INFINITE DRY-RUNNING AND SOLIDS-HANDLING VIA COMBINING MAG-DRIVE AND GAS SEAL TECHNOLOGY

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ABSTRACT

This paper presents installation and field experience of the barrier sealless pump design. Barrier design is a combination of a sealless mag-drive design and gas seal technology. Gas seal separates the fluid end from the power end, thus allowing infinite dry running operation and solids-handling capability to 40 percent particulates. Since the back end is separated, fluid-lubricated journal bearings are not required, and antifriction ball bearings, are used instead—providing stiff rotor construction, with L^3/D^4 ratio as low as 9.3, which significantly minimizes deflections of the shaft. At the same time, a containment shell ensures leak-free operation and no spills, even in the unlikely event of barrier seal failure. Installation of the gas barrier design improved the mean time between failure for a tough application at a major chemical plant from six days to two years, and the pump continues to operate successfully. This paper, and the accompanying presentation, goes over details of the design, application, and installation experience by the pump manufacturer, a distributor, and plant engineering, operating, and maintenance personnel.

INTRODUCTION

A historical evolution in fluid sealing methods, to contain it, leak-free, inside the pump, can be classified as three approximate time stages:

- Before 1970s—Packings, as a main sealing method
- 1980s—Evolution and significant improvements in mechanical seal designs
- 1990s—Sealless (mag-drives and canned motor pumps) designs

The reason for a continual pressure to minimize, or eliminate, pump leakage is natural. It reflects the exponential growth in a variety of chemicals that are being introduced on the market daily, and this impacts personal safety in production, transportation, and applications, as well as the impact on the environment.

Today, the choice of a pump sealing method depends on the nature of the fluid being pumped, cost, reliability, and regulatory laws. While packed-box designs are still popular and are used in a multitude of (relatively benign) applications, single mechanical seals, these days, constitute a majority of installations at chemical plants, and similar fluid-handling (transfer or process) facilities. For more aggressive, toxic, or expensive fluids, double mechanical seals are used. These are more complex and expensive, and, with complexity, reliability becomes an issue. However, seal designs, and their reliability, have been significantly improved by the seal manufacturers, and are used successfully, with low emissions, in most applications in chemical plants (Engineering Seal Products, 1991).

Some applications, however, require not a little, but *none* of the leakage. Toxic, dangerous liquids call for not a reduction, but a complete elimination of the sealing mechanism, because, by the virtue of the design, it has a pump shaft penetrating outside the pump, under the seals. In the event of the seal (single or double) failure, the process fluid ends up spilled on the ground. Sealless pumps are a requirement for such applications.

Because of these reasons, by the late 80s, the euphoria of the "sealless-saves-all panacea" reached a peak, and a "doomsday of a mechanical seal as we know it" became a battle-cry of the fortune-telling sealless pumping community, which, actually, was shared by many end users. However, the euphoria gave place to disappointments. Sealless pumps did not turn out to be the solution against all evils, and, as they started to fail (often misapplied) in the field, came a cold shower to many former proponents. It was realized that it is not possible to apply sealless pumps on each and every application, and, as anything else, their limitations became clear, as well as their advantages. The two main limitations of sealless pumps are:

- Cannot run dry (Figure 1)
- Cannot handle solids

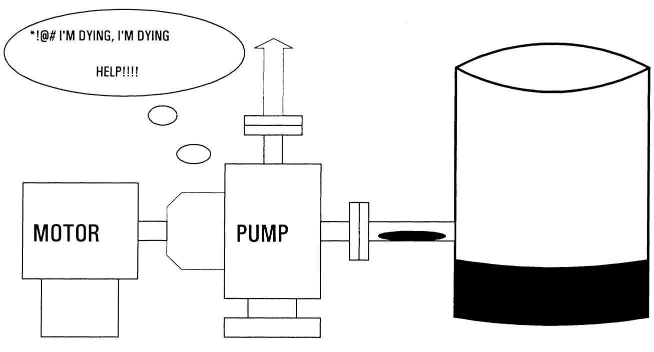


Figure 1. Sealless (Standard) Pumps Should Not Be Allowed to Run Dry.

The reason for these limitations are inherent in a basic design feature of mag-drive pumps: journal (sleeve) bearings that support the rotor are lubricated by the pumpage (product-lubrication). Lubricating fluid removes the heat from friction, as well as provides hydrodynamic film to support the rotor (which is a shaft with the impeller and an inner magnet). The process fluid also removes the heat from the eddy currents, which are generated in the metallic shell (containment) that separates the outer and inner magnets, and contains the pumpage inside the pump. As there is obviously no fluid flowing through the pump when it runs dry, there is no supply of this fluid to bearings, and, quickly, the pump fails (Figure 2).

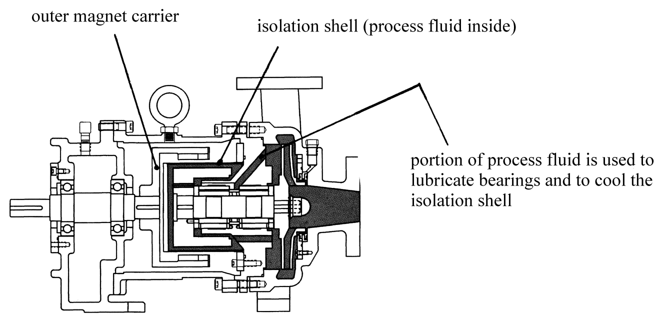


Figure 2. Mag-Drive Pump Requires Product Fluid to Lubricate Bearings and Remove Heat.

If the pumped fluid contains solids, they would get carried through the clearances between the bearings, disrupting the film, and plugging the internal passages, thus also leading to bearing failure. This is why, as users discovered, application of sealless pumps is limited to cases where the pumped fluids are clean, and no dry running (empty tanks, cavitation, flashing, etc.) takes place.

GAS SEALS VERSUS DOUBLE MECHANICAL SEALS

In either case (gas seals or regular mechanical seals), there is no assurance that, in the event of the seal failure, the pumpage would stay inside—it will not. However, a secondary seal buys some time, in the event the primary seal fails. There is a certain advantage of having an additional (secondary) seal over a single mechanical seal from the emissions standpoint: since (for the case of double, or tandem, mechanical seals) a buffer fluid is supposed to be "friendly" to the environment, its emission past the secondary seal to atmosphere is less (or should not be at all) harmful. The disadvantage is complexity and cost of the auxiliary system—buffer fluid, tank, gauges, etc. (Figure 3).

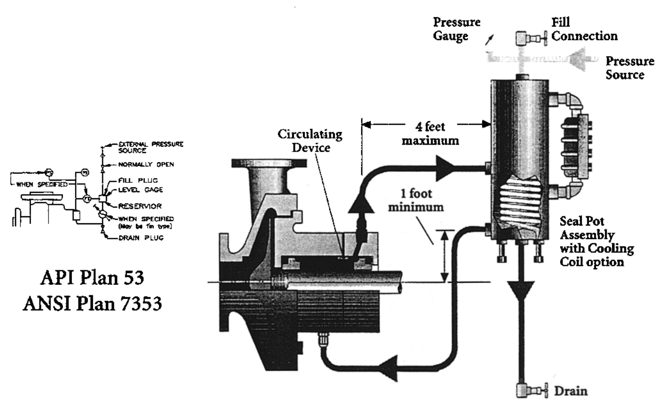


Figure 3. Dual (Double) Mechanical Seal Auxiliary Support System. (Courtesy of Flowserve Educational Services, 1999)

Gas seals are somewhat similar to mechanical double seals: there is a primary seal between the liquid and barrier gas, and the secondary seal is between the barrier gas and atmosphere (Figure 4). There are no complicated auxiliary buffer tanks and support—just a supply line to a gas source (typically plant nitrogen), and a panel with pressure regulator and gas flow indicator. Sometimes a check-valve and a gas pressure switch are added to shut down the pump in the event of a gas supply interruption, or a seal malfunction.

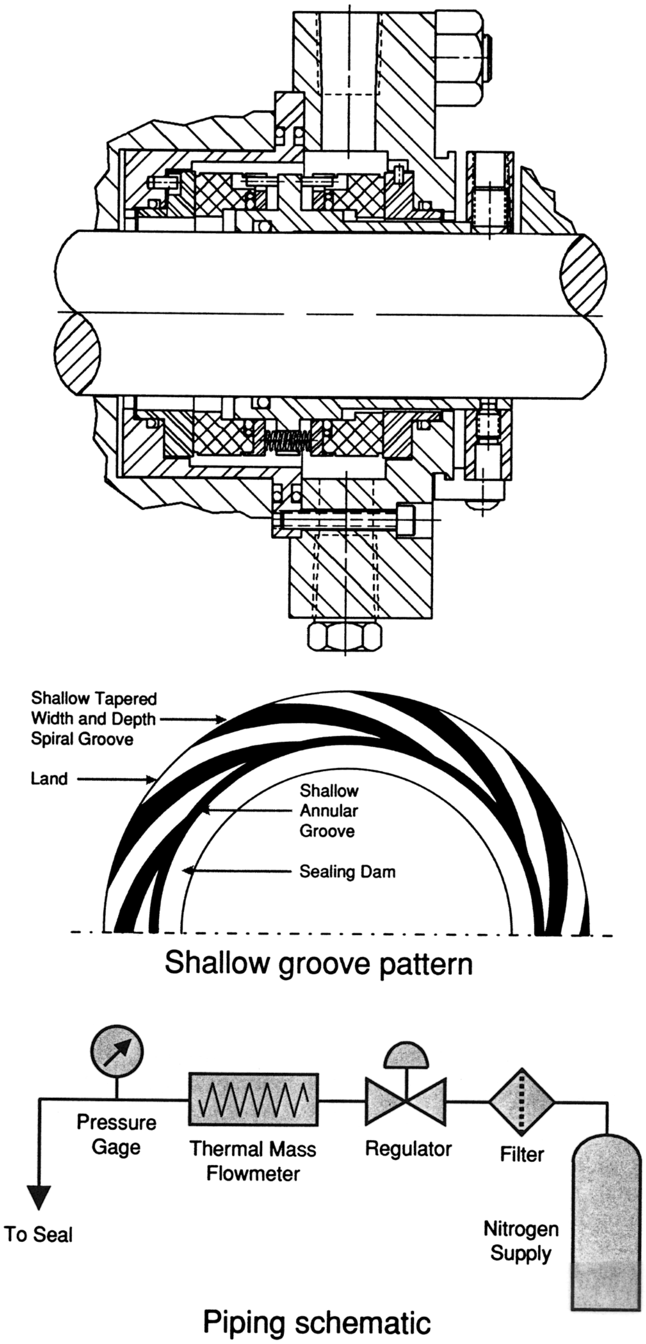


Figure 4. Typical Gas Seal. (Courtesy of Turbomachinery Laboratory, Turley, et al., 2000)

A design of a gas seal is more complex, however, because of the face geometry. While seal faces of single or double mechanical seals are plain, the faces of the gas seal are specially contoured, to provide a hydrodynamic lifting effect, to keep them separate. Lubrication of the *plain mechanical* seal faces is essentially hydrostatic (in contrast to a *gas seal* lubrication regime, which is a combination of hydrostatic and hydrodynamic (Gardner, 1969)). A very thin layer of liquid molecules is a film between the faces of a mechanical seal. This film provides lubrication and the surrounding liquid helps to convect away the heat. The thickness of this film depends on the viscosity (among other things) of a fluid between the faces. From a hydrostatic perspective, mechanical seals operate by controlling the shape of the fluid film between the faces. Both pressure and thermal deflection act to alter the shape of the fluid film from the "ideal" parallel shape. The final steady-state shape of the film is influenced by the face design and material properties and thermal losses (primarily speed and pressure). This is coupled with the selection of the seal balance (or hydraulic closing force) and spring load to determine the final fluid film thickness. In a seal running on a very light fluid (for example, propane), the film thickness is normally on the order of 20 to 30 microinches. Higher balances are used to compensate for higher opening forces generated by the flashing fluid. Higher viscosity fluids normally may run in the 30 to 40 microinch range. More viscous fluids have thicker fluid film, and if the fluid is too thin (low viscosity fluid, such as gas), the film may be too thin, and the faces could touch, generating heat, wear, and the seal may "burn up." The leakage rate (emission) from the plain mechanical seal is very small, and modern designs achieve 100 to 50 ppm emission readings, detectable only by special instrumentation.

Seals share some operating characteristics with journal bearings (*Marks' Standard Handbook for Mechanical Engineers*, 1978). For simplification, this paper will look at the effects of a journal bearing operating on a medium viscosity fluid (water) and a very low viscosity fluid (air). Fluid film thickness can then be calculated as follows:

$$h_o = 0.0341 \times \sqrt{\frac{\mu \times V \times L}{P_{avg}}} \tag{1}$$

where V is face mean velocity, L is peripheral length, P_{avg} is average carrying load, and μ is dynamic viscosity. Equation (1) applies to any fluid (liquid or gas), but, for gas, compressibility needs to be taken into account, and Equation (1) becomes more complex. Typically, hydrostatic bearings operating on turbine oil have a liquid film thickness ranging between 0.001 inch to 0.004 inch, depending on other parameters, as noted in Equation (1). For rough comparison, assume the film thickness is 0.002 inch for oil (assume average oil viscosity = 200 cP), then:

- Water (viscosity = 1 cP):

$$h_{water} = 0.002 \times \sqrt{\frac{1}{200}} = 0.0001\,inch \tag{2}$$

- Air (viscosity = 0.01 cp):

$$h_{\max} = 0.002 \times \sqrt{\frac{0.01}{200}} = 0.00001\,inch \tag{3}$$

This is why the design of a plain-face *gas* seal is a challenge: mechanical seals operate on liquid, which has substantial viscosity, lubricity, density, as well as heat conductivity—while gas has none of these benefits. As with any seal design, there must be a balance between the closing force acting on a rotary face, and an opening force, produced by the pressure of a fluid film between the faces acting on a rotary member. Gas seals are normally designed to operate with a slight net closing force (to balance against a "liftoff" force between the faces). In a standby, nonrotating condition the seal faces are closed, touching each other. The purpose of the standby, or closed-face condition is to prevent the sealed process fluid from migrating into the buffer fluid cavity. If the balance is wrong, a liquid mechanical seal may have a better "tolerance" to it: the higher force would squeeze the film between the faces, but, since liquid is essentially incompressible, the fluid film would not be reduced excessively (refer to Equation (1) above), and the excess heat buildup would still be able to be conducted out easily due to good specific heat and conductivity of the liquids. A gas seal, however, operating on a much thinner film to begin with, may get its film dangerously close to the surface imperfections resulting in film penetration and disruption, and, possibly, face contact, leading to heat buildup and failure.

A discussion above considered only plain-face seal geometries. Clearly, due to a difference in film thickness, a mechanical seal operating on fluid does not require any additional face modifications, but a gas seal does. To further assist film formation, thickness, and opening force, gas seal face geometry includes special grooves, which generate additional hydrodynamic force that separates the faces further apart, i.e., increases the thickness of a gas film. There are a variety of face geometries, but a common feature is a series of grooves, which guide the gas from the higher pressure zone to the lower, terminating with a "pressure dam," which creates separating force, keeping the faces apart (Figure 5).

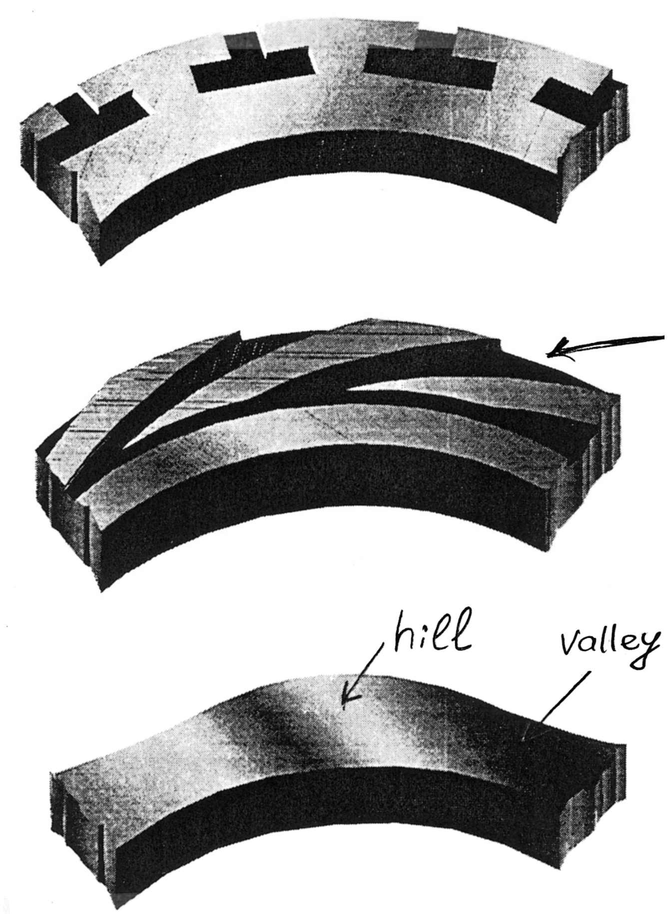


Figure 5. Example of Gas Seal Face Geometries.

A grooved pattern can be either on the rotary or stationary face (Endura Product Technology Catalogue, 1996). The gas film for a grooved design is still thinner than it is for a liquid, but significantly thicker than for a simple plain-faced (nongrooved) gas seal design (which, as was shown earlier, was extremely thin), and provides more reasonable and acceptable (about 0.0002 inch) separation between the gas seal faces.

A significant benefit of a gas seal versus regular double mechanical seal is elimination of the complicated auxiliary system, i.e., simpler and less costly.

Gas seals for pumps became popular in the mid-1990s, as a natural progression from compressor applications where they had been used for some time prior to that. It made pumps more tolerable to solids handling and dry running situations. However, the main drawback—penetration of a pump shaft under the seal and a direct lead to atmosphere, and potential spill of a pumpage if a seal fails—still remains an issue. This is where the idea of combining the benefits of the "no-spill" feature of conventional mag-drive pumps, with a gas seal (runs dry), first drew attention. Table 1 is a brief comparison between the design of double seals and mag-drives.

Table 1. Feature-Benefit Comparison Between Seals.

	Runs dry?	Handles solids?	Leak/spill-free?
Single mechanical	No	Poor	No
Double mechanical	Limited time	Yes	No
Double gas seal	Yes	No	No
Mag-drive	No	No	Yes

None of the design variations in Table 1 got a "Yes" rating for all three criteria. However, the table suggests an interesting opportunity: if a spill-free ability of a mag-drive could be somehow combined with the dry-running ability of a gas seal, such a pump would be able to have:

- Zero leak, even if the seal fails
- Runs dry, indefinitely
- Handles solids

The "barrier" design, shown in Figure 6, reconciles and combines these, at first glance seemingly contradictory, characteristics. As with any mag-drive, torque (power) from the drive shaft of the motor is transmitted through the containment shell, as a magnetic flux, driving the inner magnet, which is located on the pump (inner) shaft. So, the spill-free requirement is thus guaranteed via the virtue of a containment shell.

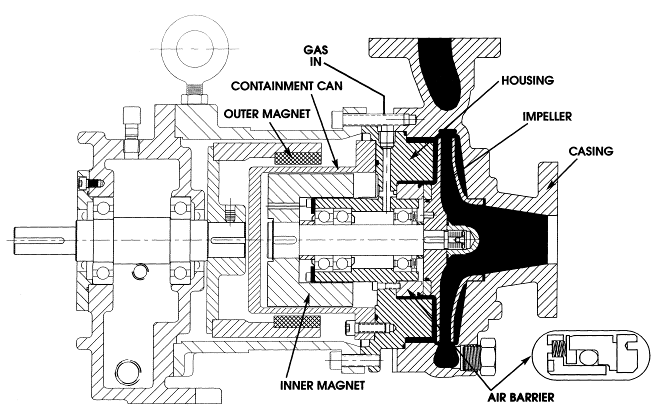


Figure 6. Combination (Barrier Design) of Mag-Drive, with Gas Seal, Results in Infinite Dry-Running and Solids Handling Ability.

A single gas seal is now added, directly behind the impeller, isolating the pumped fluid from the pump back end, which thus remains dry. Gas (typically nitrogen) is injected inside the containment. Gas is taken from the plant piping or from the nitrogen bottle, whichever is available. (Sometimes a series of bottles is used, to ensure uninterrupted supply if the plant gas is unavailable, or one of the bottles empties.)

Instead of the (product-lubricated) sleeve bearings that are used for mag-drives, a barrier design uses standard antifriction bearings, such as angular-contact ball bearings. These greased-for-life bearings, in addition to being commercially readily available, are inexpensive—but, more importantly, provide rigidity to the pump rotor, making it capable of withstanding process transients much better (refer to later discussion on L/D).

The metallic shell is replaced by a transformation-toughened-zirconia (TTZ) shell, which is nonconductive and does not require liquid to remove the heat, since there are no eddy currents generated in it. The TTZ shell is hydrostatically pressure-tested at 400 psi.

Figure 7 shows photos of a pump that was emptying a tank and was forgotten over the weekend. Monday morning, tank empty, the pump kept on running dry, with no damage.

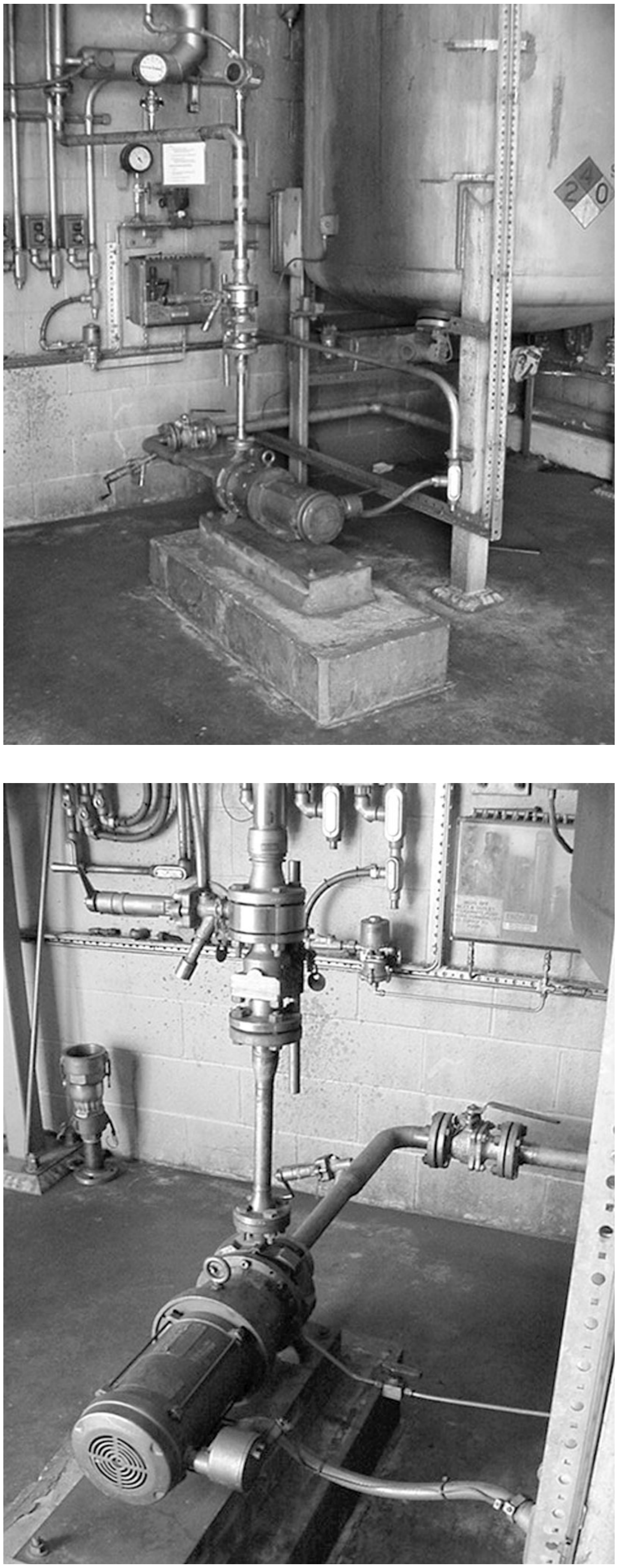


Figure 7. Pump Emptying Tank and Running Dry. (Courtesy of DuPont Company and Liquiflo Equipment Company)

MCSF AND SHAFT DEFELECTION

ANSI pumps are not recommended to run below minimum continuous stable flow (MCSF) (Nelik, 1999). Hydraulics are such that, at low flow, the hydraulic radial thrust becomes excessive. Hydraulic instabilities, at part load operation, make operation of pumps at low flows even more undesirable (Stepanoff, 1957) (Figure 8).

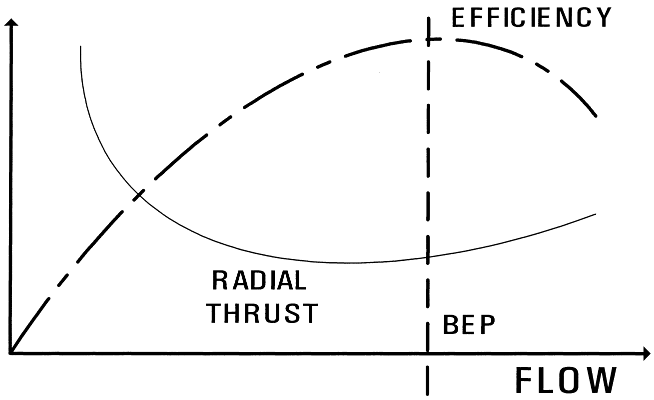
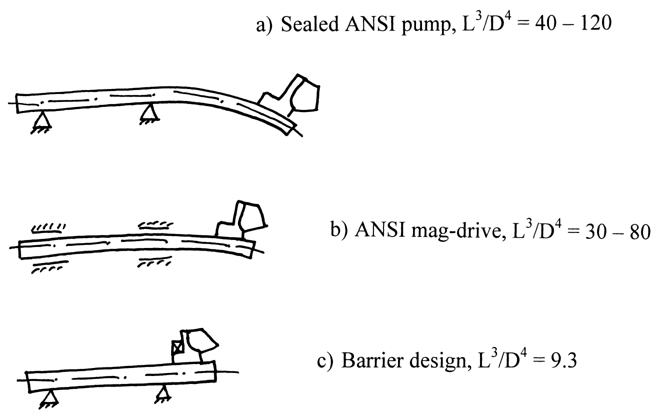


Figure 8. Hydraulic Radial Thrust—Increases Exponentially at Low Flow.

Being essentially a cantilevered beam, a pump shaft deflection is a function of the pump overhung length and the diameter. A shaft "stiffness" parameter, L^3/D^4 is a measure of the shaft resistance to the radial force (Figure 9).



$$y = \frac{FL^3}{EI} = \frac{FL^3}{E\frac{\pi d^4}{64}} = Const \times \left(\frac{L^3}{D^4}\right)$$

Figure 9. Shaft Deflection Significantly Reduced with Lowered L^3/D^4.

Positioning of a gas seal directly behind the impeller allows the radial bearing to be placed closer to the impeller than in any ANSI sealed pump design. This new design reduces the shaft deflection dramatically, allowing the pump to operate at lower MCSF as compared to the conventional ANSI designs. For example, mechanically sealed pumps or standard mag-drives are not recommended to operate below 25 percent to 35 percent of the best efficiency point (BEP) flow. Operating below the minimum flow limits, sealed pumps encounter shaft-bending and seal issues; mag-drive pumps experience thrust bearing issues. Barrier design, with its stiff shaft, is limited only by the heat generated by the action of the impeller on the process fluid in the pump (common to all centrifugal pumps). Therefore, the barrier pump can operate *almost* at the shut-off point, requiring only enough flow through the pump to dissipate the heat buildup in the fluid.

GAS SUPPLY

A limitation of a barrier design is the requirement of a gas supply never to be interrupted. Gas pressure inside the insulation shell must be greater than the liquid pressure on the other side of a gas seal. Figure 10 shows pressure distribution behind the impeller shroud. Parabolic pressure distribution behind the impeller follows a forced-vortex theory (Stepanoff, 1957), and, for a typical ANSI pump, it is approximately equal to 70 percent of the discharge pressure.

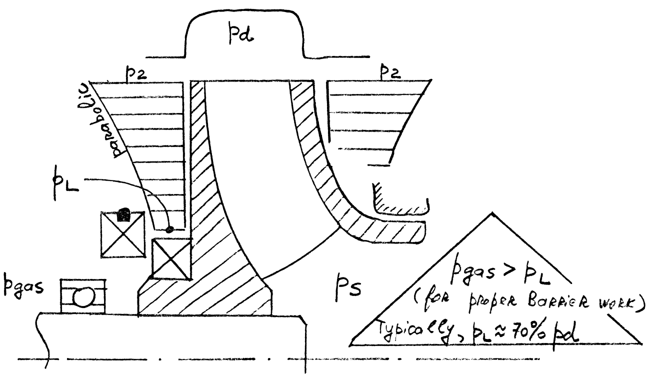


Figure 10. Pressure Distribution Behind the Impeller Shroud.

Gas pressure must be greater than liquid pressure (P_L) behind the impeller. The lower a liquid pressure P_s, the lower a required gas pressure. Plant nitrogen supply pressure is usually equal to 70 to 80 psig, and, even though it is possible to boost it up higher, it is desirable to stay under this value. This is why it is desirable to reduce liquid pressure P_L, which can be done with impeller balancing holes that connect the area P_L with impeller inlet (eye) area, making P_s pressure close to suction pressure (Figure 11).

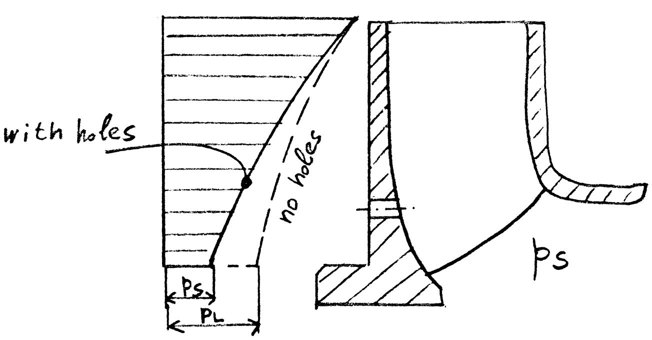


Figure 11. Pressure Reduction Using Impeller Balancing Holes.

As a rule-of-thumb, ANSI-dimensioned barrier pumps, with closed impeller and balancing holes, use the following formula to determine the required minimum gas pressure:

$$P_{gas} = P_{suction} + 35, psig \tag{4}$$

Additional design modifications, such as impeller pump-out vanes, scalloping, etc., can be a further improvement by reducing gas pressure requirement, and can be considered on a case-by-case basis. A standard barrier design for an ANSI pump application, which often has flooded suction, typically operates at 40 to 90 psig, i.e., a readily available nitrogen supply at most plants can be used.

GAS CONSUMPTION

Gas seals do not consume much gas. The gap between the seal faces is very tight, and a very minute flow of gas is sufficient to maintain gas film and to keep faces separate. Gas flow is a function of gas pressure, differential pressure (gas pressure minus liquid pressure on the other side of a seal), and other gas properties. Since, as discussed above, 40 to 90 psig is a typical gas pressure for a barrier pump, and the differential between it and liquid pressure is 20 to 30 psig, the consumption of gas is approximately 0.007 scfm (10 cc/min STP), i.e., a very tiny amount.

Most applications allow such a minute amount of inert gas (nitrogen) to enter the main flow, and, if plant nitrogen is used, the question of "running out of nitrogen" is not an issue. However, even for installations where bottled nitrogen is used, a typical 5 ft^3 bottle lasts a long time (Figure 12).

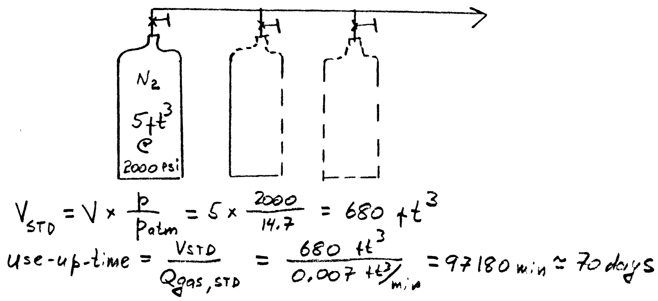


Figure 12. Typical 5 Ft^3 Bottle at 2000 PSIG Lasts Several Months.

Sometimes a bank of bottles is used, for automatic or manual switch-over, to make sure the supply of gas stays uninterrupted. Typically, the cost of nitrogen, at the above rates, is about \$10 per year (when the nitrogen is produced at the plant site).

Undetected leaks in the control panel and connecting tubing are often much higher than the leakage across the seal faces. The first choice would be to pipe nitrogen from a header. If bottles are used as the primary source, the supply must be instrumented to detect excessive leakage and low supply pressure.

Other gases may be used as well, as long as they are compatible with pumpage, are thoroughly dried, and do not affect (react with or dilute) the bearing grease. Air is used sometimes, too—however, in practice, plant air contains a high percentage of moisture (water), which, even after filtering, does not approach the dryness level of nitrogen. Moisture levels in the barrier gas become an issue when moisture condenses in the can or corrodes the radial and thrust bearings. This is an important consideration when planning to use plant air as the barrier gas.

As may be noticed from the barrier pump cross-sectional drawing (Figure 6), gas fills out the cavity under the containment shell, including the area near the grease-lubricated ball bearings. This is another reason why nitrogen is well accepted and used in just about all cases—being inert, there is no issue of reacting and diluting the grease of ball bearings or corroding the bearing races.

IDLE CONDITION

Gas pressure must always be greater than the process pressure inside the pump, in the area adjacent to the seal, on the pumpage side. This gas does two things:

- Provides hydro(gas)dynamic film for the seal faces operation (liftoff condition)
- Prevents pumpage from going back, inside the containment shell, past the seal

When a pump is shut down, the seal faces close (or touch) forming a sealing dam that ensures contact and thus provides an additional resistance to leakage into a dry area, inside the containment shell. Containment area stays "dry" by maintaining gas pressure (P_{gas}) inside greater than the liquid pressure (P_L). As long as this is maintained, there are no problems.

If the pressure of the process fluid at the seal faces is greater than the barrier gas pressure, process fluid will start seeping into the containment can. Two common conditions under which this situation may occur are:

- If barrier gas pressure is removed, or interrupted, or

- If a pump is flushed with a cleaning fluid while the pump is sitting idle, and the pressure of the cleaning fluid rises above the gas pressure in the containment can.

The result is a "negative pressure differential" trying to open up a gap between the seal faces, and to drive the fluid on the process side of the seal inside the containment shell. Even under severe negative differential, however, fluid migration into the containment can is still very slow because the seal faces are held tightly closed by the springs behind the stationary seat.

Good news, though, is that even if the fluid migrates into the containment can, the pump would still continue to operate, and no leakage to the atmosphere or spills to the ground would take place. Being a mag-drive, the barrier pump keeps the pumpage contained inside, even if the gas seal should fail. This continuation of running, if the seal fails (as described above, when gas supply is lost), is not indefinite. Eventually, as more fluid migrates inside the shell, grease in the ball bearings will become diluted, and a classical bearing failure could occur (Nelik, 1993). If the migrating pumpage is especially corrosive, it will accelerate the deterioration of the grease, and could corrode the balls and races, leading to bearing failure.

Furthermore, as another "line of defense," even if undetected (refer to later discussion on detection), the eventual failure of bearings does not automatically equate to an immediate and catastrophic failure and a spill. A concern here would be that a bearing failure would result in the eventual rotor drop on a containment shell, and its rupture—this issue is solved by design: making a radial gap between the outer diameter of the inner magnet and the inner diameter of the containment shell *larger* then the gap between the magnet and the bearing housing surface (Figure 13)—a drop of the rotor would be onto a bearing housing lip, and not onto a shell.

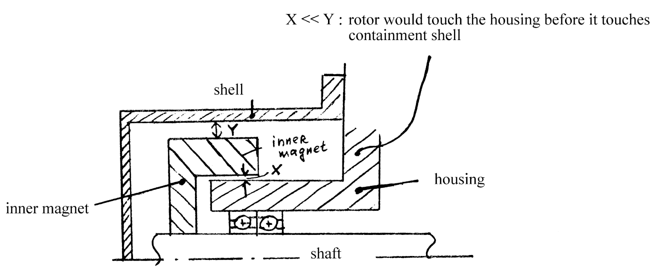


Figure 13. (Conservative Assumption) of Sequence of Events: Seal Failure, Bearing Failure, Rotor Drop; Still Maintains Integrity of Containment Shell.

FAILURE DETECTION METHODS

There are two main root causes for a possible pump failure:

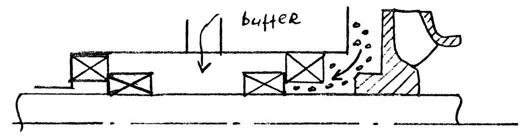
- Loss of gas supply

- Seal failure

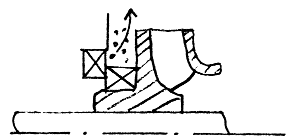
In most cases, the seal failure itself is a result of a loss of a gas supply. If gas is lost, the seal operating regime changes from the dynamically gas-lubricated to an abnormally fluid-lubricated (seal failure mode) by the liquid that enters the containment.

During normal operation, the design ensures a self-cleaning operation of a seal, as compared to conventional gas seal designs. As shown in Figure 14, barrier gas supply is from the inside to the outside diameter. Contaminant particles get centrifuged outside, and not drawn-in between the seal faces. A conventional seal draws particles into the seal face.

Even if a barrier seal fails, it still continues to repel the particles that try to approach it, but, if gas supply is interrupted, there is no longer an assistance by the differential pressure to drive the



a) Conventional double mechanical seal. Pumped particulates are drawn to seal faces, eventually causing plugging, disruption of heat removal, and failure.



b) Barrier seal design is positioned directly behind (on) the impeller. Particulates are removed by self-clearing centrifugal force.

Figure 14. Barrier Seal Gas Supply Is from Inside Diameter to Outside.

particles away. In such case, some liquid, then, with particulates dragged along, would seep past the seal faces, and would eventually fill out the containment can. From then on, lubricating regime of the seal faces is by a mixture of liquid and particulates, which would lead to seal face damage and failure.

Since such failure is still not of a "sudden death" nature, there is usually sufficient time between the interruption of a gas and ultimate failure of a pump's ball bearings. Cases have been reported where pumps continued to run for several weeks after a loss of gas, and without failure of the ball bearings. To detect this, two methods are recommended:

- Liquid sensing inside the containment

- Housing vibration monitoring, such as spike-energy methods, often used by plant maintenance and operating personnel to detect bearings beginning to skid (with races beginning to become scored and damaged), as opposed to a normal rolling mode

Liquid sensing can be done by attaching a probe at the bottom of a pump, such as a tee at the gas connection.

If proper care is taken to ensure no gas interruption at all times (running or idle), these detection measures are not necessary, but could provide additional protection for installations where gas interruption may be a problem.

USER EXPERIENCE

A chemical plant in the Midwest had a serious issue with bearings failing in their mag-drive gear pump, experiencing an average mean time between failure (MTBF) of *under seven days*. The fluid being pumped was a graphite slurry in a highly odiferous solvent that, for environmental reasons, was best contained in the process system; the slurry was also thixotropic—hence the initial selection of a magnetic drive gear pump.

Double mechanical seals were ruled out because the process solvent would have to be used as the barrier fluid, creating a situation where a no-spill condition could not be ensured.

It was discovered that one of the raw material specs allowed for traces of iron, which did not hurt the process, but which were concentrating in the can of the mag-drive pump and destroying the bearings in short order. The barrier design, on the other hand, would achieve the goal of solvent containment and, with its gas seal and conventional bearing design, provide no means for the iron to concentrate or damage the back end of the pump. Additionally, the barrier design ensured fluid would be contained inside the pump at all times, even if the barrier gas seal should fail. A pump was procured and installed in the fall of 1998, with the following history.

1. Installed in late 1998, this ANSI-dimensioned 1.5×1×6 pump recirculated 20 percent graphite slurry with 19 psig discharge pressure (Figure 15). Gas (nitrogen) pressure was set at 75 psig, with 48 ml/min gas flow, as measured by the gas flow meter, integral with a gas panel. On January 17, 1999, plant engineering informed a pump manufacturer and their distributor, that "…a pump is doing quite well. It is operating with N2 flow around 70—increasing as the tank level goes down, and decreasing as the tank level goes up." The operators and engineers were quite pleased with the pump performance. However, operators were somewhat uneasy with the fact that gas flow was changing, and requested an explanation.

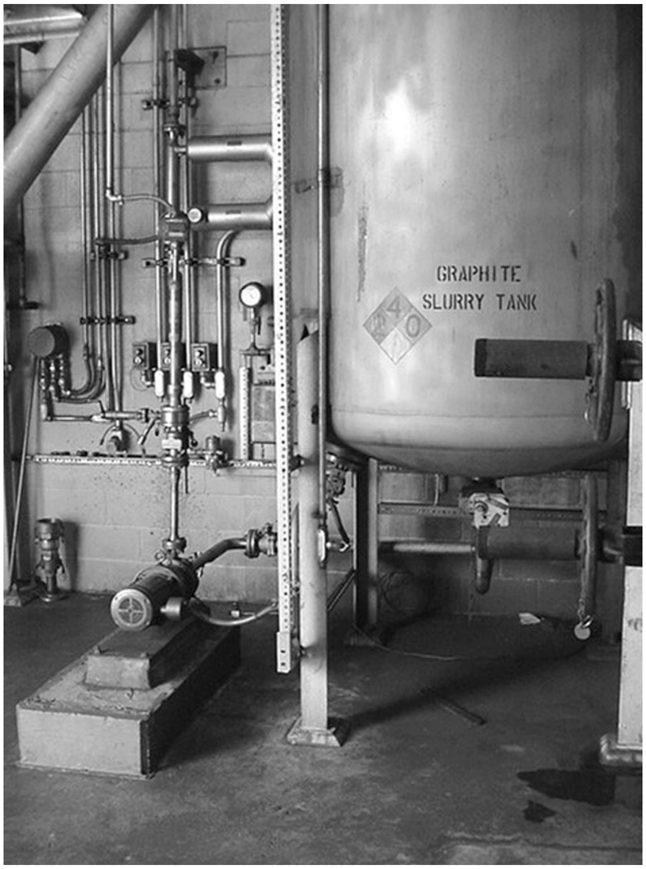


Figure 15. Pump Recirculates Slurry, with Occasional Batch Charge into the Process.

2. With the tank level fluctuating, there were periods of operation when the seal gas flow was nearly zero. In late January 1999, operators heard a "chattering" noise, and the pump was shut down. Upon disassembly, it was found that the gas seal was damaged. The pump was sent to the factory for analysis and repair. Failure analysis indicated that the gas supply was either interrupted, or somehow prevented from flowing normally between the seal faces. At that time, a more conventional spiral groove seal design was in use (Figure 16).

Since the exact cause was still unknown, the factory decided to provide a spare module (pump less casing), to ensure quick turnaround for the inplant replacement in the event the failure were to happen again. It was important to do everything possible to maintain plant uptime, while engineers, on both sides, were working on analyzing a possible reason for failure.

As a note, a definition of a "pump failure," in plant terms, meant that the pump had to cease service. Despite the shutdown, the good news was that no actual leaks, or spills, took place at any time, due to the sealless mag-drive nature of the barrier design.

Figure 16. Originally Supplied Conventional Gas Seal Design.

3. The rebuilt pump operated until May 1999, when the pump noise was again reported by the operators. Again, the pump was shutdown and inspected. Interestingly, no internal seal damage was found, but the pumped fluid had apparently seeped passed the barrier seal, and eventually caused the inner bearing to fail. It was unclear whether or not gas interruption took place at some time during operation.

The pump was sent to the factory and this time was retrofit by a new seal design, with improved seal face geometry, for the increased liftoff capability and lower gas pressure requirements. In addition, the impeller was redesigned to include expeller vanes on the back side. The expellers help reduce the pressure where the process fluid meets the seal, and they help minimize the amount of solids that can flow toward the seal (Figure 17).

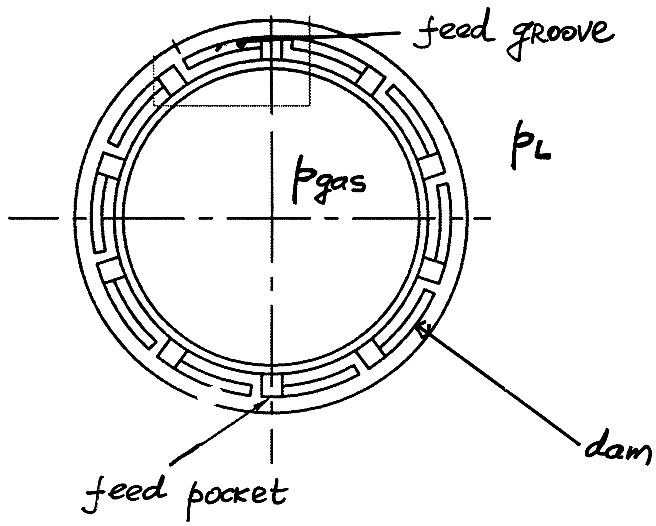


Figure 17. Improved Seal Design; Special Grooves for Better Liftoff, Lower Gas Consumption, Less Supply Pressure Requirements.

4. The rebuilt pump was installed and restarted. By then, operators had also improved the method of preparing and adding a slurry mixture to the tank. Instead of adding graphite first and then pyridine, the procedure was reversed. This way, the resultant slurry always tends to be more diluted, and does not settle as badly in the area of immediate proximity to the seal on the process side—which was believed to eliminate, or improve, a slurry settling problem in the seal vicinity, and possibly preventing gas from properly flowing between the faces.

By June, operators had a good handle on running the pump, and a record of pump pressure, tank level, gas pressure, and gas flow was being taken periodically. Studying the trend graphs, engineers were able to determine the times when the gas flow was getting dangerously low, requiring the gas pressure to be above that available at the plant. The improved seal design, with lower pressure requirement, was certainly a step in a right direction.

5. Despite these improvements in operating the pump system, intermittent "chattering" noise was still reported by the pump operators. Before another failure, it was decided by the pump distributor and the user to remove the pump from service for analysis. After removal of the pump from service and inspection of the mechanical seal, it was discovered that the stationary seal face had been "skipping" across the metal pins that were meant to hold it in place (antirotation feature). The stationary seal had chipped, damaged pin openings. The root cause was that the pins were not tall enough to hold the stationary seal in place. The seal was hopping off the pins during process pressure fluctuations. This resulted in seal damage and eventually failure of the pump.

The pump was rebuilt with more robust stationary seal face holding pins and placed back into service. After this change there were no more failures or "chattering" noise. As a result, the pump ran perfectly.

6. Up to the date of this writing (July 2001), the pump has accumulated a total of 5000 hours running time, and continues to operate well. The total number of shutdowns, during over two years experience was five times. At no time, however, was there any spill of the product, or any substantial damage to the pumps. The majority of the shutdowns were at the beginning, and, as the plant has gained a better understanding of the pump and process requirements and the seal holding pin issue was resolved, the pump became a reliable piece of equipment that has solved a difficult problem at the plant.

LESSONS LEARNED

- Manufacturer/distributor assistance, support, and training of the maintenance and operating personnel are critical, and a must. A full day of intensive training for all involved, including assembly, engineering, and operating personnel, and including actual demonstration of a dry-running pump operation, can solve numerous issues and answer major questions upfront.
- The learning curve is usually three months. This was the case at this plant, as well as an estimated average at other installations of this pump design.
- The pump is best applied for troubled installations, especially where conventional mag-drives cannot be used because of the dirty streams (example of the process fluid with contaminants is shown in Figure 18) or dry-running possibility. Another prime opportunity for problem-solving by this pump design is problematic seals—especially where upgrading to double mechanical seals is costly, complicated, or unacceptable from the potential environmental spill issues.
- Make sure that gas (nitrogen) is not interrupted at any time. Do not turn the gas off, even when a pump is not operating. Drain the pump completely before turning the gas off for maintenance or reassembly. Gas is the last thing to be turned off, just prior to disassembly.
- This pump is especially helpful in cases involving:
  - Nasty, toxic, or expensive chemicals.
  - Up to 40 percent of solids.
  - Possibility of dry-running.
  - Cavitation, NPSH runout, and flashing.
  - Curve "racing"—the pump shaft rotor stiffness resists deflections due to high radial thrust at low flow operations.
  - Applications where pump feeds flow control valve that will run the pump to near zero flow and/or well out on its curve.
- The mechanical gas seal is the critical part of the air barrier pump. The short stationary seal holding pins were the problem with this installation. By working together, the manufacturer, distributor, and the end-user were able to resolve this issue and further improve on the barrier design.

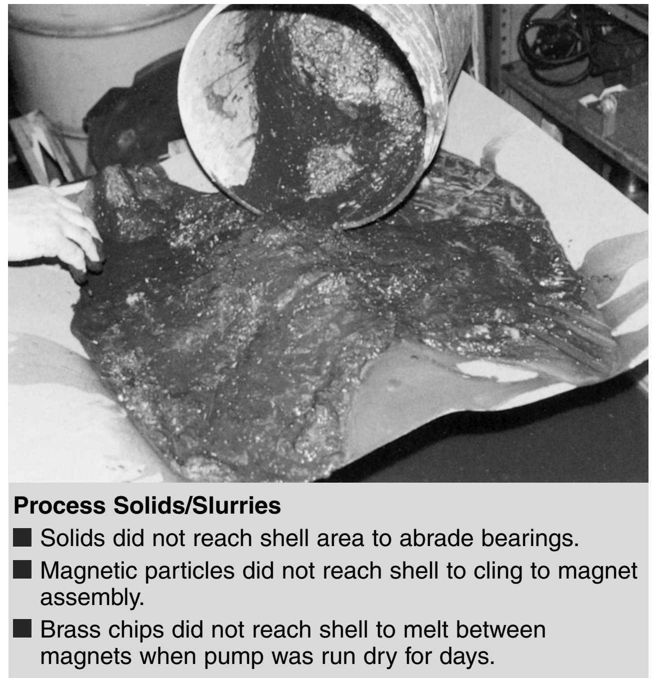


Figure 18. Extremely Contaminated Process Fluids Can Be Pumped with Barrier Mag-Drive Pumps.

ACKNOWLEDGEMENTS

The authors wish to thank their colleagues and management for providing valuable comments, reviewing the manuscript, and editorial work on figures and illustrations.

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