraulic losses are inherent in all centrifugal pumps. However, the efficiency of impellers is improving with better design, and, with the use of more powerful magnets, the size of the inner magnet assemblies and their resultant fluid friction losses are shrinking. Remember, though, that since the efficiency will always be zero at shut-off, the temperature rise will rapidly increase as the flow rate approaches zero.

(2) Radial and thrust bearing friction account for the smallest portion of efficiency losses. For example, a 1 1/2 x1x6 ANSI pump will lose 0.1-0.26 hp to bearing friction. This represents only 3-9% of the shut-off power of the pump.

(3) Eddy current losses. In metallic magnetic drive pumps the containment shells are usually made of a nickel alloy (e.g., Hastelloy®) or stainless steel. Both are electrical conductors. These stationary shells are placed between the two sets of rotating magnets within their powerful magnetic fields. When a magnetic field moves past a conductor such as a containment shell, eddy currents are generated. Generally, the eddy current loss for a 0.060”

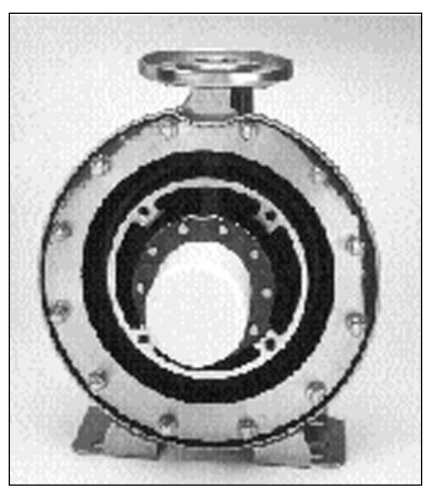


Photo 2. A Zirconia containment shell

thick nickel alloy containment shell is about 15% of the magnetic coupling rating. If a 10 hp coupling is used, about 1.5 hp or 1.1 kW of power is directly transferred to the liquid at the containment shell. This would be equivalent to equipping the pump with a 1.1 kW heater. Additionally, the heat generated in the containment shell remains essentially constant regardless of pump flow rate. This 1.1kW can increase the temperature of water flowing at 1 gpm by approximately 7.5°F. Cooling of the containment shell and prevention of flashing are the primary constraining factors for metallic mag-drive pumps at low flow rates.

Performance curves for a non-metallic mag-drive pump are shown in Figure 2. The efficiency curve includes all losses. As with all pump performance curves, the efficiency is zero at zero flow. If we take values from the efficiency and TDH curves and insert them into the temperature rise equation, given above, we can calculate the temperature rise for this pump. Figure 3 illustrates the temperature rise for flow rates of 0-20 gpm. If a maximum temperature rise of 10°F is specified, then this pump can be operated at 2 gpm for water. Slightly higher minimum flow rates may be required for other liquids due to their lower heat capacities. Figure 4 provides the minimum continuous flow rates for a series of three non-metallic mag-drive pumps. These flows are based on a temperature rise limit of 10°F for pumping water.

Dry and Semi-dry Running

Dry running and semi-dry running are the most common causes of failures in magnetic drive pumps. Damage can be caused by excessive heating in metallic containment shells and/or by mechanical or thermal shock at the bearings and shaft.

If a metallic containment shell is used and a pump runs dry, the shell will be rapidly heated by eddy currents, with temperatures rising to nearly 1000°F. This temperature rise is so quick that even jogging the pump (e.g., to check rotation) is not recommended

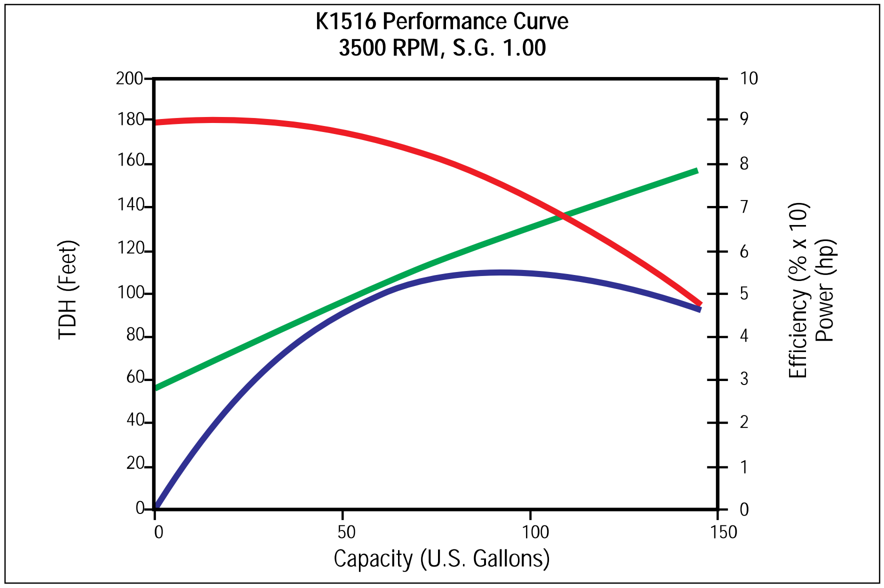


Figure 2. Performance curves for ANSI 1 1/2 x 1 x 6 size non-metallic magnetic drive pump

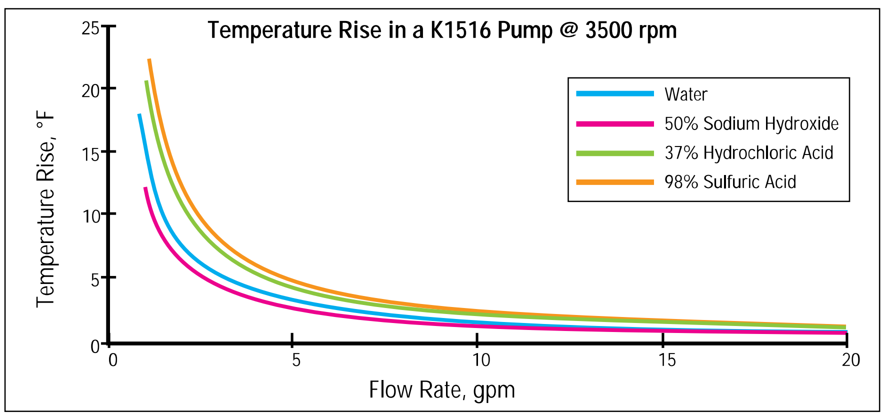


Figure 3. Temperature rise in same ANSI pump

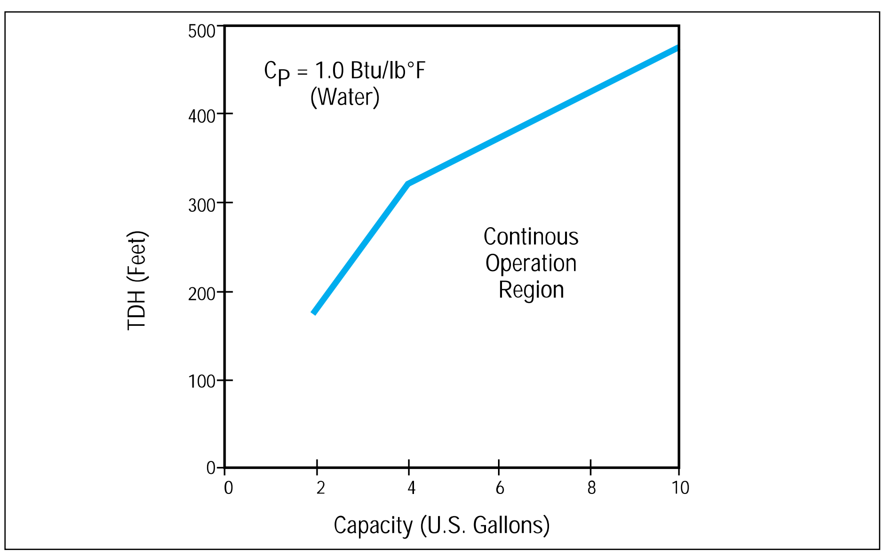


Figure 4. Minimum continuous flow for non-metallic magnetic drive pumps

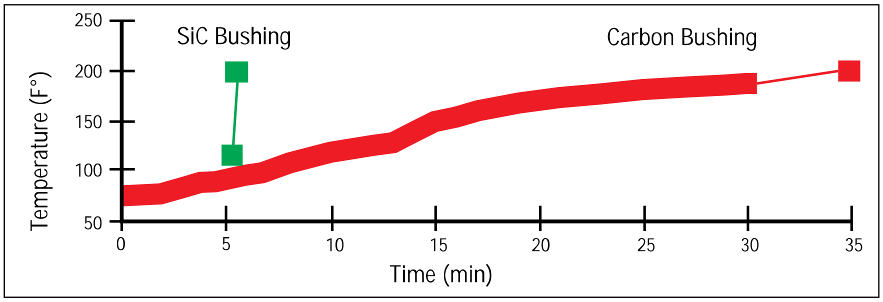


Figure 5. Shaft temperature U.S. time

when there is no liquid in the pump. A few minutes would be long enough to demagnetize the magnets completely and ruin the shell.

Non-metallic containment shells generate no eddy currents and therefore no heat, and so do not experience this kind of failure. Some manufacturers provide non-metallic containment shells for their metallic pumps. (Photo 2.) Typical materials of construction are zirconia ceramics or plastics.

The critical component with respect to dry running in pumps with non-metallic containment shells are the bearings. The bearings are designed to operate while wetted by the pumpage and will exhibit a very low coefficient of friction in this condition. Some material combinations such as a carbon bushing on sintered silicon carbide will maintain a relatively low coefficient of friction even when dry. Such combinations are thus more forgiving in instances of dry running. Sintered silicon carbide against sintered silicon carbide is the best bearing material when wet, but it will typically show a sudden increase in friction level when bone dry.

The resultant increase in shaft temperature is shown in Figure 5. The temperatures were measured using a non-metallic mag-drive in a broken suction application. Since the carbon/graphite bushing has a lower coefficient of friction when dry, the pump shaft temperature increase gradually. The silicon carbide bushing exhibits the same temperature rise in the beginning, but then an almost instantaneous temperature rise will occur. Both material combinations provide plenty of time to shut down the pump before damage occurs provided proper monitoring equipment has been installed.

During dry running the pump bearings and shaft will become hot. If while hot, cool liquid is reintroduced to the pump, the bearings and shaft may fail due to thermal shock. Different ceramic materials have far different thermal shock limits. For example, alumina ceramic can survive only a 200°F thermal shock while sintered silicon carbide is safe up to 600°F. Silicon carbide can resist thermal shock because it has a very low coefficient of thermal expansion combined with a very high thermal conductivity. This allows the material to equalize in temperature very quickly while undergoing very small thermal strains. Some manufacturers provide other bushing/shaft combinations. However, in an actual service such as unloading, the coefficient of friction is unpredictable. In a laboratory test, the carbon bushing/silicon carbide shaft combination allows extended periods of dry running if the bushing is new and clean, or if the pump suction and discharge are open and air flows freely to cool the shaft. In practical operation, dry running is not recommended in magnetic drive pumps because a) the majority of magnetic drive pumps are still using metallic containment shells, b) residue on the shaft and bushing from a previous pumping operation can change the friction coefficient, c) abrasive matter in the fluid can alter the bushing and shaft surface finish, d) vibration from other equipment may match natural frequency of the pump rotating parts, and e) piping usually restricts cooling by air flow through pump.

Monitoring, which is strongly recommended, can include a pressure switch, flow switch, electrical current monitor or electrical power monitor. A pressure switch or flow switch is reliable as long as liquid is clean and extra electrical wiring in the pump field is feasible. An electrical current monitor or electrical power monitor is very popular since no extra wiring into the pump field is required and the device can be easily placed outside the hazardous area. Exercise caution if the motor is selected for the pump's maximum requirements but actual operation is at a very low flow rate. When a motor runs at less than 50% of rated load, current monitoring will not be sufficiently sensitive. This is due to the characteristics of a motor. However, motor input power remains sensitive below 50% of rated load. For this reason sensing the performance changes between a pump operating at a low flow condition and the same pump running dry will require a power monitor. These devices can be conveniently installed at the motor starter box.

However, power monitoring may not be sufficiently sensitive to detect the changes between very low flow operation and shut-off. These situations will require the use of a flowmeter or differential fluid temperature measurement between the pump inlet and discharge.

CONCLUSION

It is essential for pump users to determine if the minimum flow rate specified by the pump manufacturer is constrained by a radial load on an impeller or by heat rise. In the case of a radial load limitation, the specific gravity of the pumped liquid must be taken into account. If heat rise is the major limiting factor, the specific heat of liquid must be taken into consideration. These two factors are unrelated. ■

Kaz Ooka is President and co-founder of Ansimag, Inc. He has a B.S. in Mechanical Engineering from Tokyo Denki University. Manfred Klein is Chief Design Engineer for Ansimag, Inc.. He holds a Bachelors and Masters degree in Mechanical Engineering from Carleton University in Ottawa, Canada.

Sealless Options Optimize Solutions

If zero leakage is the goal, sealless pump options can help tailor-make the solution.

By Robert C. Waterbury, Senior Editor

Leak free? Zero emissions? Hermetic sealing? When environmental protection needs and hazardous substances raise questions, sealless centrifugal pumps often provide the answers. The term "sealless" generally describes a class of pumps that do not allow fluid leakage into the environment. And although this de-scription covers a number of pump types, the two most prominent examples are the canned motor pump (CMP) and the magnetic drive pump (MDP).

According to David Carr, marketing specialist at Sundstrand Fluid Handling, neither the CMP nor MDP requires a dynamic shaft seal to contain the pumped fluid. Instead, a stationary containment shroud isolates the pumpage from the ambient environment. In the case of the magnetic drive pump, power is transmitted across the stationary shroud through magnetic lines of flux. These induce rotation in the impeller shaft and thus avoid the leakage that is a normal byproduct of mechanical face seals.

MAGNETIC DRIVE PUMPS

Magnetic drive pumps are simply centrifugal pumps with an integral magnetic coupling between the driver and the liquid end. The magnetic coupling replaces the seal chamber or stuffing box so that the liquid end is hermetically sealed. The mechanical seal or packing is eliminated, and the only seal is a stationary gasket or O-ring.

The two main subgroups of magnetic drive pumps are the synchronous and eddy current designs. The synchronous type typically uses rare earth magnets. Because these can be adversely affected by temperatures in excess of 400°F, special auxiliary cooling provisions are often required for such applications.

The eddy current designs, according to the Kontro Co. of Sundstrand Fluid Handling, employ a torque ring that is normally unaffected by temperatures experienced in hot oil heat transfer systems. Eddy current sealless pumps feature a rotating assembly that is sealed by a containment shell. Power is transmitted through permanent magnets mechanically coupled to the driver, rather than through motor windings. Cooling water is not required because the outer magnets and antifriction bearings are remotely located.

A torque ring integrally connected to the impeller shaft forms the inner rotating element. This assembly is supported by journal bearings that are lubricated using recirculating hot oil. No precooling is required. The recirculated flow also removes the heat generated from the magnetic coupling losses.

CANNED MOTOR PUMPS

The canned motor pump consists of an induction motor whose rotor is integral with the impeller shaft. A thin metallic can mechanically separates the rotor from the windings and seals the pumpage from the stator. The rotor is supported by journal bearings that are lubricated by recirculated hot oil. Recirculation is provided by an external tube that feeds hot oil into the back end of the pump.

CMP features may include a two-bearing single shaft arrangement, dry stator and sealed junction box, both primary fluid containment and secondary leak containment shell, a controlled bearing operating environment with monitor and a minimum of required components.

Photo 1. Chempump NC Series canned motor pumps feature an electronic diagnostic system.

The benefits or advantages of using CMPs include: no shaft seals and no external leak paths, no buffer pots, no buffer or process fluid leakage disposal, no coupling or alignment problems, low noise levels and low maintenance costs.

Because high temperatures routinely encountered in hot oil systems normally exceed the limits of the motor, CMPs are designed with an integral cooling water heat exchanger. This exchanger surrounds the outside wall of the stator and removes the heat associated with motor losses. The recirculation tube also makes a single pass inside the shell of the exchanger. Thus, the recirculated pumpage is precooled to a safe temperature before entering the back end of the pump once again.

CMP AND MDP FEATURES AND OPTIONS

The first and most obvious job in selecting a sealless pump is to determine the basic system requirements and operating parameters. Operating criteria such as high or low temperatures, fluid capacities in gpm, system pressures, abrasive slurries and corrosive agents will determine not only the size and capacity of the sealless pump, but also any special construction requirements.

Once these criteria are identified, it is possible to look at the comparative features and options offered by canned motor pumps and magnetic drive pumps. Much of the following material is adapted from information compiled by Sundstrand Fluid Handling.

Containment. This is a major consideration if there are health and/or safety concerns. Several CMP manufacturers offer some type of double containment. This might consist of welded primary containment and hermetically sealed secondary containment. For efficiency reasons few MDP manufacturers offer secondary containment.

Most MDP manufacturers, however, do offer secondary control in the form of mechanical seals on the OMR shaft penetration. Also, an MDP typically provides a thicker containment shell. This offers more resistance to corrosive or mechanical penetration. CMP primary liner thicknesses normally range from 0.022"-0.035". MDP containment shells range from 0.029"-0.060".

These differences are offset somewhat by the ability to monitor bearing and internal rotor positions. CMPs, by design, are easier to monitor than MDPs. This means that they are perhaps more apt to detect extreme bearing wear prior to containment shell contact or penetration.

Solids/slurry handling. Standard CMP and MDP products will accommodate moderate amounts of solids. Optional designs for both are capable of handling higher slurry concentrations. With a slurry modification, CMPs can deal with concentrations in the range handled by most standard centrifugal pumps. The MDP, however, cannot be isolated as easily as a CMP and thus requires a different internal bearing-mag coupling flow path than offered normally.

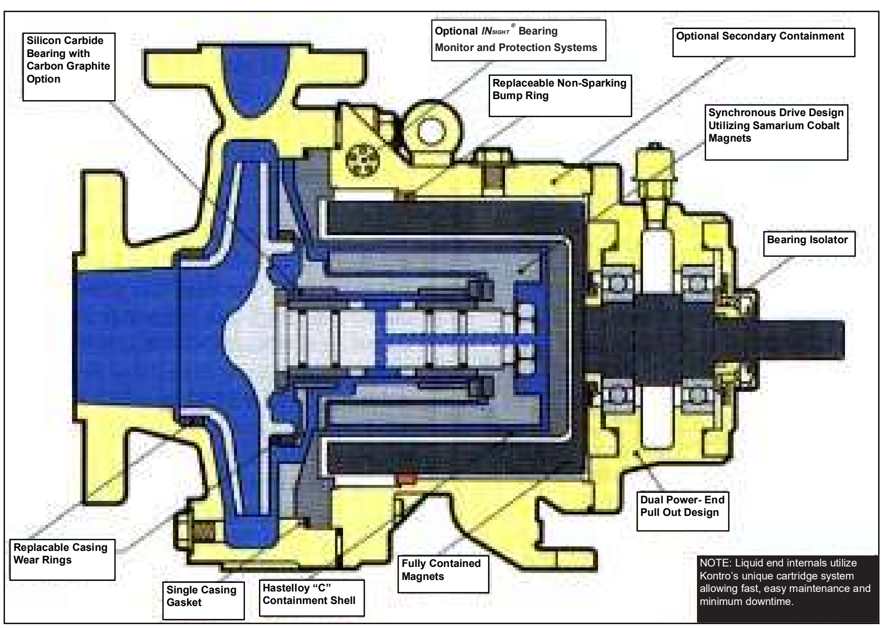


Figure 1. Today's generation of magnetic drive pumps offers advanced features and materials of construction, as shown in this example of Kontro's A-Range design.

Heat input. Heat is added by hydraulic and drive inefficiencies in either type of sealless pump. For our purposes we can consider the hydraulic efficiencies to be the same for both types.

Thus, for CMP design, a high efficiency motor is 80-85% efficient. But due to the ease in isolating the motor area, CMPs can offer optional configurations to control the fluid temperature, pressure or both to prevent product vaporization. However, because a CMP motor tends to be one large insulated mass once the unit is shut down, it allows heat soak to occur. This permits the process fluid to be heated to higher temperatures at potentially lower pressures than during normal operation. The result could lead to flashing of the contained fluid. Also, the pump may vapor lock if restarted.

Synchronous MDPs are known to be adversely affected by temperatures above 400°F. But a synchronous drive is also more efficient than an eddy current drive. A nonmetallic shell is more efficient than a metallic shell. The synchronous drive MDP with a metallic shell has a drive efficiency of 80-85%, which is comparable to the CMP. However, it lacks the insulated mass around the containment shell and is less susceptible to heat soak.

Cooling requirements. Both CMP and MDP can be configured for operation at elevated temperatures. Different designs require temperature limitations based upon specific components and therefore must be evaluated on a case-by-case basis.

Jacketing. Steam or hot oil jackets can be added to either type of pump to maintain the proper product temperature. This ensures that the pumpage will be liquid at all times. CMP designs allow jacketing of the pump case, stator and rear bearing housing. Most MDPs can have jacketed casings that add heat in the area of the containment shell without fully encapsulating it.

High suction pressure. Optional designs are available for both CMPs and MDPs in applications where the suction pressure and maximum allowable working pressure requirements exceed standard pressure design capability.

Modifications in the CMP design for high pressure applications include the use of primary containment shell backing rings, thicker secondary containment shells, additional pressure containment bolting and high pressure terminal plates. The double containment feature of CMPs is maintained even for high pressure.

Modifying the MDP for high pressure applications includes use of a thicker containment shell and additional pressure containment bolting. Typically, the CMP is more efficient than the MDP in high pressure applications due to its increased primary containment thickness.With thicker primary containment, the MDP shows greater hysteresis losses (inefficiency) than the CMP.

ANSI conformance. Although some CMPs are available with ANSI dimensions or hydraulics, most are not. Many MDPs, however, offer ANSI dimensions and hydraulics. Further, most manufacturers of sealed and sealless ANSI pumps offer interchangeability between their pumps' wet ends and bearing frames.

Efficiency. CMP wire-to-water efficiency is defined as hydraulic efficiency times motor efficiency. As mentioned in the discussion of heat input, a typical CMP motor is 80-85% efficient.

MDP wire-to-water efficiency is defined as hydraulic efficiency times motor efficiency times coupling efficiency. Again, the magnetic coupling is 80-85% efficient. However, containment shell metallurgy (or lack thereof) and magnetic coupling type play major roles in coupling efficiency.

Wet end hardware, however, further complicates the efficiency discussion. Impeller and casing geometry play a vital role in hydraulic efficiency. A Barske design (open radial blade impeller and diffuser discharge) is typically more efficient in low flow/high head hydraulics. A Francis design impeller, enclosed with backswept vanes and an increasing radius volute, is more efficient at moderate-to-high flows with low-to-medium heads. The user, therefore, must thoroughly evaluate wire-to-water efficiency for an accurate comparison.

Internal clearances. Clearances between bearing ID and mating surfaces are typically the same (0.003-0.007"), although differences between other rotating parts do occur. Typical CMP clearances between the rotor and stator liners vary by manufacturer from 0.018-0.044" radially. MDP clearances between the inner magnetic ring and containment shell are usually 0.030-0.045" radially. This larger clearance gives the MDP the advantage of allowing more bearing wear prior to containment shell contact.

Bearing monitoring. Although different monitoring methods are available for MDPs and CMPs, the design of the CMP lends itself more readily to real bearing monitoring. A CMP bearing monitor can provide axial, radial and liner corrosive wear indication. Bearing monitoring features vary by manufacturer, and care must be exercised in selecting a sealless pump manufacturer with the desired monitoring features.

Bearing material options. Most sealless manufacturers offer a hard bearing – typically silicon carbide – running on silicon carbide or tungsten carbide. Alternatively, a soft bearing is normally carbon graphite running on stainless steel. Because the materials are the same, the mounting and lubrication plans assume greater importance.

Number of bearings. CMPs have two bearings; MDPs have six (including the motor). Most experts feel that fewer is better. However, a properly designed, applied and operated MDP will last just as long as a properly designed, applied and operated CMP.

Noise. CMPs have no motor fan and thus produce less noise.

Space. A CMP with its integral pump and motor occupies less real estate (has a smaller footprint) than a comparable MDP with its pump, coupling and motor. However, close coupled MDPs that may require no more space than a CMP are available.

Orientation. CMPs can be mounted vertically or even hung on a pipe. Few MDPs can be mounted vertically.

Temperature fluctuations. Both CMPs and MDPs can effectively handle wide temperature variations in the pumpage. Manufacturers should be consulted in evaluating specific operating conditions, however.

Initial cost. For general duty service (clean, cool, non-volatile) an MDP is usually lower in price than a CMP. As the application becomes more tortuous and difficult, however, pricing reaches parity and may even favor the CMP.

Installation cost. A standard CMP or close coupled MDP with no auxiliaries (coolers, flush systems, etc.) will have lower installation costs than a frame mounted MDP. This is due to the smaller footprint and minimal foundation requirements.When optional CMP configurations are considered, the cost of cooling lines, reverse circulation lines and flush lines may equal the cost of an MDP. Instrumentation also adds cost to both types of installation, so situations must be evaluated individually.

Simplicity. Although the number of pump parts may vary by manufacturer, CMPs generally have fewer parts. As optional configurations and more parts are added, however, they can easily equal the number of parts in an MDP. Items adding parts and complexity include auxiliary impellers, tilting washers and heat exchangers.

While some MDP manufacturers may add such equipment, most offer only two basic varieties: discharge-to-suction recirculation and discharge-to-discharge recirculation. Adding a standard electric induction motor as well helps some users accept the sealless technology.

Field serviceability. CMPs have bearing monitors to predict the need for bearing changeout prior to containment breach. CMP bearings are typically easy to replace.

MDPs have thicker containment shells to prevent breach. However, they have limited ability to monitor lubricated bearings to determine when routine maintenance is required. Ease of bearing replacement also varies with manufacturer.

If a primary containment shell is breached, even the most serviceable CMP must be sent outside to be "recanned." CMP manufacturers have countered by offering spare rotor/stator sets at 60-80% of the cost of a new pump.

A containment shell breach of an MDP usually results in the purchase of a spare process-wetted rotating assembly and a containment shell. Pricing is typically 60-80% of the cost of a new pump.

Coupling alignment and venting. The typical CMP design has the impeller mounted directly on the motor shaft inside the containment area. Coupling and coupling alignment problems are therefore nonexistent.

Most MDP installations use frame mounted motors that require coupling alignment. Many MDP suppliers, however, offer close-coupled designs that nearly eliminate coupling alignment problems.

Both MDP and CMP designs are typically self-venting back to the process piping and do not require additional external lines.

PRODUCT OFFERINGS

Because sealless pumps are often used in severe duty applications, monitoring and diagnostic options are important aspects of many installations.

Monitoring and diagnostics. A monitoring and diagnostic system called IntelliSense from Crane Chempump displays the position of the entire rotating assembly in its NC Series of canned motor pumps. The system provides precise, real-time wear data with an accuracy of 0.001" radial and 0.002" axial. This information enables users to plan simple parts replacement long before costly failure occurs. The entire diagnostic system is isolated from the process fluid and is therefore not a sacrificial part that requires replacement. The display unit can be hand held, mounted near the pump at eye level for easy viewing, or located in a remote control station.

Teikoku canned motor pumps offer the Teikoku Rotary Guardian (TRG), which is available either as a built-in meter or as a remote panel meter. It monitors the running clearances between the stator and rotor, bearing condition and rate of wear, reverse rotation, loss of phase and short circuit conditions.

Sundyne offers a remote mechanical bearing monitor. It allows remote alarm or shutdown of the pump signalling a need for bearing maintenance to protect the motor from damage. A dry operation protection meter is also available. In services such as tank unloading, the meter detects low load in time to shut down the motor to prevent dry operation. A similar system called INsight from Kontro provides radial and axial bearing wear monitoring in its line of magnetic drive pumps. Kontro also offers a port for thermocouple or RTD temperature monitoring, an amperage monitoring system to protect against cavitation or dry running, a sensor port for vibration monitoring as well as liquid and pressure sensing options.

Jacketing. Complete or partial jacketing of the pump case, motor stator and rear bearing housing is offered by Sundyne to control temperature when heating or cooling is required. Teikoku and other manufacturers offer certain pump lines with built-in heat exchangers as well as motor cooling jackets for high temperature applications.

Crane Chempump offers three lines of high temperature CMPs:

- GH – external circulation for fluid temperatures to 650°F without liquid cooling
- CH – internal circulation for fluid temperatures to 650°F without liquid cooling
- GT – motor isolation with external cooling for fluid temperatures to 1000°F

The motors in the GH and CH models incorporate a high-temperature insulation system. The motor in the GT model employs an integral liquid-cooled heat exchanger.

Inducers. Teikoku, Sundyne, Crane Chempump and other manufacturers offer a wide selection of inducers to meet net positive suction head (NPSH) requirements. It is not unusual, according to Teikoku, for one of its pumps to operate at a specific suction speed of more than 13,000.

Casings and impellers. Single and double volute casings are options used in CMPs from Buffalo Can-O-Matic and other manufacturers, depending upon pump size and service requirements. Likewise, open and closed impeller designs can be specified as options on many models depending upon operating requirements.

Bearings. In addition to various manufacturer options, there are special bearing constructions to take into account. Buffalo Can-O-Matic, for instance, features spring-loaded, self-adjusting and self-lubricating tapered carbon graphite motor bearings. These are designed to distribute and automatically compensate for bearing wear. This maintains concentric rotation of the rotor-impeller assembly, prevents mechanical contact between the rotor can and stator can, and improves both operation and maintenance. Segmented bearings of self-lubricating carbon gra-phite construction are used on large Can-O-Matic pump motors. Again, they are spring loaded to compensate for wear.

Other construction options. Most manufacturers offer optional construction materials for certain components. Buffalo Pumps, for instance, provides casings in a choice of ductile iron or 316 stainless steel. Likewise, impellers are available in a choice of cast iron, bronze or 316 stainless steel according to the requirements of the application.

Chempump lists hardened rotor

journals, pressurized circulation systems, sealless junction boxes and explosion-proof CMP designs among the options in its G Series line. And Teikoku provides a large number of adapters to accommodate many different pump and motor combinations.

DRIVING FORCES

Environmental protection and zero emissions of hazardous substances continue to drive the use of sealless CMP and MDP technology. Add to that, however, the philosophy of continuous process improvement and reduced maintenance, and there is ample reason to expect that sealless pump technology has a bright future. It is appropriate not only for new installations, but also for many retrofit applications. And standardized ANSI dimensions can help make sealless technology a highly cost-effective alternative as well. ■

Vertical Turbine Pumps Power Petrochemicals

Verticals are a popular choice for low NPSH applications, versatility of construction and minimal floorspace requirements.

By Herman A. J. Greutink

Petrochemical plants need water, whether from local water systems, deepwell pumps, rivers, lakes or oceans. The vertical diffuser pump normally plays a major role in providing service water, cooling water and occasionally fire pump service. This type of pump is also used in oil field production as well as oil field pressurization. Process fluids ranging from crude oil to liquified petroleum gas and other liquids (sulphur, for example) are moved by vertical pumps in one or more stages of extraction or production.

Frequently, liquified petroleum gas, propane, butane and anhydrous ammonia are supplied from underground caverns.

In-plant pumps with very low NPSHA are also apt to be the vertical turbine type. To economize on length of barrel or can and reduce installation and pump costs, many vertical pumps are supplied with a first stage low NPSHR impeller. Some users require that the suction specific speed be limited to perhaps 11,000. This limitation may be highly important for certain types of pumps, but if a properly designed impeller is used in a vertical turbine pump application, experience shows that the suction specific speed can range up to 15,000 without creating problems. Range of operation over the pump curve and recirculation possibilities must be considered.

SUPPLY WATER PUMPS

Following are brief descriptions of various vertical diffuser pumps used in the petrochemical field:

Deepwell pumps (Figure 1) are commonly used to raise water from underground aquifers. Lineshaft pumps are either oil lubricated or water lubricated, and they are built mainly to AWWA standards. Submersible motor driven pumps are also used. Materials of construction are mainly steel or cast iron for heads and bowls and bronze for impellers.

Service water pumps (Figures 2 and 3), used to pump from ponds, lakes, rivers or oceans, are generally larger than deep well pumps. Pumps intended for fresh water intake have steel columns and heads, cast iron bowls and bronze impellers. For pumping brackish or sea water, coated standard materials are normally used. Experience dictates whether coated standard materials will offer acceptable life. Otherwise, stainless steels (316, 316L, duplex stainless) or nickel aluminum bronze may be specified.

PHOTO COURTESY OF JOHNSTON PUMPS, INC.

A 500 hp, 1800 rpm product pump operates in a midwest chemical plant.

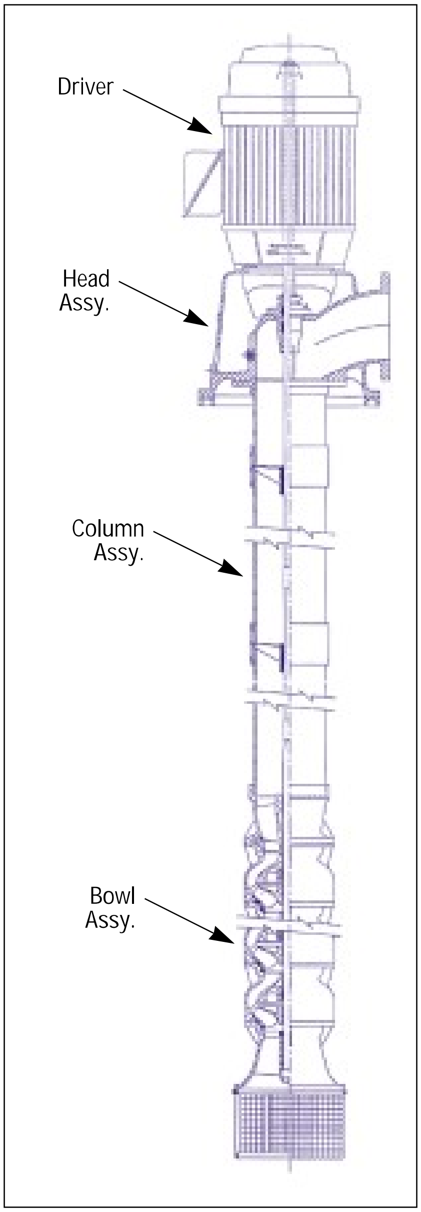


Figure 1. Variation on water lubricated deep well turbine pump

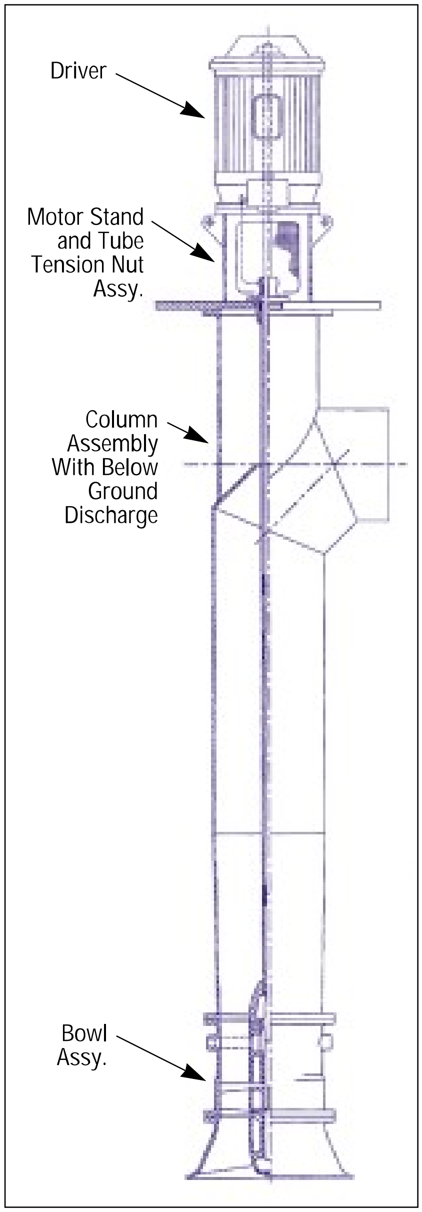


Figure 2. Supply and drainage pump, axial flow (propeller) from 5'-20' of head

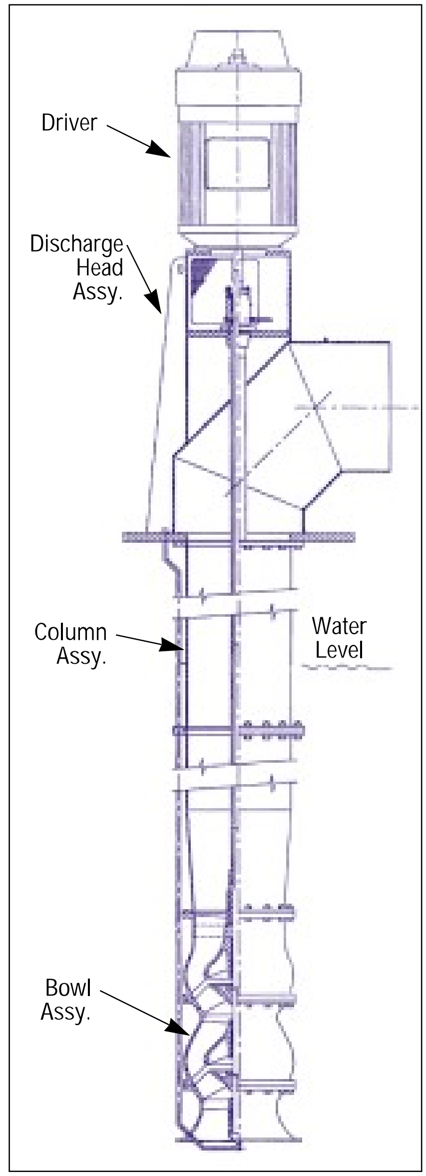


Figure 3. Mixed flow type service water, plant water, with heads of 20-60' per stage

Vertical cooling tower and plant supply water pumps (Figure 4) follow fresh water material standards and construction. Lineshaft pumps for moving water are normally built with a packing box for sealing. Lately, however, there have been requests for water pumps built with mechanical seals. Perhaps this is due to packing box maintenance requirements. There is an art to using packing boxes properly (they need to leak some water). Of course, mechanical seals also need proper maintenance and installation. But if either type of sealing technology is used and applied properly, maintenance labor can be greatly reduced.

Vertical turbine pumps used in re-pressurization of oil fields pump produced water at high pressure – up to approximately 4500 psi. Because this water is sometimes very corrosive, stainless steel bowl assemblies are commonly used, and barrels and heads are made mainly of coated steel (Figure 5).

There is an application in which no water will show up at the surface and pressure is throttled off below the surface back to the water supply. This method can be used when pumping from atmospheric or vented bodies of liquids.

PUMPING RAW STOCK AND PRODUCT

Most pumps that transport product or raw stock in the petrochemical industry are built to API

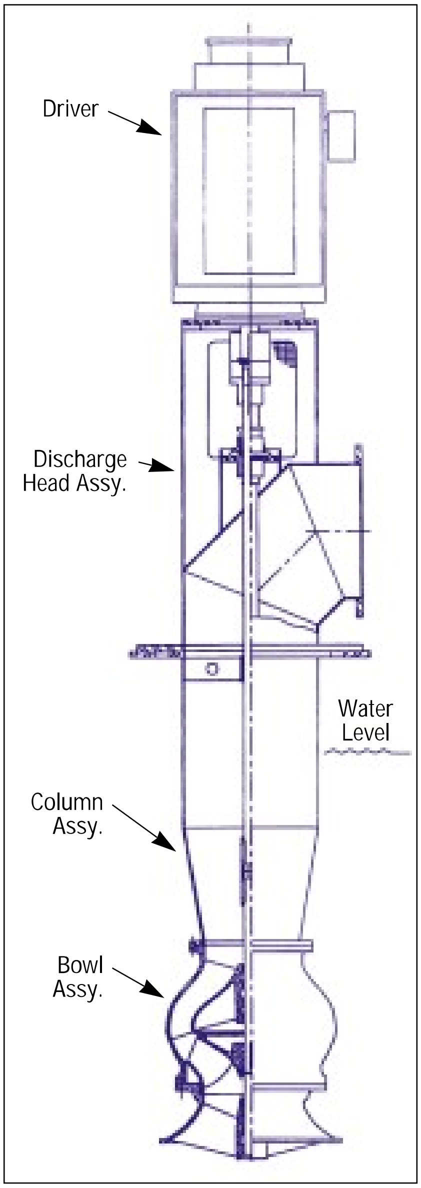


Figure 4. Service water/cooling tower pump with heads from 50' and up per stage

610. Many users add custom requirements.

In general, the specifications are tight, and extra attention to detail must be paid to the pump's proper design and fabrication. Most vertical turbine pumps in these applications are built as barrel or can pumps (Figure 6) with mechanical seals. Pressure containment construction conforms to ASME section VIII.

Sealing and bearing clearance specifications also must be followed closely. The mechanical seal

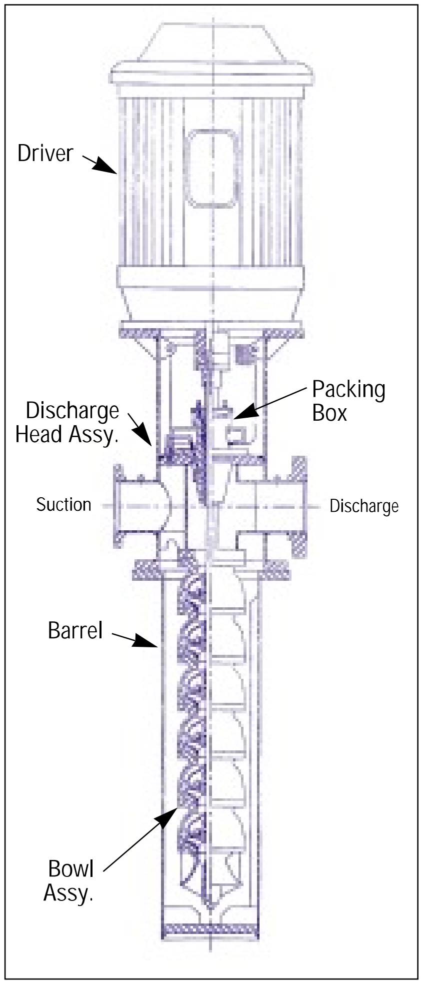


Figure 5. Oil field pressurization pump. High pressure multiple pumps (barrel or can) can be used in series.

configuration depends on the product pumped and in many cases must adhere to strict environmental protection rules. Most of these pumps have motor drives with thrust bearings in the motor, although some requirements call for the European style thrust bearing in the pump. The reason for the latter is better mechanical seal maintenance with less run-out. However, this increases total head and motor assembly, which could aggravate vibration problems. It also increases maintenance difficulty as more parts must be disassembled to get to the mechanical seal.

Pipeline pumps may be horizontal multistage pumps, but boosters from the tank farms to the pipeline pump are preferably vertical can or barrel pumps (Figure 7). These offer best utilization of storage capacity. The available NPSH can be very low – putting the first stage at a level where the tankage can be pumped empty to zero NPSH available and at the same time supply the necessary NPSH through the vertical barrel or can type pump to the horizontal pipeline pump. High pressure pipeline pumps also can be built economically using vertical multistage pumps up to approximately 2000 hp.

Extremely high or low temperature liquid applications are designed to individual requirements. For instance, sulphur is handled by vertical turbine pumps with steam or hot oil jacketing from top to bottom of the pump. Hot oil circulating pumps (for heat transfer) are built to withstand forces caused by temperature differentials and pipe loading.

The pump specifier should remember that material requirements per API 610 are aimed mainly at horizontal centrifugal pumps. The requirement for cast steel is all right for most horizontal pumps, but for turbine pump bowls and impellers it can create problems that are expensive to correct. A better way generally is to make the bowls and impellers out of stainless steel, especially the 300 series and duplex stainless.

Practically all vertical turbine pumps are made to order. Off-the-shelf items consist only of standard bowls, impellers, cast iron heads and some smaller pieces like threaded shaft couplings. As a result, communications among user, consultant and manufacturer must be very good to get the best equipment for the application.

CONSTRUCTION OPTIONS

As mentioned, construction materials vary according to fluid characteristics. Materials are often selected according to the corrosiveness and abrasiveness of the pumped liquid. The following are some construction variations.

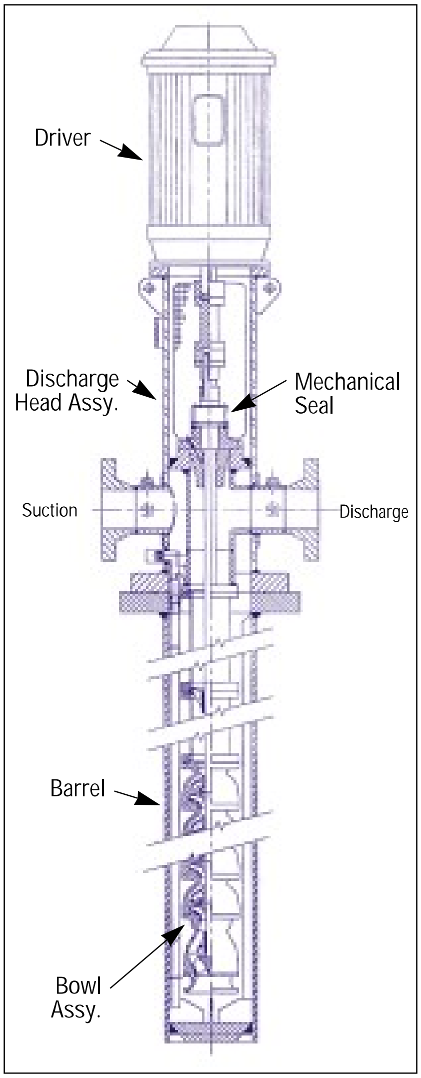


Figure 6. High pressure LPG pump (barrel or can)

Below-base discharge and/or suction is one option to the more prevalent above base-discharge design. The figures show variations in packing boxes and mechanical seals.

Product lubricated lineshaft bearings can be used as an option to oil lubricated or clean water flushed lineshaft bearings. Rubber bearings and bronze bearings are used most often. Rubber is used successfully on hard shafting or hard-faced shafting in water with abrasives such as sand. Do not use rubber bearings in petrochemicals that attack rubber. We also see the use of metal-filled graphite bearings in light ends and

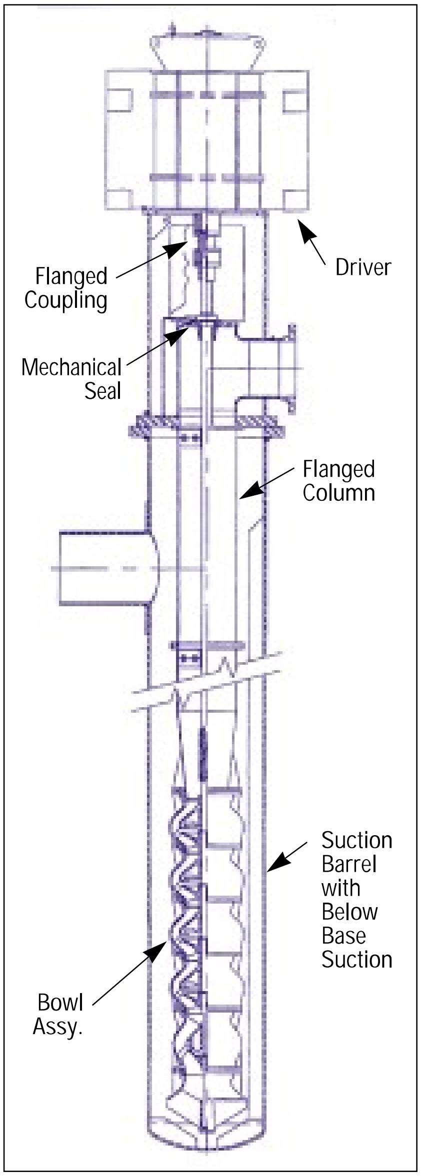


Figure 7. Pipeline or pipeline booster pump

condensate pumps. Where abrasives are present and rubber cannot be used, hard bearings (from nitronic 60 to carbide) can be used on properly selected shafting such as 17-4 pH, nitronic 50 or hard-faced shafting.

Cast iron bearings can be used in oils. It is also possible to coat cast iron bearings for corrosion protection in water pumping applications.

Constructions shown in the figures can be altered to suit the requirements of the system in which pumps will be used. Turbine pumps have been built to run upside down, horizontally or at an angle. The variations of driver systems and construction systems are endless. Any type of driver with right angle gears, vertical motors (hollow or solid shaft) and submersibles can be used.

Obviously, multi-staging allows one to reach the head required at a given capacity (it's like connecting pumps in series). However, complete pumps can be put in series, too. For instance, driver, construction and shaft limitations of a 12" multistage pump may limit a pump to perhaps 1500 psi. If 4500 psi is needed, just put three pumps in series.

INSTALLATION

Installing vertical turbine pumps is a matter of keeping all the pieces or sections in good shape. First, all the pieces that have to fit together to make a vertical turbine pump run properly must be manufactured with three basics in mind: straightness (of shafting), concentricity and parallelism. This is true for all the pieces that fit together to build a total pump. If these considerations have been well addressed, one normally should be able to turn the shafting by hand after the impellers have been lifted off the seat. Once this condition has been reached, the pump must be set to turn in the right direction when the power is applied. Most pumps of the vertical turbine, mixed flow and axial flow type turn counter-clockwise when viewed from the top. But some very old units may turn clockwise – so pay attention.

For pumps with threaded lineshaft, the motor/driver rotation must be checked before the driver is hooked up to the pump. In the case of slipfits between motor and shaft, remove the piece that forms the slipfit before checking rotation. Otherwise, the pieces may gall and cause serious and expensive damage.

Check the pump's performance, vibration and runout when it is put in service. At times, runout can also be checked before running. In most cases the pump will operate satisfactorily, but sometimes problems that require immediate attention arise.

Excess runout often can be corrected by changing coupling parts 90 or 180° (where bolted couplings are used). It is more difficult to correct if the shaft is bent or some registers are not true or some faces are not flat. Check driver base and shaft and go from top to bottom. Also, dirt between butting faces frequently causes major problems!

Vibration. Considerable runout may be accompanied by vibration. If there is no runout problem, however, it is necessary to determine what causes the vibration. First disconnect the driver from the pump and run the driver. If it does not vibrate, then check the pump. If the driver still vibrates, do the following: push the stop button or (in case of gear drive) slow down the engine drive and check vibration while slowing down. If vibration diminishes in line with speed, it is a balancing problem with the driver. Get it rebalanced. If vibration immediately disappears when slowing down, it is probably a critical speed problem, which may have several causes: 1) the design is such that the driver/head assembly has a critical equal or close to the operating speed, 2) pipe loading, 3) foundation loading. In the case of electric motors, an electric unbalance between leads can cause vibration, but this is rare. Try to find out what 2) and 3) are doing by just loosening the anchor bolts a couple of turns. Many pumps exhibit 2) and 3) as problems. If so, it may be necessary to rework the piping connection. In any case, if the vibration problem is not easily solved, get the pump manufacturer involved.

Sometimes a piece of wood or other matter gets into the pump and causes vibration. Maybe it can be backflushed out. And no matter what, the sump and the piping must be clean from the start.

When vibration begins to show up after years of running, it's time to pull the pump and replace bearings and repair it as new. Keep in mind the three musts: concentricity, parallelism and straightness.

In a particular installation of four identical barrel pumps, one unit persistently vibrated. The motor and even the pump were replaced with still no improvement. The investigation, however, identified that the grout under the barrel flange was totally inadequate for the vibrating pump. In other words, weaker support put the pump in a natural frequency mode equal to the running speed. This has been seen in other installations where pumps were mounted on beams or slabs that happen to lower natural frequencies to the pump speed.

When using variable speeds, the pump may be in a natural frequency mode at one of the speeds. If so, it is best to block out that area of speed in the controls.

Operation. When pumps are selected, built and installed properly, they will have a reasonable life expectancy. If a pump's life is not up to reasonable expectations, its selection, use, construction and installation must be thoroughly checked. Performance can be changed to suit head capacity requirements; construction and materials can be upgraded as necessary; and bearing systems, including the thrust bearing systems in the driver, can be improved. Many vertical pumps will start in upthrust. So, the bearing system must be built to handle that, and in some cases the pump may be required to run in continuous upthrust. The system can be built for that, too.

Sometimes particular pumps continually vibrate or rattle or run out and repairs have to be made frequently. That's when time must be taken to solve the problem properly, whatever the problem is – from flow to the pump, head, capacity and all materials and construction in between!! ■

Herman Greutink is Vice President and Technical Director for Johnston Pump Company (Brookshire, TX). Among other professional affiliations, he is a longstanding member of Texas A&M's International Pump User's Symposium Advisory Committee and the Hydraulic Institute, and he frequently contributes to Pumps and Systems *on the subject of vertical turbine pumps.*

Chopper Pumps Digest The Solids

These tough workhorses eliminate plugging problems in heavy-duty applications.

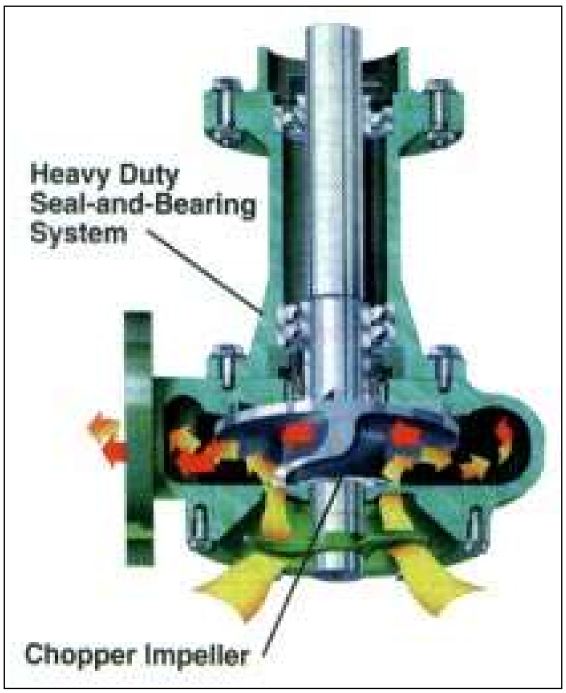
By John Hayes

If the hydraulic performance characteristics are right, chopper pumps can be used in any industrial or municipal application that involve pumping solids-laden slurries. They are a cost-effective means of eliminating pump plugging problems and optimizing system perform- ance. Before discussing specific applications for chopper pumps, however, let's look first at the design details that make chopper pumps unique.

DESIGN DETAILS

A chopper pump is a centrifugal pump that uses sharpened semi-open centrifugal impeller blades to cut against a stationary bar across the full diameter of the inlet. This bar is known as the "cutter bar," and this style of chopping and pumping is known as "positive chopping." All incoming solids too large to pass through the impeller are chopped prior to entry into the pump, thus eliminating any possibilities of pump clogging. Other items critical to the success of a chopper pump are a severe duty seal-and-bearing system, hardened wear parts and a history of successful pump installations by the manufacturer. Solids cutting by the impeller and cutter bar occurs when:

1. The suction created by the rotating impeller pulls material into the center areas of the pump impeller. To reach the impeller center areas, liquid and the material entrained in the flowstream must pass through the cutter bar openings in the lower suction plate located just below (or ahead of) the impeller.
2. As material passes through the cutter bar openings into the low pressure areas of the impeller, it gets caught between one of the rotating impeller blades and one of the two stationary shear surfaces that are cast into the cutter bar. These surfaces extend all the way across the intake opening in the cutter bar and divide it into two segments. Positive chopping is required of all solids entering the pump prior to pumping.
3. The leading edge of each impeller blade is sharpened and machined flat on the blade face where it runs next to the cutter bar. This forms a cutting edge. The material caught between the impeller blade and cutter bar is severed.



The advantage of the chopper is that pumping and chopping are integrated into one efficient system. The chopping is done right where the pumping is done. The material naturally fits through the flow passages of the impeller and casing. If the material won't fit through, the pump keeps chopping it until it does. This is an example of positive chopping.

BENEFITS

Following are some of the benefits of positive chopping.

1. Positive chopping allows large, troublesome materials to pass through the pump, eliminating downstream plugging of valves, heat exchangers, nozzles or other pumps.
2. A chopper pump often can replace two pieces of equipment, a comminuter (or pregrinder) and a "non-clog" pump. This approach is extremely cost-effective because the maintenance costs on comminuters alone can be very high.
3. Chopping material in the pump produces a more homogeneous slurry and reduces pipeline friction.
4. Chopper pumps can handle materials in sumps that no other pump can handle.
5. A severe duty seal-and-bearing system that incorporates double row thrust bearings and a mechanical seal reduces down time by handling the heavy workload of chopping and pumping solids reliably.
6. Hard, wear-resistant pump parts hold up to the rigors of chopper pump service. Standard pump impellers and cutter bars made of cast alloy steel heat treated to 550 Brinell provide extended service life in most applications.
7. Selection of a manufacturer with extensive experience in severe duty applications helps assure the user of dependable service. Chopper pumps are not a commodity

item. Every application is unique in its severe nature and duty. Procurement of inexpensive equipment or equipment that is "new" to the market could provide disappointing results. Always ask the supplier about the history of the product and its applications.

WASTEWATER APPLICATIONS

Lift Stations. Small residential lift stations are equipped with submersible grinder pumps, and larger lift stations with non-clog pumps. However, lift stations can experience an unusually high concentration of solids such as hair, rags or plastics that cannot be reliably handled by these conventional pumps. If heavy solids loading is anticipated during the engineering stage, many of the larger lift stations include a comminuter ahead of each pump. Chopper pumps have solved many lift station plugging problems and eliminated the need for comminutors. In some instances chopper pumps have directly replaced existing pumps without the need for repiping.

Chopper pumps for lift stations can be sized for hydraulic requirements without regard to minimum sphere passing diameters. Generally, non-clog pumps must be sized according to maximum anticipated sphere size, with hydraulic considerations secondary. For example, a requirement of 200 gpm at 130' tdh might normally dictate a 3" pump hydraulically. But a 3" non-clog pump cannot provide the high head and meet a 3" sphere requirement at the same time. Therefore, the engineer and/or manufacturer would be forced to use a 4" pump. This results in higher costs – due not only to the larger pump, but also to lower efficiency and higher power consumption. However, because a chopper pump reduces the size of solids before they enter the pump, sizing of the pump is based mainly on hydraulic requirements, with little consideration given to sphere size.

Septage Receiving. Septage is comprised of concentrated solids from septic tanks plus rags and plastics that can plug conventional non-clog pumps. Chopper pumps eliminate plugging problems by chopping all incoming solids prior to pumping. Septage receiving pit pumping is a very tough challenge.

Because conventional non-clog pumps often require oversizing to handle solids, another means of pump protection is to pre-grind all materials with a comminutor upstream of the pump. This is an unnecessary added cost in installation, operation and maintenance when a chopper pump, which is one piece of equipment, does both jobs.

Sludge Transfer and Recirculation. One particular problem with sludge pumping is the "roping" of hair and other stringy materials created by prerotation within the piping ahead of non-clog pumps (especially vortex pumps). Chopper pumps eliminate roping by chopping the solids as they enter the pump. Another common problem is pumping grease-and-hair balls or other reformed solids. In digester recirculation, passing a grease-and-hair ball from the bottom to the top accomplishes nothing. If the reformed solid is chopped during the recirculation process, then the chopped solids have a high surface to volume ratio and are digested faster.

Chopper pumps can handle a higher solids content than conventional non-clog pumps. In normal treatment plant processes this means that sludges can be more concentrated without exceeding the pump's capability.

Chopper pumps also reduce downstream pipeline plugging and friction losses associated with high solids content. The chopping of these solids creates smaller solids with sharp edges that tend to scour the inner walls of discharge piping. Friction is reduced due to maintaining full pipe diameter, rather than choking flow with grease build-up on the pipe walls. Chopping and shearing of sludge tend to reduce viscosity, further reducing friction loss.

Digester Scum Blankets. Both aerobic and anaerobic digesters tend to form scum blankets when conventional mixing methods are used. These blankets inhibit methane production and eventually require more frequent cleaning of the digester. A vertical chopper pump with a recirculation nozzle and sealed deck plate can be installed through an existing manhole opening in the top of the digester and used to chop and mix the scum blanket. The object is to inject supernatant through an adjustable nozzle into the scum blanket approximately 1 foot below the surface. With the pump located near the periphery and the nozzle aimed at a proper angle to the wall, the action of the nozzle forces the blanket to rotate. Once the blanket is mixed, methane gas production will increase. The pump then can be used intermittently to keep the scum blanket mixed. This can increase digester capacity up to 40% and increase gas production up to 300%.

Photo 1. By slurrying the scum blanket with a recirculating chopper pump, Tony Kucikas of Nut Island WWTP rejuvenated the digester and saved Boston $1.5 million in cleanout costs.

The action of chopping and conditioning material to reduce particle size is of proven value in sewage treatment plants. If digestible material is in smaller particles, then the surface area of these particles is relatively large in comparison to their volume. Bacterial action can then be more effective and rapid. Plants that have used chopper pumps for digester recirculation and/or scum blanket mixing have seen that particle size reduction increases both the rate of decomposition of digestible material and digester gas production.

Clarifier Scum. Inherent problems with clarifier scum include plugging and air binding. The chopper pump addresses plugging problems. Air binding, on the other hand, must be addressed indirectly.

Inherent recirculation around the chopper pump inlet usually causes enough mixing to keep air binding from occurring in scum pits with short retention times. If a scum pit has a long retention time, then the scum may concentrate and form a blanket on the top. As the pit level is pumped down, this blanket can block the pump suction. In this case, a recirculation nozzle should be used to pre-mix the scum pit prior to pump-out. Also, because chopper pumps have heavy-duty oil bath lubricated bearing and seal systems, the scum pits can be pumped completely down to the pump inlet without damaging the pump bearings or seal due to loss of coolant or lubricant. This allows full scum removal during each pump-down. By adding a motor low current monitoring system in the pump controls, the "OFF" function can be based on low motor current draw rather than an on "OFF" mercury float switch. Once the liquid level drops to the point where air enters the pump inlet, the current draw will drop off, and the low current relay will shut the pump off.

INDUSTRIAL APPLICATIONS

Food Processing. Chopper pumps can be applied in almost all food processing waste handling operations. Applications include vegetable waste with whole vegetables and stalks, poultry and turkey parts with feathers and whole birds, beef processing with hair and fleshings, fish carcasses with offal, and any other food processing wastewater. Slurries are generally pumped directly into trucks for hauling to feedlots or land application, sometimes first across a separation system to reduce water content.

Wood Products. There are many applications within the wood products industry where chopper pumps greatly reduce downtime due to pump clogging. First, raw logs are stacked in sorting yards where rainwater runoff sumps can collect bark, limbs, rocks and large quantities of dirt. This requires a heavy duty pumping system, and is ideal for a chopper pump. Next, the logs might be debarked, leaving a slurry heavily laden with bark, most of which floats. Recirculation nozzle systems added to the chopper pump help to suspend the bark while pumping. Then, the logs are either sawn in a sawmill, or chipped in a pulp mill. Runoff sumps collect the chips and sawdust, requiring further handling of solids too difficult for a standard non-clog pump. Finally, waste pulp or broke is also easily handled by a chopper pump.

Hydropulper rejects in wood products recycling are the most severe pumping applications in all industry. Rejected materials from the recycled bundles can contain wire and plastic strapping, large quantities of plastic sheeting or wrap and wood from pallets – all suspended in a heavy pulp slurry. Due to the heavy nature of these slurries, chopper pumps are generally oversize, and motor horsepower is increased to handle the heavy chopping load.

Photo 2. A vertical chopper pump handles bark, wood, rocks and dirt in a pulp mill log yard stormwater runoff sump.

Other Industries. The world of heavy industry contains numerous other special applications for chopper pumps. The steel, chemical, automotive, contractor, mining, sand & gravel, and petrochemical industries all use chopper pumps in applications involving pumping of waste solids with the occasional unknowns. Anytime a waste sump must deal with the unplanned worker's glove, pieces of wood crating, rocks, bottles, glass, cans, plastics or other items larger than the sump pump is designed for, then a chopper pump is applicable.

FINAL CONSIDERATIONS

All of the applications discussed here center on the pump's ability to handle solids from a pumping standpoint. However, we must not forget that chopper pumps can also eliminate seal and bearing failures observed in other pumps. Because the chopper design requires heavy shafting, an added benefit is longer life resulting from stronger parts and less vibration. Quite often seal failures in conventional pumps are associated with solids wrapping or binding at the impeller or seal. This can cause severe vibration that is transmitted through the shaft and seals to the lower bearing. This results not only in seal and bearing failure, but also can introduce moisture into a submersible motor. The heavier shafting and short overhang of the chopper pump bearing and seal design addresses this problem and reduces maintenance costs as a result.

More industries, municipalities and engineering firms alike are discovering the economics of applying chopper pumps in applications in which conventional pumps have historically failed. These failures are generally due to plugging or seal-and-bearing failure, and all contain hidden costs that must be addressed. The largest fallacy of the "low bid" system is that low initial price does not mean lowest overall cost. More often than not, "low bid" equipment has a higher failure potential than properly specified and purchased equipment. The solution starts with the user's request to obtain equipment that will operate maintenance free, and it ends with the foresight of those with purchasing authority to think toward the future. ■

John Hayes has a BS in Mechanical Engineering Technology from Clemson University, has worked with chopper pumps for over 12 years and is presently the Marketing Manager for Vaughan Co., Inc.

So You Need Pumps For A Revamp!

Here are tips for specifying and selecting the right centrifugal pumps.

By J. T. ("Terry") McGuire

Successfully specifying and selecting pumps for a unit revamp requires many of the same disciplines as for a new unit, but with a difference. The difference is that a full-scale working model is available for examination and analysis. Taking advantage of this opportunity can lead to employing pumps that consume less energy and have a longer mean time between repair (MTBR). This, in turn, lowers plant operating costs and can raise plant output, hence revenue, through higher plant availability.

In a process unit, pumps move liquid and raise its pressure to allow the process to run, but their role is fundamental and their interaction with the system a critical factor in their performance. This last point, interaction with the system, leads to the first step in specifying and selecting pumps for a revamp.

PUMP SYSTEM INTERACTION

A centrifugal pump operates at the capacity determined by the intersection of its head capacity curve and the system's head capacity curve (Figure 1). At this point the energy added by the pump equals the energy required by the system. Note in Figure 1 that the energy required by the system is often increased by throttling across a control valve to regulate the pump's capacity. Also note that this means of flow control is feasible only with kinetic pumps – those that add energy by raising the liquid's velocity.

Displacement pumps (Figure 2), deliver essentially a fixed capacity at a given speed and thus add only the energy needed to move that capacity through the system. Care is therefore needed to ensure that this energy never exceeds the mechanical capability of the pump. In other words, displacement pumps must always be installed with a full capacity relief valve upstream of the first valve in the discharge system. And the relief valve must have an accumulation pressure (rise above cracking pressure to achieve full flow) that keeps the pump's discharge pressure and corresponding power below the maximum allowable.

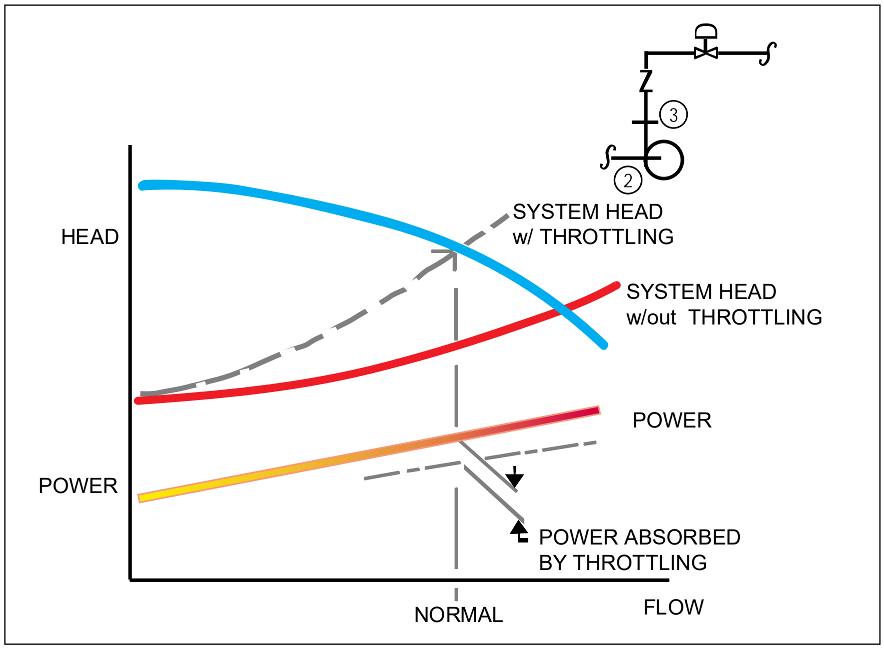


Figure 1. Centrifugal pump vs. system with control valve

LEARNING FROM THE EXISTING INSTALLATION

The system energy requirements for new units are estimated using various assumptions and margins. When centrifugal pumps are used, the system designer usually relies on a control valve to balance system and pump energy at the desired flow rate. In engineering a revamp, it is possible to determine the actual system characteristic accurately and thus avoid energy losses due to conservative assumptions and margins. In most cases these losses are far greater than those resulting from differences in efficiency among various selections for the same duty.

Determining the actual system head requires accurate measurement of:

- pump flow rate at one condition
- pressure at the pump's suction

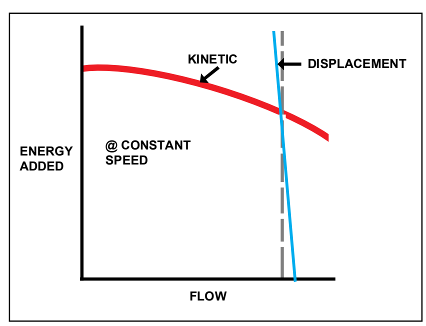


Figure 2. Flow regulation of kinetic vs. displacement pumps

and discharge and in the suction and discharge vessels

- liquid levels relative to a common reference in both suction and discharge vessels
- pressure drop across the control valve, if used, taking care to measure the downstream pressure some 10 diameters from the valve to avoid the influence of any flow distortion

To make use of the pressure measurements, it is also necessary to determine the pumped liquid's specific gravity (SG) at each measuring point. This can be determined from liquid temperature as long as the liquid being pumped is known with certainty.

Using Figure 3 as a reference, the system head is:

$$H_{system} = \frac{(P_4 - P_1)2.31}{SG} + (Hz_2 - Hz_1) + HL_{1-4} \quad (1)$$

and the pump's total head is:

$$H_{pump} = \frac{(P_3 - P_2)2.31}{SG} + Hz + HL_{2-3} + \frac{\Delta V^2}{2g} \quad (2)$$

where Hz is the correction for gauge elevation, if any, HL_{2-3} the friction loss between the suction and discharge pressure gauges, and $\Delta V^2/2g$ the difference in velocity heads at the points of suction and discharge pressure measurement. The friction loss is significant when there are elbows, valves or reducers between the gauge and the pump. The difference in velocity head usually matters only when the pump head is low and there is a difference of more than one pipe size at the points of pressure measurement.

Subtracting the static head components from the pump head, Figure 4, yields the system friction head, HL_{1-4}. When a control valve is used in the system, the head being lost to throttling across the control valve is calculated from the measured valve pressure drop, then subtracted from the total system friction to give the head lost to friction in the piping, including entrance and exit losses.

Recognizing that the head lost to friction varies as the square of the flow rate, the equivalent system friction at several other capacities now can be calculated and the system head characteristic plotted (Figure 4). If the static head varies with time, as it often does in a transfer process, then the range of system heads can be plotted after allowing for maximum and minimum liquid levels in the suction and discharge vessels.

The other critical aspect of the system to be verified using the measurements already made is the NPSH available at the pump. For measurements at the pump suction, again referring to Figure 3, the equation is:

$$NPSHA = \frac{(P_2 + P_a - P_{vap})\, 2.31}{SG} + Hz + \frac{Vs^2}{2g} \quad (3)$$

where P_a is atmospheric pressure at site, P_{vap} is the vapor pressure of the pumped liquid at the pumping temperature, Hz is any correction for gauge elevation to the pump's reference level, and Vs is the fluid velocity at the point of suction pressure measurement. The pump's reference level is the shaft centerline for horizontal machines and the centerline of the suction nozzle for vertical machines.

Because NPSH is also equal to:

$$NPSHA = \frac{(P_1 + P_a)\, 2.31}{SG} + Hz_1 - HL_{1-2} \quad (4)$$

it is possible with the measurements already made to calculate the friction loss in the suction side of the system. And then following the same procedure used for the system head, the system NPSHA characteristic (Figure 4) can be developed. If the liquid level in the suction vessel can vary with time, the range of NPSHA can be plotted in the same manner as the range of system heads.

With the net system head now known accurately, the head needed to move the required flow and allow flow control can be minimized. At the same time, an accurate NPSHA characteristic eliminates hidden margins. This means an appropriate NPSHA margin can be set for the application, facilitating selection of an optimum hydraulic design. Keeping the pump head to the minimum necessary lowers energy consumption. And an optimum hydraulic selection can contribute to both lower energy consumption and longer MTBR.

Before preparing a pump specification, two more factors must be addressed at the site. The first is suction piping. Many problems are caused by poor suction piping. A unit revamp is an opportunity to correct this. The important features of the suction piping layout are the

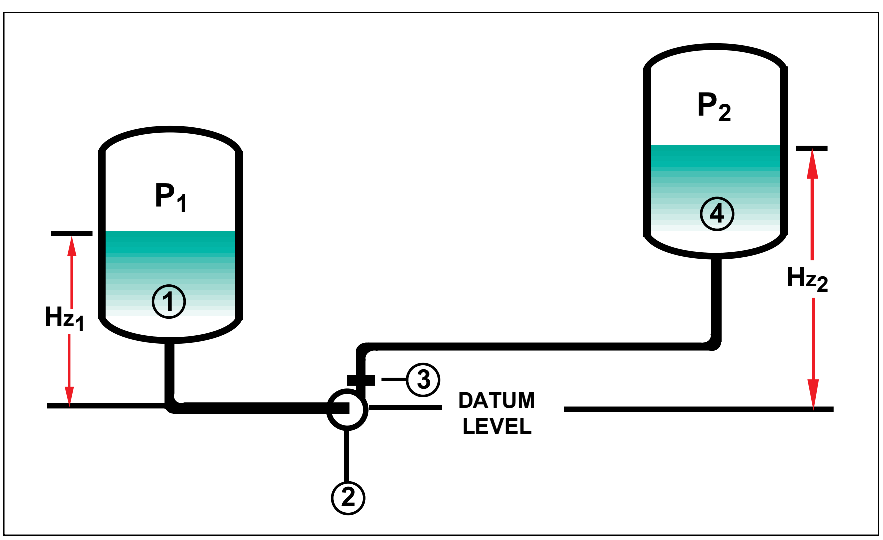


Figure 3. Hydraulic gradient

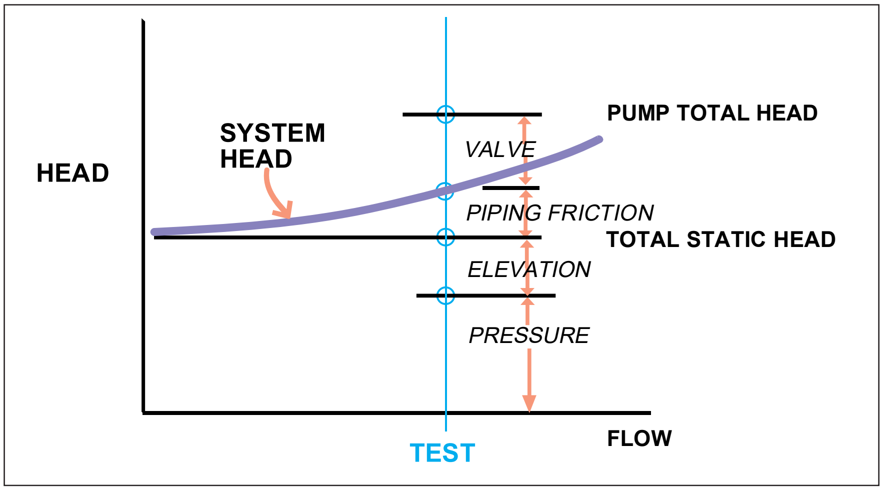


Figure 4. System head from measurements

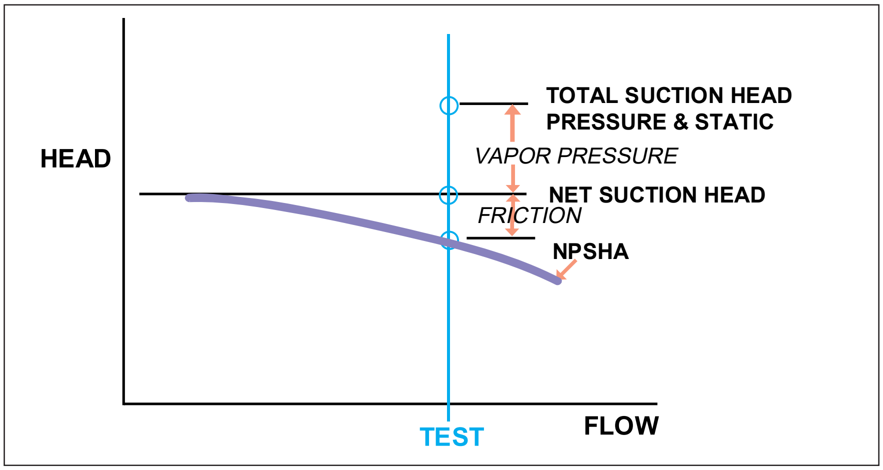


Figure 5. NPSHA characteristic from measurements

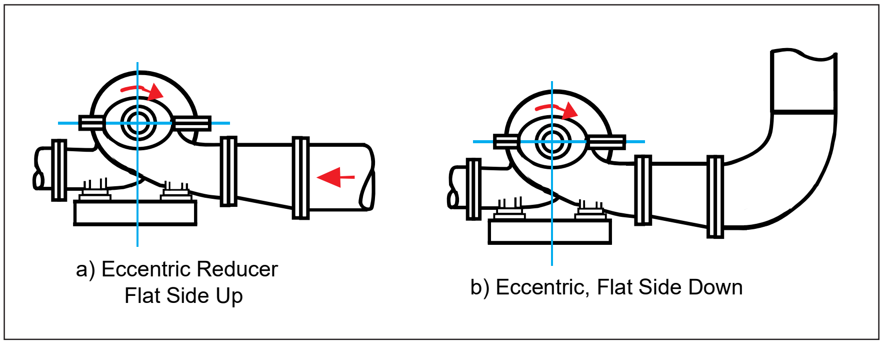


Figure 6a & b. Correct installation of reducers

orientation of reducers, the proximity of elbows to each other when in different planes, the orientation of the elbow immediately upstream of double suction pumps, suction piping slope and submergence over the vessel outlet.

When the suction piping rises to the pump, reducers in horizontal runs must be eccentric and installed flat side up (Figure 6a). In suction piping from above the pump, horizontal reducers can be concentric or eccentric, installed either flat side up or flat side down (Figure 6b). A concentric reducer is necessary for end suction pumps of high Ns or S or both. Eccentric reducers installed flat side down are used by many designers to eliminate low points in the piping, which can accumulate dirt.

At normal suction piping velocities of 7-8 fps, two elbows in series with their planes 90° apart should be separated by 10 diameters and should have the reducer downstream of the second elbow (Figure 7). Separating the elbows in this manner largely dissipates the flow distortion produced by the first elbow before it reaches the second. This avoids development of a swirl at the outlet of the second elbow. A reducer placed downstream of the second elbow helps dissipate the flow distortion and any swirl that may have developed.

The last elbow in the piping to a double suction pump must be in a plane normal (at right angles) to the axis of the pump's shaft (Figure 8a). An elbow in a plane parallel to the pump's shaft axis (Figure 8b) will cause uneven flow into the impeller. This can result in higher power consumption, noise, vibration, premature erosion of one side of the impeller and thrust bearing failure.

Moving away from the pump, the suction line must not have any high points that might accumulate air or vapor leading to reduced flow or even cessation of flow from air binding. And back at the suction vessel, the submergence over the vessel outlet must be sufficient to prevent vortexing (Figure 9) or the outlet must have an effective vortex breaker.

The second factor to be addressed while at the site is the pump's service history. This can be obtained from the plants' maintenance department. What needs to be looked for is evidence of problems in the pump's application, materials of construction or mechanical design.

Evidence of poor application might be indicated by frequent shaft seal and bearing failures, rapid wear at the running clearances, frequent shaft failure, noise and vibration or

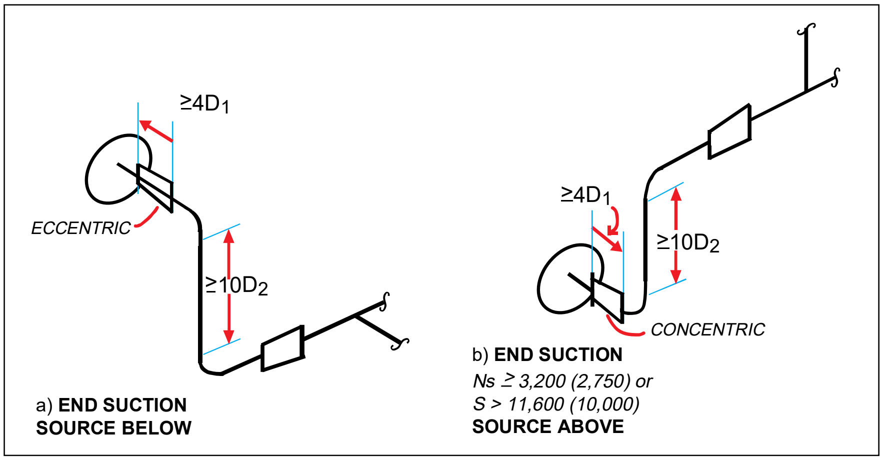


Figure 7. Suction piping layouts with two elbows in series

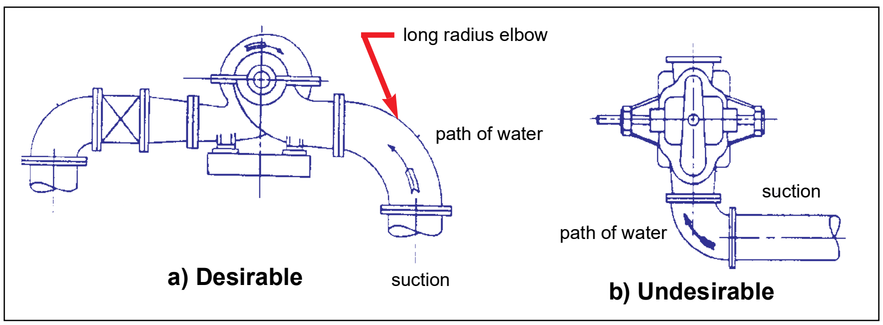


Figure 8a & b. Elbows at the suction of double suction pumps

premature impeller erosion. These are all symptoms of prolonged operation at low flows. Whether this is the case can be determined by comparing known flow rates with the pump's performance curve to see where it has been operating relative to the pump's best efficiency point (BEP).

Improved construction materials are warranted if the pump has a history of component failure due to general corrosion, corrosion-erosion, erosion, fatigue or erosion-fatigue. It is often difficult to differentiate among these causes of component failure, so it may be necessary to consult a metallurgist. In some cases changing materials may not be enough. It may be necessary either to correct a problem in the process, such as lowering the concentration of abrasive solids or bringing the pH closer to neutral, or to change to a more suitable type of pump.

Mechanical design is suspect only when the influences of application and the pumped liquid have

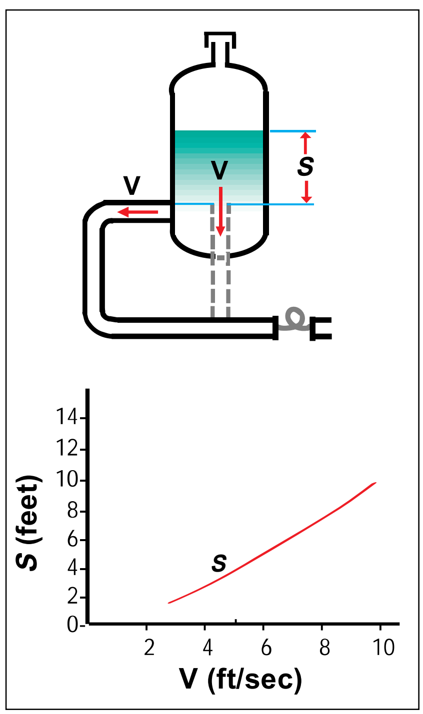


Figure 9. Submergence over suction vessel outlet

been eliminated. (This may be the reverse of common practice, but is the sequence to be followed in troubleshooting pumps.) Strain caused by piping loads is a major cause of mechanical problems. If the pump has had a high incidence of seal, bearing, coupling or shaft failures, the cause may be piping loads. The question then is whether the piping loads are too high or the pump not stiff enough. A computer analysis of the as-built piping is the first step in resolving this question. If the piping loads are reasonable or high but can't be changed, a switch to a pump with higher piping load capacity may be necessary.

Short MTBR caused solely by pump mechanical design is rare in modern designs, but not uncommon in designs dating back 30 years or more. The usual difficulties are rotor stiffness, rotor construction, bearing capacity, bearing cooling, bearing housing stiffness and casing and baseplate stiffness. These problems typically manifest themselves as frequent seal, bearing and shaft failures and rapid running clearance wear. Most of these are also symptoms of poor application, so care is needed in sorting out the true cause of the problem.

PUMP OPTIONS FOR UNIT REVAMP

Armed with accurate data on the system head and NPSH available, and knowing whether the suction piping or pump need correction as part of the revamp, it's time to look at what has to be done and how best to do it.

First, data developed by the process designer must be checked against the actual system head and NPSH available characteristics and corrected if necessary. As indicated, the questions to be answered at this

Application	NPSH Margin % $NPSHR_3$
Water, cold	10-35 (1)(2)
Hydrocarbon	10 (2)
Boiler feed, small (3)	50

Table 1. Typical NPSH Margins

NEED	OPTIONS				
	Rerate	Add	Replace	Materials	Construction
Lower flow	Δ				
Higher flow - small	Δ				
Higher flow - large		Δ	Δ		
Corrosion resistance			Δ	Δ	
Erosion resistance			Δ	Δ	Δ
Better mech design			Δ		Δ

point are the following. How much pressure drop does the control valve need to function reliably? Is it more economical to change to a variable speed pump? What NPSH margin is needed to ensure rated pump performance and expected life? The first is a question for the valve designer. Table 1 provides a starting point for the third.

Notes:

1. depends on size, higher margin for larger pumps
2. minimum 3 feet
3. up to 2500 hp at 3600 rpm
4. U_1 greater than 100 fps

New service conditions for the unit revamp can be met by exercising three options:

- rerate the existing pump or pumps
- buy an additional pump or pumps of the same design
- buy pumps of a new design

These choices may, in turn, be influenced by the operating history of the existing pumps. Table 2 summarizes the needs developed from investigation of the existing pumps and the usual options for satisfying them. Each of the options is then discussed, starting with hydraulic considerations.

Rerating the existing pump is the simplest course. To be successful, the rerate must be designed to meet the new conditions of service and at the same time overcome any deficiencies in the original application, such as being oversized for the normal flow or having a suction specific speed that is too high.

Adding a pump, either in parallel or series, is one way to achieve substantially higher flow or head, respectively. To do so successfully requires care. For pumps in parallel the fundamental rule is that the pumps operate at the same head; therefore, the combined head capacity characteristic is developed by adding capacities at the same head (Figure 10).

Two cautionary comments are needed here. First, to share flow reliably, the head of each pump must rise continuously to shutoff, or to the minimum bypass flow in the case of multistage pumps. Second, the increase in flow with each additional pump depends on the steepness of the system head characteristic (Figure 10). When the system head curve is steep, as with reference curve B in Figure 10, the increase in total flow with two pumps is quite small. In this case the flow per pump can be well below BEP, with the result that the MTBR of the pumps is reduced. At the other extreme, if the rating of the pumps is also increased by changing to a larger impeller, the

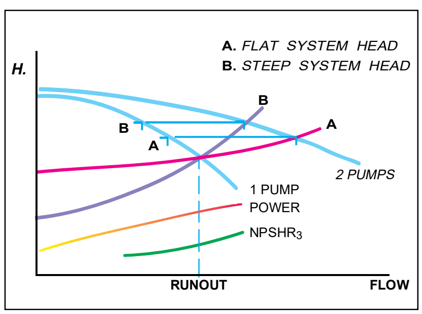


Figure 10. Pumps in parallel

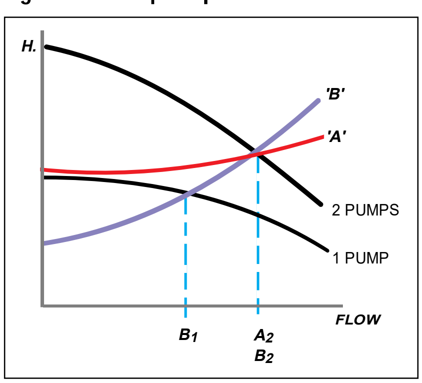


Figure 11. Pumps in series

runout NPSHR and power of a single pump must be checked to ensure that the pump has enough NPSH margin and that the driver will not be overloaded (Figure 10).

Pumps in series operate at the same capacity unless flow is taken from between them. Their combined head capacity characteristic is therefore developed by adding heads at the same capacity (Figure 11). Using series pumps against a system with high static head (curve A in Figure 11) poses the risk of flow cessation if one of the pumps shuts down. This possibility must be taken into account in calculating the degree of redundancy built into the system and in designing the pump control system.

Beyond this hydraulic consideration are two mechanical issues relating to pressure containment. First, the casing of the second pump must have a maximum allowable working pressure (MAWP) greater than the maximum discharge pressure developed when both pumps are running at shutoff with maximum suction pressure. Second, the shaft seal(s) of the second pump will, in most designs, be sealing suction pressure. Unless the pumps are separated in elevation, the suction pressure of the second pump is close to the discharge pressure of the first. This must be recognized in selecting the shaft seal.

Buying new pumps is the most complex option, but it is necessary when the new service conditions are beyond the capability of the existing design.

Regarding mechanics, a hydraulic rerate of older pump designs typically is combined with a mechanical upgrade to raise MTBR. Most manufacturers now have standard upgrades available for pumps ranging from single stage overhung to multistage.

If the existing pumps have suffered corrosion or erosion abnormal for the service and the class of pump, a change of materials should be considered. On the other hand, if the corrosion or erosion appears more related to the type of pump than to the service, changing the pump might be more economical in the long run.

In rare circumstances it will be clear from an existing pumps' service history that it is the wrong configuration for the service. Rerating such a pump to an even higher energy level would simply aggravate this condition. A high energy overhung pump in a severe service is a typical example. So as not to jeopardize the success of the entire project, a pump that is clearly not suitable for its service must be replaced.

Once an option is selected, the next step is to prepare the specification. The essential rule for a good specification is to keep it simple. Many a good solution has turned into a purchasing nightmare to the detriment of the revamp project because those preparing the pump specifications forgot this simple rule. The elements of a simple specification are:

- a one or two page data sheet
- scope of supply summary, supplemented by a terminal point diagram if needed
- agreed upon terms and conditions

A more detailed discussion of this phase and the equipment purchasing options can be found in the article "Pump Buying Strategies" by J.T. McGuire in the January, 1993 issue of *Pumps and Systems.* (Available as part of The Pump Handbook Series from the publishers of *Pumps and Systems.*) ■

J.T. ("Terry") McGuire is director of marketing for the Huntington Park Operations Division of the Ingersoll-Dresser Pump Company.

Pumping Hot Stuff – Another Perspective

Here are some general considerations for "hot" applications and, specifically, heat transfer fluid services.

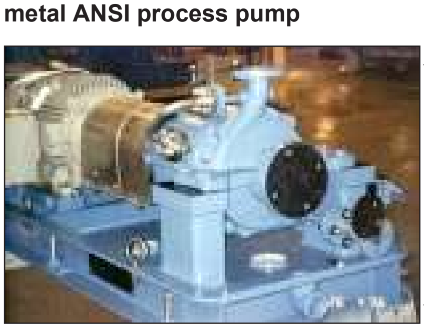
By Pumps and Systems Staff

Many words can mean different things to different people – and equipment manufacturers. This truth is important to appreciate when selecting pumps for your application. Probably because manufacturers are often "industry oriented," this situation seems to occur regularly in the case of high temperature services. For example, some pump suppliers serve the chemical process industry (CPI) primarily while others sell mainly to petroleum refiners. In the former case, ANSI standard pumps (Photo 1) are the norm while API (American Petroleum Institute, Photo 2) standards are the design of choice in the latter. Per ASME B73, the specification applicable to ANSI models, pumps should be designed to handle temperatures up to 500° F. (Some manufacturers have provisions for higher limits, though non-metallic materials may be limited to relatively low temperatures in the 200° F range.) API pumps are available for much higher temperatures. A number of vendors make hybrid or specialty equipment that does not fit either standard but is appropriate for high temperature services.

What are the primary differences between the two broad categories of pumps relative to handling high temperature liquids? The most apparent is the pump mounting method. ANSI pumps are foot mounted. API units are pedestal, or centerline mounted. This means the pump is effectively mounted at the center-line of the shaft, allowing for thermal growth of the pump in high temperature services without impacting the alignment with the driver. A foot mounted pump has only one way to grow – up from the base. The result will often be misalignment to the degree that bearing life will be compromised. Differences in material and flange standards between the product types are also evident. ANSI pumps are commonly available with ductile iron, stainless or other alloy casings and 150lb flat-faced flanges. Steel is sometimes an alternative, but it is rarely available with a short lead-time. On the other hand, steel is the "lowest grade" casing material offered by API pump manufacturers, and 300 lb raised-faced flanges are also typical. In any application where the risk of fire is significant, such as a refinery service, the effect of thermal shocking of the casing material during fire fighting efforts must be taken into account. The last thing petroleum processors would want is to provide additional fuel to a fire as a result of a cracked iron casing. API pumps also have more extensive cooling options – even the mounting pedestals are sometimes cooled.

(PHOTO COURTESY OF GOULDS PUMPS)

Photo 1. Example of a horizontal metal ANSI process pump



(PHOTO COURTESY OF INGERSOLL DRESSER PUMPS)

Photo 2. Example of an API process pump

Another major consideration when selecting pumps for high temperature services is the sealing system. Packing may be used for some applications, like boiler feed – in which case a smothering gland would be employed. In the case of mechanical seals, a film of liquid is needed to cool and lubricate the seal faces. While the high temperature liquid being pumped may be appropriate for this purpose, often it is not suitable due to lack of lubricity. This necessitates an external seal flushing arrangement. Consideration must be given when selecting the fluid used to avoid contamination of the process liquid. A combination flush and quench gland might also be needed. The quenching function involves an externally supplied fluid, such as steam, to contain the pumpage that passes the seal faces. If it is important that the pumped liquid not reach the atmosphere, either a multiple seal arrangement –

possibly with barrier fluid system – or a sealless pump may be required. This latter category includes canned motor and magnetically driven units.

We do not have space here for exhaustive explanations of various types of high temperature services. By exploring one of the most critical pumping applications in process plants, however, we can illustrate the importance of these considerations in the handling of hot liquids. That category of pump services involves heat transfer.

If process temperatures are not maintained (often within very narrow ranges), the result can include plant shut-downs and/or substantial losses in product and production. Restarting the process can be an extremely complicated and time-consuming procedure. The typical method to keep those temperatures where plant operators want them involves pumping one of several heat transfer fluids (HTF's) in "hot oil" recirulation systems. These fluids have a common trait – they are expensive. Also, many are noxious and even toxic to plant personnel – or potentially detrimental to the environment.

Looking at the options available to a buyer of pumps for an HTF service will serve to illustrate some of the complicated issues facing the decision maker. Several factors related to costs and operation will need to be evaluated. For example, if a mechanically sealed pump is selected, the pumped fluid will likely prove to be an excellent seal flush because of its typically high lubricity and flash point, although it will probably need to be cooled before being injected at the seal faces to assure reasonable seal life. This is usually done by taking a small amount of the pumpage off the discharge and piping it through an appropriate cooler. The small amount of fluid that passes the seal faces will evaporate into the atmosphere. Without the use of a quench, formation of crystals at the seal faces can be a problem resulting in shortened seal life. Over time some make-up amount will be needed. In the case of a seal failure, obviously there could be a significant amount of HTF lost, and arrangements to handle and replace this material need to be made. A popular mechanically sealed pump used for HTF service in recent years is the Dean model R400 (photo 3.) This design has many API pump features, such as centerline mounting and bearing frame cooling provisions, but it is priced between ANSI designs and those that fully comply with API-610. A number of manufacturers make pumps of this latter type. If applicable, corporate policy or government regulations relative to these fugitive emissions may necessitate provisions for monitoring and reporting, as well as specific containment and processing procedures for pumpage that leaks.

Alternatively, and likely because of these regulatory controls and corporate mandates, many buyers are opting for one of the two general categories of sealless pumps. Though canned motor and magnetically driven designs typically have a somewhat higher first cost, they offer operational and, in some cases, coincident cost advantages. The canned motor arrangement is, in effect, an induction motor with its rotor serving as the pump's shaft. A thin metal can separates the pumpage from the stator. The rotor bearings are journal type and lubricated by the pumpage. In the case of HTF, as for the mechanically sealed pump design, a small portion of the pumpage is often cooled – typically through an integrally mounted cooler, but one that in this case is used to cool the motor windings. A number of manufacturers make canned motor pumps suitable for heat transfer service. Magnetically driven pumps fall into one of two sub-categories: synchronous and eddy-current. The first group utilizes one of two rare earth materials for both its external and internal magnet assemblies. Neodymium is significantly affected by heat, especially in the range of heat transfer fluid services, making this material unsuitable in high temperature applications. More appropriate in this temperature range is Samarium Cobalt, which, while the more expensive of the two materials, is impacted by high temperature operation to a significantly

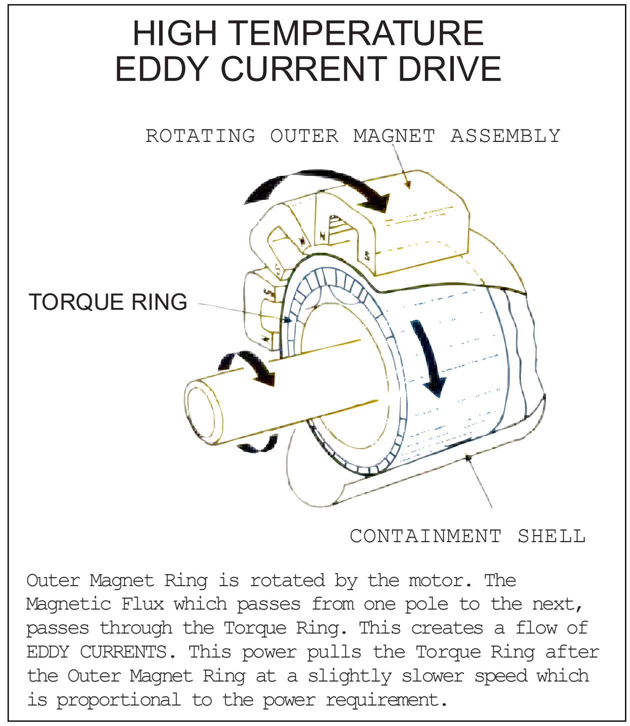


Figure 1. Kontro, torque ring design


(PHOTO COURTESY OF DEAN METPRO)

Photo 3. The Dean R400A mechanically sealed pump is used for HTF service

lesser degree. Several manufacturers serve the HTF market with appropriate units. The second group – mag drive pumps – isn't really a group. The Kontro product line manufactured by Sundstrand Fluid Handling offers a unique torque ring design (Figure 1) that is unaffected by heat in the range for this application. As in the typical mag-drive pump, permanent outer magnets are coupled to a driver. Operating outside the fluid, they see reduced temperatures and are not affected by the heat as similar magnets would be in an internal assembly. What makes this design different is that power is transferred not to a set of inner magnets but to a rotating element consisting of the impeller, shaft and integrally mounted torque ring (with a coil much like that in an electric motor.) This assembly runs in product lubricated journal bearings.

In all sealless options the elimination of noxious emissions and potential leaks from seal failures simplifies or negates the need for internal corporate or external regulatory reporting, as well as provisions for containment and subsequent handling of fugitive emissions. The improved environment for operators is often cited as a significant advantage of using sealless pumps. Make-up requirements of expensive HTF lost through seal operation and occasional seal failures are eliminated. Interestingly, the somewhat lower efficiencies of sealless pumps actually contribute heat to the HTF, reducing the load on hot-oil heaters. This is offset by cooling water requirements needed for many canned motor pumps, though manufacturers do offer optional features, such as ceramic insulation, to allow higher temperature operation without cooling in lower horsepower sizes. Without the need for cooling of the recirculated lubrication fluid, a net heat gain is provided to the system in the case of this latter group of canned motor pumps as well as the mag-drive and torque ring designs.

As you can see, there are many factors to consider when selecting a pump for HTF or other high temperature services. First cost or electrical power requirements are two obvious considerations, but often the less apparent issues will have a significant bearing on the purchase decision.■

The Power of Speed and Staging

The energy consumption of most centrifugal pump applications can be reduced by optimizing the specific speed. This can be done with either staging or shaft speed.

By Dave Carr and Bill Mabe

Centrifugal pump decisions naturally require purchasers to judge the suitability of alternative selections. Pumps comprise the second largest segment of rotating equipment, and never have such decisions been more critical. Intense competition dominates nearly all of today's markets. The minimization of expenditures is a constant focus with organizations intent on survival. Energy costs associated with pump drivers are a routine target in the continuous effort necessary to maintain market viability. Clear recognition of the most efficient pump options to meet process "flow versus head" will arm pump users with essential tools to attain reduced operating and maintenance costs. Gains in centrifugal pumping efficiency can be readily realized through the optimization of speed and staging. We will present a practical approach to understanding these issues using a simple format tabulated over a broad range of flows and heads.

Energy and Efficiency

As with all laws of nature, the key factor in minimizing the energy per unit time required to perform work on an on-going basis is the efficiency of the process. The same is true with centrifugal pumps. In U.S. units, the power equation is {HP = H x Q x S.G./ 3960 / η} where HP is horsepower, H is the total dynamic head of the pump in feet, Q is flow rate in gallons per minute, S.G. is the specific gravity of the pumped liquid (compared to water) and 3960 is a constant. The pump's efficiency is usually represented by the Greek symbol eta (η) and is expressed as a decimal (i.e., 75% is written as 0.75). To lower consumed power effectively, the efficiency component is the only variable factor in this equation for a given set of process conditions. Options do exist when selecting or beginning the design, however, since efficiency is a function of the pump's specific speed.

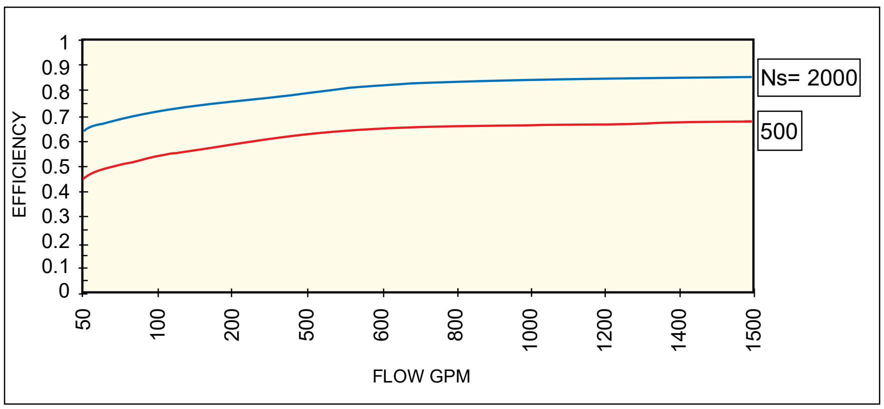


Figure 1. Efficiency as a function of capacity and specific speed

Specific Speed and Efficiency

Much has been written about "specific speed" (NS), a term that relates the pump performance to shape or type of impeller. It is not our intent to review the background of the term, but a few basic premises must be explained. The first is that a pump's best efficiency point (BEP) corresponds to the head, flow, and speed combination matching the impeller's design specific speed. Thus, the closer a match between a pump's normal operating point and the BEP, the less energy it will consume.

The second premise is that, inherent to the definition:

$$N_S = N \times Q^{0.5}/H^{0.75} \qquad (1)$$

specific speed is directly proportional to rotating speed (N) in revolutions per minute and the square root of flow (Q) in gallons per minute while inversely proportional to the three quarter power of head (H) in feet. Finally, there are physical constraints to the best efficiency that trend flow – that is, there is increasing efficiency with increasing flow. This relationship is depicted in Figure 1 showing an expected band of achievable efficiency within a flow range representative of the vast majority of pump installations. It will be shown later that the effect of flow is more accurately related to the size of the pump. This is an important concept when considering the potential efficiency to be gained by increasing the specific speed by raising the rotating speed.

Staging Reduces Energy

Staging is one way to increase the efficiency for a given set of hydraulics.

By dividing up the head by the number of stages (Z), the effective specific speed is increased by $Z^{3/4}$. As a result, the overall efficiency approximately increases according to the relationship shown in Figure 1. When referring to this figure, use stage head to calculate the specific speed.

Speed Reduces Energy

Specific speed is directly proportional to the shaft rotating speed according to Equation 1. As the design speed is increased beyond typical electric motor speeds, the specific speed also increases for a given head and flow. As discussed earlier, efficiency generally increases with an increase in specific speed and flowrate. One could conclude that it is always better to increase the speed of a pump to maximize the specific speed and the efficiency. As we shall see later, however, higher speeds may not always be the best solution. Futhermore, most published data, such as that shown in Figure 1 taken from Karrasik (1985), can be seriously misleading for high speed pumps.

High speed pumps are generally smaller than low speed single stage or multistage pumps. Euler's equation shows us that head rise is proportional to impeller tip speed for a given exit blade angle. Tip speed is the product of the impeller tip diameter and the shaft rotating speed. Once the designer fixes the exit blade angle, the impeller diameter for a fixed design head decreases with increasing speed. Figure 2, established from experience with a large number of high speed rocket engine turbopumps, shows that the efficiency of a small pump is consistently lower than the efficiency of a larger pump. Consequently, it is not accurate to use Figure 1 directly for high speed pumps without correcting the data for size effects.

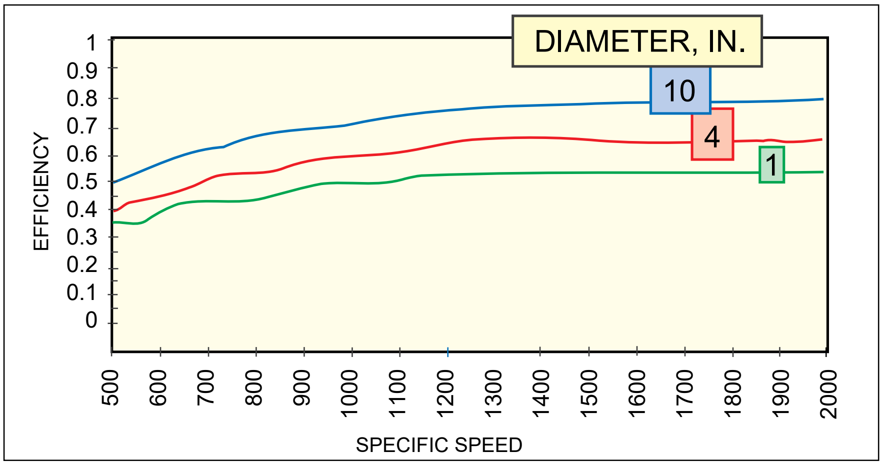


Figure 2. The effect of size on efficiency

The effect of impeller size on hydraulic efficiency at a fixed specific speed is due to surface finishes, friction factors, vane blockages, and internal leakage clearances that can not be practically scaled down from a large size. As a result, the percentage of hydraulic losses for small pumps is significantly larger than that for larger pumps. For impeller diameters greater than 10 inches, the size effect is small and generally insignificant. For diameters less than four inches, the efficiency penalty is particularly severe. The optimum speed for a given head and flow is sometimes a tradeoff between the efficiency gain from increased specific speed and the efficiency reduction due to smaller impellers. For most commercial pump applications, the gain from increased specific speed generally offsets the loss in efficiency due to size effects at higher design speeds.

Speed or Staging?

Tables of average hydraulic efficiencies (Table 1 and Table 2) have been calculated over a range of typical heads and flows for commercial applications using data adapted from Anderson (1980). The tables can be used to compare efficiency at different speeds and numbers of stages. These efficiencies are for common motor speeds of 1800 and 3600 rpm. The tables also include data for shaft speeds of 7200 and 14,400 rpm routinely obtained with commercially available speed increasing gearboxes. The calculations include size effects.

Note that increasing flowrates and shaft speeds both tend to produce higher pump efficiencies for a given head rise. At a given flowrate, the efficiency generally increases with increasing shaft speed. It should also be obvious from the tabulated values that reducing the head per stage increases the efficiency. Efficiencies tend to maximize at the higher flowrates with speed and multiple stages having minimal effects. The shaded portions of the table are usually impractical selections that one should avoid. For these applications, choose either multistage pumps, higher speed pumps, or mixed/axial flow impeller designs.

Overall Power

The most prevalent pump drivers in use today are two or four pole AC electric motors. Speed in excess of those capabilities, however, requires the use of alternative equipment. Turbines are routinely used in process plants that have low to medium pressure steam to attain speeds as high as 10,000 rpm. Mechanical gearboxes, both integral and free standing designs, are the norm with electric motors, but belt and sheave arrangements can also be used. Speed ratios as high as 10:1 are possible with some gearboxes giving the pump designer more than enough flexibility to optimize specif-

TABLE 1

CENTRIFUGAL PUMP HYDRAULIC EFFICIENCY

	HEAD PER STAGE, FT 250				HEAD PER STAGE, FT 500				HEAD PER STAGE, FT 750			
	SPEED, RPM:				SPEED, RPM:				SPEED, RPM:			
FLOW, GPM	1800	3600	7200	14,400	1800	3600	7200	14,400	1800	3600	7200	14,400
50	0.43	0.55	0.62	0.63	0.27	0.44	0.55	0.60	0.17	0.36	0.49	0.57
75	0.49	0.61	0.66	0.66	0.35	0.51	0.60	0.64	0.26	0.43	0.55	0.61
100	0.54	0.64	0.69	0.68	0.41	0.55	0.64	0.67	0.31	0.48	0.59	0.64
125	0.57	0.67	0.71	0.69	0.44	0.58	0.66	0.68	0.36	0.52	0.62	0.67
150	0.60	0.69	0.72	0.70	0.47	0.61	0.68	0.70	0.39	0.54	0.64	0.68
175	0.62	0.70	0.73	0.71	0.50	0.62	0.69	0.71	0.42	0.57	0.66	0.69
200	0.63	0.72	0.74	0.71	0.52	0.64	0.71	0.72	0.44	0.58	0.67	0.70
225	0.65	0.73	0.75	0.71	0.54	0.66	0.72	0.72	0.46	0.60	0.69	0.71
250	0.66	0.73	0.75	0.72	0.55	0.67	0.73	0.73	0.48	0.61	0.70	0.72
275	0.67	0.74	0.76	0.72	0.57	0.68	0.73	0.73	0.49	0.63	0.70	0.73
300	0.68	0.75	0.76	0.72	0.58	0.69	0.74	0.74	0.51	0.64	0.71	0.73
325	0.69	0.76	0.77	0.72	0.59	0.70	0.75	0.74	0.52	0.65	0.72	0.74
350	0.70	0.76	0.77	0.72	0.60	0.70	0.75	0.74	0.53	0.66	0.73	0.74
375	0.71	0.77	0.77	0.72	0.61	0.71	0.76	0.74	0.54	0.66	0.73	0.74
400	0.71	0.77	0.77	0.72	0.62	0.72	0.76	0.75	0.55	0.67	0.74	0.75
425	0.72	0.77	0.78	0.72	0.63	0.72	0.76	0.75	0.56	0.68	0.74	0.75
450	0.72	0.78	0.78	0.72	0.63	0.73	0.77	0.75	0.57	0.68	0.75	0.75
475	0.73	0.78	0.78	0.72	0.64	0.73	0.77	0.75	0.57	0.69	0.75	0.75
500	0.73	0.79	0.78	0.72	0.65	0.74	0.77	0.75	0.58	0.70	0.75	0.76
525	0.74	0.79	0.78	0.72	0.65	0.74	0.78	0.75	0.59	0.70	0.76	0.76
550	0.74	0.79	0.78	0.72	0.66	0.75	0.78	0.75	0.60	0.71	0.76	0.76
575	0.75	0.79	0.78	0.72	0.66	0.75	0.78	0.76	0.60	0.71	0.76	0.76
600	0.75	0.80	0.79	0.72	0.67	0.75	0.78	0.76	0.61	0.71	0.77	0.76
625	0.75	0.80	0.79	0.72	0.67	0.76	0.78	0.76	0.61	0.72	0.77	0.76
650	0.76	0.80	0.79	0.72	0.68	0.76	0.79	0.76	0.62	0.72	0.77	0.77
675	0.76	0.80	0.79	0.72	0.68	0.76	0.79	0.76	0.62	0.73	0.77	0.77
700	0.76	0.80	0.79	0.72	0.69	0.77	0.79	0.76	0.63	0.73	0.78	0.77
800	0.77	0.81	0.79	0.71	0.70	0.78	0.80	0.76	0.64	0.74	0.78	0.77
900	0.78	0.82	0.79	0.71	0.71	0.78	0.80	0.76	0.66	0.75	0.79	0.77
1000	0.79	0.82	0.79	0.71	0.72	0.79	0.80	0.76	0.67	0.76	0.80	0.78
1250	0.81	0.83	0.79	0.70	0.75	0.81	0.81	0.76	0.70	0.78	0.81	0.78
1500	0.82	0.83	0.79	0.69	0.76	0.82	0.81	0.76	0.72	0.79	0.81	0.78
NOTE:		250 > Ns > 4000 AND DIAMETER > 15 IN.										

TABLE 2 CENTRIFUGAL PUMP HYDRAULIC EFFICIENCY

	HEAD PER STAGE, FT				HEAD PER STAGE, FT				HEAD PER STAGE, FT			
	1000				1250				1500			
	SPEED, RPM:				SPEED, RPM:				SPEED, RPM:			
FLOW, GPM	**1800**	**3600**	**7200**	**14,400**	**1800**	**3600**	**7200**	**14,400**	**1800**	**3600**	**7200**	**14,400**
50	0.09	0.30	0.45	0.54	0.03	0.24	0.41	0.51	0.00	0.20	0.37	0.48
75	0.18	0.37	0.51	0.59	0.12	0.33	0.47	0.57	0.07	0.28	0.44	0.54
100	0.24	0.43	0.55	0.62	0.18	0.38	0.52	0.60	0.13	0.34	0.49	0.58
125	0.29	0.46	0.58	0.65	0.23	0.42	0.55	0.63	0.18	0.38	0.53	0.61
150	0.32	0.49	0.61	0.66	0.27	0.45	0.58	0.65	0.22	0.41	0.55	0.63
175	0.35	0.52	0.63	0.68	0.30	0.48	0.60	0.66	0.25	0.44	0.57	0.65
200	0.38	0.54	0.64	0.69	0.32	0.50	0.62	0.68	0.28	0.46	0.59	0.66
225	0.40	0.56	0.66	0.70	0.35	0.52	0.63	0.69	0.30	0.48	0.61	0.67
250	0.42	0.57	0.67	0.71	0.37	0.53	0.64	0.70	0.32	0.50	0.62	0.68
275	0.43	0.58	0.68	0.72	0.38	0.55	0.65	0.70	0.34	0.51	0.63	0.69
300	0.45	0.60	0.69	0.72	0.40	0.56	0.66	0.71	0.36	0.53	0.64	0.70
325	0.46	0.61	0.70	0.73	0.41	0.57	0.67	0.72	0.37	0.54	0.65	0.71
350	0.47	0.62	0.70	0.73	0.43	0.58	0.68	0.72	0.39	0.55	0.66	0.71
375	0.48	0.62	0.71	0.74	0.44	0.59	0.69	0.73	0.40	0.56	0.67	0.72
400	0.50	0.63	0.72	0.74	0.45	0.60	0.69	0.73	0.41	0.57	0.68	0.72
425	0.50	0.64	0.72	0.74	0.46	0.61	0.70	0.74	0.42	0.58	0.68	0.73
450	0.51	0.65	0.73	0.75	0.47	0.62	0.71	0.74	0.43	0.59	0.69	0.73
475	0.52	0.65	0.73	0.75	0.48	0.62	0.71	0.74	0.44	0.59	0.69	0.74
500	0.53	0.66	0.73	0.75	0.49	0.63	0.72	0.75	0.45	0.60	0.70	0.74
525	0.54	0.67	0.74	0.76	0.49	0.64	0.72	0.75	0.46	0.61	0.70	0.74
550	0.54	0.67	0.74	0.76	0.50	0.64	0.73	0.75	0.46	0.61	0.71	0.75
575	0.55	0.68	0.75	0.76	0.51	0.65	0.73	0.76	0.47	0.62	0.71	0.75
600	0.56	0.68	0.75	0.76	0.51	0.65	0.73	0.76	0.48	0.63	0.72	0.75
625	0.56	0.69	0.75	0.76	0.52	0.66	0.74	0.76	0.48	0.63	0.72	0.76
650	0.57	0.69	0.76	0.77	0.53	0.66	0.74	0.76	0.49	0.64	0.72	0.76
675	0.57	0.69	0.76	0.77	0.53	0.67	0.74	0.76	0.50	0.64	0.73	0.76
700	0.58	0.70	0.76	0.77	0.54	0.67	0.75	0.77	0.50	0.64	0.73	0.76
800	0.60	0.71	0.77	0.77	0.56	0.68	0.76	0.77	0.52	0.66	0.74	0.77
900	0.61	0.72	0.78	0.78	0.57	0.70	0.77	0.78	0.54	0.67	0.75	0.77
1000	0.63	0.73	0.79	0.78	0.59	0.71	0.77	0.78	0.56	0.69	0.76	0.78
1250	0.66	0.75	0.80	0.79	0.62	0.73	0.79	0.79	0.59	0.71	0.78	0.79
1500	0.68	0.77	0.81	0.79	0.64	0.75	0.80	0.79	0.61	0.73	0.79	0.79
NOTE:		250 > Ns > 4000 AND DIAMETER > 15 IN.										

Example 1.

Head = 500 ft
Flow = 300 gpm

At 3600 rpm:

Single stage efficiency = 69%
Two stages (250 ft/stage) = 75%

Or, at 7200 rpm:

Single stage efficiency = 74%

Specify either increased speed or staging to improve efficiency effectively.

Example 2.

Head = 750 ft
Flow = 100 gpm

Single stage efficiency:
At 3600 rpm = 48%
At 7200 rpm = 59%
At 14,400 rpm = 64%

Note the dramatic increase in efficiency by choosing a higher speed pump for this typical single stage application.

Example 3.

Head = 1000 ft
Flow = 175 gpm

At 3600 rpm:

Single stage efficiency = 52%
Two stage efficiency = 55%
Four stage efficiency = 70%

Note the increase in efficiency available with a four stage pump. Alternatively, one can select a single stage pump at higher speed.

At 14,400 rpm, efficiency = 68%

ic speed and, thereby, pump efficiency. Mechanical efficiencies generally decrease somewhat with increasing speed ratios. Regardless of which approach is taken, the efficiency of the drive train must be factored into the pump evaluation to ensure that maximum overall efficiency is attained.

How to Use the Tables

Use the tables to determine the potential for improving the efficiency of a pump for a given head and flowrate. Both the effects of staging and higher speed can be evaluated quickly. The following examples will illustrate:

For head, flow, or speeds not given explicitly in the table, use linear interpolation. Accuracy is sufficient for most comparison purposes.

Concluding Remarks

Today's marketplace presents ample opportunity for pump users to conserve energy through the optimization of pump efficiency. Pertinent information regarding the use of speed and staging can assist with an educated pump selection. On the surface, it may appear that all increases in rotating speed or a higher number of stages will result in comparable increases in efficiency, as a result of higher specific speeds. Remember, however, that the optimum speed or number of stages for a given set of hydraulics may be a tradeoff between efficiency gains and losses as a result of specific speed and size effects. A disciplined use of the accompanying table will give the pump user an appreciation of these tradeoffs, particularly as they impact the search for minimum consumed power.■

Dave Carr is a Project Manager with Sundstrand Fluid Handling (Arvada, CO). A graduate of Purdue University with a B.S. degree in Mechanical Engineering Technology and M.S. in Management, he has held various positions within the company's sales, marketing, product engineering and product development departments. Bill Mabe has been an Engineering Manager with Sundstrand for more than 20 years and is primarily responsible for new product development. A graduate of the University of Missouri at Rolla with a Mechanical Engi-neering degree, he is a member of the advisory committees for Pumps and Systems magazine and the International Pump Users Symposium.

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A Guide to ANSI Centrifugal Pump Design and Material Choices

Here's a pragmatic look at the commercially available design variations and material choices now being offered on pumps for corrosive/erosive and ultrapure service.

By Dan Besic

Over the past decade a number of significant articles in the trade press have focused on the wide variety of available pump designs that affect flow characteristics, service life, maintenance, safety, product purity, operating costs and other factors. Too often these erudite articles – written by highly specialized hydraulic, mechanical or materials engineers – are more valuable to pump designers and manufacturers than to those who specify, purchase, operate or maintain the equipment. This observation is particularly valid in the area of corrosive, abrasive, hazardous or ultra-pure fluids.

The adage that necessity is the mother of invention is a good explanation for the extensive design and material variations available.

The whole field of exotic metal and non-metallic pumps has been and continues to be market driven. The standard metal pump market is commodity oriented. In the pumping of neutral fluids such as potable water or ordinary wastes, the key factors are initial costs, availability and service. Design standardization and available modifications are aimed at cost reduction and operator-friendly aspects. Those responsible for price, delivery and dependable service are the main decision makers.

All of these purchasing influences are important when it comes to purchasing or specifying pumps for critical service applications. However, the handling of acids, caustics, solvents, halogens, salts, contaminated water, process fluids, ultrapure water or reagent grade chemicals and pharmaceuticals demands insight into design variations and material choices that are often specific to the fluid being handled or the operation involved. Pump design is still market driven but the speed with which new designs are translated into product availability at acceptable costs is controlled by the size of the market for the solution being offered. Plant engineering and maintenance personnel play critical roles. So do consulting and system engineers planning new installations or upgrading existing ones.

Historical Overview

Centuries of pumping experience involving water and pH neutral or mildly corrosive liquids lie behind the standard design of horizontal centrifugal pumps. As new materials of construction were developed, pumps became more compact, more efficient and more resistant to mildly acidic or alkaline solutions, as well as to various atmospheric conditions and temperature variations. With the growth of the chemical processing industries, pumps manufactured of stainless steel (types 304 and 316) became industry standards, with the molybdenum bearing type 316 holding sway because of its broader resistance to corrosion. As the market demanded even greater resistance, particularly to sulfuric acid, Alloy 20 became prominent.

For even greater resistance to heat, corrosion or abrasion, higher nickel alloys and various exotic materials were made available. Costs were driven up as alloy content rose.

Plastic pumps came into being about 50 years ago when heart/lung operations made it necessary to handle blood without contaminating it or destroying fragile red blood cells. The flexible liner, peristaltic type rotary pump developed for this life-saving critical application utilized an acrylic material called Lucite for the casing or pump body and a pure gum rubber as the flexible component. These were the only two parts in contact with the fluid. The gentle "squeegee" action on the blood and the use of noncontaminating nonmetallic materials solved the potential problem of red blood cell destruction and helped save lives. This unique sealless pump design filled an unmet need for transferring corrosive/ abrasive fluids at low flows without corrosion, contamination or seal leakage. Coincidentally, there was a major investment by DuPont and others to develop an extensive line of synthetic elastomers suitable for the liner. Industry researchers also sought to provide solid chemically resistant plastics such as Teflon, reinforced fluoropolymers, polyethylene and polypropylene for the casing. The mix and match potential for casings and liners (a dozen of these are now standard) has metamorphosed this "invention" for a particular problem into an almost universal design, a relatively low cost do-everything, seal-less plastic pump.

In another application, pump users handling bromine demanded improved service life and better worker environments than they were getting from stainless steel or nickel pumps. The answer found by pump designers relied on a new fluoropolymer, trade named Kynar, for all structural components. The market driven need for this corrosion and abrasion resistant material has so universalized its application that it has since become a standard recommendation for centrifugal pumps in an extended list of difficult-to-handle chemical-related processes, as well as those demanding high purity, such as pharmaceuticals or chemicals.

More recently, the demand for non-metallic ANSI horizontal centrifugals dimensionally interchangeable with metal ANSI pumps has become sufficiently large to enable pumps made of polypropylene, polyvinyl chloride and to be cost-effectively manufactured for a wide range of processing operations, as well as for water treatment and wastewater handling. For all these and similar market related reasons, non-metallic pumps in a variety of thermoplastic and thermoset formulations have emerged as serious alternatives to metal ones. The burden of reasonable choice, in many cases, is no longer linked to a particular innovative manufacturer of a specific pump in a specific material for a specific application. Users now have a choice among many metals and non-metallics.

The purpose of this article is to present an overview of commercially available design variations and material choices now being offered to users and specifiers of centrifugal pumps for corrosive/erosive and ultrapure service. When in doubt, there is no substitute for direct interchange with the manufacturer and your own or their experience with the specific fluids and service conditions involved with an application.

Centrifugal Pump Design

Taking the ANSI end suction centrifugal process pump as the standard, let us compare the design characteristics of the critical components made with thermoset and thermoplastic pumps, assuming for the moment that all these material choices are satisfactory for the application. First, a word about plastic-lined metal pumps. Plastic-lined metal pipe is proving to be an economical approach to transferring corrosive fluids, but when it comes to pumps, linings are not as easy to apply or maintain. They are, however, on the market. Generally, pumps are produced with a corrosion resistant lining applied in the wetted areas. Most common designs are those with fluoropolymer or thermoset linings approximately 1/8-3/16" thick. These pumps may be subject to early failure from erosion, pin holing, peeling or physical damage from solid particles, as well as from temperature fluctuation. Resistance to corrosion and abrasion in a pump is a lot more complicated than it is in a pipeline.

Since plastic lined pumps do not typically offer service life comparable to that achieved in pumps having solid molded plastic pump casings and impellers, the design variations of lined pumps will not be covered in this article.

Casing

Dimensionally, all pumps meeting ANSI B73.1 specification for process pumps are the same. The choice is yours. You can remove a metal pump and replace it with a plastic one, or vice versa, without changing the piping. As the need for a non-metallic pump to replace the stainless steel pumps in corrosive service became evident, it was natural for the manufacturers to offer designs made of various fiberglass reinforced plastics. The thermosets have physical properties closer to metals than the thermoplastics, and they combine this with many of the weight and chemical resistant advantages of non-metallics. Pressure ratings at ambient temperatures for flanges in thermoset pumps are equivalent to 150 lb. metal flanges of the same dimension. The pressure-temperature gradient is a linear factor and starts to degrade the material above 100°F, so if fluid temperatures run higher, some pump manufacturers will add metal backup rings to the thermoset support head, or they have support heads which include a backup ring.

Thermoplastic pumps designed so that the flanges and casings are completely supported and protected by heavy sectioned metal armor present no problem with respect to the pressure temperature gradient. Any of the available armored ANSI thermoplastic pumps will conform to the standards without modification.

Nozzle Loadings

As a general rule, the allowable nozzle loadings for thermoset pumps are lower than those indicated for metal pumps. Published data suggest that these pumps be exposed to nozzle loads 1/2 to 2/3 that of similarly sized stainless steel pumps. Armored thermoplastic pump designs conforming to ANSI B73.1 specifications have the discharge pipe and flanges reinforced (metal armored) so that they can accommodate the same nozzle loadings as the steel pumps they replace. Connections to metal pipe put the full load on the metal armor rather than on unsupported plastic as is the case with thermoset designs.

Shafts

Most non-metallic pumps use stainless steel or other stress proof steel shafts. Regardless of which material is used, it is recommended that the shaft be completely sleeved in plastic to isolate it from the fluid in the wetted area.

High strength carbon steel can be safely used when the shaft is so sleeved. The sleeve can be made of injection molded glass reinforced polyphelene sulfide (PPS) when the corrosive is relatively mild, but for best results shaft assembly should be made of the same material as the pump. Thermoplastic pumps should have sleeves of PVC (Polyvinylchloride), PP (Polypropylene), or PVDF (Polyvinylidene fluoride). The shaft sleeve in a horizontal centrifugal pump should be independent, not welded to the impeller. When so designed, a damaged shaft sleeve can be replaced by simply removing the impeller and then the sleeve. This approach is recommended because most users do not have plastic welding capability.

Mechanical Seal

For maximum corrosion resistance it is important that mechanical seal configurations be completely non-metallic, or mounted so that the inboard or wetted non-metallic seal face is exposed to the fluid. Since seal changes represent a significant maintenance item, it is critical that the pump design permit easy inspection of the seal without disassembling or removing the impeller shaft. In standard centrifugal pump designs which are not made to ANSI specifications, or which do not permit back pull-out, the available work area for seal inspection and maintenance is crucial. In some ANSI, as well as non-ANSI pumps, a sliding bar design, which permits backing up the primary seal for inspection without affecting shaft alignment, keeps downtime at a minimum.

Externally mounted seals facilitate maintenance of the pump/seal area by allowing personnel to inspect seal placement visually. This permits proper setting of the seals without relying, with fingers crossed, on shims or measurement. Pump stuffing box designs, which permit the widest choice of readily available single or double mechanical seals, provide a definite advantage.

Shaft Deflection

Over the years the complicated formula for shaft deflection, an important component for comparing potential seal life from centrifugal pumps, has been abbreviated L^3/D^4. This relationship between the length of the shaft overhang from the front or inboard bearing to the impeller and the diameter of the shaft at the seal face is generally satisfactory when comparing two metal pumps of similar design. However, it is useless and misleading when comparing a metal pump with a non-metallic one. The reason is simple. The full formula takes into consideration the diameter and the weight of the impeller. Impeller diameters of engineered plastic pumps are equal to or only slightly smaller than metal ones, but impeller weights are quite different. The lighter weight of the plastic impeller and the corresponding reduced downward thrust or force on the impeller end of the shaft are ignored by the L^3/D^4 abbreviation. These variables significantly affect the length/diameter ratio, making the common acceptable ratio (50) meaningless when applied to thermoset or thermoplastic pumps. Don't get caught in this trap. Insist on knowing the actual vibration level at the bearings – the actual shaft deflection at the seal face, not the "shaft stiffness" factor that the simplified formula provides.

Cost Factors

There was a time when plastic pumps were considered less expensive substitutions for metal ones. Those days are long gone. Non-metallic pumps for corrosive/erosive/hazardous or other specialized services are carefully engineered products with initial costs competitive to similar pumps of type 316 stainless steel. Cost advantages, however, become appreciable for those applications involving the higher alloyed materials and for pumps requiring titanium, nickel and other exotic metals. Published cost comparisons indicate that plastic pumps tend to be 20% lower in price than those of Alloy 20 (a nickel-chromium-copper alloy originally developed for handling sulfuric acid), and half or less than the cost of Hastelloy C, the high nickel stainless alloy developed for severe corrosive service.

Initial cost is but one factor, however. Service life is even more significant. Plastic pumps have been in service sufficiently long in highly corrosive/abrasive applications for the purchaser or user to ask for data on anticipated service life based on previous installations. Laboratory tests on metal or plastic "coupons" prove to be poor guides to pump life because they usually involve immersion time under static conditions. Corrosion or wear rates for parts rotating at 1800 or 3600 revolutions per minute, or stationary parts handling fluid flows to 5000 gallons per minute, are not comparable to those based on laboratory tests. Ask the pump manufacturer for actual case history data to be sure.

There are other cost factors to be considered when comparing metal versus plastic pumps, or one plastic pump versus another. Metal to plastic weight comparisons impact on costs involved in shipping, installation, disassembly/assembly, and time related problems such as galling or seizing of threaded components. Seal life, spare parts requirements and general maintenance requirements must be considered. In all of these areas, the non-metallics offer clear cut advantages because of their lighter weight (1/2 to 1/8 the weight of metal), lower component costs, non-corroding characteristics and longer seal life.

The differential in cost between thermosets and thermoplastics is negligible if we compare polypropylene and fiberglass reinforced plastic (FRP). Comparative service life is more critical, but this has to be based on the particular thermoplastic and specific thermoset. Generally speaking, homogeneous thermoplastics have much broader chemical resistance than thermosets, and thermosets offer physical properties higher than those of thermoplastics. For example, if the superior corrosion and abrasion resistance of the fluoropolymers is required, the higher cost of fluoropolymer thermoplastics such as PVDF or ECTFE is often repaid many times over by the extended service life of and/or product purity advantages these fluoropolymers offer.

Maintenance Factors

Standard, routine preventive maintenance programs are advisable for all pumps regardless of materials of construction or design factors. Your own service experience is the best guide to proper scheduling.

Where the installation is substantially different from what you have been involved with, it is wise to rely on the specific experience of the manufacturer. When shifting from metal pumps to plastic pumps, consider the following:

A. If you are connecting thermoset horizontal centrifugal plastic pumps to metal pipes, you may need to use expansion joints to reduce nozzle strain, particularly if there is a significant difference in the modulus of elasticity of the two materials. This may not be necessary for amored.

B. For vertical centrifugal pumps used in deep sumps, make sure the design accommodates the differential in axial thermal expansion between the long plastic support columns and the available impeller clearance. This, however, is not a factor for horizontal pumps.

C. Although non-metallic flanges are dimensionally the same as metal ones, if they are unsupported you may need metal washers under the bolt heads and nuts. Casing bolting for non-metallics requires careful adherence to torque specified by the manufacturer. This is true for all pumps but not as critical with metal or metal armored plastic ones.

D. Do not steam pressurize thermoplastic or thermoset pumps. This can cause components to be pressurized beyond ratings.

E. Protect outside mechanical seal with a seal guard or cover.

Comparative Material Characteristics

It is not the purpose of this article to concentrate on the long list of sophisticated materials of construction or erudite design factors over which the user has little or no control. These are basically the responsibility of pump manufacturers. Pump users need not be metallurgists or materials engineers. Their concern is and should be with understanding the relationship between available construction materials and the service conditions faced. Pump designers have an endless variety of materials from which the critical wetted end components can be selected. In reality, however, users are limited to choosing from among a relatively short list of commercially available materials. Exotic metal or customized composites may be required for pumping problems that can't be solved with materials that can currently be produced economically. Limited production potential often makes it difficult to produce pumps of these materials at reasonable cost and delivery schedules. The pump materials listed below are readily available for your consideration. Familiarity with the basic characteristics of these materials will help you select the most cost effective material for your applications.

Materials of Construction

Metals

Stainless Steel (type 316): A general purpose austenitic chromium (18%), nickel(8%), molybdenum (3%) alloy providing broad resistance to a long list of acids, caustics, solvents. Widely used for its atmospheric corrosion resistance and in products requiring sterilization. Not recommended for sea water, brine, bromine or strong oxidizing acids.

Alloy 20: Originally developed to resist various concentrations of sulfuric acid for which type 316 stainless steel wasn't suitable. This stainless alloy has higher nickel (28%) than chromium (20%) content, less molybdenum (2%), plus copper (3%). It offers good resistance to dilute and strong, as well as mixed acids, sulfate and sulfites, sulfurous and phosphoric acids, chlorides and brines.

Hastelloy "C": With an increase in the nickel content to approximately 50%, this nickel, chromium (22%), molybdenum (13%, tungsten (3%) alloy is recommended for uses in which the stainless steel and 20 alloys fail. It offers much greater resistance to oxidizing acids and acid mixtures, wet chlorine, chlorides, mineral acids, sea water and plating/pickling solutions.

Thermosets

Fiberglass Reinforced Plastic (FRP): Although there is an endless variety of thermoset formulations, many indicated by trade names, the most common standard formulations consist of vinyl ester and epoxy resin reinforced by glass fibers. Since the two components bring different values to the composite – vinyl for corrosion resistance, epoxy for solvent resistance – competitive formulations can be customized for a particular service. Thermoset materials generally offer broad resistance to most acids, caustics, bleaches, sea water and brine. Special formulations are available for mild abrasive service. Thermoset pumps of FRP have higher physical properties than those of thermoplastics and are available for flows to 5000 gpm, twice the flow of current thermoplastic offerings.

Thermoplastics

Polyvinyl Chloride (PVC): Widely used in chemical processing, industrial plating, chilled water distribution, deionized water lines, chemical drainage and irrigation systems. Good physical properties and resistance to corrosion by acids, alkalies, salt solutions and many other chemicals. Not suitable for solvents such as ketones, chlorinated hydrocarbons and aromatics.

Chlorinated Polyvinyl Chloride (CPVC): Its chemical resistance is similar to, but slightly better than, PVC. It is also stronger. Excellent for hot corrosive liquids, hot and cold water distribution and similar applications.

Polypropylene (PP): A light weight polyolefin chemically resistant to organic solvents as well as acids and alkalies. Generally not recommended for contact with strong oxidizing acids, chlorinated hydrocarbons and aromatics. Widely used in water and wastewater applications and for laboratory wastes where mixtures of acids, bases and solvents may be involved.

Polyvinylidene Fluoride (PVDF): Strong, tough and abrasion resistant fluoro-carbon material. Resists distortion and retains most of its strength at elevated temperatures. Recommended for ultrapure water and reagent chemicals. Resistant to most acids, bases and organic sol-

PVC	Polyvinyl chloride	140°F
PP	Polypropylene	185°F
PE	Polyethylene	200°F
CPVC	Chlorinated polyvinyl chloride	210°F
FRP	Fiberglass reinforced plastic	250°F
PVDF	Polyvinylidene fluoride	275°F
ECTFE	Ethylene chlorotrifluoroethylene	300°F
PTFE	Polytetrafluoroethylene	500°F

Table 1. Temperature

vents and equally suited for handling wet or dry chlorine, bromine and other halogens.

Ethylene Chlorotrifluoroethylene (ECTFE): Resists an extremely broad range of acids, alkalies, organic solvents and combinations of them, as well as other corrosive and abrasive liquids. Also resistant to oxidizing acids and hydroxides. Ideal for ultrapure water applications.

Polytetrafluoroethylene (PTFE): This crystalline polymer is the most inert compound known. It has useful mechanical properties at elevated temperatures. Impact resistance is high, but tensile strength, wear resistance, and creep resistance are low in comparison with other engineered plastics. Its coefficient of friction is lower than almost any other material.

Selection Criteria

Temperature: Temperature parameters are not critical in determining the choices among the various metals. Metallurgically, the differences may be great, but all of the metals considered for handling corrosive fluids are stable at most operating temperatures. When it comes to the plastics, however, anticipated temperature fluctuations are very critical. Table 1 provides upper temperature limits for the nonmetallic pump materials currently available.

Weight: The specific gravity of the material can be a significant variable when comparing metal to plastic pumps because of weight-related costs such as shipping, installation, support structures and in-plant repositioning. It is also vital in the comparison of various plastics with each other. As anyone who has lifted a fluoroethylene polymer casing can tell you, not all plastics tip the scale in the same way. A PVDF casing may weigh twice as much as one molded in polypropylene. Of course, when comparing metal-armored plastic pumps or plastic-lined pumps with all metal ones, the apparent differences are minimized. Table 2 shows the comparative weight of the materials of construction. If the weight factor is significant, ask the manufacturer for total pump weights. You may need these to assist in installation and piping.

Abrasion Resistance: Strange as it may seem, stainless steel pumps are relatively poor compared to plastics when it comes to resisting wear from abrasive fluid streams. A major reason for this is that the oxidized surface which protects passivated chromium-nickel stainless steel from corrosion is continuously removed by abrasive particles. The smooth,

Material	Weight Loss/ 1000 Cycles
Polyethylene (UHMW)	5 mg.
PVDF	5-10
PVC	12-20
PP	15-20
CPVC	20
Stainless Steel	50
FRP	388-520
PTFE	500-1000

Table 3. Abrasion resistance (Tabor Abrasion Tester)

Weight	Specific Gravity
PP	.91
PE	.92 - .94
PVC	1.30
CPVC	1.49
PVDF	1.75
ECTFE	1.75
PTFE	2.14 - 2.20
FRP	3.4 - 5.0
316 Stainless	7.9

Table 2. Weight differentials of construction materials

Material	Tensile Strength (psi)	Hardness R = Rockwell D = Shore B = Barcal	Impact (IZOD)
PE	3,500 - 5,600	R 35-40	1.5 - 12.0
PP	4,000 - 5,000	R 80-110	0.5 - 2.2
PTFE	2,000 - 5,000	D 50-55	3.0
PVDF	5,500 - 8,250	D 80	3.6 - 4.0
ECTFE	6,500 - 7,500	D 75; R 93	No break at 73°F
PVC	6,000 - 7,500	R 113	0.4 - 2.0
CPVC	7,500 - 11,000	R 121	0.6
FRP	10,000 - 13,000	B 35-40	–

Table 4. Strength of the non-metallics

uniform interior surface of a molded thermoplastic casing is a significant factor in reducing friction and turbulence, both of which contribute to wear. The FRP materials rely on the epoxy/vinyl ester to contain the glass. If the surface is subject to fast flowing process streams containing solid particles, abrasion can be severe. The reinforcing fibers may be exposed causing degradation of the composite or a wicking/bleeding action that can contaminate the fluid. For these reasons, homogeneous thermoplastics offer much greater resistance to abrasion. Table 3 shows typical weight loss of the materials being considered.

Strength of Non-Metallics: The differential in tensile strength and impact resistance between metal and non-metallic pumps may be of academic interest, but insofar as users are concerned, it is significant to keep in mind that unarmored non-metallic pumps need more care in handling than metal ones. Metal armored thermoplastic pumps can be treated as metal ones, but thermoset pumps require a bit more attention. Reinforced epoxy/vinyl composites can be brittle (like cast iron), so precautions should be taken to provide protection from falling overhead objects, fork lift trucks or careless handling. Tensile, hardness and impact strength of the various non-metallic materials are shown in Table 4.

Corrosion/Chemical Resistance: When pump specifiers and users are asked why they are considering non-metallics, corrosion resistance is by far the number one reason given. The resistance of the basic construction material is significant in terms of service life, safety, maintenance and, in many cases, the purity of the product being pumped. This latter criterion is essential for those using deionized water, reagent grade chemicals or other ultrapure fluids in their manufacturing or processing operations. When metallic or other contaminants cannot be tolerated, this resistance is critical because it may seriously affect the quality or the value of the product being produced.

Many handbooks and materials engineering articles give comparative corrosion resistance data for an exhaustive list of metals and plastics. These are rated against page after page of different chemicals at varying concentrations and at various temperatures. The tables are helpful guidelines, but for the most part, they are based on corrosion or deterioration when the subject material is immersed in the fluid under static and unchanging conditions. Conditions of service in the real world are seldom that uniform or controllable. These lists and tables are no substitute for actual pumping experience – your own and that of your suppliers'. Described below are highlights of some interesting pumping operations in which material selection played a critical role.

These examples may help you see your own pumping problems in a different perspective. If this article encourages you to review your company's pump purchasing and maintenance picture with a view to how you might extend service life, reduce parts inventory, simplify maintenance procedures, improve product quality, meet current or anticipated environmental regulations, or even feel comfortable about your current operating equipment and procedures, it will have served its purpose.

Examples:

1. Sulfuric Acid Waste Stream Containing 100 Micron Fines

This corrosive/erosive mixture was destroying 316 stainless steel pumps. The short service life and costly downtime could not be tolerated. When the switch was made to polypropylene pumps to handle the 3 pH abrasive wastewater, the problem was solved.

2. Corrosive/Abrasive Mine Water

Acid run off with a pH of 1-2 from coal mines in South Africa were severely corroding stainless steel pumps. The answer was found with a combination of non-metallics. Pump casings were supplied in polypropylene, but a PVDF impeller was specified because of the superior abrasion resistance of this fluoropolymer. To reduce problems with early seal failure, a Teflon (PTFE) packed gland and product flush arrangement was used instead of the standard single seal product flush. The pump has been handling 16-18 million gallons of acidic fluid per week at 1800 gpm. No problems.

3. Etching Glass with Hydrofluoric Acid

A major glass manufacturer experienced severe pump maintenance problems when it came to handling a slurry of hydrofluoric acid with a high content of proprietary gritty compound. Original equipment utilized a composite fiberglass pump with a stainless steel impeller. Service life averaged about 2 months. The plant manager tested a variety of non-metallics and finally decided on PVDF impellers for the corro-sive/abrasive service and polypropylene for the casing. To isolate the shaft from the fluid, he specified a thick sectioned PVDF sleeve and arranged to have the mechanical seal reverse mounted. The pumps are driven by 7.5 hp electric motors at 1750 rpm. The 10% production increase experienced is credited to keeping the production line operating with thermoplastic pumps.

4. Acid Mixing Plant in Alaska

An oil field acidizing and well service company in Alaska ordered a modular plant built in Texas and shipped to their Prudhoe Bay site. Some of the pumps are polypropylene throughout. Others are fitted with ECTFE impellers to handle the hydrochloric acid and zylene. Although the pumps are located indoors, the fluid temperature could be minus 60°F because the chemicals are stored outdoors. Service reports indicate no problems.

5. Circulating Phosphoric Acid

The use of lined metal pumps to handle phosphoric acid was causing extensive shutdown due to pinholes and tears that allowed the metal casings to contaminate the solution and downgrade it. Part of the problem was caused by heat generated in the pumps, which tended to destroy the thin linings. When solid PVDF thermoplastic pumps were substituted, product purity was assured and overall maintenance substantially reduced.

6. Ultra Purity for Hydrogen Peroxide Production

Specifications for these centrifugal pumps called for the use of pump casings, impellers and shaft sleeves to be made of virgin, unpigmented fluoropolymer totally resistant to 70% hydrogen peroxide being pumped at 50gpm against a total dynamic head of 80 feet. The material selected was ECTFE, ethylene chlorotrifluoroethylene, which is noted for its broad resistance to chemicals, pharmaceuticals and oxidizing acids. The seal rings were specified in the fluoroelastomer Viton and the casing jacket in Teflon PTFE.

7. Bleaching in White Paper Mills

The most corrosive applications in white paper mills involve pumping chlorine bleach. Traditionally, this has been done with extremely expensive titanium pumps. When environmental concerns required a switch from chlorine bleach to chlorine dioxide, pulp mills were able to change from titanium to epoxy vinyl ester pumps at much lower initial cost, reduced spare parts inventory and ready availability of pumps and parts.

8. Sulfuric Acid and a Caustic Chaser

A large can manufacturing division of one of the largest breweries in the world faced costly downtime due to excessive corrosion of the metal pumps transferring dilute sulfuric acid for the etch and strong caustic for the required neutralization. The decision was to change from stainless steel to PVC thermoplastic centrifugal pumps. All wetted end components were specified in the same homogeneous thermoplastic. The plastic pumps have served in this round the clock operation for more than a dozen years with only routine maintenance.

9. Hot Oil/Hydrofluoric Acid/ Caustic Mixture

A refinery using hydrofluoric acid in a process to produce high quality gasoline ran into difficult pumping problems due to the varying pH of the 150°F oil/HF/caustic mixture. They tried a variety of pumps from cast iron to 316 stainless steel, but a combination of corrosion plus impeller and seal damage from accumulated solids required pump replacement on a monthly schedule. Since installation of molded thermoplastic polypropylene pumps with double mechanical seals and pressurized water jackets using 8 to 10 gph of clear water to cool the seal jacket, the pumps have performed flawlessly.

10. Tall Oil Production in Kraft Mills

Tall oil soap, a byproduct of the Kraft paper making process. Skimmed from the evaporated cooking liquor, the soap is acidulated with sulfuric acid in a reactor vessel to achieve a pH of 2.5 - 3.5 and allowed to settle out. The tall oil rises to the top, and the spend acid can be either fresh acid (98%) diluted with water to 30% or with spent acid from the bleaching operations. FRP pump manufacturers report that the thermosets show improved service life over pumps made of 316 stainless or 20 Alloy.

11. Pumping Deionized Water for Laboratory Use

Medical laboratories utilize sealed diagnostic kits to analyze blood samples. To assure correct diagnosis, it is critical that the water used to prepare the various chemical solutions be free of contaminants. Pretreatment of the processed water requires a sand filter, deionization equipment and special fine pore filtration media to eliminate all particulates. Once the water is purified, it is protected by a fluid handling system composed entirely of chemically inert plastics. It is stored in polyethylene tanks and pumped through a closed loop system of rigid PVC pipe by a horizontal centrifugal pump with all wetted components of PVDF fluoropolymer. This system uses close coupled pumps with an integral pump/motor cantilevered shaft that enables the pump to run dry for extended periods without damage.

12. Pump Maintenance Reduced by $25,000

The extreme corrosiveness of an ammonium chloride/zinc chloride solution at 140°F resulted in the failure of Alloy 20 pumps after a year of service. In addition to the high cost of this annual pump replacement, mechanical seal failure occurred on a monthly basis requiring a minimum of 2 hours of lost process time each month. When the metal pumps were replaced with those made of virgin thermoplastic polypropylene, the following results were reported: lower initial pump cost, improved resistance to both corrosion and cavitation, an average seal life of nine months, and better than two years of service before removal of the pumps for reconditioning. Annual maintenance costs decreased by $25,000.

13. Silicon Wafer Production Demands PVDF Pumps

Production processes in the manufacture of high quality silicon wafers require the use of hydrochloric acid, hydrofluoric acid and various caustic solutions. The manufacturer had standardized on thermoset materials for the pumps, but the corrosiveness of the chemicals and mixed acid wastes proved too severe. Hazardous fluid leakage and reduced pump performance, particularly in the handling of the hydrofluoric acid waste streams, necessitated a changeover to polypropylene pumps with PVDF shaft sleeving in the wetted area. Plant management reports that leakage and capacity problems are a thing of the past.■

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Self-Priming Pumps: It's in the System!

Coordinate pump with piping system for optimum performance.

By Ray Petersen

With the advent of increasingly stringent environmental regulations pertaining to storage tank connections below liquid levels, self-priming centrifugal pumps are being utilized in a wide variety of applications. In the past, horizontal designs have been employed in these services due to the inherent advantages of centrifugal pumps. Installation was relatively simple, and if the suction and discharge requirements of a particular pump were met, satisfactory performance could be expected.

Converting a centrifugal pump application that previously operated off of a positive suction head to one consisting of a negative, or combination negative and positive suction head, requires additional criteria to be met for a successful application. The addition of a compressible fluid (air) in the suction line imposes conditions on the piping system that must be overcome for a self-priming pump to remove air from the suction line and pump fluid as a centrifugal unit is designed to do. Piping systems composed of check valves, pipe loops and liquid traps should be reviewed with respect to the pump operation during the priming cycle to assure that all air can be evacuated from the suction line prior to the pump moving liquid only.

Centrifugal Pump Operation

The basic theory of operation of a centrifugal pump – from which its name is derived – relates to centrifugal force. A rotating impeller imparts velocity energy to the fluid between the impeller vanes, and that velocity is subsequently converted to pressure energy in the volute section of the pump casing.

This increase in pressure energy (total dynamic head on a pump curve) is proportional to the centrifugal force applied to the fluid. Since centrifugal force is directly proportional to weight, the difference between the discharge head of a centrifugal pump filled with water and a centrifugal pump filled with air is in the order of 810 to 1! (This is the difference between the density of water and the density of air at atmospheric pressure.) A pump handling a mixture of air and liquid will exhibit discharge heads between these two limits.

$$\text{TDH} \sim \text{centrifugal force} = \frac{w\omega^2 r}{g}$$

w = weight
ω = angular velocity
r = radius
g = gravitational acceleration

Density water at atmospheric pressure = 62.4#/ft^3

Density air at atmospheric pressure = .077#/ft^3

From this it is clear that large static heads cannot be imposed on the discharge side of a centrifugal pump if it handles air during some phase of its operation.

Self-Priming Centrifugal Pump Operation

There are many types of self-priming centrifugal pumps. Included in this category are submerged impeller designs such as vertical and submersible pumps and converted horizontal pumps utilizing auxiliary vacuum producing equipment or suction priming tanks that allow a standard centrifugal pump to operate with a positive liquid head at all times.

The most popular self-priming centrifugal pump used commercially is the "recirculation" or "peripheral priming" type. It is characterized by a liquid reservoir either attached to or integrally constructed with the pump casing (Photo 1). The suction connection is usually located above the impeller centerline in order to contain and trap a volume of liquid used in the priming cycle.

(Photo courtesy of Fybroc Pump Div.)

The popular "recirculation" or "peripheral priming" type self-priming pump is characterized by a liquid reservoir either attached to or integrally constructed with the pump casing

Before the pump is initially started, the liquid reservoir must be filled manually. When the pump is shut down, a syphon breaker or internal suction check valve retains a quantity of the pumped fluid in the reser-

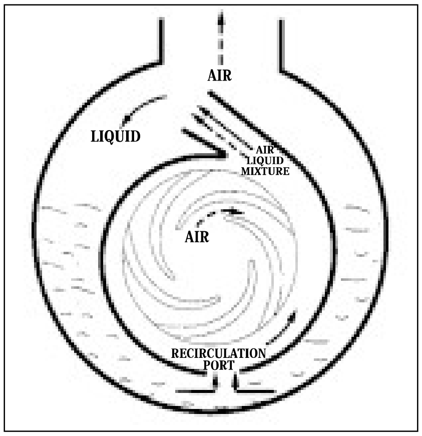

Figure 1. Internal construction of a self-priming centrifugal pump

voir for successive starts.

The internal construction of a self-priming pump is similar to a conventional centrifugal pump except for the addition of a recirculating port in the volute passage that is connected to the fluid reservoir (Figure 1). As the impeller rotates during the priming cycle, the liquid in the impeller and volute passage is discharged out of the volute into an expansion or air separation area of the liquid reservoir. However, before the air separation area of the pump can be filled with liquid and the pump considered primed and capable of generating a pressure against the discharge piping system, air from the suction line enters the impeller eye, causing a drop in pressure due to the relative densities of air and liquid.

As the impeller continues to rotate, the mixture of air and liquid moving at high velocity will draw additional liquid into the impeller and volute area through the recirculation port. This mixture is then discharged past the volute cutwater into the expansion area of the reservoir. In the expansion section, the liquid and air bubbles separate. The air, being lighter, vents upward out the pump discharge while the heavier liquid returns to the reservoir and continues to recirculate and entrain more air. This cycle continues until all the air in the suction line and impeller is evacuated, at which point the pump is primed (Figure 2).

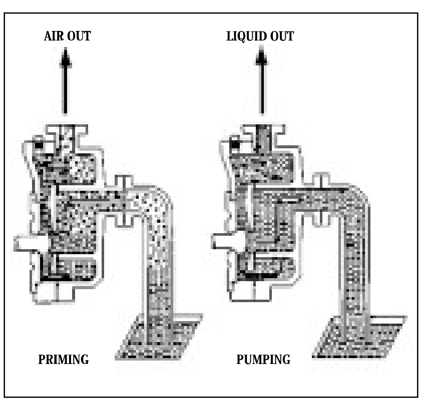

Figure 2. The self-priming cycle

With the pump primed, the reservoir is at full pump discharge pressure. Flow through the recirculation port is minimal as pressures are nearly balanced, and in some designs the port functions as an auxiliary cutwater – thus reversing flow direction. Efficiencies on this type of pump approach efficiencies on standard centrifugal pumps for the same capacity and head range.

Suction lifts up to 25ft are attainable with this design, depending on impeller diameter and speed. Close clearances between the impeller diameter and cutwater tip assure that the liquid/air mixture is discharged out of the volute area and not recirculated, which would cut down on suction lift capability. Some self-priming pumps are designed with replaceable or adjustable cutwater tips to compensate for wear. Thus, plant personnel can renew priming lift ability without replacing the entire pump casing.

Discharge Piping Systems

One important fact about the priming cycle is that the air evacuated from the suction line is at a very low pressure at the pump discharge. As noted earlier, little pressure energy can be recovered as low density air is passed through the impeller and separated from the liquid. A typical plot of pressures observed at the pump discharge during the priming cycle is shown in Figure 3.

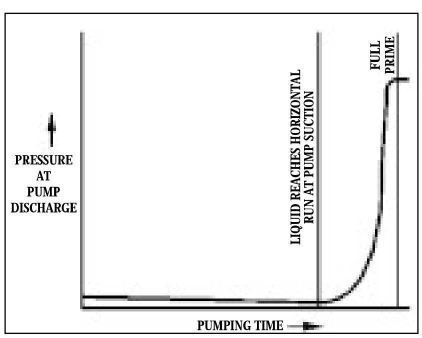

Figure 3. Typical pressures observed at pump discharge during priming cycle

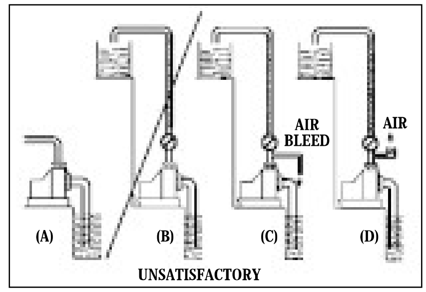

Figure 4. Discharge piping systems

Traditional applications for self-priming pumps, such as dewatering an excavation or pumping out a flooded basement, normally don't require discharge piping (Figure 4a). Air from the suction line is easily expelled at the pump discharge until the unit is primed.

However, when the application requires the addition of a discharge piping system and incorporates a check valve to prevent backflow or to stop water hammer in high vertical runs of pipe when the pump is shut down (Figure 4b), a problem arises during the priming cycle. When the pump is started, the check valve in the discharge line prevents the air from being evacuated out of the suction line because it cannot develop enough pressure to overcome the head of liquid keeping the check valve closed. With no place for the air to vent, the pump will not prime, which can lead to pump or mechanical seal damage.

Figures 4c and 4d illustrate options to prevent discharge piping

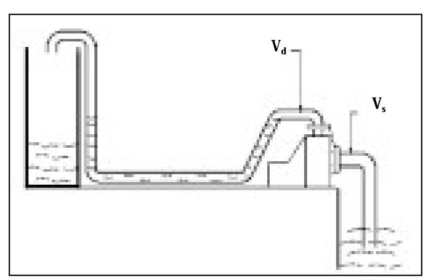

Figure 5. Example of discharge piping system containing liquid loop or trap

system backflow yet allow the self-priming pump to evacuate air from the suction line. The air bleed line in Figure 4c should not be installed below the liquid level or contain any liquid traps to impede air flow from the pump. The air release valve shown in Figure 4d allows the air to escape and seal once the pump is primed.

Discharge piping systems containing liquid loops or traps similar to those shown in Figure 5 should be avoided. If the air occupying the pump and suction line volume (V_s) cannot be added to the air in the discharge volume (V_d) without exceeding the low pump discharge pressure during the priming cycle, provisions should be made to allow the suction air to vent as in Figures 4c and 4d.

Suction Piping Systems

As applications for horizontal self-priming pumps have expanded, suction piping systems have progressed from a simple suction hose in a ditch to more complex piping arrangements designed for transferring liquids from tanks or rail cars to other storage facilities.

The suction system shown in Figure 6 is a piping arrangement for tank car unloading operation. At startup, full discharge pressure is experienced as the pump is filled with liquid. The unit continues to pump the initial volume of suction liquid into the discharge system, and when the air in the suction line is encountered, the discharge pressure drops as the pump enters the priming mode. The location of the initial volume of pumped liquid in the discharge piping system should be reviewed, as previously discussed, to be sure it does not present a restriction to the evacuation of the suction piping air. Again, a bleed line or air release valve may be necessary.

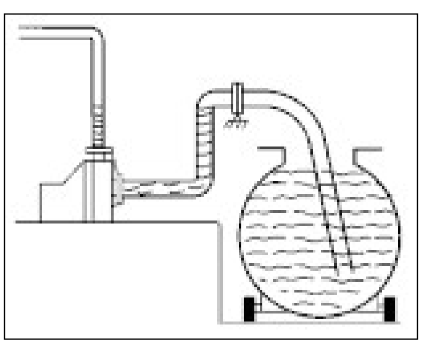

Figure 6. Suction piping system for tank car unloading

Vortexing

Vortexing is the introduction of air into a pump due to insufficient depth of liquid above the pump suction line. High velocity flow patterns at the suction pipe entrance induce vortices or whirlpools in the liquid, and these open up a channel allowing air to enter the pump.

When a tank is pumped-down to its lowest level, vortexing may occur. At this point in the pump operation, the discharge pipe system will be completely filled. To continue pump-down below this level, the air must be able to pass through the pump at reduced discharge pressures. As the liquid flow through the pump is reduced from the air drawn into the suction, the vortexing will subside and the pump will reprime itself provided the air can pass through the pump. Upon re-prime, full flow will be realized and the vortexing will reappear. This alternate primere prime cycle should be avoided since it can lead to premature bearing or seal wear if the frequency is too high.

This problem can exist in other self-priming centrifugal pumps. Submersible installations can experience difficulty if air enters the pump suction from vortexing and cannot exit because of the closing of a discharge check valve due to reduced pressure.

Suction Volume

The volume of air to be evacuated from a suction piping system should be examined to prevent pump and/or mechanical seal problems. During the priming cycle heat is being added to the fluid in the pump reservoir from the recirculation of the priming liquid. In most cases the mechanical seal is being cooled by the pumped liquid, and in the priming cycle the heat being generated by the seal faces is also being absorbed by the priming liquid. During extremely long priming cycles and high suction lifts the priming liquid can evaporate. This means the pump never reaches prime, a situation that can result in mechanical seal failure.

Data on most commercial self-priming pumps include priming time curves in addition to head-capacity characteristics. They show the time required to evacuate vertical suction lines as a function of the distance of the pump above the liquid level. These characteristics are commonly referred to as lift curves.

Lift curve characteristics are based on a liquid with a specific gravity of 1.0. If different specific gravity liquids are to be pumped, an equivalent lift should be used to determine the priming time.

equivalent suction lift = suction lift x specific gravity

Increasing the suction pipe diameter to reduce friction losses, or including long runs of horizontal pipe to be evacuated, lengthens the time required for priming.

Increasing the vertical suction pipe size results in an increase in priming time proportional to the square of the pipe diameters. Horizontal runs of suction pipe and varying pipe diameters require a numerical integration of the lift curve based on the location of the diameter changes and horizontal runs to determine the resultant priming time. If system priming times exceed the maximum times shown on the lift curve, the manufacturer should be consulted.

In some instances with large suction volumes, an auxiliary line can be installed to add make-up liquid to the pump to prevent the priming fluid from boiling off.

NPSHA/Suction Lifts

One important item to check on the suction piping arrangement of a self-priming centrifugal pump is the net positive suction head available (NPSHA). While NPSHA should be reviewed in all pumping applications, it is especially necessary on pumps operating with a suction lift. NPSHA is the absolute pressure available to push the liquid through the suction piping up to the pump suction.

NPSHA is defined as:

$$NPSHA = P_{BAR} - P_{VP} - P_{FR} + P_{HGT}$$

Where P_{BAR} = barometric pressure
P_{VP} = liquid vapor pressure
P_{FR} = suction line pressure drop due to friction
P_{HGT} = height (pressure) of liquid surface relative to the pump

Note that all the terms making up NPSHA are pressure terms and can be expressed in feet of liquid. In the case of a suction lift, the last term (P_{HGT}) would be negative as the pump is above the liquid surface.

Noting that P_{BAR} (barometric pressure) at sea level is 34ft of water; with high suction lifts, the NPSHA declines rapidly even before the vapor pressure and friction loss terms are deducted.

The resultant NPSHA must always be greater (usually with a margin of 2-3 ft) than the NPSHR of the pump to prevent cavitation. NPSHR, the net positive suction head required, is a characteristic of the particular pump. It is determined by test and shown on pump performance curves as a function of flow rate. NPSHR can be viewed as a measure of the ease of moving liquid through the pump, and it increases with flow rate and speed.

Air Leakage

While leakage is not normally considered in piping design, it is being mentioned here because it is a primary source of problems in self-priming pump performance. While a self-priming pump can produce a fairly high vacuum, it is not designed to handle large volumes of air.

A small suction air leak at a high suction lift will, in most cases, prevent the pump from ever reaching prime. A self-priming pump with a 3" vertical suction line and a 10 ft suction lift that can prime in 30 seconds has an average air handling capacity of only approximately 1 cfm. The equivalent leakage area to flow 1 cfm at a 10 ft suction lift vacuum is only .002 in^2!

For this reason gaskets and seals should be in good condition to prevent air leakage during the priming cycle. Packing is not recommended for use in sealing self-priming pumps since it is prone to leakage under the negative operating pressures encountered during priming. If priming difficulties are experienced and air leakage is suspected, a common practice is to isolate the suction piping system from the pump and check the pump for blank vacuum. Blank vacuum gauge readings should be greater than the suction lift requirements. Typically, they run in the 20-25 in.Hg vacuum range.

Summary

Horizontal self-priming centrifugal pumps have all of the inherent advantages of standard centrifugal pumps such as:

- low initial cost
- high capacity
- high suction lifts
- relatively high discharge heads
- ease of installation
- ease of operation
- ease of service
- ability to handle dirty or solids laden liquids

In addition, a self-priming pump can evacuate the air in the suction line prior to pumping liquid. However, while it can produce high lift vacuums, it cannot discharge the evacuated air against any back pressure.

The key to a successful installation is to match the pump to the application, then match the piping system to the pump. Avoid the two common pitfalls of self-primer installations – namely regarding the pump as an air compressor and imposing back pressure on it during the priming cycle, and allowing air leakage in the suction line or pump seal area. Following these guidelines will result in relatively trouble free operation and allow the pump to systematically remove the air out from the suction line. This will result in efficient pumping of liquid, and upon shutdown, the pump will retain enough liquid to repeat the cycle.■

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Know the Inside Story of Your Mag Drive Pumps

Accurate monitoring of temperature and pressure is key to reliable operation.

By Harry Schommer

Editors Note:

Mr. Schommer, of Waldkraiburg, Germany prepared this article for Pumps and Systems. To assist our U.S. readers not entirely familiar with SI (metric) units, we have included an english/metric conversion chart at the end of this article.

The pump shafts in sealless magnetic driven pumps are held in place by sleeve bearings. Therefore, these bearings must be located in the pumped liquid. Today, the most common bearing material is pure silicon carbide. Silicon carbide bearings work without problems in low viscosity liquids such as chemicals, hydrocarbons, solvents, acids, all kind of hydroxides and also in abrasive pumpage. With an additional diamond-layer, the material provides dry running capabilities.

The widely used term "process lubricated bearings" is not quite correct since there are no lubrication grooves applied and there is no defined flow through these bearings provided in conventional sleeve bearing design. Liquids such as propane, ethylene oxide or methylene chloride provide no lubrication capability. Similar to the situation between the faces of mechanical seals, only a stable fluid film is required between the slide faces. The stability of this film depends on temperature and pressure. If temperature rises inside the pump above vapor temperature of the pumped liquid, vaporization will break down this film. Under these conditions, the bearings will run dry and fail sooner or later. A reliable temperature monitoring system is required to avoid this situation.

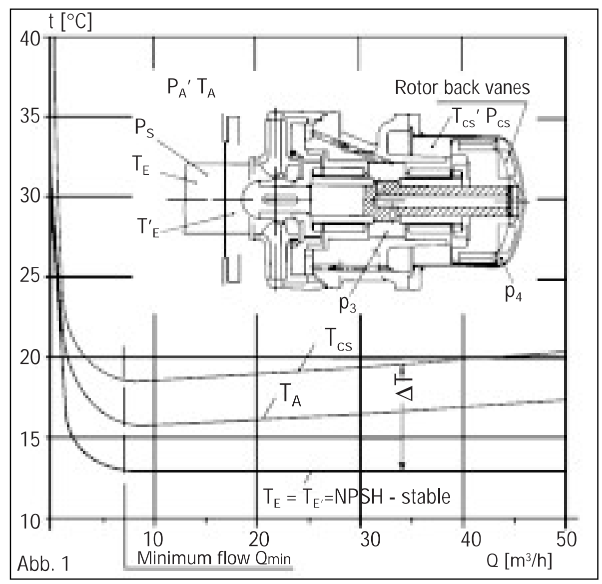


Figure 1. Internal circulation, temperature behavior

Besides vaporization of fluid inside the pump, dry running of an empty pump is the worst operating condition. Because of the starved suction, there is no flow to any part of the pump. Although the diamond-layer of the SiC-bearings will tolerate this situation because no hydraulic loads are acting, the built up heat which cannot be dissipated because of the starved internal circulation will lead to demagnetization of the coupling.

Magnetic drive centrifugal pumps with metallic containment shells

Temperature Rise in Single Stage Volute Casing Pumps with Magnetic Couplings

Internal Circulation

Sealless pumps with magnetic couplings and metallic containment shells generate eddy currents that lead to heat and cause temperature rise of the pumped liquid in the containment shell. In order to prevent this, heat must be dissipated through an internal cooling flow. This cooling flow – branched off as a partial flow from the main flow and led through the gap between internal rotor and containment shell – is shown in Figure 1.

The circulation flow is drawn from the discharge side behind the impeller, led into the chamber between the slide bearings and through the pump shaft via the rotor back vanes, and returned to the discharge side. This arrangement pressurizes the slide bearings and the containment shell with nearly the full discharge pressure, and helps to avoid flashing of the liquid in this area caused by heating up the product. Where the temperature increase is critical, the maximum pressure P_4 prevails. It should be noted that there is no heated liquid flowing back to the suction side or impeller. Therefore, no negative influence on the NPSH required will occur. For this type of pump, handling of volatile liquids is not a problem.

In pumps without rotor back vanes or rear impellers, the internal cooling flow is driven by the pressure gradient within the pump from discharge side to suction side, i.e. back to the impeller eye. In this case, problems may arise in pumps handling volatile liquids. Sufficient NPSH reserves must be available to accommodate the heat-conditioned rise of the pump's NPSH value. Exact temperature measurements of the cooling flow after passing the magnet area are also impossible.

Temperature Rise, Minimum Flow Conditions

Figure 1 further displays the temperature behavior in a volute casing pump with a magnetic coupling. The pump size is 50/200, 2900 rpm; magnetic losses are 3,0 kW and the pumped liquid is water. When reviewing the temperature curve it must be considered that the flow leading through the magnet chamber is dependent only on the geometry of the rotor back vanes and on the speed. This means that, independent from capacity and differential head, a stable circulation flow exists that adopts the dissipated heat and leads it into the main flow. Since the magnetic losses of a given magnet coupling will not change during operation at constant speed, an almost stable temperature increase ΔT is produced in the range to the right of the minimum flow. However,

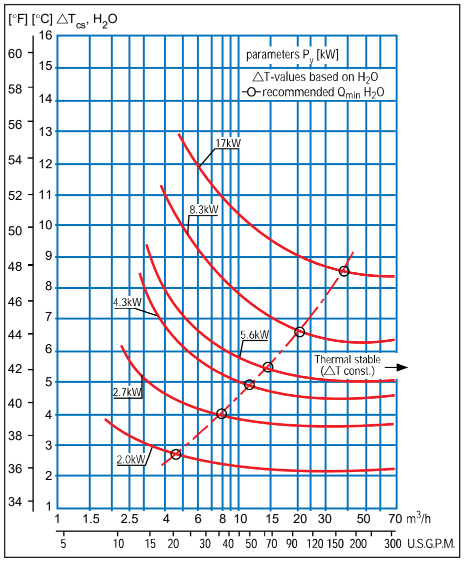


Figure 2. Temperature increase

if the minimum flow drops below, temperatures will rise remarkably. This is the reason why these pumps cannot be operated against closed discharge valve. Experience has shown that most of the slide bearing damages are a result of neglecting this fact. If process conditions dictate this operation, a bypass line must be installed from discharge to suction vessel.

Numerous temperature measurements on magnetic drive pumps with different sizes, rotor diameters, containment shell materials and speed have proved a direct relation between the capacity Q, the magnetic losses P_v and the temperature ΔT_{cs} inside the containment shell. These relationships are displayed in Figure 2, based on water at 20°C. Determination of the actual temperature increase inside the containment shell for other liquids depends on the product.

$$\Delta T_{cs,product} = \Delta T_{H_20} \cdot \frac{spec.heat H_20}{spec.heat\ product} \cdot \frac{density H_20}{density\ product} \quad [°C]$$

Knowing the inlet temperature T_E, the containment shell temperature T_{cs}, product is determined as follows:

$$T_{cs,\ product} = T_E + \Delta T_{cs,product} \quad [°C]$$

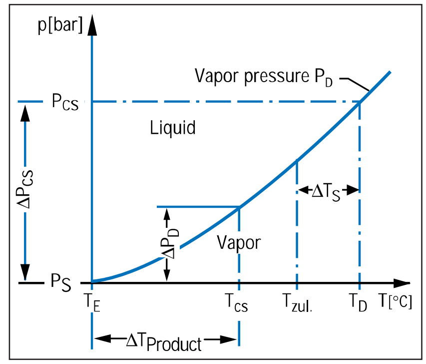


Figure 3. Containment shell, vapor pressure curve

Maximum Allowable Containment Shell Temperature

When handling volatile liquids or products with a vapor pressure that complies with the pump suction pressure P_s, the relation between containment shell temperature, containment shell pressure and boiling point of the liquid must be taken into consideration. Only when the operating conditions are not beyond the boiling point – that is, liquid is not flashing inside the containment shell – can safe operation be guaranteed (Figure 3). The condition point must always be in the liquid state.

Basically, the pressure rise ΔP_{cs} inside the containment shell during operation must always be higher than the heat-conditioned rise of vapor pressure ΔP_D of the product. To determine the maximum allowable containment shell temperature T_{zul}, the vapor pressure curve of the product must be available (Figure 3). The boiling temperature T_D can be taken from the intersection point between containment shell pressure at duty point of the pump and the vapor pressure curve. By adding a certain safety margin ΔT_s, the maximum allowable containment shell temperature T_{zul} can be determined. However, it must always be higher than the calculated containment shell temperature T_{cs} in order to avoid vaporization. This means a high pressure rise ΔP_{cs} in the containment shell area provides a higher safety factor.

Pumps with internal circulation from discharge to discharge side, through back vanes or rear impellers

(Figure 1), are pressurized in the containment shell area by approximately 80% of the differential head plus the inlet pressure P_s. The containment shell pressure for this pump design can be calculated as follows:

$$P_{cs} = P_s + \Delta P_{cs} \text{ [bar]}$$

The pressure increase ΔP_{cs} depends on the rated differential head H and the density of the liquid:

$$\Delta P_{cs} = \frac{H \cdot \rho}{10{,}2} \cdot 0{,}8 \text{ [bar]}$$

$$H[mLC],\ \rho[kg/dm^3]$$

Pump series with circulation from discharge side to suction side have lower ΔP_{cs} values. Exact values can be learned from the pump manufacturer or taken from the pump data sheet.

Practical Example

A standard chemical pump with metallic containment shell (Hastelloy C) and partial circulation as indicated in Figure 1 is required for handling ammonia (NH_3). Service conditions are as follows:

- Liquid: NH_3, TE = 0 °C, P_s = 4,4 bar, Q = 0,66 kg/dm_3, C = 0,492 g/cal/°C
- Differential head 120 mLC
- Capacity 40 m³/h
- Coupling losses Pv = 2,7 kW

The expected containment shell pressure P_{cs} is to be determined first:

$$P_{cs} = P_s + \Delta P_{cs}$$

$$P_{cs} = 4{,}4 + \frac{120 \cdot 0{,}66}{10{,}2} \cdot 0{,}8 = 10{,}6 \text{ bar}$$

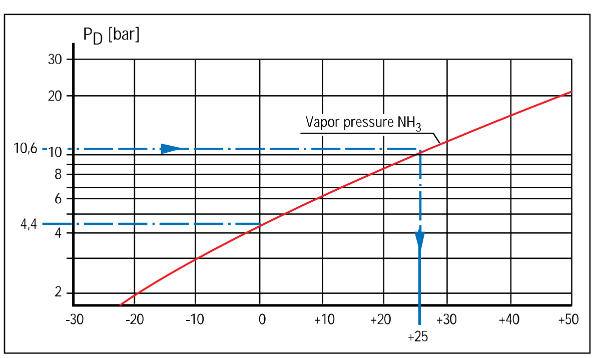


Figure 4. Vapor pressure curve, NH_3

Based on the vapor pressure curve for NH_3 (Figure 4) and on the calculated containment shell pressure of 10,6 bar, a boiling temperature of + 25°C is given which must not be exceeded during operation. For final determination of allowable temperature T_{zul}, the actual expected containment shell temperature at the thermal stable minimum flow of 8 m_3/h (Figure 2) must be calculated:

$$T_{cs} = T_E + \Delta T product$$

$$= \frac{0 + 4 \cdot 1}{0{,}492} \cdot \frac{1}{0{,}66} = 12{,}3°C$$

Considering the boiling temperature of 25 °C and the actual temperature of 12,3 °C, the allowable containment shell temperature T_{zul} can be defined at 20 °C.

Temperature Monitoring

General

Centrifugal pumps with magnet coupling are not considered electrical equipment. Level detectors or temperature control devices are not required by government authorities for these pumps, even when they are installed with explosion proof motors in hazardous areas. However, application experiences have shown that the main reason for operational troubles – besides worn antifriction bearings – is the practice of exceeding the allowable containment shell temperature. Consequences of excessive temperature rise are cavitation in the area of the containment shell and dry running of the slide bearings due to product flashing. Causes for this can be operation below minimum flow or against a closed discharge valve, blocking of the inner rotor, clogged circulation holes and/or solid particles between rotor and stationary shroud.

Therefore, it is essential to monitor magnetic driven pumps with temperature probes to ensure an automatic switch off before serious damage occurs.

Temperature Probes

For permanent control, resistance temperature probes are the preferred method of monitoring containment shell surface temperature, although this kind of protection is not activated under dry running conditions. Nor can containment shell rupture by the outer magnets resulting from worn antifriction bearings be avoided.

Typically, the probes work with a measurement rheostat of platinum, showing an electrical resistance of 100Ohm at 0°C. Temperature changes at the measuring point lead to a change of the resistance and, in turn, to a change of the voltage. If the preadjusted temperature limit is exceeded, the voltage change switches off the driver through a connected controller.

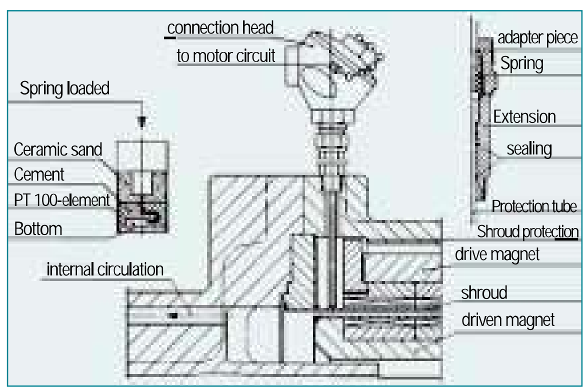


Figure 5. Temperature probe (PT 100)

Figure 5 shows a standard probe adaptable for containment shells. It has a flat bottom for sufficient contact to the containment shell surface. The element is located directly on the bottom of the probe. Continuous contact between the probe and containment shell surface is guaranteed by an integrated compression spring.

Another type of temperature probe, located in the internal recirculation flow, measures the fluid temperature after passing the magnet area. This system works sufficiently if the pump is properly filled with liquid. It protects against exceeding the boiling point of the liquid in the

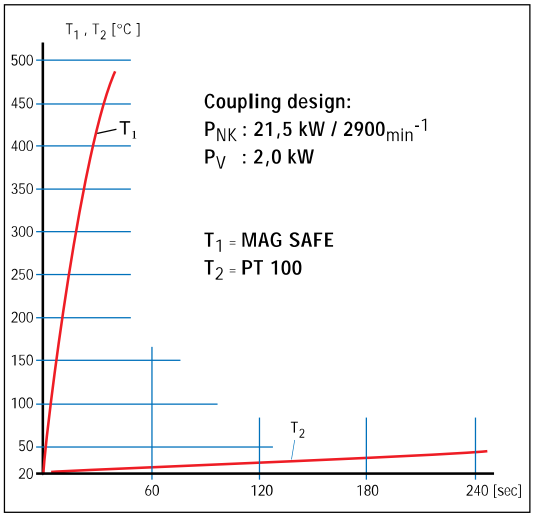


Figure 6. Containment shell surface temperature

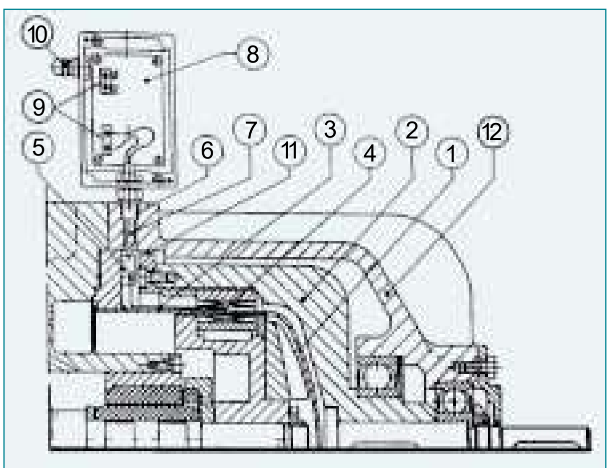
area of the containment shell caused by excessive temperature rise.

The problems of protecting pumps against dry running through temperature probes have already been pointed out. Given an empty pump, i.e., under dry running conditions, it has been proven that the containment shell surface temperature T_1 in the center of the magnet coupling (Figure 6) deviates remarkably from the surface temperature on the measuring point T_2. The reason for this are the eddy currents occurring in the center of the magnets that rapidly increase the containment shell temperature. The temperature probe at measuring point T_2 cannot detect this temperature rise in time because of the bad thermal conductivity of the shell material (18.10.CrNi or Hastelloy). Consequently, these type of probes are not able to protect magnets against overheating of an empty pump.

To obtain reliable measurements with this type of monitoring, the pump must be vented and the probe located at the reverse internal cooling flow (after passing the magnet area). This is only possible in designs with rotor back vanes or rear impellers. If the temperature probe is located in the precirculation flow (this is the case when circulating to the suction side), serious problems occur when pumping volatile liquids. With this setup, an increase in temperature is indicated only when the complete pump has already become hot – too late to prevent flashing in the magnet end.

MAG-SAFE Temperature Monitoring

Figure 7 shows the recently developed MAG-SAFE temperature monitoring system that is able to read the temperature directly on the heat source. It records the actual temperatures occurring between the magnets inside the containment shell and converts them into a linear output of 4 to 20 mA. Therefore, it is possible to preadjust through a trip amplifier any shut-off temperature within the range of -5 to 250°C.



1. Containment shell
2. Outer magnets
3. Thermocouple wire
4. Thermocouple
5. Connecting block
6. Thread connection
7. Connection piece
8. Transmitter
9. Connection clamps
10. Cable inlet
11. Protection device
12. Bearing bracket

Figure 7. MAG-SAFE temperature monitoring

Compared with common temperature control (Figure 5), the MAG-SAFE system features:

• Extremely fast reaction time to all temperature rises (Figure 6) , and switches off under dry running conditions.

• Since the containment shell is a heat source because of the eddy currents induced in it, a temperature rise can be detected before the liquid temperature in the containment shell is remarkably affected. Exceeding the boiling point can be prevented if the limit temperature is correctly set.

SI Unit	Approximate Conversion Factor	Resulting English Unit
kW	1.34	hp
m³/h	4.4	US gpm
°C	9/5 °C + 32	°F
mLC (meters liquid column)	3.28	feet(of head)

• Excessive temperatures at containment shell surface within the Ex-area can be prevented.

• The direction of the internal circulation flow has no influence on the temperature indication.

• Worn out ball bearings cause eccentric run of outer magnets and will lead, if not detected, to erosion at the protection device and containment shell rupture. With the MAG-SAFE system, drive magnets cut the connection wire 3 if such a condition is not recorded in time, and the pump is switched off before serious damage can occur.■

Harry J. Schommer is Chief Engineer of Dickow Pumpen KG in Waldkraiburg, Germany, with responsibility for the design and development of the company's sealless magnetic drive pumps. He has authored several papers on pumps and shaft-sealing systems, and conducted seminars for pump users in Germany, Europe and the Far East.

Pump Rebuilding at Avon

This refinery's successful program meets strict emission standards and enhances pump reliability at the same time.

By Stephen C. Rossi and John B. Cary

Most states have adopted regulations that limit fugitive hydrocarbon emissions from mechanical seals in centrifugal pumps. In California, limits as low as 100 ppm have been imposed. Users, faced with few choices to meet these strict standards, have turned to dual seals and sealess pumps to comply. Many users, however, have found that they can meet current fugitive emission limits with single seals by paying careful attention to detailed retrofit and repair procedures. This article discusses the rigors required to rebuild, maintain and operate pumps successfully – in most cases, with a single seal. These techniques also enhance pump reliability and have been applied to more than 100 pumps in harsh refinery environments.

Background

Pump History at Avon

The Avon Refinery is more than 80 years old. In 1913 crude oil pipe stills were built by a group of San Joaquin Valley oil producers on the Carquinez Straits near Martinez, California. Shortly after building a wharf for receiving the crude oil, they commenced construction of what is now the Avon Refinery a few miles away.

The oldest operating process plants date back to the late 1930s. Several facilities were built at the onset of World War II to support the defense effort. They were equipped with high speed centrifugal pumps almost exclusively. New plants have been added every decade since, but the average age of a centrifugal pump for the whole complex is almost 20 years.

Pumps of almost every type were installed over the years. Many were converted from packing to mechanical seals without regard to shaft flexibility and operating point. Over time poorly assembled and maintained auxiliary piping and flush systems were added, modified or abandoned. In addition, unclear or nonexistent operating procedures left operators to use their judgment. All of these factors contributed to poor pump/seal reliability.

Evolution of BAAQMD Regulations for Pumps

In the early 1970s the California Air Resources Board (CARB) mandated emissions standards for refineries. Two regional agencies were formed to monitor enforcement of the new standards – the Bay Area Air Quality Management District (BAAQMD), and the South Coast Air Quality Management District (SCAQMD). These agencies were also authorized to develop compliance standards for their jurisdictions. Historically, these standards have been much more stringent than federal emissions standards.

The first hydrocarbon emissions standards for pumps limited release of Volatile Organic Compounds (VOCs) to 10,000 ppm (parts per million), expressed as methane. The BAAQMD also required no less frequent than quarterly monitoring of pump seals. The district also mandated detailed record keeping. Fifteen days were allowed to repair pumps found over the emissions limit. The equipment could not be returned to service until the emissions limit was met. These limits were attained in most cases by replacing mechanical packing with single pusher-type shaft mounted mechanical seals.

As pumps were brought into compliance, the regulators began to "ratchet down" (their term!) on refiners. The regulation became increasingly stringent and, beginning in 1993, requirements went into effect regulating the percentage of equipment that could continue to operate and be put on a future or turnaround repair list (Table 1).

YEAR	LEAK STD	TIME TO REPAIR	%WAITING T/A REPAIR	REMARKS
1992 & prior	10,000 ppm	15 days	no limit	
Jan. 1, 1993	1,000 ppm	minimization in 24 hours, repair in 7 days	10%	Spared equip. may not be on T/A list
July 1, 1993	same as above	same as above	same as above	Nat Gas added
Jan. 1, 1995	same as above	same as above	same as above	Methane
Jan. 1, 1997	500 ppm	same as above	1%	Nat Gas added

Table 1. Summary of BAAQMD regulations for pumps

As these emissions standards went into effect, it became more difficult – and in some cases impossible – to maintain emissions levels in older style pumps. Slender shafts and long spans between bearings created too much shaft deflection at the seal faces. Small seal chambers and inadequate clearances added to the difficulty of retrofitting pumps with modern mechanical seals. At the same time, seal technologies were evolving. Cartridge mounted seals and requirements for larger seal chambers drove Avon managers to embark on a major fugitive emissions reduction program.

Initial Approach

In January 1993 the BAAQMD mandated that pump hydrocarbon emissions levels be reduced to 1000ppm. To meet this more rigorous standard, the refinery reliability group took the lead in identifying the scope of the program, with the intent of turning over the project to engineering for execution. The following approach was used for the initial scoping project:

- data collection and analysis
- seal performance testing and pump evaluation
- vendor selection

Data Collection and Analysis

Data was collected and analyzed to develop a list of pumps requiring work. At this point, the work itself was undefined. Formal fugitive emissions testing was done every quarter. Three years of data on approximately 500 pump seals was analyzed to identify all pumps that had failed their emissions test more than once per year over that period. This information was compared with the refinery's "bad actors" list and prioritized accordingly. Estimates and schedules were developed for the annual budgetary cycle. Other data sources were reviewed to gain a thorough understanding of the scope of the project. These included:

- pump importance and its effect on the process
- failure history, including mean time to repair
- maintenance cost history (total and per repair)
- bad actors lists
- hydraulic performance (NPSH and the difference between BEP and actual operating point)
- future hydraulic requirements

External information was also collected. A benchmarking study was undertaken in other West Coast refineries to determine what their experience had been with various seal designs and seal vendors.

Seal Performance Testing and Pump Evaluation

The initial data indicated that there were about 150 pumps requiring some degree of retrofitting. The majority of these pumps handled low viscosity products such as propane, butanes and light gas oils. Based on the scope of the project and the time allotted, the decision was made to rely on one seal manufacturer. This would save the time involved in competitively bidding each seal, and it would reduce the overall cost through volume purchases.

Three identical pumps in the same service were chosen for initial seal testing. These pumps were selected because they were in light hydrocarbon service and moving liquid near its vapor pressure. In addition, the pumps were in a unit that was going into a maintenance turnaround, providing an opportunity to rebuild all three pumps to identical specifications.

Three pump vendors were invited to participate in the project. They were asked to submit proposals to provide their "best available control technology" (BACT), and they were given the specifications for the test pumps. The objectives for the test project were to:

- Establish a control technology for light hydrocarbon services to be used on pumps that must be retrofitted to meet future emissions requirements.
- Provide single seals designed to meet stringent emissions standards, avoiding expensive and complicated dual seal installations.
- Document this technology and performance so it could be applied to retrofits as well as new installations.

A technical support agreement for the test program was developed for the participating seal vendors, stipulating the obligations of each to provide technical support and assistance as a full partner in seal selection, retrofits and testing.

Each vendor assigned a field service engineer to participate in the overhaul, installation, start-up and field monitoring of the test seal. The test pumps were meticulously overhauled and carefully installed. The seals were tested over a three month period, with the following data recorded daily:

- emissions readings
- suction and discharge pressure
- pumped fluid temperature
- vibration (radial and axial)
- seal flush pressure
- quench steam flow

Vendor Selection

At the end of the seal trial period, field test data were compared with results of the alliance selection process led by the plant's purchasing group. Vendor performance was weighed against several pre-established performance criteria, such as degree of technical support, seal performance, response time, level of technical expertise and experience. Long term alliance agreements were then drafted, budgets prepared and proposed schedules developed.

Project Approach

Defining the Project

After establishing the breadth of the project, Avon officials initiated a formal program to implement the work. The program was divided into phases corresponding to annual budget cycles. A project core team was assembled, consisting of a project manager, project engineer, full time vendor technical representative, draftsperson, clerk, mechanical contractors, pump alliance partner and a buyer from the Purchasing Department. Various area maintenance and operating personnel became ad-hoc participants, depending on the location of the retrofit work.

The charter of the project team was as follows:

- Develop reliable technology for pump emissions control to comply with the 1993 through 1997 emissions limits.
- Implement this technology as proactively as possible to avoid penalties and fines for non-compliance.
- Complete detailed designs required to achieve reliable 1993-1997 emissions compliance.
- Provide project management support for all phases of equipment upgrades.
- Develop and evaluate equipment hydrocarbon emissions data to prioritize and schedule optimal cost effective repairs and upgrades.
- Integrate the activities of the project with that of the maintenance and operating departments to minimize disruption in the operation of the refinery.
- Recommend and coordinate equipment upgrades meant to improve equipment reliability in conjunction with emissions compliance modifications.

Candidate Pump Evaluation

Emission levels were measured on all pumps identified as having potential VOC compliance problems. Qualitative data on all VOC program pumps were surveyed to assess the need for upgrades to the sealing system to meet 1997 regulations.

Pumps found to be above the 500 ppm limit for 1997 were considered for possible upgrade. The company's third party contractor for emissions compliance used a data base system to generate queries on all pumps with emissions levels at or above the limit. Detailed analysis of each pump system produced a breakdown of anticipated upgrades and replacements (Table 2).

Proactive Approach to Problem Pumps

A proactive approach was taken because an in-kind, reactive repair program would have increased cost to the company and created major repair backlogs when more stringent

Recommendation	Number of Pumps
Seal Upgrade only	96
Power End Retrofit	45
Complete Replacement	5
No Change	24
Total	170

Table 2. Recommended modifications

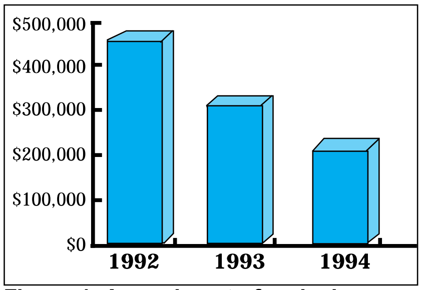


Figure 1. Annual cost of emission

"Reactive" Approach
(in-kind repairs on as needed basis)

1995: 25 emissions repairs per year at $5000 each — $125,000
1996: 25 emissions repairs per year at $5000 each — $125,000
1997: 40 emissions repairs per year at $5000 each — $200,000

TOTAL COST $450,000

emissions limits went into effect.

The cost of emissions related pump repairs made in 1992, 1993 and 1994 is shown in Figure 1. In 1994, 40 pumps required emissions related repairs. This was down substantially from 92 in 1992 and 64 in 1993. The total cost for all repairs during this three year period was more than $1 million.

The following projections illustrate the potential for savings with a proactive vs. reactive approach to emissions compliance. (Repairs were expected to remain constant through 1996, then increase by 50% in 1997 when the emission limit was reduced to 500 ppm).

Savings over this three year period was estimated to be at least $175,000 as shown in Figure 2. A complimentary maintenance cost savings was also expected due to

"Proactive" Approach
(continuation of modifications and thermal oxidizer installations)

1995: 25 emissions repairs per year at $5000 each — $125,000
1996: 15 emissions repairs per year at $5000 each — $ 75,000
1997: 15 emissions repairs per year at $5000 each — $ 75,000

TOTAL COST $275,000

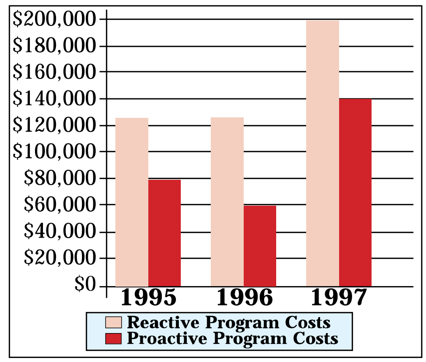


Figure 2. Project pump repair costs

increased reliability.

In conjunction with emissions history, mean time between repair (MTBR) and hydraulic performance requirements were evaluated for possible upgrade.

Experience had shown that in-kind repairs of equipment that failed emission monitoring would, in most cases, not comply with 1997 levels due to pre-existing problems such as pipe strain, unstable foundations and poor suction conditions. These conditions were corrected as part of the upgrade process. This approach reduced chronic mechanical seal failures and subsequent emissions, improved long term reliability and lowered total life cycle cost of the equipment.

Tremendous effort was expended in identifying pumps to be included in the program. Data from a number of sources was scrutinized to justify each pump's inclusion. This was an important exercise because it prioritized the upgrade sequence (shortest MTBR to longest MTBR), identified detailed scopes of work, and scheduled each retrofit to coincide with other refinery activities.

At the end of this process, a well defined list of pumps targeted for upgrade emerged. The plan was communicated to all affected refinery personnel and was used as the basis document for all project work.

Preparation of Specifications

Specifications for Pump and Seal Purchase

The project team immediately set out to establish minimum requirements for pump and seal specifications. A series of meetings was held with both the seal and pump alliance partners, as well as with company machinery specialists, to create project specific specifications. The basis for the specifications was API 610 with clarifications in the areas of fits and tolerances. A draft version of API 682, (Centrifugal Pump Shaft Sealing Systems for Refinery Services; first draft 9/92)was also used for guidance. The intention was to create an environment for the seal that would allow it to function as intended. These specifications were later adopted company-wide.

Examples of Critical Fits

Areas of concern and additional requirements to API 610 are tabled below:

Additional requirements:

- seal chamber register (radial) to shaft within 0.001 T.I.R.
- component match impeller to shaft fit to achieve a goal of 0.000 inches tight to 0.001 inches loose
- component match of rolling element thrust bearings to achieve 0.004 inches maximum axial float
- shaft sleeve bores equal to the maximum diameter of the shaft with a tolerance of +0.0010 inches to -0.0000 inches

Modifications:

- Squareness of seal chamber face register to shaft axis was reduced from 0.002 to 0.001 T.I.R.
- Sleeves will have a relief centered axially and the minimum sleeve thickness can be 0.090 inches within the relieved area.

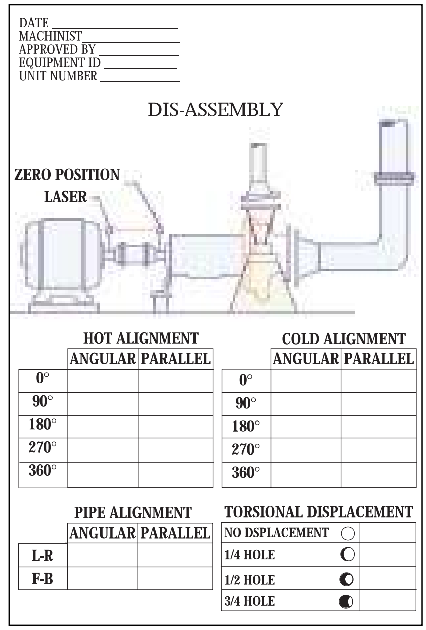
It was agreed that these specifications would be the minimum quality level expected.

Repair and Installation Specification

The quality of pump installations – including foundation preparation, grout (or lack of it), piping strain, alignment and other key factors – varied considerably throughout the plant. No company repair and installation standard existed other than the original equipment manufacturer (OEM) guidelines and a few rules of thumb. Consequently, the project team created a repair and installation specification. This document covered setting of new pumps and drivers, rebuilding of pumps, installation of retrofit kits, and seal flush, vent and drain connections.

The following are important focus areas addressed in the specification:

- "As found" pipe strain effects were checked and recorded (Figure 3) prior to pump removal, using a laser alignment tool attached to the coupling. These were checked again when final piping connections were made.
- Existing pump base was checked for voids and flatness (Figure 4). Pressure injection grouting and field machining of mounting pads were carried out when tolerances were exceeded.
- On non-retrofit pumps, new 17-4PH pump shafts were fabricated and all fits reclaimed to tolerances in the new pump specification.
- All pumps, including retrofits, were fitted with close clearance carbon throat bushings to maintain seal chamber pressure.
- New dynamically balanced, multiple disk spacer couplings with register fits for the hubs and center section were provided on all upgrades.


DATE ____
MACHINIST ____
APPROVED BY ____
EQUIPMENT ID ____
UNIT NUMBER ____

DIS-ASSEMBLY

HOT ALIGNMENT

	ANGULAR	PARALLEL
0°		
90°		
180°		
270°		
360°		

COLD ALIGNMENT

	ANGULAR	PARALLEL
0°		
90°		
180°		
270°		
360°		

PIPE ALIGNMENT

	ANGULAR	PARALLEL
L-R		
F-B		

TORSIONAL DISPLACEMENT

NO DSPLACEMENT ○	
1/4 HOLE	
1/2 HOLE	
3/4 HOLE	

Figure 3. "As found" pipe strain record form

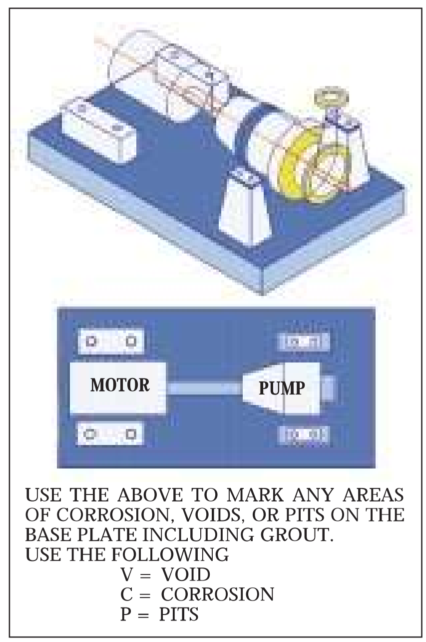


Figure 4. "As found" baseplate condition

- Completed pump and seal assemblies were leak tested before field installation. This minimized the need for rework after the pump system was filled with process liquid.

Support Systems Specification and Selection

Existing piping and seal systems

Photo 1. Piping reinforcement detail

Photo 2. Seal and pump venting procedure

were upgraded. Most of these pumps had screwed connections (potential emissions sources) throughout. As part of the upgrades for piping reliability, the following were addressed:

• All screwed connections on the pump casing or process piping were replaced with Schedule 160 nipples; the nipples did not exceed 4" in length and were gusseted in two planes, seal welded and flanged – eliminating screwed connections if possible (Photo 1).

• All tubing connections to the primary seal were 1/2" 316 stainless steel with a wall thickness of 0.065". Smooth radius bent 3/4 inch tubing was used for thermosyphon cooling when required.

To reduce seal flange distortion and make installation and removal of the seal easier, the staff made sure that the final tubing connections were not more than 18 inches long.

Vent systems were provided on seal chambers in addition to the pump case to ensure filling of the seal chamber prior to startup. These venting systems included instructions in the form of a venting procedure tag installed on nearby piping. The message addressed both seal and pump venting (Photo 2).

Instrumentation

Only essential – or remote and unattended – pump seal systems were instrumented. Typically, a level switch on a seal reservoir was the only signal back to a control room. Most installations did not warrant remote readout instrumentation since the operators were better informed of the general health of the pumps by observing them in person.

Secondary Containment (VRS)

A number of pump seal systems used existing vapor recovery systems (VRS) for 100 percent containment when it was available close to the pump installation. In many remote locations total containment was required, but VRS was not available.

Alternative Technologies

Where total containment was required and the only choice was a nitrogen pressurized dual seal arrangement, an alternative involving new emissions control technology was used. This technology utilized compressed air passing through a jet ejector to pull emissions from pump seals or barrier fluid reservoir vents through a flameless reactor that thermally oxidized the VOC's to water vapor and carbon dioxide, as shown in Photo 3. Four units have been installed with good success; the system is 99.99% effective in reduction of VOC's.

Photo 3. Thermal oxidizer system

Each unit currently controls emissions from 10 pumps. Future expansion up to 20 pumps is possible. These systems are expected to avert as many as 15 emissions-related repairs per year. Many non-compliant pumps are scheduled to be connected to a thermal oxidizer within the next year, possibly eliminating the need for further modifications.

Part 2

Seal performance is affected by numerous internal and external forces. How a pump is sized for an application and how it is actually operated have a significant impact on seal life. In fact, both of these factors can shorten seal life in a pumping system. Mechanical problems such as misalignment, unbalance and flatness, as well as poor concentricity and perpendicularity are fairly well understood and relatively easy to control. Hydraulic forces, on the other hand, are generally not as well understood or recognized by personnel responsible for daily pump operation and maintenance. Furthermore, it is usually more difficult to remedy a hydraulic problem since it may often relate to the original design of the system.

Shaft deflection and vibration caused by unbalanced hydraulic forces can be very destructive to a pump and severely diminish seal life. Before embarking on a project to improve seal performance, it is imperative that the pump's hydraulic performance be verified. The closer a pump operates to its best efficiency point (BEP), the longer the seal will last. This has been demonstrated many times in the field and was recently proven analytically in a computer model specifically designed to predict seal life based on a pump's proximity to BEP. A discussion of the process for evaluating pump hydraulics is included in the appendix to this article.

Developing Alliance Partnerships

To be lasting, an alliance relationship must be profitable or beneficial for all parties. Partners need to take mutual responsibility to ensure that the desired goals are achieved. One of the first steps the newly formed Avon project alliances undertook was to develop well defined objectives along with a mission statement.

Alliance Team Mission Statement

Adopt a fundamental philosophy of decreasing mechanical seal life cycle costs through increased equipment reliability.

- Maximize equipment availability
- Manage and document change accurately and completely
- Improve data quality (new and existing)
- Obtain accurate process information from the owner/user
- Analyze the root causes of failure
- Maintain honest communication about failure by owner
- Strive for total buy-in by management and staff – down to the last person

The alliance teams also developed matrices to assess the benefit of the arrangement and continue improving it:

Dollars

1) cost of new seal purchases
2) cost of seal repairs
3) inventory reduction
4) market share

Reliability

1) number of repairs
2) mean time between failures

Contractor Performance

1) plant-wide pump survey status
2) on-time delivery
3) failure analysis submittal

Company Performance

1) appropriate paperwork
2) on-time payment
3) provides pump access

The key is to involve alliance partners in all facets of project activity.

Candidate Pump/Seal Evaluation

Both the pump and seal alliance partners participated in all facets of the pump/seal evaluation.

- When a pump was added to the list, based on emissions survey, the pump vendor and company representatives interviewed operations personnel for possible insights on performance deficiencies and operational problems. This information, along with service data, was then assembled into a file.
- Process data – including head and flow requirements – and physical data was then collected on the fluid

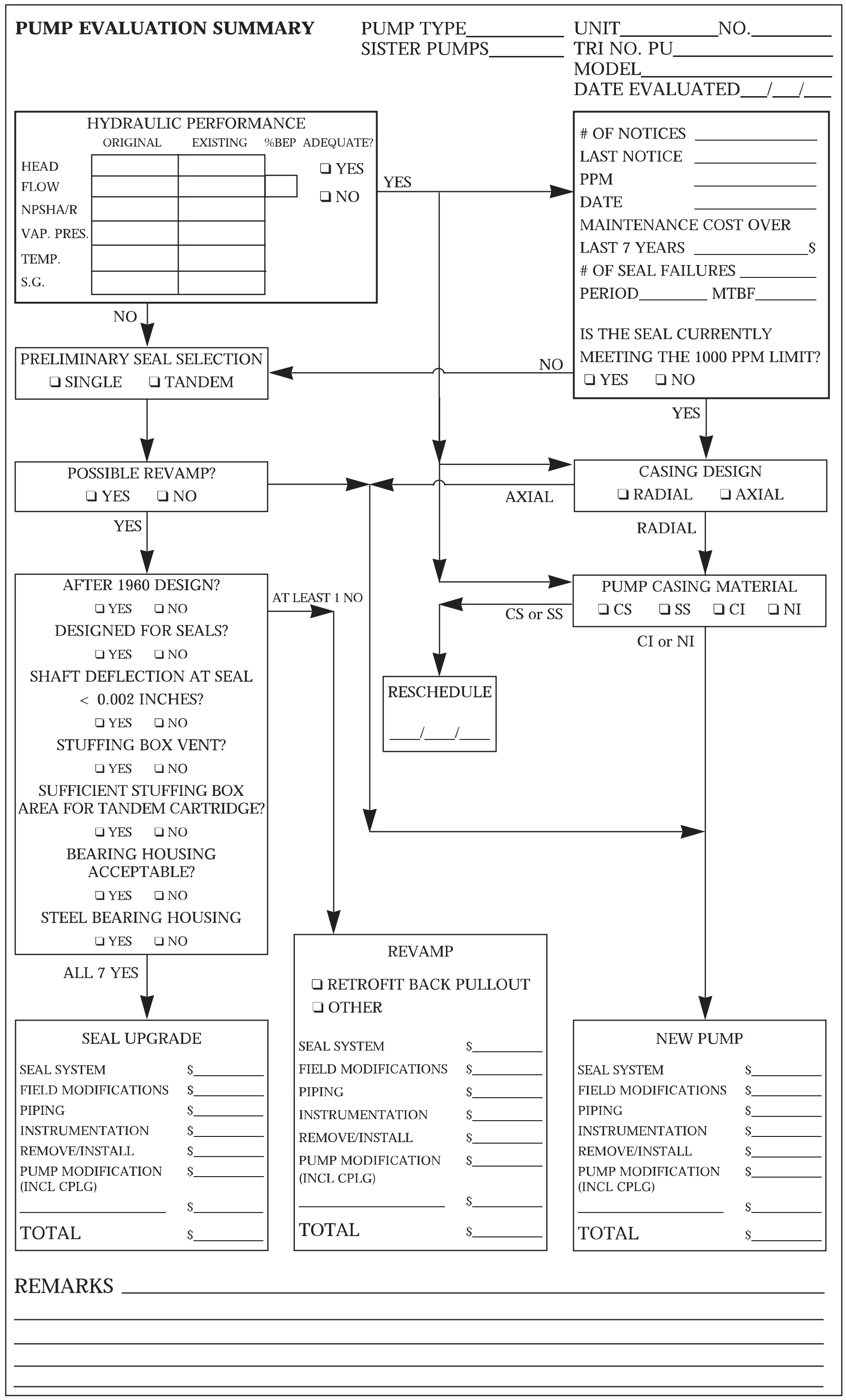


Figure 5. Pump evaluation summary

SEAL PROPOSAL ADDENDUM

PUMP#/UNIT: PU.... /
PUMP REPAIR TYPE:
SEAL:
PRODUCT:
TEMPERATURE: F
SUCTION PRESSURE: PSIG
VAPOR PRESSURE: PSIA
DISCHARGE PRESSURE: PSIG
SPECIFIC GRAVITY:
API PIPING PLAN:
SUCTION RETURN?:
QUENCH?:
PROPOSAL#:
TOSCO CATALOG#:
PUMP TYPE:
NUMBER OF BOXES:
COMMENTS:

SIGN-OFF: STEVE ROSSI________
GIL TIGNO________

Figure 6. Seal proposal addendum

being pumped. The information, which included vapor pressure and solids concentration, was then summarized on a Pump Evaluation Summary form (Figure 5) and used to determine the best fix based on pump type, emissions, maintenance history and performance data.

Selection – The pump and seals alliance consultants submitted proposals for the agreed upon upgrades. Attached to each was a seal proposal addendum (Figure 6) that provided design details for construction. The seals consultant also prepared a new seal order checklist (Figure 7) to further define the construction details. When field measurements were required, the pump was taken out of service and checked to ensure that all the components fit precisely.

Installation & Startup – After the installation was completed, a QA/QC evaluation was made, and the Pump Commissioning Check List (Figure 8) was signed by the project representative and the operator prior to startup. The seal vendor usually witnessed the startup and recorded initial emissions levels.

Living Program Maintenance – As part of the long range compliance strategy, data is still being collected on all VOC equipment to determine future direction. Some areas where the alliances are now concentrating their efforts include:

- Providing on-going training for maintenance and operations personnel featuring detailed information on seal installation and operation. It is expected that this training will greatly increase the MTBR and reduce life cycle costs of pumping systems throughout the refinery.
- Continuing to develop records on seal life, failure analysis and life cycle costs, with focus on solutions to bad actor pump/seal systems.
- Incorporating seal and pump parts and repair services under a single manufacturer for each.

NEW SEAL
ORDER CHECKLIST - FOR OVERHAUL PUMPS

PUMP NUMBER: ____________

FIELD MEASUREMENTS

1) Physically verify: [ ]
Shaft diameter_________ Bolt circle__________
Stud size______________ First Obstr__________
Gage Ring dist_________ Bolt Orientn_________
2) Suct Press_________ Disch Press _________ [ ]
3) Rotation from driver end - CW/CCW [ ]
4) Temperature _________
5) Make a diagram of the bearing web and the existing seal piping. Where can new seal piping be located? [ ]
6) Note cage ring tap location(s). Can the seal box be vented through the cage ring taps? [ ]
7) Is the existing seal the same model as indicated by the files? [ ]
8) What is the O.D. of the current seal gland. [ ]

EVALUATIONS

1) Verify the vapor pressure if possible. Make sure that the box pressure is sufficient to keep adequate vapor suppression. [ ]
2) Design the seal flush piping system including orifice sizing and throat bushing clearance in order to get the required flow and vapor suppression. [ ]
3) Make sure there is adequate room for an O-Ring groove and multi-port injection between the box bore and the inside the stud holes. (especially important if box is to be bored) [ ]
4) Verify that the seal selected will fit. OK any box boring that will be required with Tosco and pump manufacturer. [ ]

Figure 7. New seal order checklist

FUGITIVE HYDROCARBON EMISSIONS PROJECT PUMP COMMISSIONING CHECK LIST

PUMP NO.:____________ **DATE:**______________

MACHINISTS: Name:____________________________

1. COUPLING GUARD SECURE ____
2. HOLD DOWN BOLTS INSTALLED PUMP ____ MOTOR ____
3. FLANGES PROPERLY MADE-UP ____
4. JACKING BOLTS BACKED OFF ____
5. SEAL DRIVE COLLAR BOLTS TIGHTENED ____ TORQUE: ______________
6. SEAL SETTING PLATES ROTATED AWAY FROM COLLAR ____

COMMISSIONING ENGINEER: Name:______________

1. PROPER OIL LEVEL PUMP ____ MOTOR ____
2. PRESSURE GAUGES INSTALLED/ORIENTED ____
3. PIPE PLUGS INSTALLED ____
4. GASKETS INSTALLED ____
5. SMALL BORE PIPING SUPPORTED ____
6. NOMALLY-CLOSED VALVES ***CLOSED*** ____
7. ORIFICE PLATE INSTALLED WITH INSCRIBED TAB ____
8. COOLING WATER FLOWING
9. VENTING PROCEDURE SIGN POSTED AND VALVES TAGGED ____
10. AREA CLEANED UP ____
11. LOCKS/TAGS REMOVED ____
12. VENT PROCEDURE DELIVERED ____
13. SCREWED PIPING LEAK TESTED ____

IF THERE ARE ANY PROBLEMS WITH THIS PUMP AFTER COMMISSIONING CALL STEVE ROSSI AT EXT.3263

Figure 8. Pump commissioning check list

Conclusions and Recommendations

Accomplishments

- All of the 102 pumps modified in the project beginning in 1991 met 1993 emission limits.
- More than 60% of these pumps had initial emissions levels of 1000 ppm or more, and the MTBR initially averaged 8 months; after retrofitting the MTBR has increased to an average of 16 months.
- The alliances established criteria for cost effective procurement of pumps and seals, and they provided accessibility to the most current technology resources.
- An engineering standard for further VOC pump upgrades and a repair and installation standard for pumps and seals was put in place.

A key contributor to this success was the ability of those involved to view the solution as an overall system of modifications. Pre-existing conditions such as pipe strain, unstable foundations and misalignment were corrected to eliminate vibration, stresses and distortions. As an added benefit, the reliability and safety of the equipment improved, thus lowering equipment life cycle costs.

Correlation of Bad Actors to Emissions Compliance

From 1990 to 1992 numerous in-kind maintenance repairs were made on equipment in response to emissions violations. Most of these repairs lasted only 3 to 6 months before another violation notice was received. The upgrades undertaken have in many cases doubled or tripled the time between emissions failures and eliminated chronic reliability problems.

Lessons Learned

There are three primary causes for premature seal failures and/or excessive vapor emissions from upgraded pumps:

• installation errors
• changes in the chemical composition of the pumpage
• operational and hydraulic problems (such as dry running and cavitation)

The first problem is the most controllable. The others are more challenging and require continuous education and training.

In addition to initial equipment installation, an improved focus on equipment reliability through troubleshooting to resolve premature failures is needed. This is expected to take the form of additional training for both maintenance and operating personnel, revision of operating procedures, and continuous measurement of MTBF and life cycle costs.

A skilled team of dedicated experts can rebuild a pump perfectly and still fail to achieve the final objective if the system is not started up and operated properly. A number of details must be attended to in order to achieve success:

• Include process operators in the installation process. Have the production department assign responsible operators to the start-up team. Communicate their responsibilities for a proper start-up and continued operation. Then conduct training on any special requirements of the seal system. Use seal and pump partners to develop materials and provide training.

• Develop pre-startup checklists that include the following procedures:

a. Steam, flush and purge the pump casing prior to introducing product. (Minimize the time spent doing this to avoid contamination and overheating in the seal chamber.)

b. Prepare for hot alignment checks (P.T. > 300°F and steam turbine driven pumps).

c. Review existing pump start-up instructions.

• Prior to starting a pump, gather responsible core team members together, including alliance partners. Review the start-up procedure and the duties of each team member. Develop a start-up checklist that incorporates the following information:

a. Pump start-up procedures (including venting of all air and vapors from the seal chamber prior to and during start-up.)

b. Expected normal, minimum and maximum operating parameters (flow, temperature, pressure, viscosity, cooling, etc.)

c. Performance parameters, including suction and discharge pressures, flow temperatures, suction strainer differential pressure and so on.

d. Program for continuous monitoring after start-up.

e. Troubleshooting guidelines for operators and mechanics.

• Pump and seal alliance partners should be full participants with users in the successful commissioning and operation of retrofit pumps. As stated, they conditionally guarantee their equipment if all repair, installation and start-up conditions are met. For this project, the seal alliance partner guaranteed that fugitive emissions levels would not exceed BAAQMD limits for three years of continuous operation.

• The ability to exercise a warranty is dependent on good documentation. Post start-up documentation requirements must be agreed to with alliance partners as part of the initial parameters of the arrangement. As a minimum, the following data should be collected:

Fugitive emissions levels – Initially, the project team collected this data monthly until levels stabilized, at which time the monitoring was turned over to the contractor responsible for collecting quarterly compliance data.

Vibration data – This information is also taken more frequently in the beginning, to catch infant mortality-type failures. When readings stabilize, then routine (documented) monitoring can resume.

The importance of documentation can not be overstated. Proper documentation [API Standard 682, Shaft Sealing Systems for Centrifugal and Rotary Pumps, First Edition, October 1994] is required throughout the entire process from start to finish. The minimum requirements are listed below:

Seals

1. Completed API Standard 682 data sheets
2. Cross sectional drawing of all seals (modified typical)
3. Schematic of any auxiliary system (or systems) including utility requirements
4. Electrical and instrumentation schematics and arrangement/connections
5. Seal manufacturer qualification test results, if specified
6. Detailed cross sectional drawings of all seals (specific, not typical)
7. Detailed drawing of barrier/buffer fluid reservoir (if included)
8. Detailed bill of materials on all seals and auxiliaries

9. Material safety data sheets on all paints, preservatives, chemicals and special barrier/buffer fluids
10. Installation, operation and maintenance manuals
11. Pre and post start-up checklists
12. Routine performance monitoring data sheets

Pumps

1. Completed API Standard 610 data sheets
2. As-found and as-built specifications (rebuilt pumps)
3. Pump manufacturer performance test results, if specified
4. Detailed cross sectional drawings of all pumps (specific, not typical)
5. Detailed bill of materials on all pumps and auxiliaries.
6. Material safety data sheets on all paints, preservatives and chemicals
7. Installation, operation and maintenance manuals
8. Installation checklists
9. Pre and post start-up checklists
10. Routine performance monitoring data sheets

On-going Program Performance

In closing, the importance of continuing to work within the alliance partnerships cannot be overemphasized. These relationships require constant nurturing and attention. Resistance to using the alliance will be an ongoing issue for the core team members. The alliance must constantly review the performance of the partnership itself and compliance with stated goals and objectives. A formal and periodic review process should be formulated.

Additionally, long term issues such as how to provide continuous improvement (CI) to the alliance relationship are important. There are many facets to CI, but it typically involves empowering employees to pursue improvements actively. It also means providing technical support at the front line, rigorous root cause failure analysis and use of advanced analytical techniques. Along with CI is the need to maintain the new way of doing business. This includes purchasing quality spare parts, maintaining quality control and standardization, doing meticulous pump and seal overhauls, developing consistent, detailed documentation, and retaining a highly skilled and motivated work force.■

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Evaluating Sealless Centrifugal Pump Design & Performance

Caught in the sealed versus sealless crossfire? This article sheds light on the appraisal process so you can determine the best fit for your service.

By Dave Carr

Sealless pumps have been available for more than 50 years, yet it has been only during the last decade and a half that they have gained popularity in North America. Sealless pumps are used much more often in the European and Japanese markets. It is not coincidental that North America's interest in this category of pumps coincides with legislation aimed at protecting the work place and the natural environment through increased safety and minimization of fugitive emissions.

Many reviews have previously been published regarding the two drive technologies that dominate this class of sealless pump – canned motor and magnetic couplings. Paramount to both designs is the fact that no dynamic shaft seal is required to contain the pumped fluid. Rather, a stationary containment shroud is used to isolate the pumpage from the ambient environment. This is possible because power is transmitted across the shroud through magnetic lines of flux that induce rotation of the impeller shaft. The significance of this design feature is the fact that sealless pumps can negate leakage that is a fundamental byproduct of the typical mechanical face seal. The corresponding benefits to personnel and the environment are obvious.

The chemical processing industry (CPI) is a leader in the adoption of sealless centrifugal process pumps and a model for their use in North America. The CPI is currently experiencing a heightened awareness of volatile organic compound (VOC) and volatile hazardous air pollutant (VHAP) emissions as a result of growing regulatory requirements. Today's environmental regulations also have related monitoring and reporting costs in association with the 1990 Clean Air Act. Sealless pumps fit within the broadest exemption from emission instrument inspections due to their compliance with the *without an externally actuated shaft penetrating the pump housing* definition [Ref.1]. These costs vary from plant to plant but, in aggregate, can account for a substantial budgetary allowance with equipment that is pumping volatile liquids. Likewise, they represent an input that should be included within an assessment of "total evaluated costs" (TEC) – i.e., all costs accrued from inquiry through the life of the pump.

Numerous studies show that shaft seals are the principal cause of failures in chemical process pumps. One such study [Ref. 2], of more than 1250 operating pumps (and more than 2500 installed pumps), indicates that approximately 50% of the recorded primary failure causes were the result of leaking seals. Additionally, officials for a major chemical manufacturing plant [Ref. 3] have concluded that their pump repairs are 30 to 60% less costly/frequent with canned motor and magnetic driven sealless pumps than with chemical duty pumps equipped with multiple mechanical seals. It is, therefore, important to recognize the role that maintenance/operating availability should play in the pump evaluation process. When factored into the TEC equation, it has often been cited as the component that sways a decision toward a sealless offering.

Consequently, the need for a succinct "sealed versus sealless pump" decision process has surfaced. The following discussion reviews a logic stream that can be used to assist in this exercise. It is reasonable to expect that the outcome of any analysis model can not be considered absolute since customer-specific application and experience-sensitive inputs are often required to make an informed equipment selection. The basic tactic, however, can be used to determine when a sealless design is a viable choice.

Sealless Hydraulic Capabilities

Performance capabilities to 2000 gpm and 750 feet TDH (though not necessarily simultaneous) are representative of widespread sealless pump experience and coincide with

SEALLESS CENTRIFUGAL OPERATING CHECK LIST

Liquid Types: *hazardous, or volatile vapors, toxic, flammable, corrosive heat transfer fluid*

Flows:	< 2000 gpm < 750 ft TDH Field experience typically limited to head/ flow combinations that fit within about 200 hp
Temperatures:	< 750°F (without auxiliary cooling) < 850°F (with auxiliary cooling) > -140°F (specialized designs to -260°F)
Viscosity:	> 0.15 cP < 200 cP (Note: Lower viscosities may require bearing design changes to support shaft)
Solids Content:	< 150 Micron 3-5 wt %

single suction, single stage, centrifugal pump capabilities. Production designs are available to further extend this region, but field experience has been typically limited to head and flow combinations that fit within 200 horsepower. Requirements outside of this range will likely mandate the use of larger single stage, multistage or high speed centrifugal pumps or positive displacement designs. The vast majority of sealless manufacturers' equipment is built around 150 and 300 pound ANSI flange ratings, which are consistent with the needs of a typical chemical processing plant.

Once it has been determined that your service conditions roughly fit into the range of today's sealless centrifugal pumps, an appraisal process to determine the best fit can begin. Pump users are particularly challenged when operating with liquids that are characterized by one or more of the volatile, toxic, flammable or corrosive definitions. In the CPI, this represents a broad range of fluids that include acids, alkalis, salts, esters, hydrocarbons, monomers/ polymers, alcohols, ethers, halogenides, nitrogen/sulfur compounds and even some extreme water conditions. Some characteristics common to these broad definitions are a propensity for the fluids to be unstable, poisonous, noxious, dangerous, destructive or chemically reactive with air. They may also precipitate components, solidify easily, or need to be handled at extreme temperatures. Sealless pumps are well suited for these applications, but some basic application criteria need to be established to ensure a troublefree installation.

Look at the Liquid

A qualification of the pumped liquid is the first major decision point when considering the use of a sealless pump. Early applications were relegated to services addressing a need to meet emissions limits for volatile fluids (VHAP/VOC) or where mechanical seals had proven ineffective. Often those applications were associated with situations in which fluid volatility was a significant application parameter. It should immediately be recognized, however, that a simple change from sealed to sealless options is not appropriate without a corresponding analysis of the strengths and suitability of each pump. By design, a mechanical seal face requires a liquid film to act as a cooling and lubricating medium between the stationary and rotating members. Corresponding leakage across those faces, be it ever so slight, is a necessary consequence to promote acceptable seal life. This situation is complicated with volatile fluids having vapor pressure characteristics that result in a change from liquid to vapor states at atmospheric pressure. Consequently, leakage can cause liquid mechanical seals to be a source of fugitive emissions even when operating properly.

A hybrid category worth specifically noting is heat transfer fluids (HTF), which can exhibit one or more of the above mentioned attributes. Many users believe that HTF applications represent the single most prevalent application for sealless pumps in the CPI today since those services transcend safety, environmental and maintenance issues. Any of the three criteria by itself can be used to justify a sealless pump purchase decision and, collectively, they represent a prime opportunity to exploit the leak-free nature of sealless pumps.

Only sealless and multiple mechanical seal (either liquid or gas configurations) pump designs should be considered to meet demands of liquids in the volatile, toxic, flammable, corrosive or HTF categories. Applications that fall outside of those parameters can reasonably be controlled with single mechanically sealed pumps as an alternative to more sophisticated designs.

Frigid or Fervid?

Extreme temperature conditions represent good opportunities for the use of sealless pumps. This belief is attributable to the fact that such conditions present a difficult situation with regard to maintaining the necessary seal face integrity that complements low leakage. High values are commonly associated with reference to "extreme" temperature conditions. It is generally accepted that a "high temperature" definition refers to operating levels greater than approximately 250°F (121°C), and that temperatures above this threshold act as a handicap to the operational ease, simplicity and effectiveness of liquid sealing. The development of high strength magnets and insulation materials has yielded sealless designs capable of withstanding temperatures as high as 750°F (399°C) without the use of auxiliary cooling. Current technology limits cooled versions to as high as an 850°F (454°C) rating, but these limits are continually being challenged by new product designers intent on extending application limits.

Cryogenic extremes, however, are often disregarded when considering the use of sealless pumps. Most sealless pump manufacturers rate their standard equipment for levels to -140°F (-96°C), and specialized designs are available for temperatures approaching -260°F (-162°C). A side benefit to a canned motor pump's operation with cold liquids is the fact that increased cooling capacity can increase the power transmission capability of a particular frame size. The basic advantages that attract pump users to sealless designs for high temperature services are equally pertinent for low temperatures.

The pumped liquid's sensitivity to temperature changes must also be considered to complement its discreet temperature evaluation. This includes liquids that have peculiar melting and/or freezing points, e.g. MDI, TDI and many acids, and ones that experience polymerization or crystallization with accompanying temperature changes, such as formaldehyde. Some liquids (caustics, for example) experience crystallization when they come in contact with air. This can occur as the liquid leaks across a conventional seal face or when air is drawn into the pump – e.g., in a system where the pump's suction is less than atmospheric pressure. These circumstances are troublesome to mechanical seals since the crystal residue can be abrasive to the sealing faces and inhibits face adjustments with accumulation in the secondary area. Also, in vacuum applications process disruptions can occur with the introduction of air.

Sealless pumps address these problems by nature of the pumped liquid's absolute isolation from the ambient environment and the ability to add or subtract heat from the system. Jackets are available for drive sections, bearing housings and pump cases to meet minimum or maximum temperature constraints of a particular liquid. For the successful application of pumps in such services, however, the pump user must disclose a liquid's temperature sensitivity characteristics. It is imperative as well to recognize that unobstructed flow paths, i.e., without the accumulation of solids, crystals or polymers, are essential to ensure lubrication and heat dissipation within the drive section. If a heat medium is not available for liquids that have the propensity to polymerize, crystallize or freeze, an externally flushed sealless or a multiple mechanical sealed pump should be considered.

Thick or Thin?

Centrifugal pumps are generally applied to relatively low viscosity liquids, and hydraulic-end viscosity corrections are the same for a sealed or sealless design. When high viscosity liquids are passed through the drive section, however, increased parasitic losses occur. The Hydraulic Institute specifies that an external flush fluid should be used when the viscosity exceeds 200 centipoise [Ref. 4]. Any simplified evaluation of viscosity's impact toward parasitic losses must recognize the fact that smaller/slower speed designs are much less affected by viscosity changes. The correction factor corresponding to the drive section's coolant/lubricant flow is an exponential step function, rather than a linear relationship, therefore parasitic losses must be well understood. Special designs are available to extend a sealless pump's viscous handling capabilities, but the earlier comment on the value of specific experience, be it the manufacturer's or user's, is again appropriate.

On the opposite extreme, many liquids described in the "Look at the Liquid" section of this article have viscosities that are less than that of water and can benefit from the use of sealless pumps. Included within this group are liquids such as HF acid, ammonia, fluorocarbon refrigerants, hot water, heat transfer fluids and hydrocarbons, all of which are regularly used in many CPI processes. It should be recognized that there are long-standing debates about the acceptability of such liquids with product lubricated bearings. Experience has shown, however, that they can conservatively be used, with well designed bearing systems, down to a 0.15 centipoise level, and some manufacturers have exhibited experience at even lower values. Properly designed product lubricated bearing systems result in legitimate sealless pump alternatives to multiple mechanical seals for low viscosity services.

A liquid's contamination with solids is another critical concern in the appraisal of a sealless pump's viability. Various manufacturers' literature specifies a range of 2 to 6% by weight. A 3 to 5% guideline is a slight compromise from the maximum published range, but it accurately addresses the capabilities of most manufacturers and offers a reasonable rule of thumb. Clarification with regard to the size of the solids, however, is required since sealless pumps have small passages within the drive section circuit. A 150 micron judgment limit allows approximately a 4:1 safety factor to the clearance between the rotor and containment shroud with worst case designs. Invariably, one of the smallest clearance locations is at the shaft (sleeve) to journal bearing. This clearance typically ranges between 0.001 and 0.0035 inches (radial) for many manufacturers. Therefore, further analysis is required to predict the impact of the concentration, size and abrasiveness of solids with regard to the materials used for the bearings and other components of the drive section.

Pump applications that fail to pass the critical questions discussed within the "liquid type" sections may be more economically serviced by a single mechanically sealed design. Extreme temperature and pressure conditions often prove to be an exception to the single seal decision and present a competitive opportunity for the sealless pump. There are times when the leak-free nature of sealless pumps is desirable, but the temperature sensitivity, viscosity and solids content characteristics of the pumped liquid converge to demand an examination of a barrier liquid's viability.

The reasonableness of using a barrier fluid to provide the drive section with a higher quality coolant/lubricant is analogous to clean liquids that would support mechanical seals (API-32/52/53 seal support plans). Disparities are generally the result of supply and consumption differences between the sealed and sealless configurations. Sealless pump manufacturers have become extremely innovative with flush designs for sealless drive sections,

Figure 1.

which effectively isolate the pumpage from that area. If a barrier fluid analysis finds that the sealless approach is not practical, the choice defaults to multiple mechanical seals, which will still require an ancillary support system.

The Internal Flow Circuit

Many of today's sealless pump applications are with liquids exhibiting good thermal stability. When that is the case, the basic design understanding of all manufacturers will ensure stable pump operations. Volatile liquids, however, are an application field in which additional scrutiny is required for operating success. We have previously discussed the fact that a sealless pump's internal circulation system is used to supply the bearings with lubricant and to dissipate the heat generated in the drive section. The corresponding flow paths are demonstrated in Figure 1 for a typical magnetic driven pump.

Inadequate flow and/or pressure will ultimately result in distress to the bearings and, if unchecked, lead to ultimate failures. Active participation in the equipment selection and operation processes will minimize these problems. The sealless decision must, therefore, include a thorough engineering analysis for such services so as not to trade a seal problem for one with a sealless pump's drive section.

Two critical liquid parameters that must be understood are vapor pressure and specific heat. The pump designer requires vapor pressure information at the rated temperature and at some level greater than that to understand the fluid's state as a result of pressure and temperature changes of the drive section's flow. Normally, vapor pressures at rated temperature plus 10°F (5.6°C) and 20°F (11.1°C) from suction will give the pump manufacturer sufficient information to make an informed analysis. A fluid's specific heat value is critical since most manufacturers' temperature rise analyses are derived from testing with water, which can absorb considerably more heat than many of the liquids in today's critical processes.

A thorough understanding of coolant flow, pressure and temperature must be demonstrated to support troublefree pump operation in the field. Pumps operating on volatile liquids typically exhibit minimum flow constraints that are dictated by thermal, rather than mechanical, limits. One of Murphy's laws is that pumps never operate at their design point. Therefore, the pump supplier must also conduct his analysis at off-design points to establish a recommended operating "window" – i.e., a minimum to maximum flow regime – to give good operating flexibility. There are many successful sealless pump applications with steep vapor pressure liquids attesting to the fact that reputable manufacturers possess the ability to meet these demanding services. Ammonia, isobutane and propane are examples.

The effort expended to conduct this flow circuit analysis will ensure that the drive section is continually supplied with a liquid and not a gas. This situation is analogous to the topic of dry running a pump. There are a number of design means that can be used to minimize the effects of a loss of pumped liquid, including impellers with special or no wear rings, impregnated or coated bearing materials and nonmetallic shrouds. As with any pump, whether sealed or sealless, the goal should be to prevent or at least to minimize the occurrence of a pump being operated without liquid.

The successful application of sealless pumps on numerous loading and unloading services is one example of the viability of the design under dry running circumstances. Success is conditional, however, upon the minimization of the dry run duration and having sufficient instrumentation to protect the machine as configured. With a sealless pump, it must be recognized that the drive and pump sections may run dry and equal care should be exercised to prevent either occurrence. Recent technological advances have been made with non-invasive electrical devices that can sense the fluid condition – liquid, gas or two phase state – within the sealless pump's drive section, thereby ensuring reliable pump operation.

Instrumentation

Too often sealless manufacturers are confronted with installations in which no precautions have been made, supplied instrumentation has not been installed or excessive instruments (which yield unnecessary complications for field personnel) have been employed. A balanced instrumentation plan is recommended to preclude the possibility of these occurrences.

Flow measurement is the surest way to guard against a loss of pump flow, but it is typically not available in most chemical plant installations. A pseudo measurement of flow is a change in the driver's power requirement. For electric motors greater than approximately 10 horsepower, a current sensing device works well, but smaller motors may be better protected with watt meters. This protection should not, however, be used as a minimum flow protection device! Power or current sensing does not have the sensitivity to act in that manner but can readily indicate the onset of a "no-flow" condition.

Pump flow and power instruments will not indicate a vaporization state in the drive section that has been caused by blocked flow passages or insufficient cooling. A temperature detection device, such as an RTD or thermocouple,is commonly used to indicate that condition and can be positioned to sense the temperature of the containment shroud.

It is generally accepted that a nominal 20°F (11.1°C) setting, greater than the calculated temperature rise, will protect against a dry run occurrence. Further, insufficient cooling will cause a CMP to experience a rapid increase in the stator winding temperature, a fact that highlights the need to always wire its thermostats.

To meet electrical safety standards, Europeans have used the science of ultrasonics to ensure a "wet" condition within the drive section of their canned motor pumps. Innovative adaptations to this technology are finding their way into sealless pump designs in North America to guard against overheating a motor or magnet assembly. Similarly, instruments are now available to indicate a canned motor pump's direction of rotation without physically seeing the shaft turn. This advancement addresses a long-standing complaint by users resulting from instructions to observe a dry run bump start to guarantee proper motor wiring.

A number of manufacturers now offer bearing wear detectors for use with carbon bearings. These instruments can be either electrical or mechanical and do a good job of alerting the user to an impending minimum material condition. Consequently, routine maintenance can be scheduled in a manner consistent with production needs. Growing competition in today's sealless pump marketplace has prompted manufacturers to take a more active role in the application of pumps, and their counsel should be sought when developing a sealless pump instrumentation plan for the first time.

This article has attempted to demonstrate that sealless centrifugal pumps are a valuable asset in the arsenal of today's pump users. The fact that they completely eliminate fugitive emissions complements increasingly strict environmental and safety objectives. To capitalize on a sealless pump's cost effectiveness and inherent reliability, however, applications should be well understood. Wisdom in this area results from a combination of complete customer input, a manufacturer's thorough technical analysis/application expertise and adequate protective instrumentation. Worldwide experience shows that these pumps should not only be considered for new installations. Many conversion opportunities exist for the retrofit of existing, poorly performing pump installations. Manufacturers' standardization with ANSI dimensions facilitates this process and makes retrofitting a cost effective alternative to living with leaking pumps.

There will continue to be a need for mechanically sealed pumps that operate outside of the hydraulic capabilities of sealless equipment. Movements toward a continuous improvement philosophy, increasing recognition of sealless pump maintenance advantages, continued developments in materials and significant strides toward pricing parity can be expected to further encourage the worldwide utilization of sealless technologies.■

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Vertical Motor-Under Pumps Expand Their Range

By Pumps and Systems Staff with Gaylan Dow of Hayward Tyler, Inc., and Joe Campanelli of Air Products & Chemicals, Inc.

If you asked two individuals, one an experienced utility machinery engineer and the other an experienced process machinery engineer, to describe an integral motor sealless pump (think canned motor pump for now) you probably will get two very different descriptions. Like the proverbial elephant and the blind men trying to understand it through touch, sealless pump technology is perceived differently at different ends of the marketplace. This article highlights the features of a type of sealless pump utilized extensively in the utility industry. It also explains how many of these features are being used with benefits in process environments.

Most specifying engineers and pump users are familiar with the sealless pump/motor combination design of a canned motor/centrifugal pump of end suction configuration with a single primary radial impeller that does the work. It is mounted horizontally on a bracket or baseplate, and, short of being compact in design and without a coupling, it would look similar to most process pumps. If you pressed harder, you might learn that it uses a mechanism known as "hydraulic balance" to locate the impeller pretty much in the center of the volute.

Another class of sealless integral motor pumps has been used for decades in the utility industry. In addition to boiler feed pumps found in all conventional power plant boilers, many boilers are designed to use pumps that force the water to circulate instead of relying on natural convection circulation. With thousands of units installed worldwide, the vertical motor-under pump is one of the unsung success stories in the pump industry.

Proven Design for Utility Service

Boiler water circulation is arduous duty, with pressures around 3000 psig, temperatures of about 650 °F and horsepower to 1600. The pump that has evolved to suit this duty is fabricated into and suspended by the plant pipe work (Figure 1). In this setup the motor is slung beneath the pump and is, in turn, supported by a hot neck, which thermally isolates the pump and motor. The whole assembly, not tied to any baseplate or foundation, is free to move with thermal expansion of the plant. In many instances the pumps will move more than one foot from cold start-up to hot running condition. This movement represents relief of piping strain which would normally have been imposed on the pump through the nozzles.

Water at boiler pressure is circulated through the motor and heat exchanger by a secondary (auxiliary) impeller that is designed into the thrust bearing. Because of the large horsepower involved, these units are designed to maximize the efficiency of the motor. However, though they are highly efficient for fluid-filled motors, even a small fraction of a large horsepower yields a significant amount of waste heat that needs to be disposed of.

This precludes the use of a canned type approach with a liner that separates the winding from the fluid in the bore of the motor. The heat generated would create unacceptably high temperatures in the winding cavity.

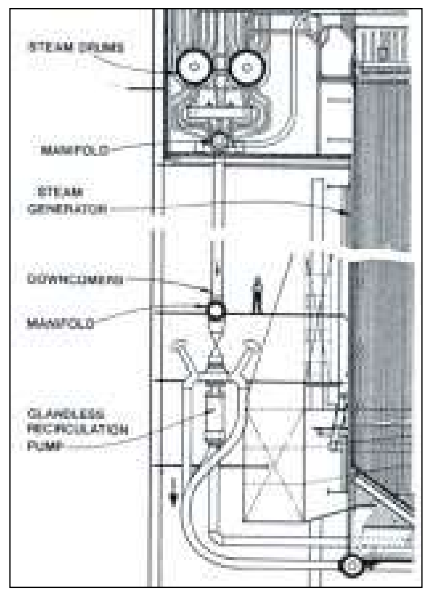

Figure 1. The glandless recirculation pump is fabricated into and suspended by the plant pipe work.

Consequently, a design approach similar to that used in many types of bore hole submersible pumps is used. The windings are coated with a high dielectric strength polymer and immersed in the water itself. This is termed a WSU (wet stator unit) (Photo 1) as opposed to a DSU (dry stator unit/canned motor), and it ensures that a large surface area is cooled by the internal water flow. Additionally, with the absence of an internal liner (or can), the rotor to stator gap is increased, and a failure point (damage to the can) is removed.

Photo 1. Wet stator unit. A secondary impeller incorporated in the motor rotating element circulates water through the motor cavity to the top of the heat exchanger and back to the lower part of the motor. This recirculation dissipates the heat generated by motor electrical losses.

A special power feed-through is utilized. It forms both the terminal for connection to the power supply (460V to 6.6 KV) and part of the pressure vessel of the motor.

Practical Advantages

Advantages of the motor-under design include the following:

• The unit is continuously self venting. Most conventional horizontal canned motor pumps with hot necks and heat exchangers have the latter located horizontally above the motor. On hot standby, this arrangement promotes thermal siphoning and keeps the motor insulation system within temperature limits. The setup will often require a venting operation from the high point (the heat exchanger) when initially filled. This operation is not required with the motor mounted beneath the pump.

• The pump remains full until intentionally drained. These are large pumps that are full of fluid and therefore extremely quiet. There have been reported incidents of boiler circulator pumps being left on while the boilers have been completely drained. Although this is not recommended, having the motor (and therefore bearings) surrounded by fluid minimizes the potential for damage during dry pump running.

• The design utilizes a hydrodynamic thrust bearing to locate the rotor axially. Because of the rotor's massive weight on boiler circulators, generating and closely controlling the large thrusts necessary to accomplish axial hydraulic balance is impractical. Therefore, the pump is designed to thrust positively in one direction, and it incorporates thrust bearings capable of accepting thrust loads in all cases. This provides the added benefit that the pump has a wide latitude of operation over the curve because there is no hydraulic balance to upset.

• Radial bearing loads are reduced by the weight of the rotor. Additionally, a multi-vaned diffuser in a concentric casing is sometimes used to further minimize radial loads.

• Suction conditions are optimized. Rather than mounting an elbow on the pump suction as is typical of an end suction pump, a longer run of straight pipe can come down directly into the pump suction.

• Design and construction costs are minimized for hot systems. The piping goes in, the piping goes out. Much like a motor operated valve, there is no need for long runs of pipe, numerous elbows or expansion joints to relieve the piping strain on the pumps. This greatly simplifies the analysis and design of the piping system. There is no foundation to design or pour to be done and no baseplate to design and fabricate. Also, there is no grouting or hot coupling alignments.

• Heat tends to travel sideways or up. Locating the motor below the hot pump end minimizes the heat exchanger load and lifetime operating costs.

• In addition to boiler circulation duty, this configuration is also used in the utility industries in attemperator spray pumps as well as in nuclear applications such as reactor internal pumps and reactor water clean up pumps. To minimize the piping involved with reactor internal pumps, the pump ends are actually located directly in the reactor vessel.

Process Versatility

Because the pump is continuously self venting, it eliminates an opera-

(photo courtesy of Hayward Tyler, Inc.)

Photo 2. A 6000 psig/300ºF canned motor-under pump in service at a chemical process plant. Large quantities of gas are injected and consumed as part of the process. This gas becomes entrained and the primary concern is loss of liquid in the pump end. The process has upset conditions with rapid pressure spikes to 6000 psig .


(photo courtesy of Hayward Tyler, Inc.)

Photo 3. This canned motor-under pump at a chemical processing facility – an end suction configuration – has to contend with entrained gas and thermal shock.

tor function such as venting on start-up – always a good idea. Additionally, some process systems (such as heat transfer) generate non-condensable vapors.

In other applications, gas that is injected purposefully as part of the process is consumed, and more and more systems have gas injected at seals and as blankets. This gas tends to collect through centrifugal force about the pump shaft and can make its way back into the motor cavity. On a canned pump with a top mounted heat exchanger, this usually requires the use of a level detection device at the heat exchanger, as well as operator intervention to vent the pump during operation. Even on canned or other sealless pump applications not requiring heat exchangers, the gas can centrifuge out around the rotor and cause possible dry running of the bearings.

In addition, because the pump remains full of liquid until intentionally drained, it is protected against dry running, and the motor is impervious to large quantities of entrained gas in the pump end. Typical applications which this benefits include tank car unloaders and batch processes in which tanks are completely evacuated on purpose.

The sealless integral motor pump utilizes a hydrodynamic thrust bearing to locate the rotor axially. A canned motor pump relies on a set of clearances about the impeller to balance the rotor assembly hydraulically and locate it axially. Theoretically, this could be accomplished in a vertical motor-under design by designing in just enough thrust to lift the rotor and locate it axially. In practice, however, using a real thrust bearing has its advantages. Positioning it in the bottom of the unit protects the pump and motor assembly from system upsets. Dry running, or the collection of entrained gas behind the impeller, negates the lift generated by the impeller. Thermal shock dramatically changes the internal pump wear ring clearances, which temporarily and drastically alters the dynamics of a thrust balance design. Also, history has shown that despite efforts to control water chemistry, many boilers are loaded with oxides, which have a tendency to wear clearances and block balance holes, upsetting the thrust characteristics of the impeller. In various process applications rust, scale, catalysts or product solids can be present, and these can create similar circumstances. Any of these conditions would cause a hydraulically balanced unit to lose lift or thrust upward enough to generate considerable wear and damage to the pump and eventual failure.

Radial bearing loads are reduced with the motor-under pump. This can be a real advantage for processes in which fluid viscosity is low. And for service in hot systems, these pumps are cost effective to install. The same design constraints of trying to bolt a massive hot piping system to grade through the pump are present in process plants as well. The piping wants to move, and it is better to let the pump move with it.

Vertical motor-under canned motor pumps can also be attached directly to a chemical reactor. This setup further reduces piping (and thus lowers costs) as well as minimizes the heat tracing requirements.

Conclusion

A vertical motor-under canned pump can be useful in applications requiring zero leakage, including situations in which large quantities of gas are present or the pump runs dry. It can also provide reliable service for hot systems in which significant thermal strain can be imparted to the pump flanges, or on systems which run both hot and cold and alignment is an issue.

If other service requirements can be met, the motor under design can also lower plant construction costs and be an advantage when floor space is at a premium – such as on oil production platforms or in multi-level chemical processes. Last, it is also useful in situations in which the minimization of piping is attractive, such as those requiring heat tracing, or for lethal services.■

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Upgrading Boiler Water Circulation Process

By Joe Campanelli, Air Products and Chemicals Inc.

Process waste heat boiler circulating pumps have traditionally been horizontal pumps of either end-suction, overhung impeller design (for very low flowrates) or double-suction, impeller-between-bearing, single-stage units for higher flow applications. Boiler water circulation is a very demanding service, with the pump taking its suction directly off the boiler steam drum. Due to the elevated suction pressure and temperature duty, mechanical seal problems are frequent, as are other maladies commonly associated with high pressure/temperature pumps, including:

• Pump casing deflection. This is caused by thermally induced pipe strain resulting in rotor rubs, seal leakage/failures, bearing failures, casing cracks, casing splitline leakage and piping cracks.

• Thermal stratification in hot standby units. This results in rotor bows, seal leakage, internal rubs on startup and rotor lockup.

• Seal leakage and seal maintenance problems. These are constant issues on boiler circulating pumps – often requiring replacements and additional spares. And catastrophic seal failures, while rare, do occur and can expose operating and maintenance personnel to significant releases of hot, high pressure water, which immediately flashes into a plume of steam.

The recent introduction of vertical canned motor pumps into this application has enabled our process pump users to capitalize on the extensive experience gained by Hayward-Tyler (HTI) in the utility industry. The HTI vertical canned motor pump offers a proven, robust mechanical design with optimum materials of construction. The application of these pumps to process waste heat boiler circulation service has solved the problems associated with conventional horizontal pumps and has provided significant benefits in pump reliability, maintenance and spare parts cost reductions.

Two recent waste heat boiler applications that have utilized the HTI pumps with good results are detailed in the chart.

The pumps in both applications have performed extremely well to date and will be chosen for this type of application on future projects. ■

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Facility	LaPorte, TX	Pasadena, TX
Pump manufacturer	HTI	HTI
Pump type	Dry stator	Wet stator
Pump model	3 x 4-8/DSU	12 x 14-14/200 WSU
Capacity - gpm	656	8052
TDH - ft.	170	76
Inlet Pressure - psig	745	620
Discharge Pressure - psig	802	646
Inlet Temperature - °F	510	489
Speed - rpm	3490	1780
Power - hp	72	200
Pumps installed	(4) 2 trains, 2x 100%	(2) 2 x 100%
On stream date	Jan ‘96	Oct ‘96

Centrifugal Pump Suction

Pump problems often begin on the suction side. Here is a guide for checking existing operations as well as information on proper designs for new suction systems in the planning stage.

By John H. Horwath, Ampco Pumps Co.

Pump not working properly? Chances are something is wrong in the suction sector. To begin with, there are a multitude of circumstances that can cause suction problems. To cover this subject effectively and systematically, you are being provided a format for checking existing operations as well as information on proper designs for new suction systems in the planning stage.

The system's suction arrangement must provide the energy to move the liquid to the eye of the rotating pump impeller. Until the liquid reaches the leading edge of the impeller vane, the pump cannot impart its energy to move the liquid onward.

One must establish a suction requirement for a pump that meets or exceeds the system's minimum suction availability. It is also the user's responsibility to see that the suction system provides and maintains the stated condition at the required flow and that the pump is maintained in good operating condition.

Existing Operation

When a previously successful pump-suction system fails, there are three primary areas that need to be looked at for probable cause. They are the pump, the suction and the liquid being pumped. The most common causes of failure in each sector are:

Pumps
- Leaky seal
- Reduced speed
- Pump modification
- Wrong direction of rotation
- Plugged impeller
- Worn wearing rings

System Line
- Leaky suction line
- Plugged line (including pump inlet)
- Sticky valve
- Suction height increased
- System line modification
- Corrosion or product build-up in suction line

Liquid
- Increase in temperature
- Drop in liquid level
- Aeration due to vortex or processing change
- Viscosity change
- Specific gravity change
- Other physical property changes

Once the problem is identified, it can be corrected. In the case of a processing change, it provides awareness of potential problem areas to check new requirements against the capabilities of the existing pump to see if the unit remains adequate.

The eye-balling method of measuring cannot be relied upon to define operating conditions effectively. Use proper instrumentation to measure pressure, flow, suction lifts, speeds and the temperature rise of motors.

When motor speeds are incorrect, check connections and measure voltage at the motor terminals. Remember, too, that flow and pressure readings should be taken in areas where stable accurate readings can be obtained. This usually requires 5 to 10 pipe diameters of straight piping after going through an unsettling section such as one that includes elbows, valves and reducers. In addition to measurement, be aware of sounds different than those emanating during normal operations, as well as vibrations and surges.

Planning Stage

It is during this phase that most potential problem areas can best be dealt with. Begin with careful planning and installation of the pump suction line. Design the line to provide the shortest practical direct route utilizing large radius elbows with an absolute minimum number of fittings and valves.

Suction Piping

Air pockets or high spots in a pump suction line invariably cause trouble. Piping must be laid out so it provides a continual rise or at least a perfect horizontal run without high spots from source of supply to the pump. For the same reason, an eccentric reducer instead of a straight taper should be used in a horizontal suction line. Another way to remove vapor trapped in a suction line because of a high point is to vent the sector back to the vapor space in the supply vessel. If an air pocket is left in the suction pipe when the pump is primed, it will often pump properly for a time and then lose its prime or have its capacity greatly reduced. The small pocket of air under a partial vacuum condition will expand and greatly reduce the effective flow cross-section of the pipe, thus starving the pump. Or air will be drawn into the pump, resulting in loss of prime. The suction pipe should be submerged to a depth of at least 3 feet when the water is at its lowest level.

Frequently, foot valves are installed on the end of suction pipe for convenience in priming. They should be sized to provide a flow area 50% greater than the pipe area. To protect the foot valve, pump and other equipment from being fouled by refuse such as sticks, rags, light plastics and miscellaneous somewhat buoyant solids, a screen strainer with an area at least three times that of the pipe area should be installed ahead of the foot valve.

After planning the required piping system, you should determine the NPSH available of the system. (This is a desirable procedure to follow where your height plus dynamic losses exceed 15 feet or handling a hot liquid or operation in a closed system.)

Keep in mind that a pump's location can be of utmost importance in establishing its suction requirements. Often you will find it more feasible to relocate the pump than to call for a special low NPSH pump. The cost of these changes must be weighed against the cost of a usually larger or special pump required for a lower available NPSH condition.

Net Positive Suction Head

NPSH (Net Positive Suction Head) can be defined as the absolute pressure at a datum line (normally the centerline of the impeller eye at the suction nozzle) minus the vapor pressure of the liquid (at pumping temperature) being pumped.

Proposed System

A. Proposed System
(units in feet of water)
NSPH Available=
hp ± (*) hz – hf – hvp where:

hp = absolute pressure acting on suction liquid surface

hz = liquid suction height above or below the pump impeller centerline

hf = total head losses in the suction including exit, entrance, fitting and pipe losses at the intended flow rate

hvp = vapor pressure of the pumped liquid at the pumping temperature

(*) for liquid level above centerline + applies;for liquid level below centerline – applies

Existing System

B. Existing System
(units in feet of water)
NSPH Available =

$$pg \pm ps + \frac{v2s}{2g} - hvp$$ where:

pg = gas pressure in closed tank or atmospheric pressure in open tank

ps = gage pressure reading in the pump section centerline (steady flow conditions should exist at the gage tap; five to ten diameters of straight pipe of unvarying cross-section are necessary immediately ahead of tap). The corrected gage reading is a minus term in the equation if it is below atmospheric pressure.

$\frac{v2s}{2g}$= velocity head at point of measuring ps (based on actual internal diameter of pipe) at point of pressure taps

hvp= absolute vapor pressure of the pumped liquid at the pumping temperature

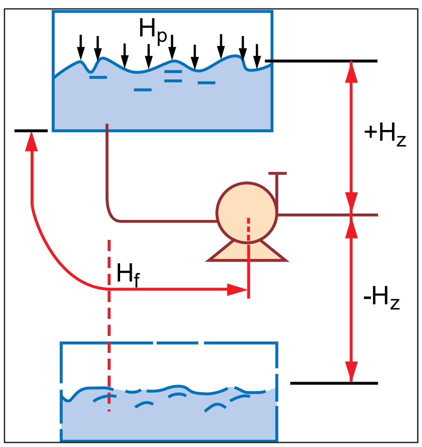


Figure 1. Proposed system design

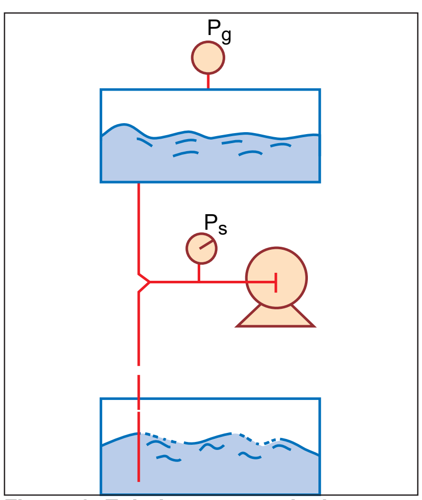


Figure 2. Existing system design

Pumped Liquid Characteristics

The liquid being pumped can have a profound effect on the pump's suction capability. Keep in mind that a centrifugal pump is not fully capable of handling gases or vapors and that it is always necessary for the absolute pressure in any sector of the pump to be higher than the vapor pressure of the liquid being pumped. Entrained air also has an adverse effect on pump performance. As little as 1% entrained air by volume can reduce pump head and capacity substantially. Under no circumstances should a standard centrifugal pump unit be expected to handle more than 3% air by volume as measured under pump suction conditions.

Substantial effort should be made to keep air out of the liquid entering the pump. This commonly occurs if a vortex develops at the suction pipe inlet and adequate submergence or effective baffling can help prevent this condition. A leak in the suction or pump stuffing box operated under a vacuum may also introduce air into the pump's suction. A well-designed entry coupled with good maintenance practices will alleviate most entrained air situations.

Dissolved gas or gas evolving from a chemical reaction can also be troublesome. Common practice calls for the use of a large diameter impeller (not necessarily a larger inlet) to meet the hydraulic performances based on handling cold clear water alone. Where the percent of air (or gas) exceeds the recommended limit for standard pumps (which may vary for different designs), it may be appropriate to consider less efficient pumps designed specifically for handling two-phase flow.

The boiling of a liquid that can occur at reduced pressures is dependent on the liquid properties – pressure, temperature, latent heat of vaporization and specific heat.

The stated suction characteristics of pumps based on cold water is standard throughout the pump industry. Determination of other liquids must usually be made through testing, although the Hydraulic Institute's Standard 14th Edition (Figure 70 in their publication) does provide correction for some liquids at temperatures up to 400°F. The same operating conditions as those with cold water are usually maintained in the absence of data. Other contributing factors are liquid surface tension, specific gravity and viscosity.

Some adjustment can be made for certain liquids, however, to avoid any chance of cavitation. The same operating conditions as with water are usually maintained. Always remember that the pressure at any point within the pump must remain higher than the vapor pressure of the liquid if cavitation is to be avoided.

Cavitation

When the absolute pressure becomes equal to or less than the vapor pressure of the liquid being pumped, bubbles consisting of dissolved gases begin to form. The bubbles are carried by the liquid flow into an area beyond the leading edges of the impeller vanes where higher pressure being developed causes the bubbles to condense and collapse, creating severe mechanical shocks. The term used to indicate this process is “implosion.” The bursting bubbles begin to damage the pump’s interior surfaces in the immediate vicinity, and this damage can be quite destructive over a period of time, depending on the pressures developed, their collapsing rate and the base material being subjected to attack.

Four common symptoms of cavitation are:

Noise – caused by the collapse of vapor bubbles as they enter the high pressure area. This is typically identified as a light hissing and cracking sound at the onset of cavitation and a rotating noise when fully developed.

Vibration – caused by the impacting of the bursting bubbles on the impeller surface, this condition can also result in a premature bearing and shaft seal failure.

Drop in Efficiency – indicates the onset of a cavitating condition. The degree of efficiency drop-off increases precipitously as cavitation increases.

Erratic Flow – commonly occurs. The severity of the fluctuating flow is determined by the degree of cavitation and pump design.

The Pump

The suction system must provide the energy to move the liquid into the eye of the impeller. The determination of a centrifugal pump’s NSPH requirement is established empirically through a series of performance tests run under specific conditions.

Of prime importance in the suction sector is the pump’s inlet vane angle. This is usually in the range between 10 and 27 degrees independent of the pump’s design. An often used flow angle is 17 degrees – a compromise between efficiency and a low NSPH requirement. For best efficiency the vane inlet angle would be nearer 27 degrees while for a low NSPH requirement one would go closer to 10 degrees. Four to six vanes are commonly used in most designs of the smaller units available from several pump vendors.

Manufacturers have developed various pump design innovations to improve suction capability. Extremely low NSPH availability may require the inlet blade angles to be reduced even further. Some commercial units provide a separate axial impeller (inducer) in the suction entry just ahead of the main impeller to further induce flow into the standard impeller’s eye.

When a low angle inlet is introduced it may be necessary to reduce the number of vanes to decrease the blockage effect. Some manufacturers reduce the length of every second inlet blade tip into the impeller passage, speed (rpm) and profile centroid radius.

Pump manufacturers today incorporate design features that provide efficient conversion of velocity energy to pressure energy, smooth cast surfaces for lower friction losses and sharp vane tips to reduce entry shock losses in the impeller eye. Smooth surfaces can also defer the onset of cavitation because they delay the formation of microscopic bubbles that form prematurely on rough surfaces prior to the inception of a cavitating condition.

Accurate determination of the start of cavitation requires very careful control of all factors that influence pump operation. A number of test points bracketing the point of change must be taken and the data plotted. Because of the difficulty in determining just when change starts, a drop in head of 3% is usually accepted as evidence that cavitation is present.

Summary

An important aid in diagnosing a pump or application problem is a maintenance file card or folder kept on the pump that lists its history of operational problems as well as parts replacement over the years. In cases where severe attacks occurred in the impeller suction area caused by erosion or cavitation, photographs of the attacked area should be kept as well as commentary describing operation at that point – be it noise, vibration, erratic flow and/or surges.

Become familiar with the terminology and procedures provided in this article since they can aid in identifying and resolving many of the problems encountered on the suction side of the pump. Understanding basic pump suction concepts will enable you to more clearly explain your situation to your pump representative when a persistent problem can’t seem to be corrected.

When purchasing a new or replacement pump for a sensitive suction service, go beyond the price, pictorial and geographical presentations and ask the following questions:

1. Were the pump’s calibration tests conducted in conformance with the Hydraulic Institute’s standards?

2. Can the vendor provide performance data specific to your anticipated suction requirements?

3. Can the vendor furnish a typical pump impeller and casing set for you to examine?■

John H. Horwath is a Senior Technical Consultant with Ampco Pumps Company in Milwaukee, WI.

Pumping Options for Low Flow/High Head Applications

Familiarity with the major designs will help you match the right one to your application.

By Patrick Rienks, Sterling Fluid Systems (USA), Inc.

There is a trend toward low flow/high head pumps in the chemical processing industry. This article will focus on several types of such pumps, describing the advantages of each and telling when they should be used. We will address the more popular low flow/high head pumps, giving basic terms and discussing one specific design in greater detail.

Low Flow/High Head

Many industrial processes require approximately 15 gpm with a pressure of 200 to 1000+ ft TDH. Many companies use standard end suction pumps in an attempt to meet these needs. Most end suction pumps, however, are not designed for this type of application – a fact that leads to premature failure.

As an ANSI pump is operated to the left of its Best Efficiency Point (BEP), several things happen. First, there is suction recirculation. Second, with even less flow, impellers, bearings and mechanical seals wear out faster. Last, in shutoff conditions, there is no flow and significant rise in temperature. All of these result in shorter pump life, less reliability, shorter seal life, and shorter MTBPM. However, several options are available to apply the right pump to the right application.

Option 1 – Special ANSI Design Impellers

Due to the large installed population of ANSI pumps in the world, many manufactures have designed special type of impellers for low flow applications. There are basically two configurations.

The first is the “Barske” design. Basically, this is a semi-open impeller with balance holes and it requires a special concentric casing. The advantage of this design is stable low flow/high head applications, and only the impeller and casing have to be replaced on an existing ANSI pump for a conversion.

The second low flow impeller design is an enclosed type that provides stable operation at low flows but has added benefits: the impeller is the only part that needs to be replaced, and there are no wear rings. Thus, an existing ANSI pump can be changed out with minimal cost and time through a simple replacement of the impeller. Figure 1 illustrates this type of design.

Side Note: One company offers a heavy duty vertical enclosed-design chemical sump pump for low flow/high head applications that offers an additional benefit. It has, as an option, balance holes in the back cover that allow the pump to run against a closed valve (dead headed) for extended periods of time with no adverse effect on pump life. This is ideal for applications in which tanks are being used for a batch process that only requires intermediate amounts of product. With the pump’s ability to run continuously, the customer can let the unit run and operate a valve to meet requirements. Also, the pump has ANSI impellers and casings as a standard, which offers increased interchangeability of parts and less inventory in the field.

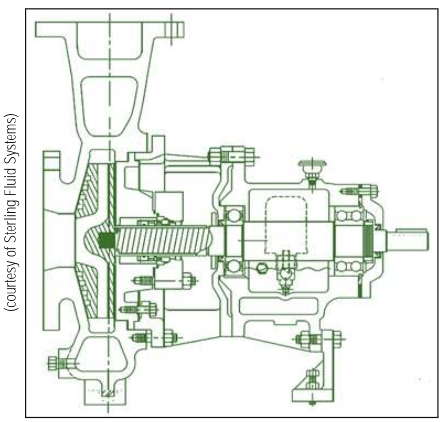

(courtesy of Sterling Fluid Systems)

Figure 1. Enclosed design for ANSI pumps

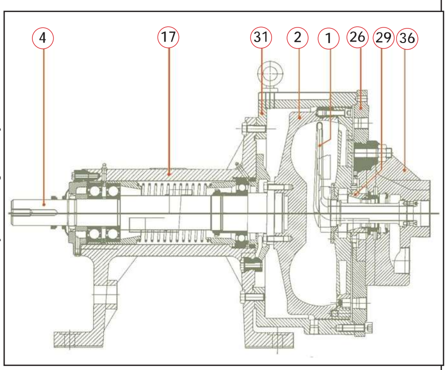
Both basic impeller designs have been available for some time from various pump companies. In addition, these designs allow for field upgrades while keeping cost and down time minimal. Furthermore, if these are purchased as new units, all other parts would be interchangeable with other installed ANSI pumps.

These designs will handle a small amount of entrained air and/or solids that may be present. The NPSHR is also less than standard ANSI pumps running at the same service. These pumps are available with all the ANSI options –e.g., jacketing and large bore stuffing boxes and a wide range of materials.

Option 2 - Pitot Tube Pumps

In a unique single stage centrifugal pump known as the Pitot tube pump, the Pitot tube is stationary, and the inner casing rotates (See Figure 2). The liquid enters the pump through the suction line, passing the mechanical seal, then enters the rotor (Part 2) where it is brought up to rotor speed. The liquid near the largest rotor diameter has a pressure that obeys the basic mechanical laws of centrifugal force.

A stationary wing-shaped Pitot-tube (Part 1), is placed inside the rotor (Part 2) which has a circular opening near the largest rotor diameter. The Pitot-tube has a double function. First, the liquid is forced to enter the tube due to lower pressure in the tube. Second, when the rotating liquid hits the specially shaped stationary tube, the liquid speed is transformed into pressure energy.



(courtesy of Sterling Fluid Systems)

Figure 2. Combitube – Pitot-tube style design

This operational principle enables the pump to generate pulsation-free flow and a stable NPSHR curve.

The Pitot tube design offers higher pressures – up to 6200 feet of TDH – whereas most ANSI pumps are limited to about 900 feet of TDH. The pressure can be significantly higher in Pitot-tube pumps because the mechanical seal (Part 29) is subjected to suction pressure only, and thus the seal pressure is low, where in an ANSI design the seal is subjected to a combination of suction and discharge pressure.

Also, the Pitot tube design has lower NPSH requirements than the ANSI style, and it can handle some solids. Pitot-tube pumps offer greater efficiency, 64% versus 35% for ANSI styles. A wide range of alloys is available, and the design provides stable flow in a wide range of applications.

And there is another advantage. To increase flow and head in most situations, one needs to change only the Pitot-tube and/or the speed. This provides a quick and inexpensive solution versus changing entire sizes of ANSI designs.

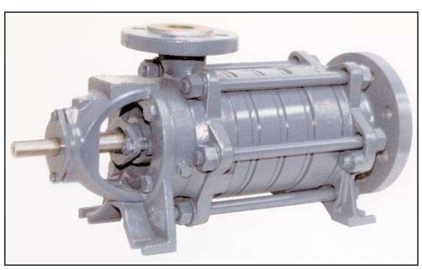
The Pitot tube design has been in industry for more than 35 years. Thus, for special applications it is a proven problem solver. Typical applications include cleaning, descaling, injection, boiler feed, process handling and hydraulic and spraying systems in chemical, plastic, paper, steel and other industries.

Option 3 - Side Channel Pumps

The side channel design is used more in other parts of the world than in the USA, but it is starting to draw domestic interest. Photo 1 shows a typical side channel design. Figure 3 shows a cross-sectional view.

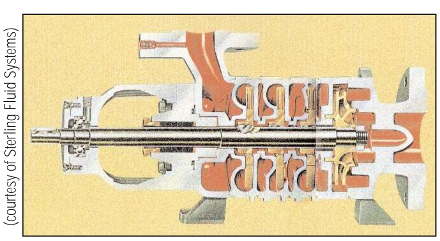
This proven end suction multi-stage design brings the liquid into the eye of the impeller. The inlet stage is centrifugal, and the impeller is enclosed, optimizing suction requirements and providing superior NPSHR values.

Additional performance is achieved by the open impeller stages behind the inlet stage, each of which generates pressures many times greater than standard centrifugal pumps running at the same speed. The special configuration of the stages gives the pump excellent gas handling capabilities and self priming.



(courtesy of Sterling Fluid Systems)

Photo 1. CEH side channel pump

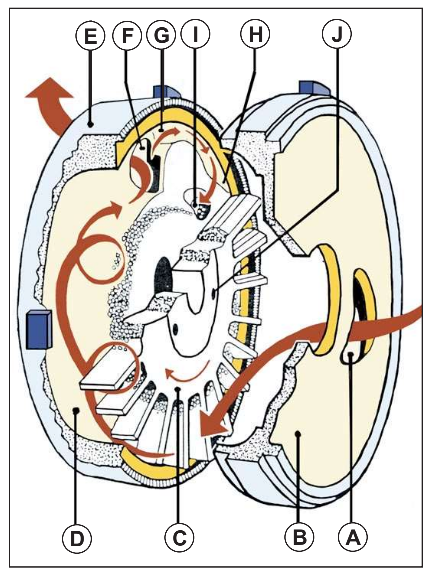


(courtesy of Sterling Fluid Systems)

Figure 3. Cross section of CEH side channel impeller stage

Also, unlike ANSI designs, side channel pumps will not vapor lock. ANSI will handle approximately 7% entrained gas vs. 50% for the side channel design. For example, this pump can handle volatile liquid up to 150 gpm at 1200 feet TDH with an NPSHR of less than 1 foot. No other design can accomplish this. Additionally, it can achieve suction lifts of more than 25 feet, has "floating" impellers that eliminate end thrust on bearings, and can operate at low speed and under 25" Hg vacuum or greater for long periods. The side channel pump is a compact design available in a wide range of materials.

Figure 4 illustrates the design principle of lateral channel stages in turbine pumps. This design can be made in one to eight stages. From the diagram it can be seen that liquid or



(courtesy of Sterling Fluid Systems)

Figure 4. Cross section of CEH side channel impeller stage

vapor or mixtures enter the stage through the inlet port (A) in the suction intermediate plate (B). Although not shown, it should be noted that the internal face of this plate is flat, not channeled as in conventional turbine pumps. The mixture, once it encounters the rotating impeller (C), makes several regenerative passes through the unique side channel design shown (D), located in the discharge intermediate plate (E).

Due to centrifugal action, the liquid, being the heavier component in a mixture, is forced toward the periphery of the chamber, whereas the lighter vapor or air collects near the center at the base of the impeller blades. Most of the liquid exits through the discharge port (F), with the remainder then guided along the mini channel (G), which eventually dead ends at point (H). There the liquid is forced to turn toward the impeller hub, thus compressing any vapor or entrained air at the base of the impeller blades. This compressed vapor or air is now forced through the secondary discharge point (I), after which it rejoins the major portion of the liquid that was discharged through discharge port (F).

Thus, the problem of continual air or vapor build-up within the chamber has been overcome, and the liquid vapor mixture continues on to the subsequent stage and is eventually discharged from the pump. Where multiple stages are utilized, they are staggered radially to bring about balance and minimize shaft deflection. The impeller in each stage, although keyed for radial drive, is allowed to “float” axially, thus assuming a running position in the equilibrium brought about by balance holes (J), which are appropriately positioned in the impeller hub. The floating action of the impeller also eliminates axial thrust on the pump’s external ball bearings.

This design, also available in a mag drive version, can be applied to the same services as Pitot-tube and ANSI style pumps, but it can be additionally applied to self priming applications. Another option is a barrel design enclosing the entire pump for added safety. However, due to the lateral gap between the vane wheel impeller and the guide disk in this design, these pumps should not be used where solids may be present.

The barrel pump is also available in many materials, including non-metallic PAEK.

Summary

There are many possibilities for low flow/high head applications. We have explained some basic guidelines. However, if you have a specific application, its best to ask the applications group of a pump company to review the data in order to select the right pump for your needs. In today’s changing industry, there are many solutions to help you keep your process running, lower your costs and greatly increase your MTBPM by selecting the right pump for the job.■

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Anti-Friction Bearings in Centrifugal Pumps

A detailed overview of bearing design, along with application and maintenance tips.

By William E. (Ed) Nelson, Turbomachinery Consultant

Introduction

Bearings used in centrifugal pumps are categorized according to the direction of the forces they absorb, either radial or axial thrust. Most small centrifugal pump bearings use anti-friction bearings, ball or roller types, because they can be designed to handle a combination of both radial and axial thrust loads.

Anti-friction bearings use balls or rollers instead of a hydrodynamic fluid film to support a shaft load with minimal wear and friction (Figure 1). Cleanliness, accuracy and care are required when installing ball bearings. The ball bearing is a piece of precision equipment manufactured to extremely close tolerances. To obtain the maximum service from it, the shaft and housing must also be machined to rigid tolerances. For example, locating shoulders must be at right angles to the shaft centerline so the bearing will be squared with the shaft, and the housing bores must be in almost perfect alignment to ensure that the bearing will not be forced to operate in a twisted position. Maintenance of ball bearings is simple. Protect the bearing from contaminants and moisture, and provide proper lubrication.

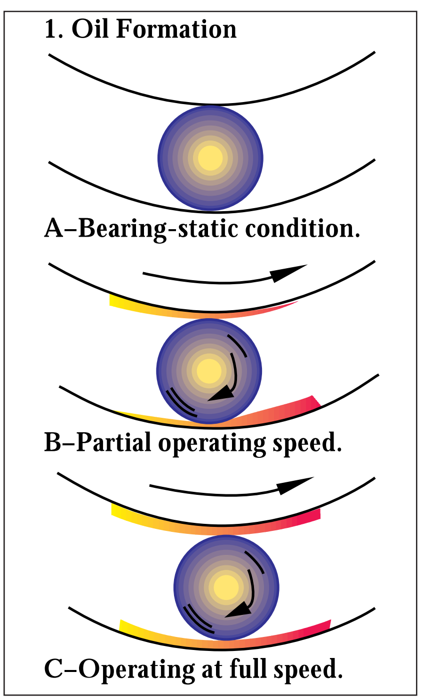

Figure 1. Theory of ball bearings

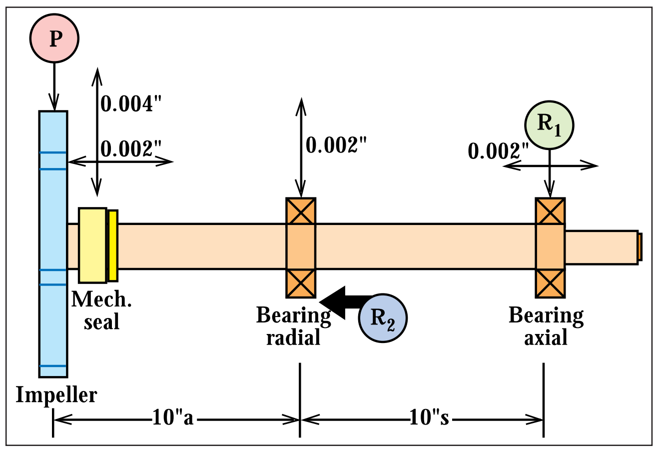

Figure 2. Loads on radial and thrust bearings

Bearing Loads

Assuming that the above conditions are satisfied, the life of a ball bearing depends on the load it must carry and the speed of operation. The loads on pump bearings are imposed by the radial and axial hydraulic forces acting on the impeller.

In any two-bearing system, one of the bearings must be fixed axially while the other is free to slide. This arrangement allows the shaft to expand or contract without imposing axial loads on the bearings. At the same time, the arrangement locates one end of the shaft relative to the stationary parts of the pump. Generally, the outboard bearing of between-bearings design pumps, or the closest one to the coupling on a back pull-out design, is fixed axially. The inboard bearing is free to slide within the housing bore to accommodate thermal expansion and contraction of the shaft as shown in Figure 2. Since the outboard bearing is fixed in the housing, it must carry both axial and radial thrust. The axial thrust is considered to be acting along the centerline of the shaft and therefore is the same at the outboard bearing as it is at the impeller. The radial and axial loads combine to create an angular load at the outboard bearing.

Radial thrust acting on the impeller creates radial loading on both bearings. The magnitude of the load at each bearing can be determined by the use of Figure 3 and these equations:

$$R_1 = \frac{P \times a}{s}$$

$$R_2 = \frac{P(a+s)}{s}$$

Where:

R = Radial load on bearings 1 and 2 (pounds)
P = Radial thrust on impeller (pounds)
a = Distance from the centerline of the impeller to the center line of the inboard radial bearing (inches)
s = Distance from the centerline of the inboard radial bearing to the centerline of the out board bearing (inches)

Radial loads come from other sources as well. The weight of the rotating assembly (shaft, sleeve and impeller) gives one load. Unbalance and external misalignment of the shaft give still another. The weight of the overhung coupling also creates a bearing load. Pump designs should limit the shaft deflections at the seal face to under 0.002 inch at the worst conditions. For single stage horizontal pumps, this will be with the maximum impeller diameter at “shut off” conditions – i.e., closed discharge. For larger double suction pumps, this load might well occur at the far end of the performance curve. Attention paid to cutwater clearances, Gap “B,” and “overfiling” of impeller vanes can reduce some of the hydraulic loads.

Ball Bearings Types

A ball bearing normally consists of two hardened steel rings and several hardened balls utilizing a separator to space the rolling elements and reduce friction as shown in Figure 3. The many ball bearing designs used

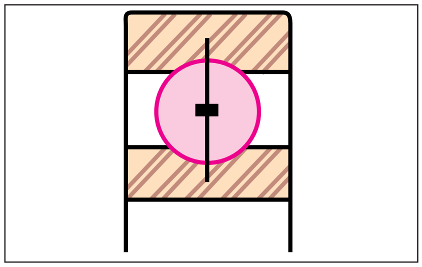

Figure 4. Conrad bearing design

in industry are classified according to the type of loading they receive: radial, thrust and combined. Sizes and classes of precession of bearings are governed by the Anti-Friction Bearing Manufacturing Association (AFBMA) and by the Annular Bearing Engineers Committee (ABEC). There are five ABEC Classes, 1, 3, 5, 7 and 9. Class 1 is standard, and Class 9 is high precision. Pump bearings generally are Class 3, loose fit. ABEC Class 9 is factory order only and has no longer bearing life or higher speed rating than ABEC 1.

Three types of bearings are generally used in centrifugal pumps:

Conrad Type – The Conrad type – identified also by its design features as the deep-groove or non-loading groove type – is the most widely used (Figure 4). A general purpose bearing, it is used on electric motors or wherever slight axial movement of the shaft is permissible. The deep-groove rings enable this bearing to carry not only radial loading, for which it is primarily designed, but about 75% of that amount of thrust load in either direction, in combination with the radial load. API 610 Standard “Centrifugal Pumps for Petroleum, Heavy Duty Chemical and Gas Industry Services” pumps require that single row or double row radial bearings be Conrad type with Class 3 or loose fit. This permits enough flexibility to let the shaft correct for any misalignment between the housing and the shaft.

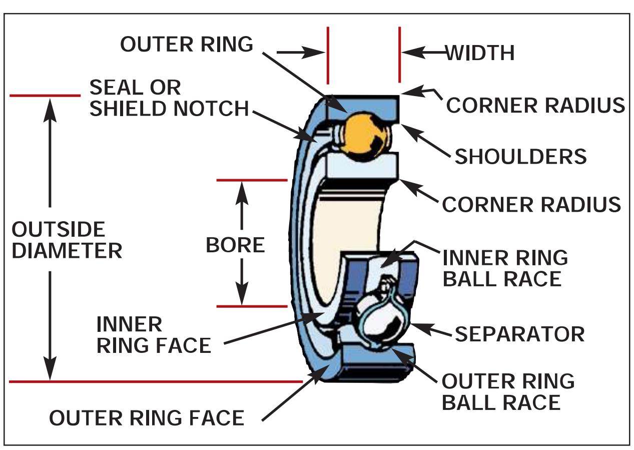

Figure 3. Elements of a ball bearing

Maximum Capacity Type–Another radial bearing is the maximum capacity or filling-slot type. It is provided with a filling notch that extends through the ring or ring shoulders to the ring way, permitting a larger number of balls to be placed between the rings than can be done in the same size Conrad bearing. The supposed advantage of these bearings (larger load carrying capability) has vanished with the availability of better steels and lubricants. Filling slots often are not precision machined and can enter the ball contact area of the rings. This will result in early bearing failure. Several bearing manufacturers consider a filling slot bearing to be unreliable and discourage their use. The 1981 and later editions of API 610 prohibit their use although they are still permitted by ANSI specifications.

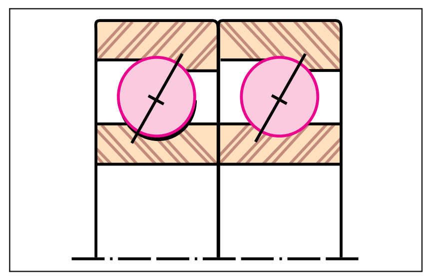

Figure 5. Angular contact bearing design

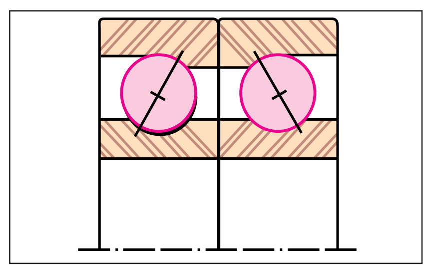

Figure 6. Back-to-back duplex mounting of angular contact bearings

Angular Contact Type – This design allows the carrying of high radial loads in combination with thrust loads from 150 to 300 percent of the imposed radial load (Figure 5). Unlike the Conrad design, the contact angle is not perpendicular to the bearing axis. Several different angles are available, offering a variety of radial and thrust loadings. Filling slots are not used in this design, and

Contact angles for 7000 series duplex bearings					
	Contact angle				
Bearing manufacturer	20°	25°	30°	35°	40°
A	X			X	X
B			X		
C		X		X	
D					X

Figure 7. Contact angles of various bearing manufacturers

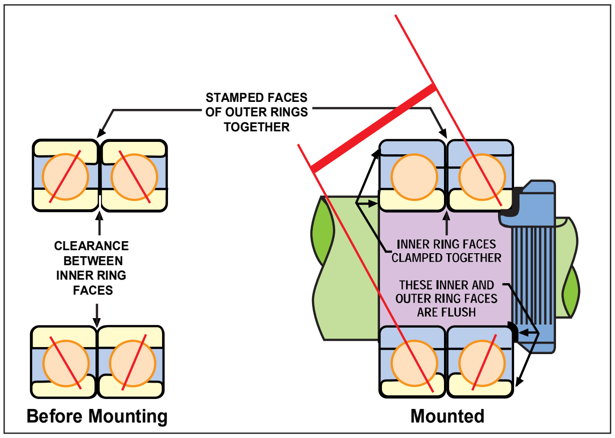

Figure 8. Back-to-back (DB) mounting moment arm

thrust loads can be imposed from one direction only.

Duplex Type – These are identical angular contact bearings placed side by side. The contacting surfaces must be ground to generate a specified preload (Figures 6). This special grinding allows the two bearings to share loads equally. Without it, one bearing in the pair would be overloaded, the other underloaded.

API pump specifications require duplex 40 degree contact angle thrust bearings mounted back- to-back with a light (100 pounds or 45 kg) preload as the best choice. While the requirement of a 7000 series, 40 degree contact angle, light preload bearing should be a fairly tight specification, it is not. First, there are three 7000 series bearing designs – light, medium and heavy. There is about a 50% change in capacity from one design to another. Second, some manufacturers use more than one contact angle. The contact angle is the source of considerable confusion as shown in Figure 7. There are no standard designations to identify the 40 degree angle.

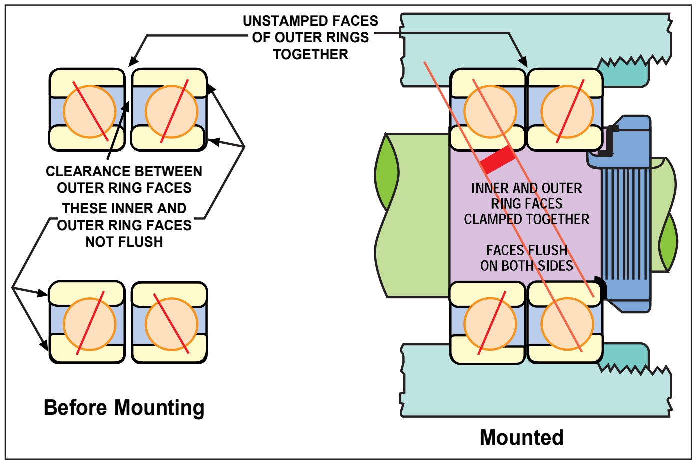

Figure 9. Face-to-face (DF) mounting moment arm

Not all ball bearing companies manufacture the 40 degree angle angular contact bearing. If the local bearing supply house is not on its toes, pump repairs could be made with non-specification bearings. The 40 degree contact angle gives an 18 to 40 percent increase in capacity over the 30 degree angle depending on bearing size. This differential is a very important piece of information, and unfortunately, it is communicated or designated in industry by a confusion of suffixes. The numerical code used in bearing identification is mostly standard among the various bearing manufacturers. However, the alphabetical prefixes and suffixes are not. When identifying bearings from codes for the purpose of inter changing bearings, care should be exercised that the meaning of all numbers and letters is determined so an exact substitution can be made. Most manufacturers supply cross-reference tables for identifying equivalent bearings. A very good source is the bearing manual of the AFBMA.

Mountings

There are five types of duplex angular contact bearing mountings, although only two are commonly used. Rigidity of the shaft and bearing assembly depends in part on the moment arm between ball-contact angles of duplex bearings. Within reasonable limits, the longer the moment arm, the greater its resistance to misalignment.

API 610 calls for the DB or back-to-back mounting of angular contact bearings. They are placed so that the stamped backs of the outer rings are together. In this position the ball-contact angles diverge outwardly, away from the bearing axis. With DB bearings the space between the diverging contact angles is extended (Figure 8). Shaft rigidity and resistance to misalignment are correspondingly increased.

DF bearings are intended only for face-to-face mounting. They are placed so that the faces (or low shoulders) of the outer rings are together (Figure 9). Ball-contact angles thus converge inwardly, toward the bearing axis. With DF bearings the space between the converging contact angles is short. Bearing-shaft rigidity is relatively low. However, this arrangement permits a greater degree of shaft misalignment than other mounting methods. Some older multistage pumps use this mounting arrangement.

Special Designs

Centrifugal force in the unloaded thrust bearing causes its balls to move out of their intended track and operate on a skewed axis. The balls begin to slide rather than roll during rotation. The increased friction that results reduces the viscosity of the oil film, accelerates wear of

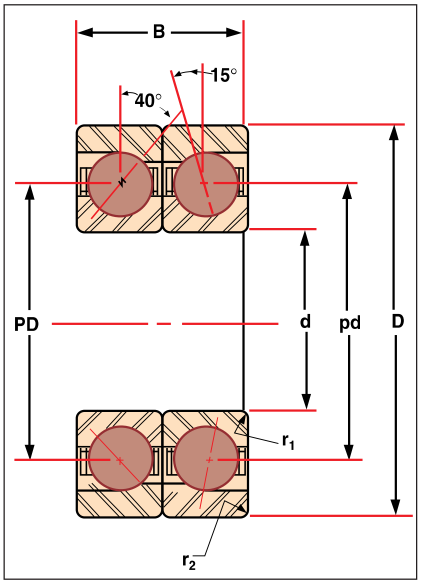

Figure 10. Special design duplex angular contact bearings

the raceways and leads to early failure. Some recent work in bearing selection indicates that adherence to the API specification of 40 degree bearings mounted DB may not be optimal.

For single stage pumps in which the thrust action is steady and in one direction at all flows, the use of a 40 degree angular contact bearing to absorb the primary thrust and a 15 degree angular bearing for any reverse thrust has extended the life of pump bearings. The 15 degree bearing decreases the tendency for ball sliding and increased friction. The bearings also have machined bronze retainers to reduce internal friction further. The bearings are packaged in pairs and are marked so that when they are mounted they will accommodate the primary thrust load (Figure 10). This arrangement is better in some but not all applications.

Recommendations for bearing use are summarized as follows:

A. Pump Type: Overhung single suction pump and low speeds, 1750 cpm or below, with any load conditions

Bearing Style: 40 degree duplex, DB mounting, or 40 and 15 degree

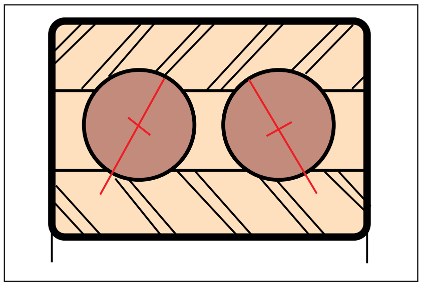

Figure 11. Double row bearings

B. Pump Type: Overhung single suction pump and high speeds, in excess of 1750 cpm, with large thrust loads coupled with radial load

Bearing Style: 40 and 15 degree, DB mounting

C. Pump Type: Double suction between bearings and high speeds, in excess of 1750 cpm, mostly radial loads with low thrust loads

Bearing Style: Duplex 40 degree, DB mounting

Double-Row Type

Essentially, the double-row bearing is an integral duplex pair of angular contact bearings with built-in preload (Figure 11). It resists radial loads, thrust loads or combined loads from any direction. Two basic types are available, corresponding to the face-to-face and the back-to-back mounting of conventional duplex bearings. Avoid using two double-row bearings on the same shaft because this makes the mounting too rigid. The bearings on each end of the shaft will tend to impose loads on each other.

While double-row bearings can be constructed with angular contact, such designs require a filling slot for assembly of at least one row. API 610 prohibits the use of the design because of its greater vulnerability to failure in reverse thrust applications.

Bearing Misalignment Capability

The ability to tolerate misalignment between the bearing housing and the shaft is dictated by ball and ring geometry. Table 1 is a chart showing the relative capabilities of three bearing types. Knowledge of these relative capacities can help a maintenance engineer or supervisor make substitutions to get out of many bearing problems. Note the Conrad bearing's rated radial capacity and speed limit are taken as unity for comparison purposes.

The angular contact bearing can carry almost double the Conrad's radial rating in thrust. The Conrad can only carry 75% of its radial rating in thrust, and the self-aligning ball bearing can carry only 20%. This means that thrust load on a self-aligning ball bearing is prohibited. Note the angular misalignment capability of the various bearings. The Conrad can withstand 15 minutes; the self-aligning ball can take 16 times as much or 4 degrees. Values in this chart are for comparison purposes only. Actual catalog values for load ratings and limiting speed should be used.

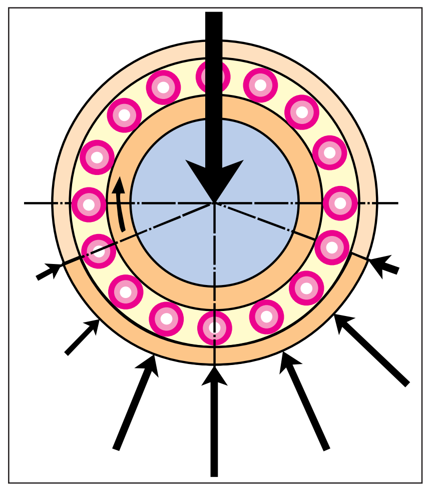

Figure 12. Load zones and retainers of a ball bearing

	Average relative ratings			
Type	Radial	Thrust	Limiting speed	Misalignment
Conrad type	1.00	0.75	1.00	± 0 deg 15’
Angular contact 40°	1.00	1.90	1.00	± 0 deg 2’
Self-aligning	0.70	0.20	1.00	± 4 deg

Table 1. Capacity for various bearing designs

Other Bearing Problems

There are a number of problems associated with anti-friction bearings utilization that impact pump reliability.

Retainers

A retainer ring or cage is used to make all the balls of a bearing go through the load zone (Figure 12). The most common retainer material is low carbon steel (1010 analysis) attached by fingers, rivets or spot welding. Riveted or spot welded steel strip retainers are more subject to fatigue failures. When a bearing ring or cage is misaligned, the balls are driven up against the ring shoulder, the top ball to the left and the bottom ball to the right. The center balls on each side, at this particular point, tend to stay in the center of the ring because balls in this position relative to the misalignment are not thrust loaded. The net effect of this action is to flex the ring in plane bending. As the inner ring turns, a cyclic retainer bending stress occurs.

The load on the retainer pocket is also cyclic. At the high thrust positions, the retainer exerts the maximum force in maintaining the ball space. Since the retainer and ball are in rubbing contact, the thermal load is at its highest on one side (no thrust load point). In this manner, the retainer is subjected to both a flexing and thermal cyclic load that can lead to fatigue cracking at retainer stress points. Notches and rivet holes may form. Due to the rubbing contact between the retainer and balls, the lubrication requirements here are more critical than for the rolling contact between the balls and rings. Shock loading of the bearings also causes retainer failure at the pockets.

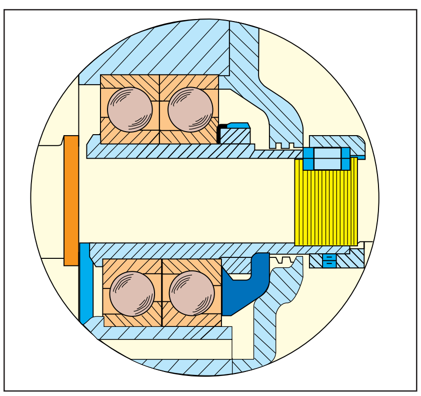

Figure 13. Typical bearing carrier design

Some manufactures use pressed brass, machined bronze, machined phenolic and molded plastics in an effort to reduce the heat generation. The relative desirability of bearing separator types is as follows:

1. phenolic
2. machined bronze
3. pressed brass strips
4. pressed steel strips
5. riveted steel strips

Bearing Carriers

In order to use a larger radial bearing and still be able to remove the impeller or mechanical seal from the shaft, some manufacturers utilize a bearing carrier similar to the one shown in Figure 13 on between-bearing pumps.

The carrier is a shouldered sleeve with a small clearance between it and the shaft. The radial bearing is then shrunk onto the outside diameter of the carrier. The problem with this design is that if the bearing begins to heat up, due to lack of lubrication or some other reason, the carrier also heats up – expanding until it comes loose. At this point, even though the bearing has not failed yet, the carrier may be free to spin on the shaft, a condition that will eventually cause the shaft to bend or fail. Current API 610 specifications require that the bearings be mounted directly on the shaft. There are a lot of carriers still in service.

Snap Rings

Snap rings are flat, split washer-like devices used by some manufacturers to position components axially – e.g.,ball bearings and seal sleeves on shafts.

As with bearing carriers, there are two major problems in using snap rings. First, removal requires the use of a tool that is not normally found in the pump machinist's tool box. When used in a shaft, they must be positioned in a groove. The addition of a radial groove in the shaft effectively reduces the diameter and may weaken the shaft. When used in a bearing mounting, the rings permit considerable end float of the bearing. Current API 610 specification prohibits the use of snap ring mounted bearings.

Bearing Arrangements

Different arrangements of anti-friction bearings can handle various loadings imposed on the pump. Pump design is crucial in determining possible bearing arrangements.

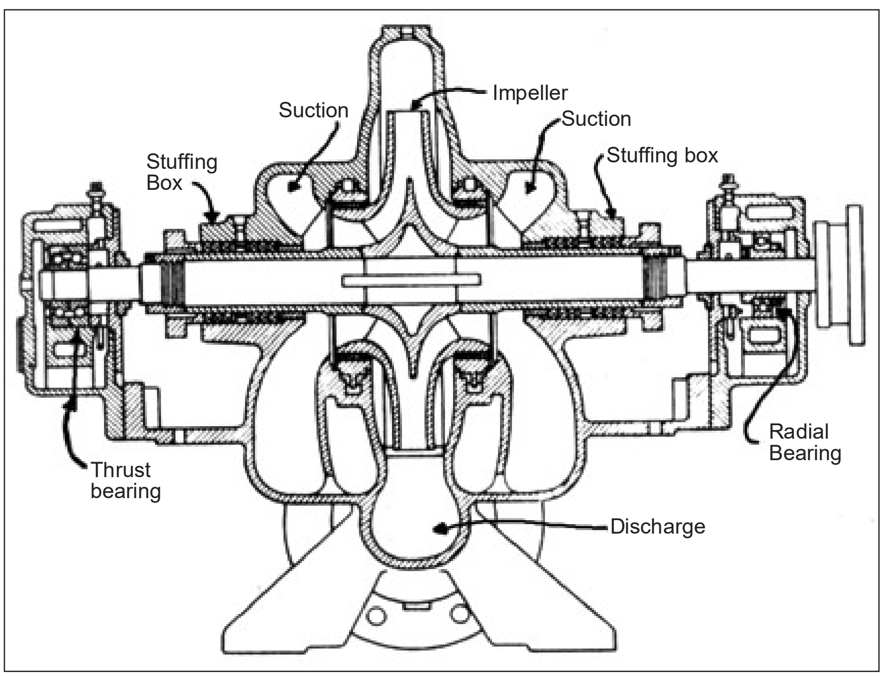

Figure 14. Typical pump bearing arrangements – between bearing pump

Horizontal Pumps

Overhung impeller pumps usually employ ball bearings only. In a typical bearing housing arrangement (Figure 2), the radial ball bearing is located adjacent to the impeller or inboard position. It is arranged to take only radial loads. The thrust bearing is located closest to the coupling and usually consists of a duplex pair of angular contact bearings. The bearings are mounted back-to-back so that axial thrust load can be carried in either direction. This duplex bearing pair carries both the unbalanced axial thrust loading as well as radial load.

In between-bearing pumps, the ball radial bearing and the ball thrust bearing combination have individual bearing housings (Figure 14). The radial bearing is normally located at the coupling end of the pump. The ball thrust bearing is located at the outboard pump end. The thrust bearing must be secured axially on the shaft to transmit the axial thrust load to the bearing housing through the bearing. The bearing is usually located against a shoulder on the shaft and locked in place by a bearing nut. This means that the shaft diameter under the thrust bearing is less than the shaft diameter under the radial bearing. Thus, by mounting the radial bearing on the inboard (or coupling) end of the pump shaft, a larger shaft diameter is available to transmit pump torque from the coupling to the impeller. The thrust bearing, on the other hand, is locked axially in the thrust bearing housing, the radial bearing is axially loose in its housing to allow for axial thermal growth.

A popular combination for between-bearing double suction pumps consists of journal type radial bearings and a ball thrust bearing. In such an arrangement, all radial pump loads are handled by the journal radial bearing. The ball thrust bearing is mounted in the thrust bearing housing such that only axial loads are carried by the thrust bearing. The housing around the ball thrust bearing is radially loose. A metallic strap is employed on the outer rings of the thrust bearing. This strap locks into the bearing housing to prevent rotation of the outer rings. Such an bearing arrangement is useful in higher horsepower and higher speed applications where ball radial bearings would be impractical due to speed, load and lubrication limitations. Because the ball thrust bearing is located on the outboard end of the shaft, the shaft diameter under the ball thrust bearing can be relatively small since no torque is transmitted from this end of the shaft.

Vertical Pumps

Most vertical pumps differ from horizontal pumps in that the entire axial thrust, consisting of axial hydraulic forces as well as the static weight of the pump and the driver rotor, is supported by the driver thrust bearing. Therefore, the sizing of that bearing becomes a joint effort between the pump manufacturer, the driver manufacturer and the end user.

Many end users require that this bearing be rated to handle at least twice the maximum thrust load, up or down, developed by the pump in a worn condition with two times the internal clearances it had when new. This requirement came into effect after users experienced problems in the field that result from the following facts or principles.

1. The calculation of pump thrust is not highly accurate.
2. Pump thrust increases as internal clearances increase.
3. The thrust load varies with the vertical position of the impellers with the casing(s).
4. The thrust load varies with flow. (In some cases it may even reverse direction.)

A reasonable margin should be provided between the driver thrust bearing rating and the maximum calculated pump thrust.

Motors for vertical pumps are available as solid shaft or hollow shaft units. On hollow shaft units, the pump shaft extends upward through the motor shaft and is supported at the top of the motor shaft. Clearance is provided between the outside diameter of the pump shaft and the bore of the motor shaft. On solid shaft units, a solid coupling is furnished by the pump manufacturer to provide rigid attachment between the pump shaft and the motor shaft extension. Thus, the pump shaft is retained radially by the lower motor bearing. This is generally considered to be a better arrangement for most vertical pumps since the shaft runout will be less and thus the seal or packing life will be longer. Further, larger diameter pump shafts can be coupled to solid shaft motors than can pass through the bore of hollow shaft motors, and this also provides increased shaft rigidity.

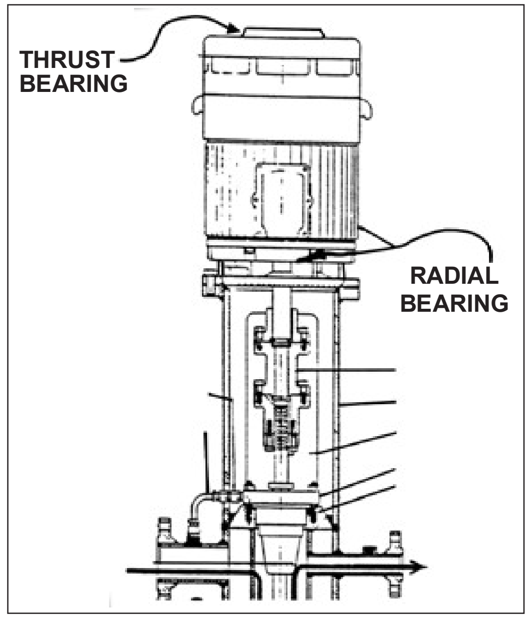

Figure 15. Vertical pump motor – solid shaft

Because of potential field problems, the thrust bearing should be mounted in the driver top bearing housing, farthest from the solid coupling and the pump (Figure 15) . In the event that a thrust bearing fails, any subsequent drop in the driver/pump shafts could result in a mechanical seal failure that could release hydrocarbons to the atmosphere. Also, these could be ignited by a hot bearing.

Ball Bearing Fits

Unfortunately, many pump manufacturers do not indicate the proper bearing fits for shaft and housings to guide shop repairs. The original dimensions of both the housing and the shaft will change with time due to oxidation, fretting, damage from locked bearings and other causes. Every bearing handbook has tables to aid you in selecting fits. The vibration effect of looseness on the bearing fits is different for the housing and the shaft.

Housing Fits – Ball bearing fits in the bearing housing are, of necessity, slightly loose for assembly. If this looseness becomes excessive, vibration at rotational speed and multiple frequencies will result. Do not install bearings with O.D.'s outside of the given tolerance band since this might result in either excessive or inadequate outer race looseness. Table 2 shows rules of thumb.

1.	Bearing OD to housing clearance - About 0.00075 inch loose with 0.0015 inch maximum.
2.	Bearing housing out of round tolerance is 0.001 inch maximum.
3.	Bearing housing shoulder tolerance for a thrust bearing is 0 to 0.0005 inch per inch of diameter off square up to a maximum of 0.002 inch.

Table 2. Rules of thumb for housing fits

Shaft Fits – A loose fit of the shaft to the bearing bore will give the effect of an eccentric shaft, at a one times running frequency vibration pattern. The objective of the shaft fit is to obtain a slight interference of the anti-friction bearing inner ring when mounted on the shaft. The bearing bore should be measured to verify inner race bore dimensions. Do not install bearings with an I.D. outside of the given tolerance band since this might result in either excessive or inadequate shaft tightness. Table 3 gives rules of thumb.

1.	Fit of bearing inner race bore to shaft is 0.0005 inch tight for small sizes: 0.00075 inch tight for large sizes.
2.	Shaft shoulder tolerance for a thrust bearing is 0 to 0.0005 inch per inch of diameter off square up to a maximum of 0.001 inch.

Table 3. Rules of thumb for shaft fits

Detection of Anti-Friction Bearing Defects

Anti-friction bearing defects are difficult to detect in the early stages of a failure because the resulting vibration is very low and the frequency is very high. If monitoring is performed with simple instrumentation, these low levels will not be detected, and unexpected failures will occur. The vibration frequencies transmit well to the bearing housing because the bearings are stiff. Detection of defects is best done using accelerometers or shock pulse meters.

There are guidelines for evaluation of bearing deterioration. For example, a ball passes over defects on the inner race more often than those on the outer because the linear distance around the diameter is shorter. There are four dimensions of a ball bearing that can be used to establish some feel for its condition:

1. A defect on outer race (ball pass frequency outer) occurs at about 40% of the number of balls times running speed.
2. A defect on inner race (ball pass frequency inner) causes a frequency of about 60% of the number of balls multiplied by running speed.
3. Ball defects (ball spin frequency) are variable with lubrication, temperature and other factors.
4. Fundamental train frequency (retainer defect) occurs at lower than running speed values.

A simple check for verification of poor bearing condition is made by shutting off the pump and observing that the high bearing frequency remains as the pump speed reduces. This high frequency signal will normally remain until the pump stops. The frequency indication is normally from 5 to 50 times the running speed of the machine.

In next month's conclusion to this article we will examine various lubrication methods for anti-friction bearings, as well as the role of bearing protection devices, labyrinths and magnetic seals.

PART II

This second half of our series examines ways to keep your anti-friction bearings in top operating condition. Grease, oil flood, ring-oil and mist lubrication systems are all detailed, as well as some advantages and disadvantages of three bearing housing protection devices.

Anti-friction pump bearings can be either grease or oil lubricated. Failure from lack of effective lubrication, either in type or quantity, constitutes a major source of bearing difficulties. The primary purpose of oil, or the oil constituent of grease, is to establish an elastohydrodynamic film between the bearing's moving parts as shown in Figure 1. This film results from a wedging action of the oil between the roller elements and raceways. The formation of the film is, to a major degree, a function of the bearing operating speed and, to a lesser degree, the magnitude of the applied load. Lubrication for ball or roller type bearings can be developed in three ways:

1. full elastohydrodynamic oil film — no metal-to-metal contact
2. no separating oil film — metal-to-metal contact all the time
3. mixture of the above methods (boundary lubrication) — occasional metal-to-metal contact with an oil film present only part of the time

While the surfaces of bearings are highly finished, there are, nevertheless, small surface imperfections. Use of correct viscosity lubricant ensures development of a full oil film between rotating parts. In boundary lubrication, metal-to-metal contact occurs, and friction wear develops. If the bearings are operated with the correct viscosity lubricant for the speeds and loads involved, a full elastohydrodynamic film will develop between the rotating parts (Figure 1).

Under these conditions the oil film is thick enough to separate the bearings completely, despite the unevenness of their surfaces.

Since there is no metal-to-metal contact with full film lubrication, there is no wear on the bearing parts. The only time metal-to-metal contact occurs is on startup, or when the bearing is brought to rest. A lubricant with a viscosity too low for the operating loads and speeds permits the moving parts to penetrate the oil film, which results in their making direct contact. In boundary lubrication, this metal-to-metal friction causes wear of the surfaces to increase rapidly, as the film is frequently ruptured. Viscosity requirements for both ball and roller type bearings are expressed in terms of DN value, a factor used to compare the speed effects of different sized bearings. The DN value is obtained by multiplying the bearing bore size in millimeters by the actual rotating speeds in revolutions per minute. Once determined, the DN value is checked against standard tables to determine which viscosity oil to use.

It is important to remember, however, that the advantages of proper viscosity can be offset by high speeds if too much lubricant is placed in the bearing and housing cavity. At high speeds, excessive amounts of lubricant will churn the oil and increase the friction and operating temperatures of the bearing.

Lubricants also help protect highly finished bearing surfaces from corrosion and, in the case of grease, aid in the expulsion of foreign contaminants from the bearing chamber through periodic regreasing.

Increased operating temperature reduces oil viscosity and film thickness and accelerates deterioration of the lubricant. Petroleum based lubricants operated beyond the 200°F range will experience a 50% reduction in life for every 18 degrees above this level. At high temperatures, the more volatile components of oil and grease begin to evaporate and can carbonize or harden within the bearing cavity.

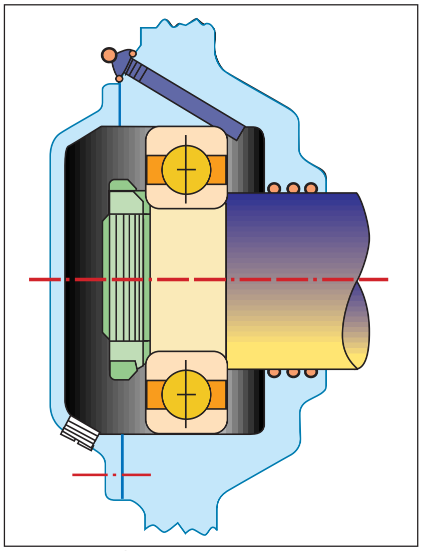

Figure 16. Grease lubricated pump bearings

Grease

Grease lubrication is normally limited to small, low horsepower, noncritical pumps which operate at relatively low speeds and temperatures (Figure 16). The grease can be located in the bearing housing surrounding the bearing or packed in the bearing and then sealed.

Oil Flooded

A more common lubrication system for centrifugal pumps is the oil flood (Figure 17). In such an arrangement, the bearing housing provides a sump, or oil reservoir. This sump maintains a level of oil at or near the centerline of the lowest ball of the bearings, kept constant by means of a constant level oiler. There are two problems associated with this type of lubrication. First, if the level is too high, frothing and foaming may occur, generating heat within the reservoir. Second, there is a very small range between the proper level and the bottom of the balls, below which the bearings are dry.

Ring-oiled

Ring-oiled systems are often used to lubricate anti-friction bearings in larger horizontal pumps where, because of speed or loads, a simple flood system is not adequate. Rings of a diameter larger than the shaft ride on top of the shaft and dip into the oil reservoir below (Figure 18). These rings are located axially between, but adjacent to, the bearings. The rotation of the shaft causes the rings to rotate and carry oil from the reservoir up to the shaft. The oil is then thrown from the shaft by oil flingers, located next to each oil ring, directly into each bearing to assure complete lubrication. As the oil is circulated through the bearings, it is returned to the oil sump. For proper system function, the oil level in the reservoir must be maintained so that at least 1/4 to 3/8 inch of the ring bore is immersed.

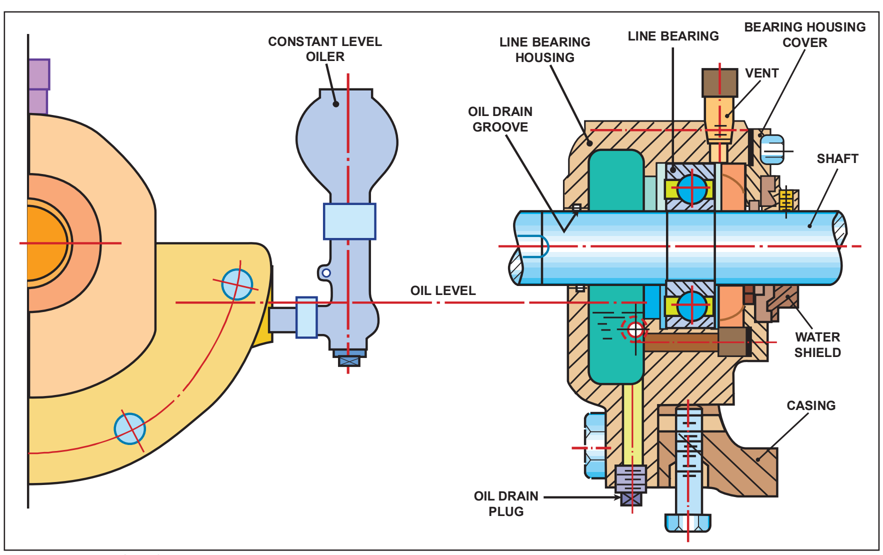

Figure 17. Oil flood lubrication

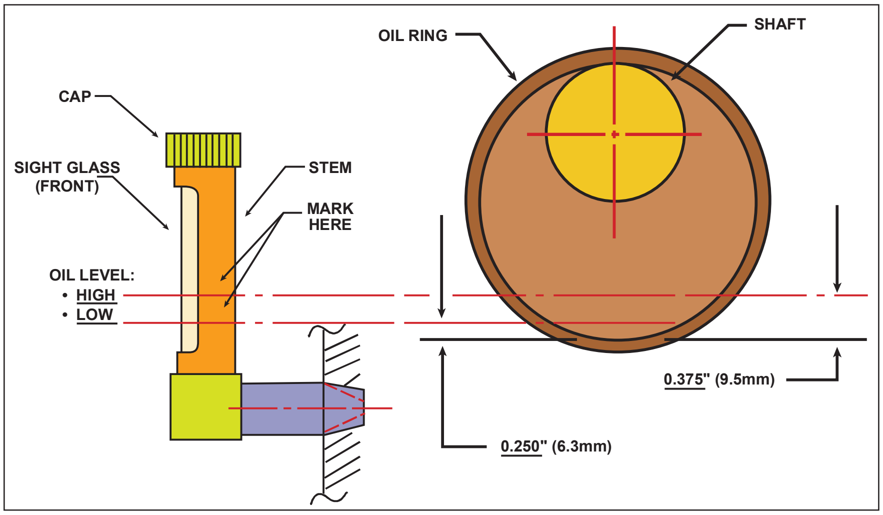

Figure 18. Ring oiling

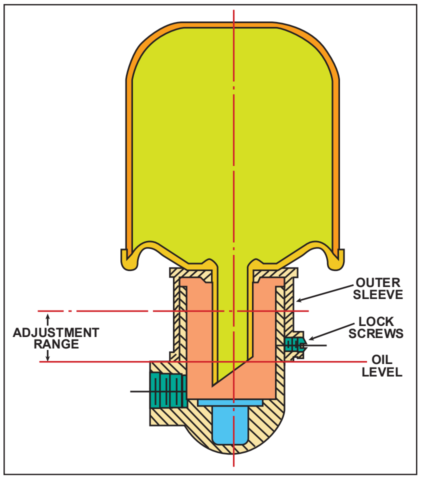

Figure 19. Constant level oiler

The constant level oiler (Figure 19) is a device used with both flooded and ring-oiled lubrication systems and acts as a small reservoir for extra oil while maintaining a predetermined level in the bearing housing. The constant level oiler uses a liquid seal to keep the oil in the bearing reservoir constant. When the oil level in the bearing recedes due to consumption, the liquid seal on the spout is temporarily broken. This lets air from the air intake vents enter the oiler reservoir, which releases oil until the liquid seal and proper level are re-established.

Unfortunately, each oiler installation is slightly different, so some thought must go into setting this position. The proper level is usually indicated (by either a name plate, casting mark or stamp) on the side of the reservoir.

Oil Mist Lubrication

The basic concept of the oil mist lubrication system is dispersion of an oil aerosol into the bearing housing. The equipment necessary for such a system is shown in Figure 20. There are two types of inlet fittings or reclassifiers of the oil particles. They differ in the degree of coalescence from essentially none for the pure mist to partial for the purge mist as shown in Figures 21 and 22.

Pure Mist

In pure mist lubrication (Figure 21), an oil/air mist is fed under pressure directly into the bearing housing. There is no reservoir of oil in the housing, and oil rings are not employed; the pressurized mist flows through the bearings. The moving components of the ball bearings produce internal turbulence, causing impingement and collection of oil upon the surfaces of the ball bearing. Vents are located on the backside of each bearing to assure that the mist travels through the bearings, and a drain in the bottom of the bearing housing prevents the buildup of condensed oil as shown in Figure 23.

The advantage of pure mist is that it creates an uncontaminated environment in which the bearings can operate, and protects them from adverse environmental conditions while effectively eliminating heat buildup. The oil mist system will follow the path of least resistance. The back-to-back mounting of the angular contact thrust bearing will have the most windage, so most of the flow of a single inlet fitting will go through the radial bearing, which has less windage and hence less resistance to flow. As a result, heavily loaded bearings may require two spray inlets. The mist must flow from the inlet fitting through the bearing, then to the vent. For duplex angular contact bearings, the flow should be in the same direction as the thrust (Figure 23). The position of the vent and the center spray can be interchanged. Vent area should be about twice the reclassifier bore area. This will create a slight back pressure in the bearing housing, which keeps dirt out. All vents should carry approximately an equal portion of air in multi-vent installations. Different sized reclassifier orifices are needed

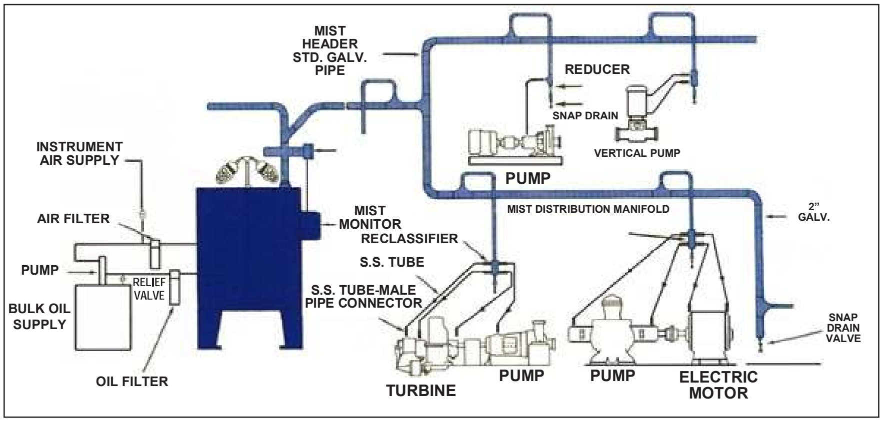

Figure 20. Basic oil mist system

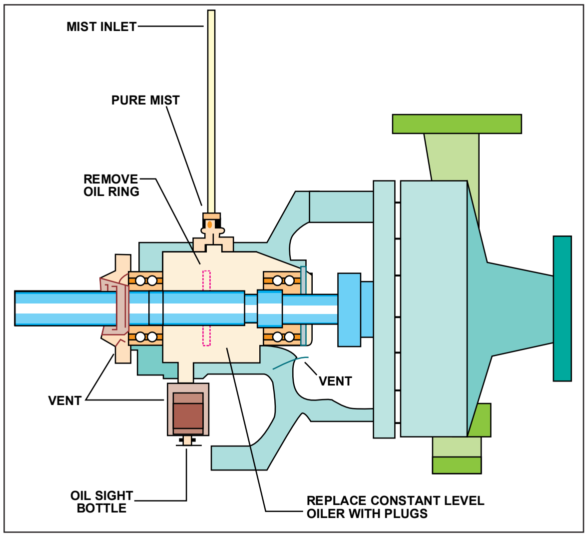

Figure 21. Oil mist lubrication - pure mist

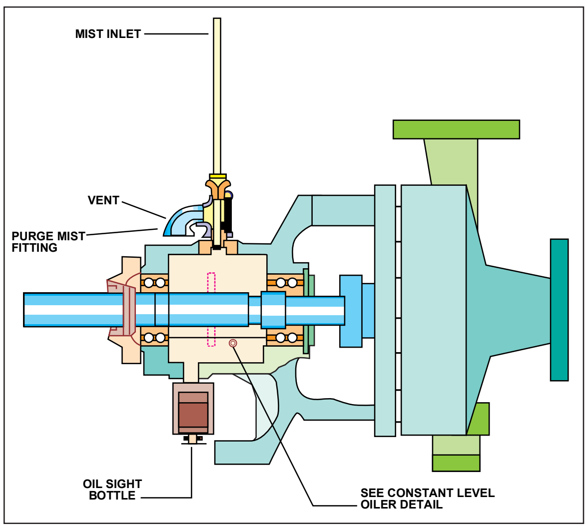

Figure 22. Oil mist lubrication - purge mist

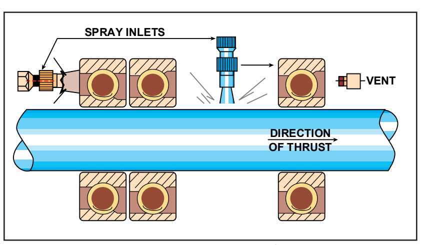

Figure 23. Flow pattern of oil mist

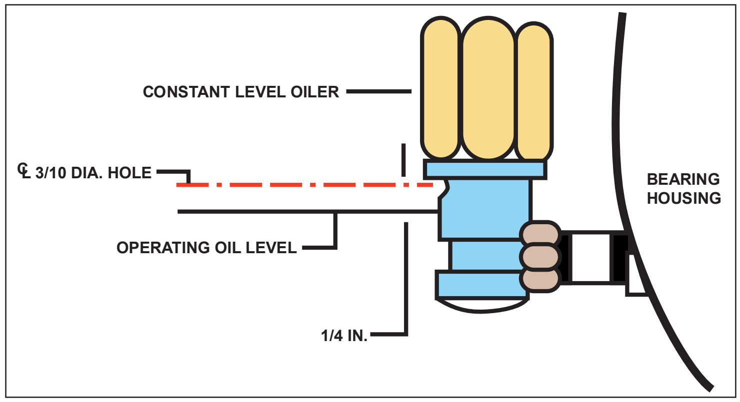

Figure 24. Vented oil sight glass bottle

according to bearing size.

During the mounting process, bearings must be heated to about 250°F to go on the interference fits of the shaft. Most of the oil on the bearing will flow off. To replace this oil, the pump bearing should be either:

1. Reoiled by filling the bearing housing with oil up to the shaft level. The shaft should be turned 3 or 4 revolutions so that the bearing is coated and the oil drained out of the housing.

2. Connected to the oil mist system and operated 8 to 12 hours to "plate out" an oil film on the bearing.

Purge Mist

Another version of oil mist lubrication is called purge mist (Figure 22). This system is employed in conjunction with a conventional oil flood or oil ring lubrication system. It combines the advantages of the positive oil circulation created by the oil rings or oil flood system with the pressurized uncontaminated oil mist system. When this combination is employed, a constant level oiler with overflow feature is used to prevent buildup and flooding of the bearings, which can result in excessive heat buildup (Figure 24).

Bearing Housing Protection Devices

There is a close relationship between the life of rolling element bearings and mechanical seals in pumps. Liquid leakage from a mechanical seal may cause the bearings to fail, while a rolling element bearing in poor condition can reduce seal life. Only about 10% of rolling element bearings achieve their three to five year design life. Rain, product leakage, debris and wash-down water entering the bearing housing contaminate the bearing lubricant and have a catastrophic effect on bearing life. A contamination level of only 0.002% water in the lubricating oil can reduce bearing life by as much as 48%. A level of 0.10% water will reduce bearing life by as much as 90%. To improve the conditions inside a bearing housing, various types of end seals are used. In almost every case, the normal operating life and quality of the end seal is not nearly as good as that of the rolling element bearings. Improving the quality of the end seals will increase the life of rolling element bearings.

Lip seals

The advantages of the frequently used lip seals are low initial cost, availability, and an easily understood technology. New lip seals provide protection in both static and dynamic modes. Their major disadvantage is short protection life due to wear of the elastomer. Life expectancy of a common single lip seal can be as low as 3,000 hours, or three to four months. Thus, while a bearing is designed to last from three to five years of continuous operation, the lip seal will provide protection for only a few months. The temperature limits of lip seals are - 40°F to 400°F (-42°C to 203°C) for Viton.

Labyrinths

Labyrinths are devices that con-

SUMMARY OF BEARING HOUSING PROTECTION DEVICES			
	Lip Seal	Labyrinth	Magnetic Seal
Wet Sump	Possible Short Life	Acceptable	Acceptable
Purge Oil Mist	Possible Short Life	Acceptable	Preferred
Pure Oil Mist	Possible Short Life	Acceptable	Acceptable
Grease	Acceptable	Acceptable	Not Recommended
Vertical Shaft Positions	Acceptable	Top Position Only	Top Position Only
High Humidity or Steam with Temperature Cycling	Possible Short Life	Acceptable	Preferred
Direct Water Impingement	Possible Short Life	Special Design	Preferred
Dirt and Dust Atmosphere	Very Short Life	Acceptable	Acceptable

Table 4. Summary of bearing housing protection devices

tain a tortuous path, making it difficult for contaminants to enter the bearing housing. Unfortunately, there are both well designed and poorly designed labyrinth seals. The advantages of labyrinths are their non-wearing and self-venting features. With no contacting parts to wear out, a labyrinth can be reused for a number of equipment rebuilds. Because the labyrinth provides an open, however difficult, path to the atmosphere, the bearing housing vent can be removed and the tapped hole plugged with a temperature gauge.

The disadvantages of labyrinths include a higher initial cost than lip seals and the existence of an open path to the atmosphere, which can enable contamination of the lubricant by atmospheric condensate as the housing chamber "breathes" during temperature fluctuations in humid environments. Also, they do not work as well in a static mode as in a dynamic, rotating mode.

The temperature limits of labyrinths are determined by the elastomers driving the rotor and holding the stator in place, the same as for the lip seal.

Magnetic seals

Magnetic seals use a two-piece end face mechanical seal with optically flat seal faces held together by magnetic attraction. They have a design life equivalent to mechanical seals and rolling element bearings and can be repaired. The major advantage of magnetic seals is the hermetic seal they provide for the bearing housing. Because of the positive seal, other arrangements must be made to allow for the "breathing" that results from expansion and contraction of the air pocket above the lubricant during normal temperature changes. Disadvantages of magnetic seals include higher initial cost and a shorter life than the almost infinite life of a labyrinth. Magnetic seals are generally not recommended with dry sump, oil mist lubricated bearing housings or grease-lubricated bearings. The upper operating temperature limit of magnetic seals is lower than that of labyrinth seals, in the range of 250°F (121°C).

Table 4 summarizes suggestions for the application of bearing housing end sealing devices with various types of lubrication systems and environments. Bearing life can be extended by improving the environment of the bearing housing, and this can be accomplished simply by improving the end seals.

Conclusions

The effective working life of a pumping system is influenced by many factors that are not necessarily apparent to the facility engineer. Look at all possibilities that can cause a premature failure, not just the bearings. However, anti-friction bearings — their installation and the environment they operate in — are a major factor in pump life expectancy. Well cared for bearings can extend mean time between failures (MTBF).■

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Centrifugal Pumps Operating in Parallel

When it comes to pumps and flow, one plus one doesn't necessarily equal two.

by Uno Wahren, Consultant

Often it happens that a user buys a pump for a given system. Later its capacity proves to be inadequate, and operations people request that capacity be doubled. A common mistake in this situation is to purchase another pump of equal capacity to add to the system while using the same discharge piping configuration. After the new pump has been installed, it becomes apparent to everyone's chagrin that the flow has not doubled.

The problem is that the operating point has shifted because of increased friction losses in the discharge piping system due to higher fluid velocity.

The tools available to analyze a pump system are the head-capacity or pump performance curves and the system curve. Pump suppliers provide head-capacity curves with their pumps as shown in Figure 1. On the y-axis they plot the head, or pressure. The x-axis shows the flow. Pump efficiency curves and often the NPSH required are also shown. The buyer plots the system curve on an x-y axis as in Figure 2. This curve represents the required discharge head and the friction losses in the discharge piping. The x-axis shows the flow in gpm and the y-axis the friction losses in feet. The desired discharge head is a straight, horizontal line, since it remains constant from minimum to maximum flow.

When selecting a pump, the buyer specifies the differential head and the capacity. This is the operating point of the pump. There are many reasons to have pumps operating in parallel. The most common is flexibility. If only one pump is used for a service and that pump breaks down, the pumping system or process shuts down. A spare pump is therefore often mandatory in critical systems. The spare pump can be identical to the main pump. Project specifications often demand 50% sparing; one or two pumps in parallel supply the total flow, with an additional pump as a spare. This gives extra flexibility for maintenance and in case the process flow varies.

Buying pumps for a new system is fairly straightforward. The first step is to determine the desired flow, the required head, the desired pipe diameter and length, and the valves and fittings included in the system. Next, plot a system curve. The y-axis shows head losses, the x-axis flow. Let's say a particular application requires 900 gpm at a constant head of 82 feet. Suction is flooded. This constant head may be the pressure of a pipeline from which the liquid is to be pumped, or the elevation of a reservoir, tank or pressure vessel. The optimum discharge pipe diameter is determined to be 8". The last step is to plot the friction losses against flow.

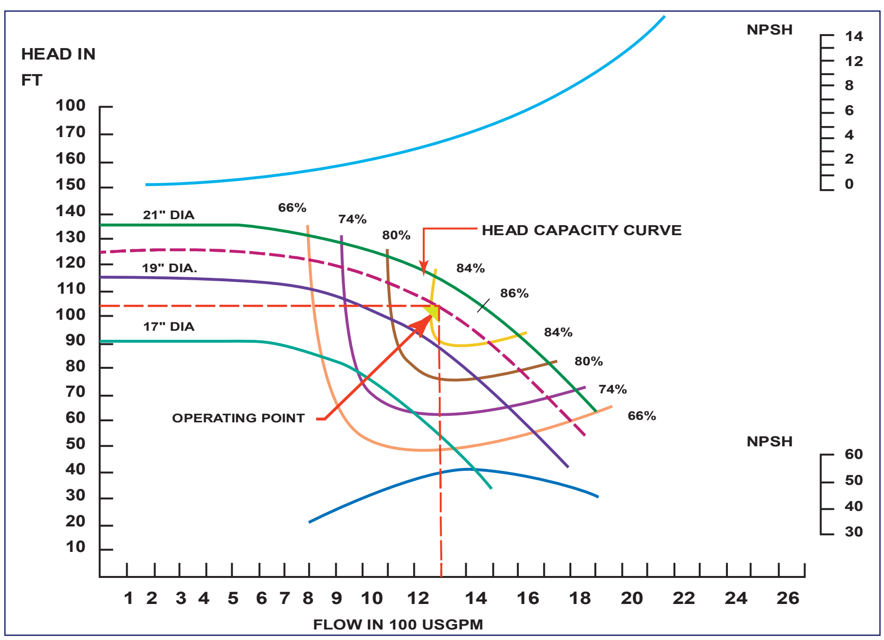

Figure 1. Typical head-capacity curve

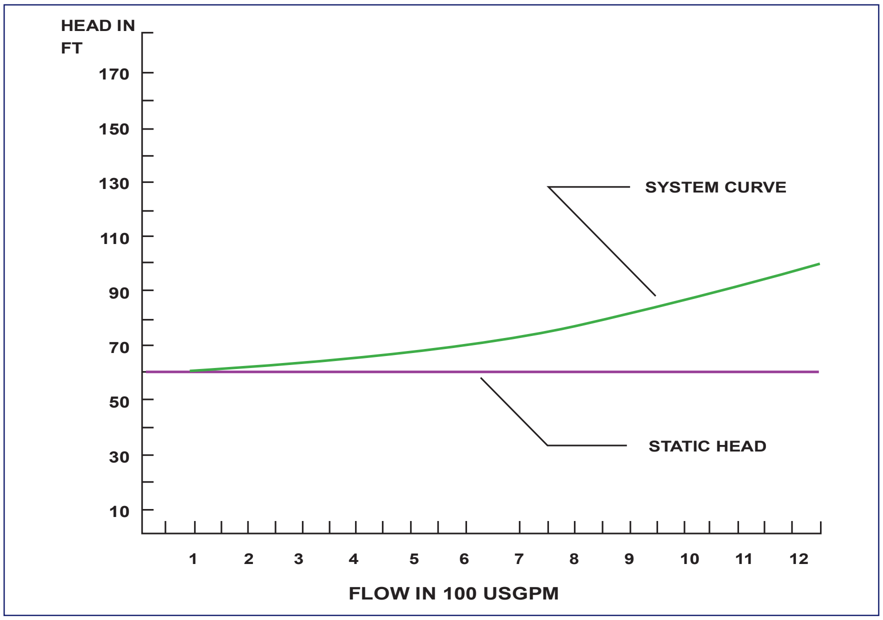

Figure 2. System curve

Figure 3 shows a straight line parallel to the y-axis drawn at 900 gpm and intersecting the system curve at 82 ft. This particular system requires a pump that delivers 900 gpm with a discharge head of 82 ft. Two pumps each delivering 450 gpm, or three pumps each delivering 300 gpm, will achieve the same result. In this case, the decision is to use two pumps running in parallel, each with a capacity of 450 gpm.

Each pump operating alone delivers approximately 520 gpm at only 68 ft. This is adequate because the end of the line or static head is 60 ft. If the calculated system head curve is above actual, it will be necessary to throttle the pump flow, using either the discharge isolation valve or an eventual control valve on the discharge line.

Suppose a system has one pump delivering 450 gpm against 68 ft. The pump discharge line is 6" new steel pipe. The fluid is water with a specific gravity of 1.0. There is a request to double the flow. In a case like this it is very common to request another identical pump — one that will deliver 450 gpm at 68 ft. That is a mistake. With the addition of the second 450 gpm pump, the combined (1 and 2) head-capacity curve will bisect the system curve (Point B) at 790 gpm. The total head required at that flow is 78 ft as shown in Figure 4.

If the new pump is already bought and installed, flow can be increased by changing the discharge piping from 6" to 8". This is usually not an economical solution. The discharge line may be long. Replacing valves and fittings may be expensive. Another solution is to buy a third pump and run the three pumps in parallel. This can also be expensive and adds to maintenance costs. Changing the impellers to a larger diameter that will still fit into the pump casing may also solve the problem.

The characteristics of a centrifugal pump are such that with constant speed (rpm) and a specific impeller diameter, the curve will not change, regardless of the properties of the liquid being pumped. However, the curve will change if either the speed or the impeller's diameter change.

The performance curve shown in Figure 1 will change if the pump's rpm changes from 3560 to 4200 through a speed increasing gear as follows:

$$Q_1 = 450\text{ gpm};\ H_1 = 160\text{ ft}$$
$$Q_2 = 450 \times 4200/3560 = 530\text{ gpm}$$
$$H_2 = 160 \times (4200/3560)^2 = 223\text{ ft}$$

Increasing pump speed is uncommon. Pump characteristics change when speeds are higher than design specifications. As NPSH requirements increase, internal recirculation can become a problem. The shaft and bearings may not be designed for the increased torque and loads of the higher speed and horsepower requirements. A gear train means added maintenance problems. A larger driver may be required. Why get into a situation like that?

Assuming that the installed impellers have an outside diameter of 6", the maximum size impeller for that pump is 6.3" O.D. The affinity law for a constant speed pump is:

$$D_1/D_2 = Q_1/Q_2 = \sqrt{H_1}/\sqrt{H_2}$$

The flow and head for a 6.3 diameter impeller are:

$$Q_2 = (450 \times 6.3)/6 = 472\text{ gpm}$$
$$H_2 = (\sqrt{68} \times 6.3/6)^2 = 75\text{ ft}$$

(Note: Some impeller designs do not precisely agree with the affinity laws for impeller diameter changes. Always discuss this with the pump manufacturer.)

As shown in Figure 5, the two pumps with the larger impellers operating in parallel still do not deliver the full 900 gpm. In this case, the total flow might be enough for the particular process. On the other hand, it may not. Increasing the impeller size will often not solve the problem.

To buy a pump that will double the flow requires plotting the system curve against the head-capacity curves. The plot shows that at the increased flow the system will require a total discharge head of 82 ft. When the first pump (pump 1) runs alone on the system curve, the flow is 450 gpm and the head is 68 ft. Next select a pump (pump 2) which, running in parallel with pump 1, will deliver the required flow at the required total discharge head (TDH). For this example, an adequate pump is one with a flow of 645 gpm with a TDH of 72 ft. The pumps are operating in parallel on the same system curve. The pumps are sized so that pump 1's head-capacity curve intersects the system curve before point A. Therefore, pump 2 is the commanding pump. If pump 1 starts first (before pump 2) it will back off and the system will deliver only about 650 gpm against 72 ft of head.

Figure 6 shows how the two dis-

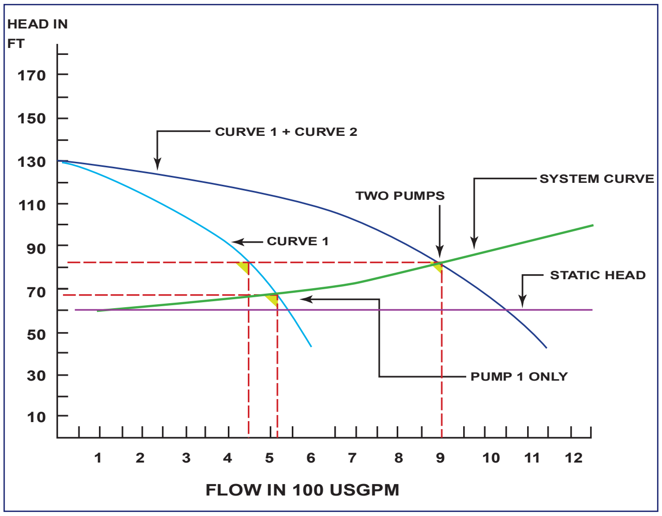

Figure 3. Two pumps in parallel

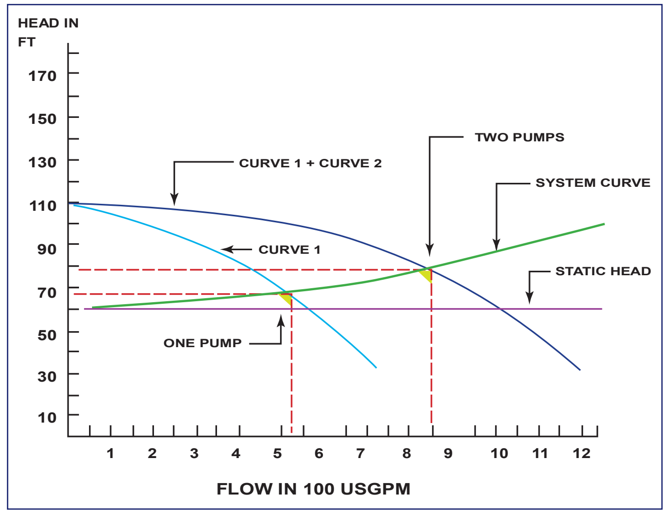

Figure 5. Larger diameter impeller pumps operating in parallel

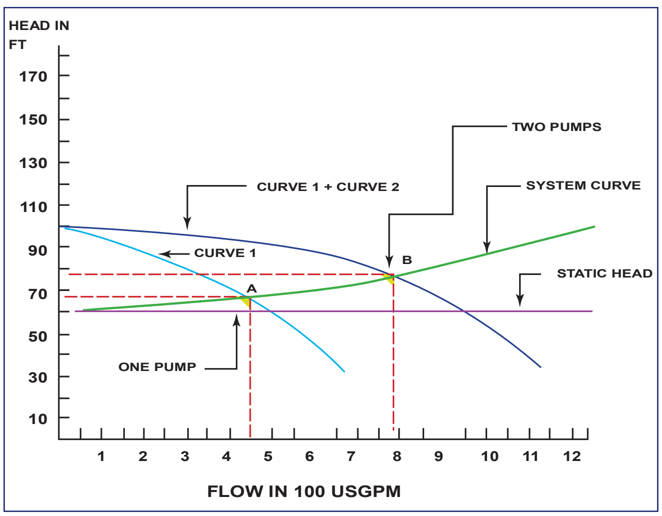

Figure 4. Addition of identical pump

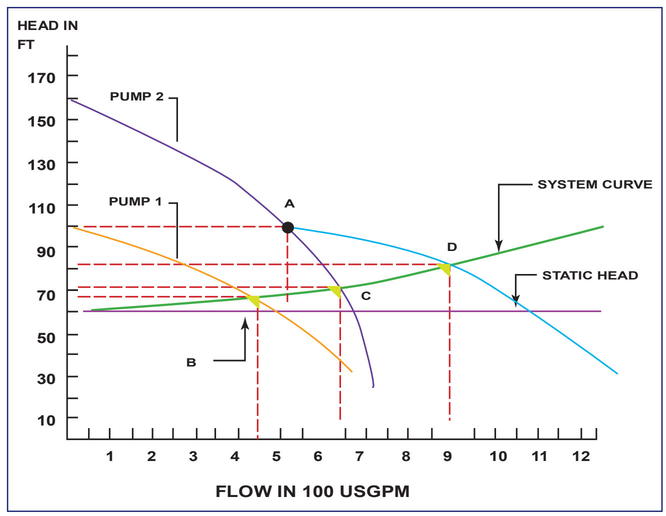

Figure 6. Two dissimilar pumps operating in parallel

similar pumps can operate correctly only if the head-capacity curve intersects the system curve on the AD portion of the head-capacity curve. The pumps may be throttled, but not further back on their curve than 525 gpm (Point A), for then pump 1 will back off, letting pump 2 deliver only its full capacity. For the system to operate, pump 2 has to start first. After pump 2 has reached full flow, pump 1 can be started. (Figure 6).

When selecting an additional pump, bear in mind that no pump will operate satisfactorily from zero flow to the end of curve flow. Pump manufacturers show minimum allowable constant flow on their pump curves. This means that below that flow, the pump is unstable. The minimum flow requirements for low head, low capacity pumps range from 20 to 25% of flow at Best Efficiency Point (BEP). It is not uncommon for large multi-staged pumps to have minimum flow requirements as high as 65% of BEP capacity.

Always pick a pump where the required flow is less than the flow at BEP. It is good practice to specify that a pump will operate satisfactorily at 120% of the rated flow, since flow requirement may increase. The higher the flow, the higher the NPSH requirement. At some point beyond the BEP, the pump curve will collapse. Operations beyond that point will cause cavitation and severe pump damage.■

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Fire Pump Systems-Design and Specification

Think your fire pumps are just like the rest of your fluid-movers? Think again. Rigorous standards and certifications make sure these life-savers are up to snuff.

by George Lingenfelder and Paul Shank, Precision Powered Products

Point of view is everything when discussing fire pump systems. For the engineering-contractor, they are relatively uncommon systems referencing specifications unheard of in conventional process units. To the purchaser, they are mandated systems that take time, space and capital away from money-making units. For users, fire pump systems are (or should be) a once a week test requirement.

But in spite of the time, space and money constraints, fire pump systems must work as required. Period. Hundreds of lives and millions of dollars in hardware and production costs rely on the performance of these systems. Fortunately, they almost always work.

Because of the critical nature of this service, one might think that industry standards could simply be invoked to insure reliable design and specification. Of course they can. They just never are. At least not without the addition of supplemental proprietary specifications that can conflict with industry standards, government regulations and sometimes with the basic system design. Seemingly minor requirements can result in the loss of a listing agency label and render a perfectly functional system unacceptable to insurers or governmental agencies.

Pumps

The basic fire pump system includes a UL (Underwriters Laboratory) or FM (Factory Mutual) listed pump, driver and controller. Pumps are rated in discrete increments starting at 25 gpm and extending through 5000 gpm. Each capacity designation is tested for compliance with NFPA 20 (National Fire Protection Association) requirements for performance and by independent testing institutions such as UL or FM for design, reliability and safety. A single pump can be certified for more than one operating point, but it must meet all performance and design criteria at both points.

In the area of basic hydraulic performance, a pump must deliver at least 65% of rated discharge pressure at 150% of rated flow to achieve NFPA 20 acceptance. So a 1500 gpm pump rated at 150 psig must deliver 2250 gpm at a discharge pressure of not less than 97.5 psig. This operating range then establishes the design parameters for the piping system and fire fighting equipment. Another requirement establishes that the maximum shut-off head must not exceed 140% of design head.

Both horizontal and vertical centrifugal pumps are available as listed pumps. (For brevity, we will use the term "listed" to refer to any equipment certified by UL, FM, the Canadian Standards Association (CSA) or other agency as suitable for fire pump system application.) While there are some obvious design differences between the requirements for these two styles, the performance prerequisites remain un-changed. It is important to note that listed pumps are not designed or manufactured to API 610 standards.

Photo 1. Fire water pump with diesel engine and air start with a nitrogen back-up system

Materials of construction vary with the type of system and the fire fighting fluid. The standard cast iron case with bronze impellers and wear surfaces is the most common in land-based applications. This material is generally suitable for fresh water and sometimes brackish or even salt water services, depending on the length of service and anticipated life of the unit. Even in fresh water services, it is critical to consider casing corrosion rates when choosing materials. For more aggressive fluids, many manufacturers offer various grades of bronze, including zinc-free and nickel-aluminum bronze. Higher alloys of 300 series stainless steel and Alloy 20 may also be available for very corrosive services or situations in which the expected project life is extremely long.

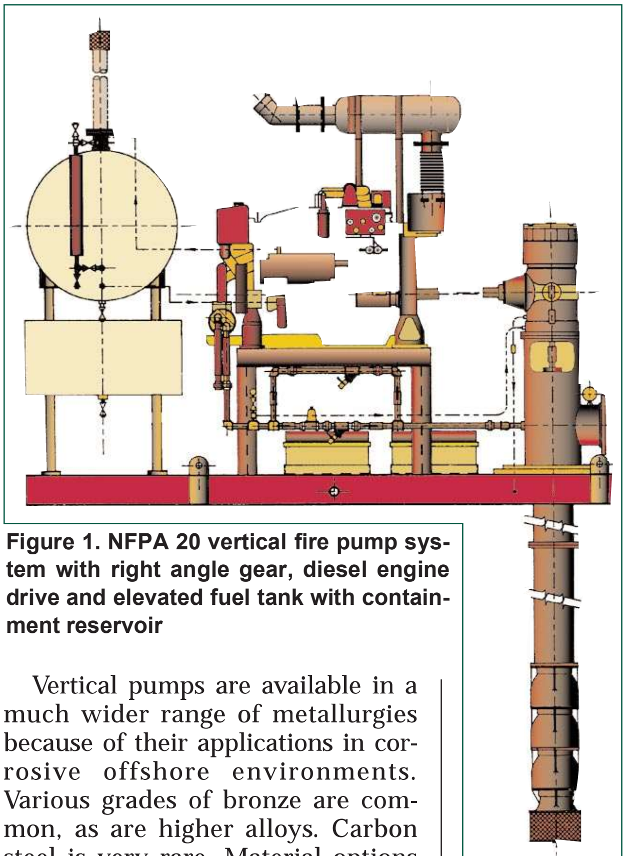

Figure 1. NFPA 20 vertical fire pump system with right angle gear, diesel engine drive and elevated fuel tank with containment reservoir

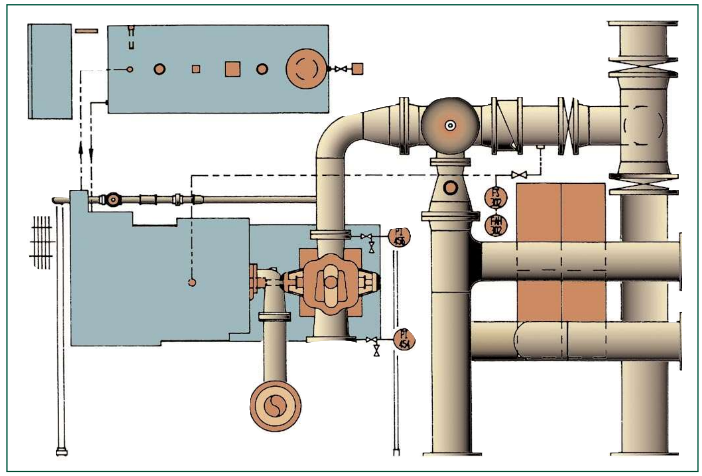

Figure 2. Plan view of horizontal fire water pump system with diesel engine drive, discharge piping main relief valve and discharge cone

Vertical pumps are available in a much wider range of metallurgies because of their applications in corrosive offshore environments. Various grades of bronze are common, as are higher alloys. Carbon steel is very rare. Material options for horizontal units include bronze casing materials, moving up to carbon steel and then austenitic steels for both case and internals. As with vertical pumps, higher alloys are also available for special situations.

From a commercial standpoint, cast iron-bronze fitted (CIBF) pumps are the basic choice. Standard aluminum bronze materials such as ASTM B148 Alloy 952 (for vertical units) generally command price premiums in the 4.5-5 multiplier range. Nickel-aluminum bronze (B148 Alloy 955) will increase this pricing by an additional 10-15%. Carbon steel (for horizontal units) will raise standard CIBF costs by 3-4 times. Stainless steel, typically 316SS, increases base costs by a factor of 5-8 depending on size. Higher alloys can increase CIBF costs by up to 8 times.

Auxiliary equipment and accessories offer many possible options for the pump. While these will be discussed in greater detail later, it is important to note that most fire water pump manufacturers only offer a limited number of materials and options for listed pumps. Pumps (like drivers and other equipment) can be listed for fire pump service only when they are manufactured in the same materials and configuration as the design tested by the certifying agency. Changes in casing materials, even when poured by the same foundry using the same patterns, can prevent a manufacturer from labeling the pump. Likewise, changing accessories, especially on engines, can result in the loss of the label.

In practice, this is usually a semantic issue and rarely becomes a significant problem, provided that the original design or unit was labeled and that the changes are clearly upgrades in materials, control and/or reliability. The importance of labeling also varies with the location and type of fire pump service, as well as responsible government agencies and insurance carriers. Most manufacturers offer labeled equipment in the most elemental fashion for application in the widest range of markets. This makes it possible for the specifying engineer and user to work together with the supplier to develop the best system for their application. Even though the end user might customize the system and add accessories that void the label, the original equipment was labeled, and this shows the intent of the user to install labeled equipment.

Drivers

While the pump is the mechanical heart of a fire water system, the driver is a critical component, frequently requiring more attention in specification and design. Diesel engines power the vast majority of fire water pumps, creating another difficulty for the specifying engineer. Most of us have limited experience with this category of equipment. They are rare in general plant design and frequently used only because the power required exceeds the available UPS (Uninterruptible Power Source). Unlike the situation with motors, the specification and selection of diesel engines cannot be foisted off on another discipline.

The basic, listed fire pump engine, offered by many manufacturers, is more than adequate in terms of reliability and service life. After all, this system will only operate once a week for thirty minutes until it is called on to run in a real emergency. Then it will operate for eight hours (the standard fuel tank sizing) or until it is destroyed in the conflagration.

A great deal of time and energy is expended on the starting system,

Photo 2. Custom fire water pump system designed for Class 1, Groups C & D and Division II. The controller features all NEMA 7 switches mounted in a stainless steel cabinet. The battery charger and battery case are purged and constructed of 316SS. The system has a primary electric start with a back-up hydraulic system.

Photo 3. Vertical fire water pump for offshore installation. Compressed air start with back-up nitrogen system and expansion receiver

which often includes back-up start systems and sometimes even multiple starting systems. Basic systems include dual sets of batteries. Also commonly available are pneumatic systems—compressed air with bottled nitrogen back-up and hydraulic starting systems. Many of the larger engines, above 200 hp, offer two ports for starters, so that two types of starting systems can be used. This having been said, most experienced engine users agree that if the unit does not start within the first three to five seconds, the likelihood of starting at all is just about nil. For this reason, it is very important to pay close attention to the starting cycle of the unit during the weekly exercise sessions. Remedy any difficulties or malfunctions at once.

Although relatively uncommon, some systems also use motors, especially when the firewater pumps are also used for water lift or wash-down services. Whether they are more reliable remains an open discussion. The simple fact of their predominance as drivers has made them more acceptable. Like other major components of fire pump systems, motors are available in UL/FM and CSA listed varieties from a wide range of manufacturers. While still not widely accepted, the most recent revision of NFPA 20 (January 1998) now requires fire pump motors to be UL listed.

Controllers

Controllers were the last major component of fire pump systems to receive certification by various listing agencies. Unlike other components, however, they come with a wide range of options and even provide remote system contacts for the other pump, driver and other options that may need to be incorporated into the control scheme.

A few areas in the design and specification of the controller should be reviewed carefully. The first is area classification. Unusual as it may seem to designers and specifiers who are unaccustomed to it, standard drivers and controllers are not rated for any National Electrical Code (NEC) area classification. Depending on the manufacturer, they can add this as an option or build a custom controller that meets a specific area's classification. Meeting a Division I or II classification usually involves adding a Z-purge system, which can be expensive relative to the cost of the controller (typically increasing cost by up to 50%), but this is currently the most cost-effective method to meet these requirements. Custom controllers, a very expensive alternative, will generally incorporate hermetically sealed or NEMA 7 enclosures for all potentially arcing devices. In addition, diesel engine control sensors and battery start systems are not intended for a classified area. Electric motors must also be specified for the area.

Pneumatic engine controllers with air start systems are an obvious alternative when area classification becomes an issue. Although not commercially available as UL/FM listed units, it is generally agreed that they are inherently explosion-proof because they have no electrical components. Custom built electric controllers, however, can meet the area classification requirements of any facility. Controllers incorporating PLC logic are also available as standard commercially available, but not as UL/FM listed units, or pneumatic controls as custom built units.

Custom manufactured controllers are becoming more common, especially in the HPI/CPI markets to meet NEC area classifications. It is important to note, however, that while these units may consist entirely of UL (or other) listed components, this does not convey a UL listing to the controller. Thus, the controller does not meet the UL218 requirements. This may appear to be a subtle differentiation, but local codes and regulations, as well as insurance requirements, may require sacrificing an area classification for a UL/FM listing.

Instrumentation

Fire pump systems are not process systems. Thus, much of the advanced instrumentation applied to process systems is relatively uncommon in the world of fire pumping. However, some systems include transmitters and other "smart" devices that provide additional information to the user. Carefully review the purpose and function of these devices to determine their value under the anticipated operating conditions and to insure that they provide valuable information without contributing to needless complication. Unlike a process system, where almost every eventuality can be understood, evaluated and prepared for, the full array of conditions under which a fire pump system will be used is almost impossible to imagine. Controls should be as simple and straightforward as possible, lest they fall victim to the law of unintended consequences.

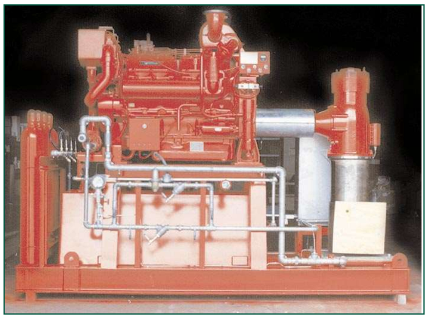

Photo 4. A 570 hp V-12 diesel engine driver for all nickel aluminum bronze fire water pump producing 3500 gpm at 180 psi.

Industry Standards

Two major organizations are the primary promulgators of standards for fire pump systems, and their roles, far from being conflicting, are complementary.

National Fire Protection Association

The National Fire Protection Association, through its NFPA 20 Standard for the Installation of Centrifugal Fire Pumps, provides the most complete set of provisions for the design and installation of various components as well as the overall system. The NFPA 20 sets standards for design and construction of all the major components in the fire pump system, including minimum pipe sizing tables, electric motor characteristics, performance testing, and periodic testing and system design. NFPA 20 further attempts to develop a safety standard or level of performance for centrifugal fire pump systems to provide a reasonable degree of protection for life and property. Under these provisions, alternate arrangements and new technologies are permitted—and in fact encouraged.

The National Fire Protection Association does not, however, certify or evaluate compliance with its various specifications and guidelines.

NFPA 20, while the source of some performance requirements, is best known for its detailed design provisions for all components commonly found in fire pump systems.

Unlike many industry standards that address only the design, manufacture and testing of specific components, NFPA 20 is a veritable 'how-to' manual for the engineer or user who needs to develop guidelines and specifications for fire water systems. Taken as a complete document with referenced texts, this standard alone will assure the purchase and installation of a reliable fire pump system. Take care in developing additional specifications, whether stand-alone or ancillary, to avoid conflicts with NFPA 20. Because of the detailed nature of this specification, add-on requirements can frequently have the effect of actually diminishing the standards.

Here again it is important to keep in mind the function of the equipment in a fire pump system. Contrary to the design considerations in process units, fire pump systems should be designed to operate reliably under the most adverse conditions for as long as required, but typically this is only a matter of hours. In a process unit, systems are designed to protect themselves with various shutdowns and monitors. Fire pump systems are designed to protect the plant and personnel even if that means operating to destruction. In other words, three hours into a fire fighting event, the vibration level of the diesel engine driver is of no significance so long as the pump continues to deliver adequate flow.

NFPA 20 includes pump design guidelines for vertical shaft turbine pumps, horizontal (both end-suction and split-case) and vertical in-line pumps. It also includes standards for motors, both horizontal and vertical, right angle gears and diesel engine drives.

Additionally, the guidelines are an excellent resource for auxiliary and ancillary equipment found in most fire water pump systems. For example, the diesel engine specifications include not only requirements for the engine itself, but also the fuel supply, exhaust system and control system operation.

Underwriters Laboratories

Underwriters Laboratories (UL) reviews equipment and systems from a performance orientation. The organization provides an independent, third party evaluation of manufacturers for a variety of components, but for fire pump systems the primary specifications for our consideration are UL448 for pumps, UL1247 for diesel engines and UL218 for controllers.

In certifying equipment to these standards, UL reviews construction design, materials of construction and overall performance. These reviews assess the ability of the equipment to perform the task for which it is to be rated. Following initial certification, UL maintains an ongoing surveillance program to insure continued adherence to design and performance criteria. UL field representatives make unannounced visits to all manufacturers displaying the UL label. If team members encounter problems or questions, they may schedule more frequent visits.

UL standards are among the most stringent in the industry, and equipment certification is necessarily very specific. So what happens when a listed product is modified to meet customer specifications or site specific requirements? If the change is clearly an upgrade to the existing listed design and not a major change in the product, UL will frequently certify the modification without re-testing. Major modifications, of course, must go through the complete certification process. The manufacturer is responsible for initiating changes to the listing.

UL448 sets the standards for certification of centrifugal pumps for use in fire water systems. As previously noted, UL448 is based on the ability of the unit to perform under the conditions of service required and according to the manufacturer's specifications with regard to total differential head (TDH), capacity, and efficiency or power requirement. Materials of construction are reviewed for strength and corrosion resistance. Each unit or size to be certified is given an operational performance test and is also hydrotested to twice the manufacturer's published Maximum Allowable Working Pressure. This is a substantial increase over pump industry standards of 150% hydrotest pressures.

After certification, UL448 requires that each unit bearing the UL mark be tested successfully for hydrostatic integrity and hydraulic performance.

Diesel engines fall under the scrutiny of UL1247. Again, the emphasis is on performance. Certification requirements include extended testing on dynamometers as well as speed control. Because of the importance of driver speed in centrifugal pump applications, speed control and overspeed shutdown operation are critical areas. Units are also tested for their ability to start under a wide range of conditions, both hot and cold.

After certification, each production unit shipped must be subjected to a dynamometer test including performance checks of the speed control and overspeed shutdown systems.

Engine and motor controller specifications are covered by UL218, which is written and administered by the Industrial Controls section of Underwriters Laboratories. While this section is closely aligned with UL's primary purpose to review equipment for fire and shock hazards, the critical nature of fire pump controllers makes performance an important concern of this standard. These units are reviewed for safety, of course, but the importance of starting a fire water pump under emergency conditions warrants a different philosophy in certifying the controllers.

Controllers are inspected with special attention to their ability to signal—that is, to notify remote personnel of any abnormal conditions in the controller system—as well as to be able to accept remote instructions for starting. Diesel engine controllers, besides providing a starting signal to the engine, must also provide charging current for the batteries, perform a weekly starting and run test of the engine and driven equipment train, as well as provide visual and audible indication of various engine failures. These include failure of the engine to start, shutdown from overspeed, battery failure, battery charge failure and other abnormal conditions. Diesel engine controllers also include pressure recorders to sense pressure in the fire protection system and confirm unit performance on demand or during weekly tests.

Electric motor control requirements also focus on the ability of the unit to start the motor drive. This overriding concern is demonstrated in unit design, for fire system controllers are different from standard motor controllers. This is perhaps best demonstrated by their use of either a listed fire pump circuit breaker or a non-thermal instantaneous trip circuit breaker with a separate motor overcurrent protective device for protection against overcurrents and short circuits. Another difference between these and standard controllers is that once a fire pump controller is under emergency conditions, it is prevented from shutdown except when it is under a condition more threatening than the fire or until conditions return to normal standby.

Surprisingly, UL lists no fire pump controllers for installation in hazardous (classified) locations. To meet Division I or II requirements, controllers must be custom designed with a suitable hazardous location protection method. They must either have a purge system or be explosion proof. Custom designs, of course, will incorporate either NEMA 7 components or enclose elements in explosion-proof boxes. The X and Z-purge systems are the most common.

For international installations, the problems are compounded because no harmonized IEC (International Electrotechnical Commission) standards exist that address fire pump controllers This situation, however, is being addressed by a joint working committee of the NFPA and UL that is developing an IEC standard that will incorporate NFPA 20 and UL218 requirements. At present, the committee has developed a draft specification and submitted it to the IEC for inclusion as a new work item proposal (NWIP). Following acceptance as a NWIP, the IEC will convene a group to review and comment on the standard for development as an IEC publication. Given the importance of international markets, this move will be an appreciated step for U.S. based manufacturers.

Conclusion

Fire pump systems are designed to protect life and property. In this area, specifiers, owners and operators need to change their perspective on equipment, controls and operation. Performance and reliability must be the priority in design, purchase and maintenance. While site specifications and purchaser requirements certainly must be considered, simpler is usually better. The ability of the system to perform reliably under the most severe conditions, for the protection of life and property, must be the driving consideration. n

Special thanks to Kerry Bell and Dave Styrcula at Underwriters Laboratories for their assistance.

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Well Pump Applications for Mine Dewatering

Choosing the right pumps means knowing drainage requirements, dewatering schedules and well construction as well as system and fluid conditions.

by Mark List, Miller Sales and Engineering

Once the economics of a mineral deposit have been determined as favorable and a decision made to proceed with mine development, significant financial commitment is placed at risk in expectation of a calculated return on investment. Many aspects of mining carry relatively high levels of uncertainty that contribute to the overall degree of risk. One important consideration is groundwater control, should it be a factor, during mining operations. Where mining must take place below the water table in highly permeable geologic material, operations would not be possible without effective control of groundwater. Several mines in the western United States have developed large well fields, up to 70,000 gpm capacity, to intercept inflow and divert groundwater from the workings.

Mine Dewatering Objectives

There are two general objectives to most mine dewatering programs: Keeping the working conditions relatively dry and maintaining the stability of the excavation or opening.

Operating costs and production rates are directly influenced by working conditions. Wet floors and working faces create poor ground conditions for heavy loading and hauling equipment. They also increase tire wear, reduce cycle times and impact tonnage factors. Safety becomes a direct issue if electric powered machinery is used. In the worst case, a submerged working level becomes inaccessible and production is halted.

The stability of open pit walls and underground openings is of vital concern for safety and economic reasons. Adequate drainage must occur in order to keep pore pressure at acceptable levels based on geotechnical stability analysis.

General Types of Mines and Groundwater Control Methods

Mines are either open pit excavations or underground excavations, or both. In some cases large scale open pit mines have succeeded prior episodes of underground mining in the same area. In other situations, underground mines are developed adjacent to or from within existing open pit mines. Driven by metal prices and advancing technology, companies have continued to explore deeper and/or more challenging geologic territory for mineable orebodies. Mine dewatering methods have evolved out of necessity in response to the increasing groundwater control requirements of contemporary mining.

Where conditions permit, open pumping from collection sumps is a standard practice. This method is commonly used in open pit mines to control surface water drainage and in both open pit and underground mines where groundwater inflow rates are small enough to be managed in this manner. Booster stations might be required employing positive displacement pumps or horizontal centrifugal pumps designed for high head dirty water, abrasive solids or slurry service. Depending on the hydrogeologic setting, however, some mines cannot be effectively or safely dewatered using this method alone.

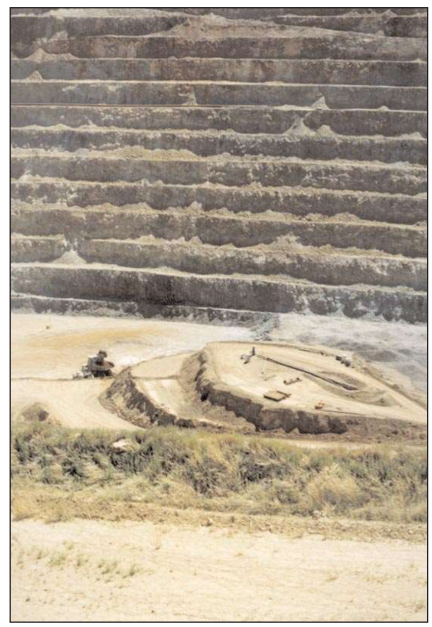

Photo 1. In-pit well in service with loading operation on left, blasted material on right, and high wall in background

Well Field Systems for Dewatering

Several mines in northern

Photo 2. Twin 800 hp, 5000 gpm vertical turbine can boosters with 42" discharge main in background

Nevada require the use of wells to intercept and lower groundwater levels around open pit and underground excavations. The orebodies associated with these mines are hosted in fractured bedrock formations along mountain ranges that contact alluvial basins high in groundwater storage. Mining companies drill large diameter deep wells in bedrock fracture zones tested for favorable production yield. Other wells are completed so as to intercept shallow recharge or promote drainage of less permeable zones.

Wells are typically located outside the open pit, but drilling sites in the pit are often unavoidable due to local hydraulic compartmentalization. A well field can be comprised of 30 or more individual wells with completion depths up to 2000 feet or greater and production casing diameters up to 24 inches. Well-specific capacities can exceed 60 gpm per foot of drawdown. Vertical turbine line shaft pumps (400 hp) are in service at setting depths of 1020 feet, and 1500 hp vertical turbine line shaft pumps are in service at setting depths of 800 feet. Single 2200 hp submersible units work at setting depths of more than 1800 feet, and at more than 2000 feet deep single 1500 hp submersible units are applied. Submersibles are also installed in series using tandem 1000 hp (2000 hp) units, tandem 1170 hp (2340 hp) units and combination 1500 hp/800 hp (2300 hp) units with the lower setting at depths exceeding 2000 feet.

Pit perimeter wells typically discharge to gathering mains at relatively low well head pressures. In-pit well pumps can be staged for the additional head required to discharge to a surface location outside the pit, or to a booster station in the pit. Vertical turbine can pumps and horizontal centrifugal pumps are in service for this application.

Understanding the Application

The uncertainties involved in developing an efficient mine dewatering program become much better understood as operations progress. Groundwater flow information available at the onset of large scale dewatering can be very complete and supported by sophisticated model simulations, but such information is usually based on field test data that cannot be conducted at a scale proportional to what will actually be undertaken. Granted, pump applications engineers are most comfortable when customers assume all risk by specifying the necessary conditions for pump equipment selection. The outcome is likely to be better for all parties involved if knowledge is shared prior to establishing the conditions for equipment selection.

Getting the Right Concept

Well field dewatering involves lowering the water pressure level in the mine area to permit safe, efficient excavation. Over the life of the mine, the change in pumping lift from the initial static water level to some future pumping water level can be large—on the order of 1000 feet or more. Individual well production capacities can decrease dramatically depending upon aquifer system characteristics. From a pump applications perspective, this means selecting equipment with initial operating points that best match starting conditions and which can be made to fit, if possible, conditions that are expected to occur as formation dewatering progresses.

Dewatering Schedule and Pumping Rates

The rate at which a mine is expected to be deepened below the static water level is an important planning factor. It is used to establish a schedule for lowering groundwater heads before excavation begins, and it is a major consideration in predicting the required overall pumping rate. The change in pumping lift over time, indicated by the dewatering schedule, is the variable component required to evaluate intermediate and final TDH conditions for pump selection. Pump capacity range can be estimated assuming that sufficient test or operating data are available to be confident in doing so.

The most reliable values for individual well production capacity and efficiency (well drawdown) are not available until the well is constructed and test pumping has been completed. However, the overall mine development schedule might not allow for the long delivery times that may be needed for special pump engineering or construction. If this potential problem is not addressed during project planning, pump equipment orders can be placed with results that are not cost-effective over the long term.

Well Construction

Well dimensions limit the size and type of pump equipment that can be installed. Although very costly to construct, large diameter deep wells can accommodate the installation of the large diameter four-pole (1800 rpm) 1500 hp and 2200 hp submersible electric motors that are required for high volume deep set applications beyond the practical setting depth limitations of line shaft pumps. Similar deep set applications with smaller casing diameters can require the use of two-pole (3600 rpm) submersible motors with an overall length of 100 feet or longer. In both situations a well must be drilled deep enough to achieve the design pump intake elevation and to accommodate motor equipment length and standard clearances.

In addition to well diameter, well alignment is of critical importance for deep set line shaft pumps. Even the closest attention to construction alignment standards, however, cannot prevent ground movement from adversely affecting well alignment as dewatering progresses. Depending on the severity of ground movement and resulting deflection, shaft vibration can result in shaft and motor bearing failure.

System Considerations

System conditions vary in response to production well field changes and discharge method modifications. A well head pressure condition can usually be determined for use in staging the well pump, but a conservative approach is often taken to ensure that the desired pumping capacity can be maintained. If required, throttling is used to impose pressure temporarily until system conditions are within pump operating conditions.

Fluid Conditions

Water temperature and corrosivity are major factors influencing the selection of dewatering well pump equipment. Water temperature can influence the type of construction and materials used in a line shaft pump, but elevated water temperature adds significantly to the cost of submersible electrical equipment and thus can be a limiting factor in selection. At one particular dewatering operation, line shaft pumps are not an option, and submersible motors rated at up to 2200 hp are operating in water temperatures of 140°F. These are oil filled motors of specialized construction sometimes fitted with heat exchangers. Because the motors are located in the lower reaches of the wells, below the pump intake, shrouds designed for adequate water flow past the motors are usually required for cooling purposes.

Unforeseen corrosion damage to pump cases, impellers and column pipe joints can ruin the best efforts in hydraulic applications engineering. Corrosion potential can sometimes be estimated up front by water quality analysis, and should be taken into account, if possible, in the specification of pump equipment materials and coatings. Unfortunately, geologic formation water conditions can and do change during dewatering. Partial aeration can occur with the rapid displacement of groundwater, and this can lead to unanticipated corrosive damage. Bronze and bronze alloys should be considered if conservatism is justifiable. If standard materials are selected, the first pump tear-down will reveal what doesn't work.

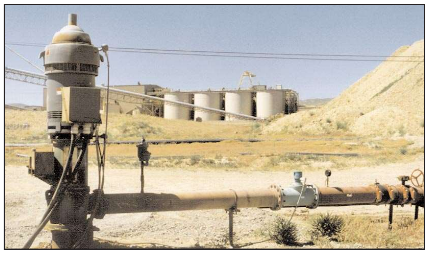

Photo 3. 400 hp vertical turbine line shaft (1020' setting) with mineral processing plant in background

Equipment Selection

Making a reasonable attempt to understand the factors that dictate initial conditions and influence future conditions is key to selecting dewatering well pumping equipment that will remain effective under actual operating conditions. Nevertheless, there are limitations, and well yields can eventually decline to the point that pumps must either be operated intermittently or replaced with lower capacity units.

Electric Submersible vs. Line Shaft Pumps

Vertical turbine oil lubricated line shaft pumps are operating successfully at setting depths of more than 1000 feet. The slower pump and driver speeds (1800 rpm or less) of these units are favored by many operators. Pump damage from abrasive particles or partially aerated formation water is significantly less intense at slower speeds. Electrical problems are much simpler to troubleshoot and correct because motors are located at the surface above the discharge head. On the downside, slower speeds require larger bowls and well casing diameters. Line shaft equipment is mechanically complex and requires special engineering and manufacturing for bowl tolerances to accommodate the effects of relative shaft elongation under high thrust loading. Well alignment problems can adversely affect shaft and bearing life or even preclude the use of line shaft equipment.

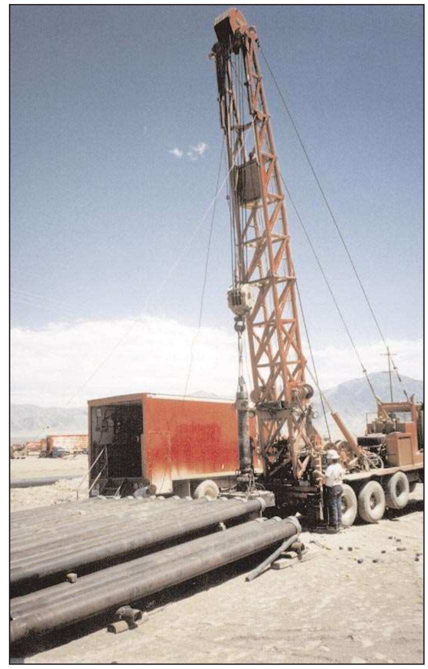

Photo 4. Pump rig installing deep set line shaft pump

High capacity submersible equipment is available in both 1800 rpm and 3500 rpm classes. Small diameter high yield wells or setting depths greater than 1000 feet generally restrict pump equipment to the submersible type. The limitation here is motor power output. Four-pole 20-inch diameter motors (1800 rpm) are available to 2200 hp for 24-inch casing applications. Slim line two-pole motors (3500 rpm) can be coupled in tandem to produce more than 1000 hp. These two-pole motor assemblies are more than 100 feet long, requiring additional well depth. Because the motors are installed below the pumps, electrical faults that occur in the down hole power cable or motor system require retrieval of the equipment string from the well for testing and repair to take place.

The Pump Curve

For a desired initial performance and estimated final performance, there is a simple rule of thumb for dewatering pump selection: start on the right side of the H-Q curve, run back to the left through the Best Efficiency Point, and plan to refit the pump end with additional stages if necessary to conform with estimated future conditions. Another method is to throttle the pump during initial operations if the range of expected conditions indicates that this will eliminate the need to pull and refit the pump end. Throttling is most common in deep set submersible applications involving relative certainty in the drawdown rate and final conditions.

Pump Mechanical Considerations

Line shaft applications involving deep settings and high thrust require special consideration for relative shaft stretch and bowl endplay requirements to establish adequate lateral impeller clearance under running conditions. Enclosing tube tension design as well as manufacturing tolerances for tube, shaft and column pipe lengths also need to be considered.

Surface Equipment

Power transformers and switching equipment are available in modular form on skid mounted platforms. Also, individual components can be custom assembled on a common skid or placed on individual pads if preferred. Mine power distribution systems often are plagued with swing loads and transients depending on the variety of electric machinery in service. Power factor correction, system protection and motor control requirements vary with the application. Vertical hollow shaft type motors used with line shaft pumps are usually 460 volt or 4160 volt. Submersible motors are typically 460 volt, 2400 volt or 4160 volt.

Installation Considerations

Line shaft pump installation can be more mechanically involved than submersible pump installation. The oil tube and shaft are usually shipped assembled in lengths of 20 feet and must be individually placed in each piece of column pipe before installation in the well. Because the column, tube and shaft assembly are run in the well casing, three threaded connections must be properly made at each 20-foot interval. The projection dimensions of the tube and shaft, which start at the pump discharge case, must be maintained over the length of the column assembly for proper fit at the discharge head and motor coupling.

Depending on the manufacturer of the submersible pump equipment, motor system and pump assembly during installation can be more or less complicated, generally requiring manufacturer's field service in addition to the installation crew and equipment. However, once the pump and motor equipment are assembled in the well and tested for continuity, column installation is a straightforward process of making one threaded joint per pipe length and securing the power cable to the column. This goes relatively quickly, especially if the pump rig can handle pipe lengths of 40 feet.

Operating Considerations

It is important to follow up on the performance of a pump after it has been placed in service. The operator will no doubt inform someone associated with the sale of the equipment if a failure has occurred or performance is not as represented; conversely, the operator will be concerned with other matters if equipment performance is acceptable. Either situation involves information that can assist the applications engineer in selecting proper equipment and recommending the most effective modifications. ■

Mark List has more than 25 years of experience in the construction and mining industries, with 10 years of practice in groundwater investigation and mine dewatering equipment design, construction and operation as Chief Dewatering Engineer for a western gold mine. He is a registered engineer and state water rights surveyor in Nevada.

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