

# SEALLESS CENTRIFUGAL PUMPS

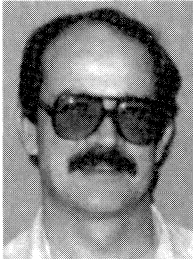
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## ABSTRACT

Magnetic drive and canned motor pumps have many common features including bearing and rotor designs, internal cooling, and lubrication. The primary differences center on the manner in which the rotating magnetic field is created and the makeup of the stationary isolation device.

Proper bearing lubrication is essential to long, reliable operation. Fluid film journal bearings are used to support the rotor and flat-faced thrust bearings to limit its axial travel.

Hydraulic balance can be used to limit the loads transmitted to the thrust bearings. This can be accomplished by either balance holes and set clearance throttling devices or pumping vanes, as in standard pumps or by variable clearance throttles that automatically adjust to maintain axial balance.

A rotating magnetic field is generated by either an electrical stator coil or a rotating permanent magnet assembly. This drives an inner rotor consisting of induction motor laminations or permanent magnets. The barrier between the inner rotor and the atmosphere contains the process, but has nonmagnetic properties to allow the passage of the field undisturbed. Eddy currents are set up in the barrier and generate heat that must be removed.

A clean cooling stream with sufficient margin from vaporization is required to carry off the heat generated by magnetic hysteresis and bearing friction losses. The pressure/temperature characteristics of the process dictate the type of circulation needed to maintain a liquid state.

Instrumentation is offered to help monitor the condition of the cooling stream and avoid deadheaded and dry-running operation.

## BEARINGS

### Fluid Film Journal Bearings

Both magnetic drive and canned motor pumps share similar rotating element designs. Rotors are typically overhung and sup-

ported in two journal bearings, the ends of which act as thrust surfaces.

Journal bearings in sealless pumps operate on the same principles as those in other rotating equipment, that is the creation by shear forces of a hydrodynamic film which supports the rotor and prevents contact between it and the bearing.

The Sommerfeld number, or capacity number [1], in Equation (1) is used to show the relationship between bearing geometry, load, speed, viscosity, and eccentricity.

$$\text{Sommerfeld Number} = S = \frac{v \cdot N \cdot D \cdot L}{C^2 \cdot W} \quad (1)$$

where

- v = Dynamic Viscosity (reyns = lbf-sec/in<sup>2</sup>)
- N = Shaft Angular Velocity (rev/sec)
- D = Shaft Diameter (in)
- L = Bearing Length (in)
- C = Bearing Diametral Clearance (in)
- W = Bearing Load (lbf)

The operating clearance in the bearing must be sufficient to prohibit contact and allow contaminants to pass. To determine the operating clearance look at the relationship between S and the eccentricity ratio in Equation (2).

$$S = \frac{(2 + e^2) \cdot (1 - e^2)^{.5}}{12 \cdot \text{Pi} \cdot e} \quad (2)$$

where

- e = Eccentricity Ratio =
- 1 - (minimum film thickness/bearing radial clearance)

At high Sommerfeld conditions, such as those obtained with low loads and high viscosity, the eccentricity is low so that the bearing operates close to the center of its clearance. Eccentricity vs Sommerfeld number is shown in Figure 1.

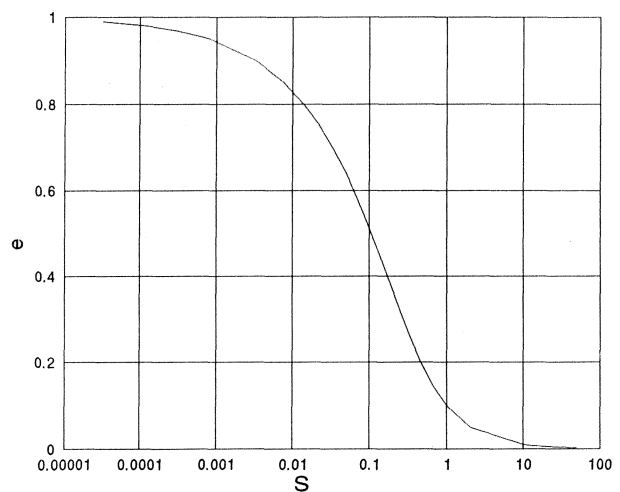


Figure 1. Eccentricity vs Sommerfeld Number.

A 2.5 in long bearing with a diameter of 1.25 in and a total clearance of 0.003 in is operating at 3550 rpm has a journal load which varies between 25 and 75 lb. The Sommerfeld numbers and resulting eccentricities for light turbine oil, water, and hexane at minimum and maximum load are shown in Table 1.

Table 1. Sommerfeld Number/Eccentricity at Different Loads and Viscosities for 2.5 in long, 1.25 in Diameter Bearing Operating at 3550 rpm with a Clearance of 0.003 in.

		25	75
	W		
Oil	1.0E-05	13/0	4.3/.01
Water	.15E-06	.2/.15	.07/.3
Hexane	.05E-06	.07/.3	.02/.9

The short Sommerfeld (Ss) or capacity number from Equation (3) is used to include the length effects on short bearing performance.

$$Ss = v * N * D * L * (L/C) ^ 2 / W \tag{3}$$

In Equation (4), the relationship between unit load P (W/L\*D), length, and bearing performance can be seen.

$$Ss = K * L ^ 2 / P \tag{4}$$

where

$$K = v * N / C ^ 2 = \text{Constant for a given } v, N, \text{ and } C \text{ (lb/in} ^ 4 \text{)}$$

Therefore, a bearing with unit load P and length L/2 will have a short Sommerfeld number equal to one-fourth that of a bearing with the same unit load and a length of L ((L/2)^2 = L^2/4). The eccentricity and load capacity of this bearing would be similarly reduced. This, in effect, is what happens when a circumferential or spiral groove interrupts the formation of the hydrodynamic wedge in the journal bearing. Some form of axial groove must be used to supply cool lubricant along the length of the bearing, but for maximum load capacity, they should be out of the loaded portion of the bearing.

Equations (1), (2), (3), and (4) are applicable for bearings with Sommerfeld numbers in normal range. When the bearings are dry or the minimum film thickness are low and the chances of contact are good, the bearings are evaluated with a pressure-velocity criteria similar to mechanical seals. The allowable P-V ranges from well over 1,000,000 to 5,000 psi-fpm, depending on the materials and construction and cooling of the bearings.

*Bearing Materials and Mounting*

Most sealless pump bearings are constructed from carbon or silicon carbide. These are usually slip fit into their stationary housing.

The sleeves can be either silicon carbide, carbon, or metallic, coated or uncoated. The metallic sleeves can be shrunk onto the shaft but are usually slip fit. The carbide sleeves must be slip fit because of their inability to take the tensile stresses from an interference fit. The silicon carbide will lose about 0.0007 in clearance to a 316 stainless shaft per inch of shaft diameter for every 100°F temperature rise. For this reason there must be sufficient clearance. Some bearing sleeves are installed on resilient mounts such as metal tolerance rings or O-rings to allow room for thermal expansion while preventing looseness on the shaft.

Silicon carbide running on silicon carbide is a good material selection if maximum wear resistance is desired. This combination is available from most manufacturers. Carbon bearings on metallic or silicon carbide sleeves is another good combination that is more tolerant of abuse or limited dry running. With a carbon or filled Teflon® bearing wear should be expected and the pump should either be routinely inspected or a bearing wear monitor used. One manufacturer used a carbon material for the rotating journal running in a metallic sleeve. This takes advantage of the higher coefficient of thermal expansion of the sleeve to allow dry running. The heat generated during the dry running condition causes increased clearance, preventing bearing lockup and allows the pump to run for a limited amount of time.

*Flat Thrust Bearings*

The flatfaced thrust bearing most frequently used contact in operation in a manner similar to a mechanical seal and have equivalent pressure/velocity limitations. If full face contact is maintained, the PV limits on these thrust bearings can exceed the limits associated with mechanical seals, because of the additional cooling offered by the radial grooves and the slight load carrying, converging wedge created by thermal distortion. At 3600 rpm, acceptable loads are in the 20 to 30 psi range operating on water. The allowable load must be decreased for lower specific gravity or lower heat capacity fluids.

The materials used in thrust bearings are similar to those used in the radial bearings. Edge loading these friable materials will cause cracking so it is important to maintain full face contact. Some bearings are mounted on a spherical seal to ensure full contact.

**THRUST BALANCE**

The thrust load in most sealless pumps is minimized by hydraulically balancing a closed impeller using balance holes and throttling rings and/or pumping vanes.

A simple system shown in Figure 2 is composed of balance holes and fixed throttling rings. The remaining load is transmitted to the thrust bearing.

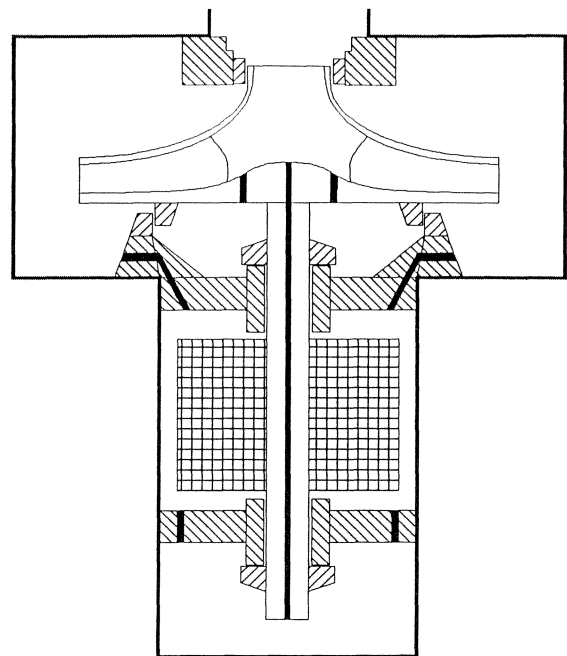


Figure 2. Thrust Balance by Throttling Rings and Balance Holes with Discharge to Suction Front to Back Cooling Circulation.

A self-balancing scheme uses balancing holes and variable orifice rings to center the rotor hydraulically. This results in a stable system that is insensitive to load changes. As seen in Figure 3, a change in rotor position causes a change in pressure drop through the variable orifice with a resulting restoring force. The balance, however, depends on a 100 percent liquid fill. Vapor will render the balancing system ineffective. Because of the need to run the orifice clearances to near zero, the thrust bearings will only contact after the variable orifice surfaces have contacted.

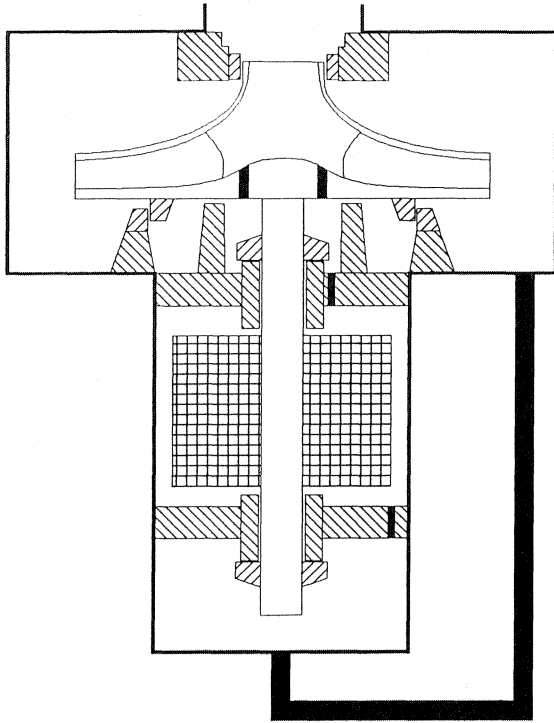


Figure 3. Thrust Balance by Throttling Rings and Variable Orifice with Discharge to Suction Back to Front Cooling Circulation.

Auxiliary pumping vanes, as seen in Figure 4, are used to increase the pressure between the rotor and containment shell to obtain additional thrust. Pumping vanes are also used on the back of an open impeller to limit the thrust load.

MAGNETIC DRIVES AND CONTAINMENT

Sealless centrifugal pumps are broken up into two categories, magnetic drive and canned motor. Both, however, used the similar principles to generate torque in the pump rotor.

The rotors contain either a laminated iron induction motor core or a permanent magnet assembly. The rotating magnetic field is created by either an induction motor stator or rotating permanent magnets. The inner rotor and outer rotating field pieces are separated by a stationary shroud made of a corrosion resistant, magnetically transparent material such as Hastelloy C-276, 316 stainless steel, ceramic, or plastic. This shell provides a barrier between the process and the atmosphere. In the case of a magnetic drive in which the rotating outer field is generated mechanically, the containment shell must have sufficient thickness to provide hoop strength against the process pressure. In the case of a canned motor pump, the stationary stator and backing rings provide strength with the relatively thin containment shell providing only process isolation.

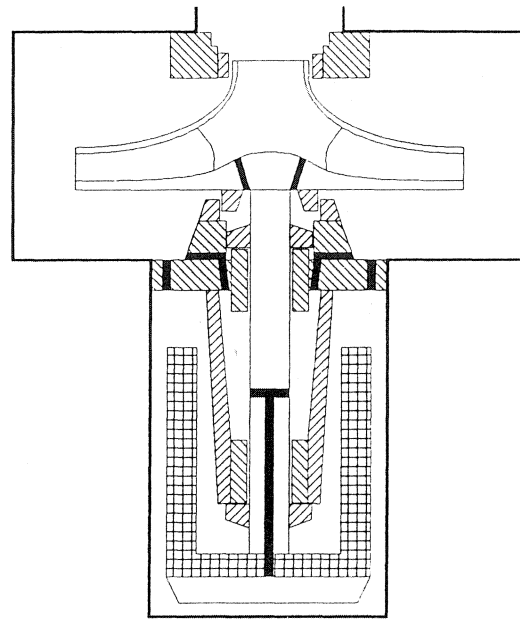


Figure 4. Thrust Balance by Throttling Rings and Auxiliary Impeller on the Back of the Rotor with Discharge to Discharge Cooling Circulation.

Drives with an laminated iron core generate torque dependent on the difference between the driving field frequency and the rotating field frequency, or slip speed. The rotating field can either be generated by an electrical stator or by rotating permanent magnets.

Drives with a permanent magnet rotor core must operate synchronously with the outer magnetic field and stay coupled to it. For this reason, the outer field on permanent magnet core rotor is generated mechanically by a rotating outer assembly. An electrically generated outer field would produce slip the instant it was energized around the zero speed rotor and would fail to transmit the required torque. The acceleration of the mechanically generated rotating field can be governed by the inertia of the pump/motor system, the hydraulic pump load, and the motor applied torque as seen in Equation (5). This must be matched against the pump rotor inertia and hydraulic load to ensure the that the torque capacity of the drive is not exceeded.

$$T = I * a \tag{5}$$

where

- T = Sum of the externally applied torques (in-lb)
- I = System mass moment of inertia (in-lb-sec<sup>2</sup>)
- a = System angular acceleration (1/sec<sup>2</sup>)

$$a = (\text{motor-hydraulic torque}) / (\text{pump inner rotor} + \text{pump outer rotor} + \text{coupling} + \text{motor inertias}) \tag{6}$$

$$a = (\text{magnetic coupling applied torque} - \text{hydraulic torque}) / (\text{pump inner rotor inertia}) \tag{7}$$

$$T_m = a * \text{pump inner rotor inertia} + \text{hydraulic torque} \tag{8}$$

At design conditions, the magnetic coupling need only transmit the hydraulic torque. T<sub>m</sub> in Equation (8) is additionally dependent

on the rotor inertia and rate of acceleration, making it necessary to minimize both in order to avoid costly oversizing of the magnet assembly. In Equation (6) it can be seen that the acceleration is inversely proportional the total inertia, making it advantageous to design a high total inertial for slower acceleration. In order the achieve both a relatively high total inertia and a minimum rotor inertia, the outer rotor is made with a high inertia.

By starting the pump in a throttled condition, the magnetic coupling can transmit the torque necessary to both accelerate the inner rotor and meet the hydraulic requirements. The coupling should be sized with enough margin to transmit maximum continuous motor torque and the end of curve horsepower. Purchasing oversized magnets with enough capacity to transmit maximum impeller diameter horsepower is usually an unjustified expense.

Of primary concern in the design of sealless pumps is the eddy current losses in the stationary containment shell. These losses are proportional to the number of poles, speed squared, flux density squared, shell thickness, surface area, and inversely proportional to the resistivity of the shell material. Several horsepower can be consumed when using metallic shells and the losses are translated into temperature and vapor pressure rise in the cooling medium. Nonmetallic materials have the advantage of very high resistivity and very low losses. These are available in both ceramic and engineering plastic construction. Both offer good electrical properties. Because of the nonductile nature of the ceramics, proof testing is suggested. The plastic containers are less abrasion resistant and may have additional temperature/pressure limits that are not in line with the remainder of the pressure containing parts.

In all cases, product compatibility with the containment shell is important. With normal section thicknesses in pumps, a low corrosion rate may be acceptable but because the shell is thin, no corrosion is acceptable. The elevated temperature properties in the high magnetic flux areas must be considered. The temperature rise in three areas of a 20 hp magnetic drive pump are shown in Figures 5 and 6. In Figure 5, temperature rise vs flow is shown for the containment shell under the magnet, for the cooling fluid, and for the exposed area of the shell typically monitored. Temperature rise vs time for the same areas with the pump deadheaded is shown in Figure 6. As can be seen, the temperature under the magnet and likely the peak temperature of the fluid is considerably higher than can be measured using available instruments. If temperature sensitivity is a concern for corrosion or reactivity reasons the actual peak temperature obtainable must be determined. This maximum product temperature is likely to be reached when the pump is shut

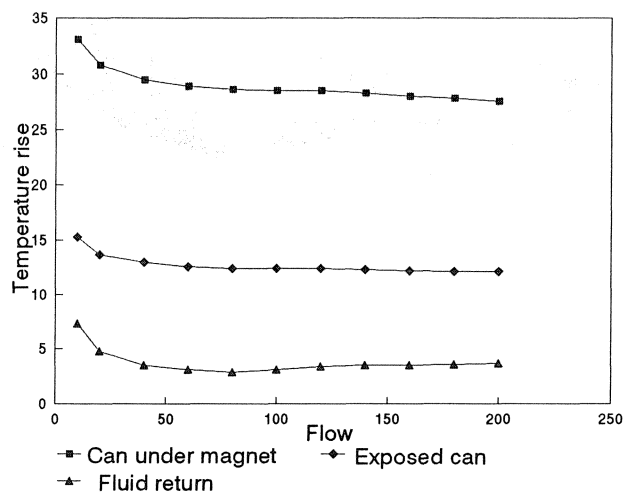


Figure 5. Temperature Rise (°F) vs Flow (gpm) in Three Locations for a 20 Hp Magnetic Drive Pump.

down and the residual liquid becomes heat soaked. An acrylic acid pump consistently experienced plugging problems due to polymerization when shut down. In one case, the heat affected zone of a welded Hastelloy C-276 shell in HCl service was attacked in this elevated temperature zone. The shell required a solution annealing process to regain the full corrosion properties in the weld area.

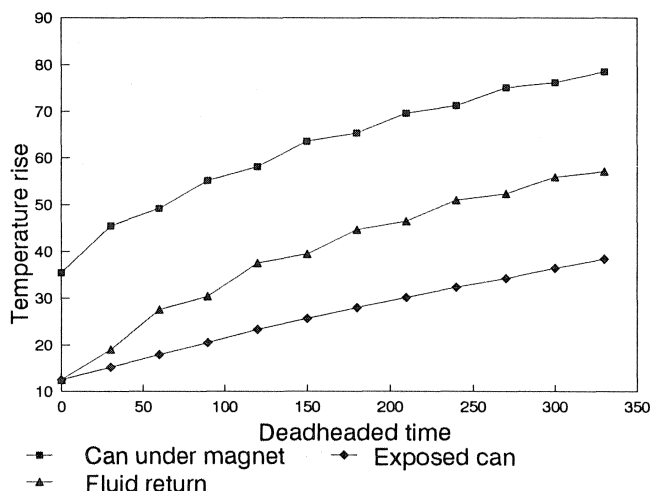


Figure 6. Temperature Rise (°F) vs Time (seconds) in Three Locations for a Deadheaded 20 Horsepower Magnetic Drive Pump.

One novel solution is a laminated container similar in concept to a motor rotor. The thin, sealing, insulated rings result in high resistance and low eddy current losses. The container is designed to keep the many rings in compression and maintain a positive seal.

Temperature is the main consideration when selecting materials for permanent magnet rotors. For service temperatures below 250°F neodymium iron magnets are common. These magnets offer the lowest cost per horsepower of any available. Between 250°F and 425°F samarium cobalt or alnico magnets are used. Samarium cobalt is more common because its higher strength allows more compact rotors to be designed. From 425°F to 750°F alnico is used. By using Alnico on the outer rotor driving an induction type inner rotor higher product temperatures can be tolerated. This eliminates the need to place the magnets in the high temperature product. The magnets will tolerate more than 425°F, but the adhesives used to attach the magnets to the rotor will not.

Temperature is also a concern when specifying a canned motor pump. Standard insulation classes are typically good for up to 400°F. External cooling and special insulations can increase the limit to 800°F.

Secondary Containment

Most pumps can offer some level of secondary containment in the event of a failure in the primary. Both static and dynamic secondaries are used.

Canned motor pumps offer inherent secondary containment from the structure used to house the stator. If the electrical terminal block is well sealed and the container otherwise is not breached, a leak to the atmosphere is unlikely. The stator cavity will become contaminated, however, if the liner is ruptured and is difficult to clear completely. This is a growing concern with the problems involved in putting hazardous materials on the road.

Magnetic drive pumps offer several variations on both static and dynamic secondary containers. Some offer a double wall containment shell. These can be either metallic or nonmetallic. The torque

capacity of the drive will be reduced due to the increased distance between the driving and driven magnets.

Mechanical seals and lip seals can also be supplied. If a dynamic seal is used and must be relied upon for secondary containment, some means of monitoring the integrity is necessary. A slight pressure can be maintained between the primary container and secondary seal and the leakage can be monitored.

## PRODUCT COOLING

Another common feature shared by magnetic drive and canned motor pumps is a cooling loop using, in most cases, the pumped product as the fluid.

A common type shown in Figures 2 and 3 takes the cooling stream off the discharge and directs it around the rotor and bearings and back to the pump suction. The return to the suction is either through a bore in the shaft or directly into the balance area behind the impeller. The disadvantage of this system is that heated fluid with an elevated vapor pressure is introduced back into a lower pressure point where it can flash if NPSH is not sufficient. Temperature rise vs flow for the cooling fluid in a 20 hp magnetic drive pump is shown in Figure 5. If the back of the container can be accessed, as with a canned motor pump, the fluid can be returned ahead of the pump through an external line. This allows time for the heated fluid to mix and cool before entering the impeller.

Auxiliary impellers are used as shown in Figure 4 in both magnetic drive and canned motor pumps to boost the cooling fluid pressure and allow it to be returned to a higher pressure where the NPSH is not a concern.

Products which contain solids or magnetic particles, which tend to polymerize, exhibit corrosive tendencies, or have other undesirable properties at elevated temperatures should be treated carefully. Solids will abrade the liners and bearings or will eventually pack into the cavities and prohibit cooling flow. There have been cases of iron particles adhering to a permanent magnet inner rotor to such an extent that the containment shell was machined through. Products which skin over such as latex or polymerize with heat such as acrylic acid are very difficult to pump. The residual heat in the metal inner parts can be enough to solidify some materials when the pump is shut down. One pump with a ceramic containment shell is being successfully applied in a temperature sensitive propylene oxide service. The cooling fluid temperature rise is about 2°F compared to an adjacent all metal pump with a 7°F rise. Most designs allow for an external coolant that can be used to introduce and filtered or otherwise clean, compatible stream. This does, however, introduce complexity and a number of connections to the system.

Whatever is used, the cooling fluid must be clean, compatible, and have sufficient heat capacity to maintain a liquid state when heated in the cooling loop.

## INSTRUMENTATION

Many types of instrumentation are available on sealless pumps and can be effectively applied monitoring the health of the pump.

On magnetic drive pumps, the most common instrument is a thermocouple or RTD measuring temperature on the outside of the containment shell. This is a multipurpose device intended to detect loss of cooling flow, deadhead, and contact between the outer rotor and containment shell and any other condition which will cause the container to heat up. Some designs allow the use of a device to directly measure the temperature of the cooling fluid as it returns to the pump. If product temperature rise is important, this instrument should be considered.

Canned motor pump and a few magnetic drive manufacturers offer bearing wear monitors. These operate either by sensing a change in magnetic flux and generating an output proportional to shaft position or by cutting a circuit or diaphragm to activate a switch or meter.

An amp or power monitor can be used to detect dry running or deadheaded conditions on either a magnetic drive or a canned motor pump and can detect a magnetic uncoupling on a magnetic drive pump. The uncoupled condition must be detected through either a temperature or load detector. Eddy current losses continue to be generated, but because the inner rotor is only turning slowly, the pressure differential required to drive the cooling flow is not generated. The result is a loss of cooling flow and an increase in can temperature. If allowed to run, the can will continue to heat up until it swells and contacts the outer rotor.

Various instruments can be used to monitor the integrity of the containment shell. These include pressure, vapor, and liquid detectors and sniffers.

Many magnetic drive pumps are becoming available with a squeal ring between the outer rotor and the pump housing. This operates with a clearance closer than the outer rotor to containment shell clearance. If the outer bearings fail, the rotor will contact the ring before the containment shell. A temperature measuring device can be placed in the ring to detect contact.

## CONCLUSION

Sealless pumps are developing a reputation for process containment and mechanical reliability. When supplied with a consistent source of clean lubricating fluid, they can be expected to give long reliable service. There is no design feature, bearing material, or clearance that will compensate for the lack of a good flush.

While the initial purchase price of a sealless pump is more than a mechanically sealed ANSI pump, the maintenance costs are proving to be lower in most cases. Problems with sealless pumps tend to be chronic, continuing until a change is made in design or process. They can, however, be expected to give long, reliable service when properly applied, operated, and maintained.

## REFERENCE

1. Wilcock, Donald F. and Boozer, E. Richard, "Bearing Design and Application", First Edition, p. 198, New York, New York: McGraw-Hill (1957).

