DEVELOPMENT AND QUALIFICATION OF A MAGNETICALLY COUPLED PROCESS PUMP FOR API 610 7TH EDITION SERVICES

by
Charles C. Heald
Chief Engineer
Paul W. Behnke
Manager of Product Engineering for Industrial and Process Pumps
and
Richard F. Gussert
Design Engineer
Ingersoll-Rand Company
Engineered Pump Division
Phillipsburg, New Jersey

Charles C. Heald is Chief Engineer of the Engineered Pump Division of Ingersoll-Rand Company, Phillipsburg, New Jersey. In this position, he is responsible for the design and development of new products along with design engineering policies, standards and procedures for engineered pump products worldwide.

Mr. Heald received a B.S. degree in Mechanical Engineering from the University of Maine (1960). Since that time, he has held various positions with Ingersoll-Rand, including Development Engineer, Senior Hydraulic Design Engineer, Manager of Pump Design, and Manager of Engineering. He is a member of ASME and the API Centrifugal Pump Manufacturers' Subcommittee (Vice Chairman), and he serves on the API 610 Task Force and the API Seals and Sealing Systems Task Force. He has authored several articles and technical publications and holds several U.S. patents on centrifugal pumps.

Paul W. Behnke is Manager of Product Engineering for Industrial and Process Pumps in the Engineered Pump Division of Ingersoll-Rand Company, Phillipsburg, New Jersey. His current responsibilities include design engineering, new product development, cost control, engineering system standards, and engineering support of the marketing, manufacturing, and field service functions. Mr. Behnke has been involved in numerous product development projects including boiler circulating pumps, process slurry pumps, power recovery turbines, barrel-type pumps, and high speed pumps.

Mr. Behnke is a Registered Professional Engineer in the State of New Jersey and an Associate member of ASME. His education includes a BSME degree from Rutgers University and graduate studies at Lehigh University, where he is currently pursuing an MBA.

Richard F. Gussert is a Design Engineer with the Engineered Pump Division of Ingersoll-Rand Company, Phillipsburg, New Jersey. His responsibilities include all phases of the development of a magnetically coupled process pump for API 610 7th Edition services. Previously, he held Application Engineer and Sales Engineer positions with the Air Compressor Group of Ingersoll-Rand.

Mr. Gussert is a graduate of the University of Wisconsin with a B.S. degree in Mechanical Engineering.

ABSTRACT

A magnetically coupled process pump configuration has been developed to provide strict compliance with API Standard 610, 7th Edition, and Section VIII of the ASME Code by rigorous reading and interpretation of these standards in light of their intent. The result is a sealless pump with provision for a backup mechanical seal (or close fitting sealing bushing) and an outer housing that withstands full working pressure as a secondary containment means. Included are optimized arrangements of recently recognized innovations, including nonmetallic (eddy current eliminating) shells, performance monitoring instrumentation, laser shaft treatments, and a circulation path to lubricate the bearings and cool the magnet area within the shell.

INTRODUCTION

In recent years, the advancement of magnetically coupled pump technology has posed new and interesting challenges to both pump engineers and pump users. Many challenges have been met with innovative designs, materials, and application techniques. For example, since 1978, rare earth magnets have enabled engineers to attain a fourfold improvement in the amount of power transmitted within a given space envelope, while increasing the drive efficiency as much as 80 to 90 percent (or more) [1]. In 1990, a unique self aligning thrust bearing design was introduced which allows semi-open impellers to be utilized in magnetically coupled chemical pumps which meet the ANSI B73.1 standard dimensional envelope [2].

The application of magnetically coupled pumps to refinery services poses yet another set of challenges. Apart from the generally severe nature of these services, magnetically coupled pumps need to meet requirements of API Standard 610, even though magnetically coupled pumps were not a category of pumps considered in the writing of the latest (7th) edition of this refinery pump standard [3]. It is the responsibility of the pump manufacturer to evaluate the severe service requirements of this equipment class, along with the industry standard specification requirements, and to perform appropriate analytical studies and testing to ensure that risks to the pump user are minimized.

The Initial Decision

The general design of magnetically coupled pumps has been described many times before [1, 5], and it is assumed at this point that the reader has a basic understanding of the concepts. In addition to these concepts, the process pump "design team" formed in 1990 had, at its inception, the benefit of two years of development, testing and production experience with magnetically coupled chemical pumps [2]. At the earliest stage, the team had to make a fundamental decision: whether to utilize the existing chemical pump magnetic drive system directly, or design a magnetic drive system specifically for API Standard 610 process pumps.

Considerable development cost and time could have been avoided by utilizing the existing chemical pump magnet drive system. For example, the same bearing housing patterns could have been used for both the chemical pump and the process pump. But, in order to do this, a series of design compromises would have been necessary, the sum of which did not address the challenges of API refinery services. Furthermore, significant differences between chemical and process pump envelope dimensions were identified (Figure 1). This compromise was unacceptable because interchangeability between magnetically coupled process pumps and conventional process pumps was judged to be an important feature. The decision was made to proceed with the design of a drive system unique to API Standard 610 process pumps, which in effect, reapplies existing magnetic drive technology without utilizing chemical pump bearing housings.

The Challenges

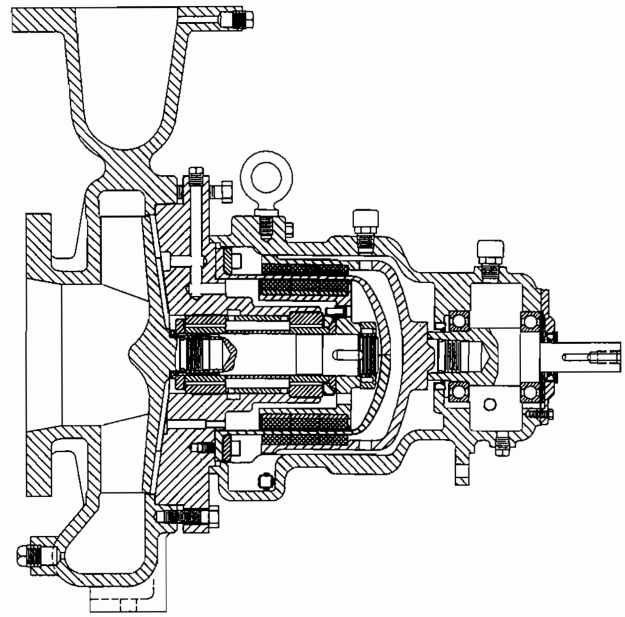
Specifically, the challenges in producing magnetically coupled pumps to API Standard 610, 7th edition requirements include five crucial areas that have been identified by surveys of sealless pump users [4]:

- pressure containment
- cooling and NPSH considerations
- pumped liquid lubricated bearings
- special instrumentation
- high temperature service considerations.

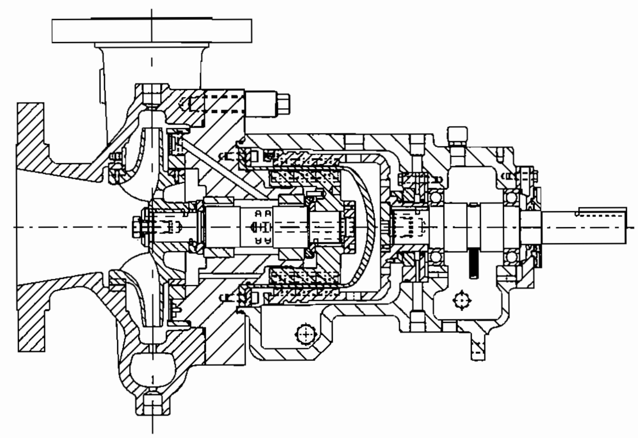
Engineering approaches for dealing with these challenges by designing magnetically coupled pumps specifically for refinery services and specification requirements are described in the following pages. These approaches add credibility to the sealless pump configuration for such services.

PRESSURE CONTAINMENT CONSIDERATIONS

API Standard 610, 7th Edition, does not specifically consider sealless pump configurations. It does, however, give very specific requirements for the "pressure casing." It is important, if a magnetically coupled sealless pump is to be considered an "API 610 pump," that it address the pressure casing design requirements as outlined in the API 610 Standard.



CHEMICAL PUMP (ANSI B73.1 ENVELOPE DIMENSIONS)



PROCESS PUMP (CONVENTIONAL PUMP ENVELOPE DIMENSION

Figure 1. Magnetically Coupled Pump Cross Sections.

The Pressure Casing

API Standard 610 requires that design stresses not exceed the values given for material in Section VIII, Division 1, of the ASME Code [7]. For cast materials, the code quality factor shall be applied. Pressure casings of forged steel, rolled and welded plate, or seamless pipe with welded cover also need to comply with the applicable rules of Section VIII. In addition, API Standard 610 requires that the "pressure casing" be designed for the pump maximum discharge pressure, at the pumping temperature, with a minimum corrosion allowance of ⅛ in (3.2 millimeters). Internal bolting must be of a material fully resistant to corrosive attack by the pumped liquid, and radially split casings must have metal-to-metal fits with confined, controlled compression gaskets.

By API 610 definition, the pressure casing "..is the composite of all stationary pressure-containing parts of the unit, including all nozzles, seal glands, and other attached parts but excluding the stationary and rotating members of mechanical seals..." If a new pump configuration (such as a magnetically coupled design) does not contain these specific components, the pump engineer must determine how the new configuration can be compared to a conventional pump assembly, how the two relate to one another, and how the API definition and design requirements apply.

The Conventional Mechanical Seal

The conventional process pump utilizes a mechanical seal to prevent pumpage from escaping out of the pressure casing along the shaft. Depending on the characteristics of the liquid (flammability, toxicity) and the environmental impact of leakage to the atmosphere, these mechanical seals and associated sealing systems can range from being quite simple to extremely complex.

In its most basic form, the mechanical seal consists of stationary and rotating sealing faces, retained by a rotating seal sleeve and a stationary seal gland, both containing static sealing materials (gaskets or O-rings). When the pumped liquid is flammable or toxic, or when leakage to the atmosphere is otherwise undesirable or intolerable, a tandem seal arrangement is typically used (Figure 2). In this arrangement, the seal faces nearest the pumpage (the primary faces) are intended to do the actual sealing under normal operation. If leakage occurs between the primary seal faces, this leakage is contained by the secondary seal faces. These secondary seal faces must be designed to the same pressure-retaining criteria as the primary faces.

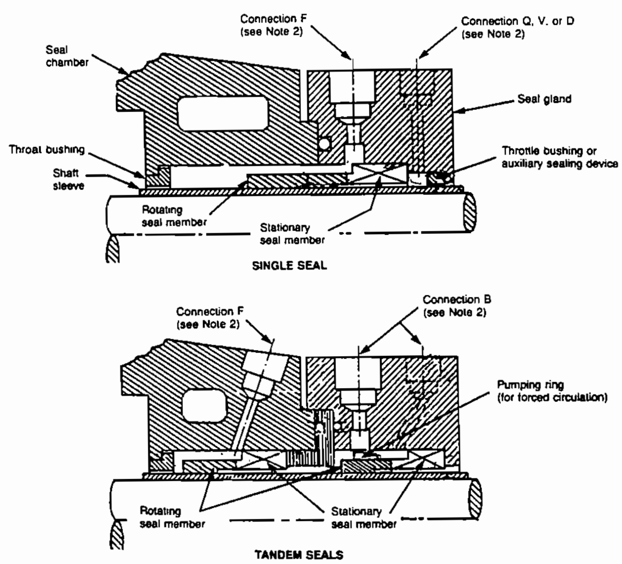


Figure 2. API Mechanical Seal Arrangements.

The seal gland which supports the stationary seal faces is, by definition, a part of the "pressure casing." It also supports the "secondary containment," which, in the case of the single mechanical seal, is a throttle bushing. This bushing provides a close clearance "seal" around the shaft (or shaft sleeve) that prevents massive leakage to the atmosphere in the event of total primary seal failure.

In normal operation, the secondary seal faces of a tandem mechanical seal must have liquid present to lubricate the faces and to carry away the heat generated at these faces. This is typically accomplished by installing a piping system between the seals which includes a tank containing barrier fluid, elevated above the seals. API Standard 610 requires positive circulation of the barrier fluid by means of a circulating device such as a pumping ring. The secondary seal piping system also allows for the installation of alarms and signals to warn if the primary seal begins to leak into the secondary circuit, and the provision for cooling coils in the barrier fluid tank to remove the heat generated at the seal faces.

If the mechanical seal is a high integrity, very low leakage design, it may not be necessary to provide a tandem arrangement, with the associated support tanks and piping systems. In this case, the seal gland is arranged with a throttle bushing, which is sometimes referred to as a "disaster" bushing (Figure 2). With such an arrangement, the area between the seal faces and the throttle bushing is typically piped (or vented) to a "safe" place.

Note that, in any form, the seal gland is always, by definition, a part of the pressure casing.

The Mechanical Seal Analogy

Having described the typical mechanical seal arrangements that are utilized by API 610 pumps today, and the arrangements that were considered in the writing of the API 610 Standard, the authors consider how the magnetically driven pump can be compared to these conventional configurations in order to establish what its design characteristics must be.

Think of the magnetically driven pump primary containment shell (the pressure containment component between the driving and the driven magnets) as replacing the seal faces in a pump with a conventional mechanical seal. API Standard 610, 7th Edition, paragraph 2.7.1.3 reads as follows:

> This standard does not cover the design of the component parts of mechanical seals; however, the design and materials of the component parts shall be suitable for the specified service conditions. The components shall also withstand the maximum discharge pressure (see 1.4.5), except in high-discharge-pressure service where this requirement is impractical. For such applications, the vendor shall advise the purchaser of the maximum sealing pressure and the seal's maximum dynamic and static pressure ratings.

The major difference between the seal faces of a conventional mechanical seal and the shell of a magnetically driven pump is that the mechanical seal faces leak (albeit small amounts when they are working properly) and the shell of the magnetically driven pump does not. The design and material of the shell must be "suitable for the specified service."

In the case of the magnetically coupled process pump, when the conventional mechanical seal "faces" are replaced with a "shell" (Figure 3), this shell becomes the "primary containment" element, similar to the primary sealing faces of the mechanical seal.

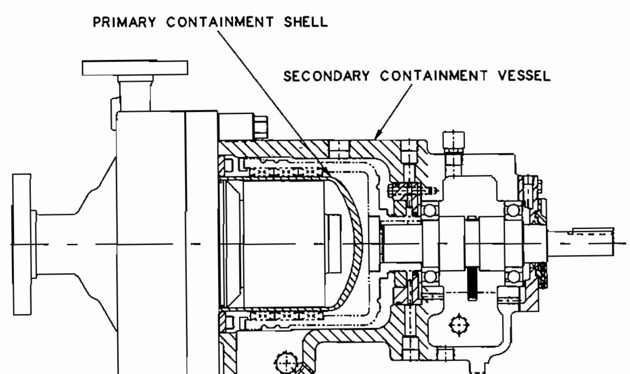


Figure 3. Closeup of Shell and Secondary Containment.

In the case of nonmetallic or composite materials, where allowable stress levels (or even realistic methods of design and analysis) are not addressed by Section VIII, Division 1, of the ASME Code, the maximum allowable working pressure may be established by a hydrostatic proof test criteria. Specifically, the maximum allowable working pressure is equal to one-half the minimum hydrostatic proof test pressure at maximum pumping temperature.

Paragraph 2.2.2 of API Standard 610 requires that ". . the casing shall be designed for the maximum discharge pressure plus allowances for head and speed increases (see 2.1.4. and 2.1.5) at the pumping temperature, with a minimum corrosion allowance of ⅛ in (3.2 millimeters)." Because the torque capability of the magnetic coupling is greatly affected by both the gap between the driving and driven magnets and the eddy current losses in the shell material itself, it is extremely important that the overall gap between the driving and driven magnets is kept to a minimum [2]. One effort to reduce eddy currents introduces fairly thick, laminated components into a larger air gap, resulting in a coupling with less torque capability [6].

To facilitate this requirement, and still maintain compliance with the objectives of the API Standard, the following approach is taken:

- The material of the primary containment shell is selected to substantially eliminate corrosion as a serious factor in the application/service. Although there are different views as to what constitutes an acceptable corrosion rate in a particular environment, 20 to 30 mils/year (MPY) are often accepted as maximum values [8]. It follows, then, that if a shell material is selected that has a corrosion rate of less than two MPY, the need for a ⅛ in (3.2 millimeter) corrosion allowance on the shell is substantially eliminated.

- Following this reasoning, the thickness of the shell is designed to comply with the stress levels as given in Section VIII, Division 1, of the ASME Code without adding the ⅛ in (3.2 millimeter) corrosion allowance.

- The primary containment shell is completely surrounded by a secondary containment vessel (see Figure 3). This secondary containment vessel is formed by the steel frame of the bearing support system for the driving magnet carrier.

The secondary containment vessel qualifies as a part of the "pressure casing" because it is designed to withstand the maximum allowable working pressure of the pump with a ⅛ in (3.2 millimeter) corrosion allowance, and it is hydrotested to 1-½ times this pressure. The design stresses used for this secondary containment vessel do not exceed the values given in Section VIII, Division 1, of the ASME Code. Thereby, the analogy between the magnetically driven pump and a conventional mechanical seal is complete from a pressure containment standpoint; the secondary containment vessel is compared to the conventional seal gland.

Secondary Sealing Devices in Magnetically Coupled Pumps

The question that remains is whether the service (or the user) requires a single seal arrangement represented by the primary containment shell and a bushing, or a tandem seal arrangement represented by the primary containment shell and a mechanical seal, given the high integrity of the primary containment shell.

In the instance where a high-integrity single seal with a throttle bushing is acceptable for the application, the comparable magnetically driven pump would be equipped with a simple close clearance sealing bushing, installed in the secondary containment structure (Figure 4). Normally the secondary containment vessel would be vented through the sealing bushing. This arrangement is the most commonly used.

To allow the magnetically driven pump to assume the features of a tandem seal pump, an optional mechanical gas seal is installed in the secondary containment vessel in place of a sealing bushing (Figure 4). This conventional gas seal is arranged such that it normally operates "open." This means that, during normal operation, the secondary mechanical seal faces do not contact. In the event of a failure of the primary containment shell, the secondary chamber would flood with pumpage. In the presence of liquid or vapor, the secondary mechanical gas seal would be activated, and

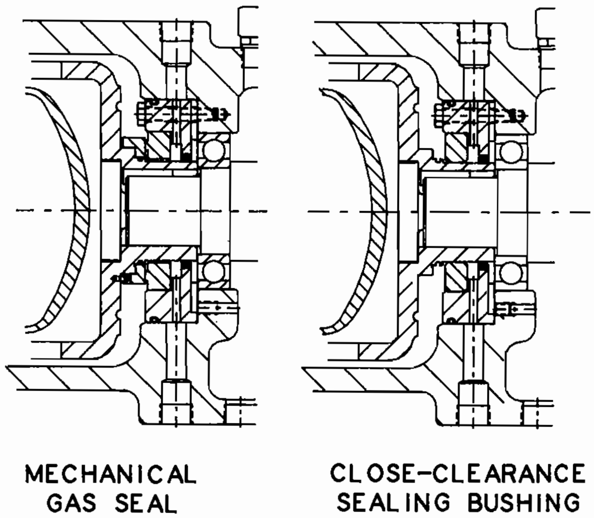


Figure 4. Secondary Pressure Containment Concept.

it would provide sealing capability similar to that of the secondary sealing element in a conventional tandem mechanical seal. The mechanical gas seal is designed to seal maximum discharge pressure—the same as the design criteria for conventional tandem mechanical seals. The difference is that the secondary mechanical gas seal is designed to normally operate without the need for a barrier liquid to carry the heat away. Because the gas seal normally operates open, the sealing faces do not contact, and no significant heat buildup occurs.

To summarize, the magnetically coupled sealless pump can be compared to a conventional pump with a conventional mechanical seal. The conventional pump with a single mechanical seal and throttle bushing in the seal gland equates to the magnetically coupled sealless pump with a sealing bushing. The conventional pump with a conventional tandem seal arrangement equates to the magnetically coupled sealless pump with a mechanical gas seal. The advantage of the sealless pump design: the primary "seal" has zero leakage.

COOLING AND NPSH CONSIDERATIONS

The magnetically coupled pump designs produced today use the pumped liquid to lubricate the bearings and to cool the area surrounding the magnets. Cooling of the magnet area is required because eddy current losses in metallic primary containment shells generate heat. As previously discussed, the amount of heat generated by the eddy current losses is a function of the containment shell material and thickness, with nonmetallic shells eliminating these losses [2]. Approximately one-half of this heat is transferred from the shell by convection to the pumpage circulated through the magnet area.

The Circulation System

Internal circulation is typically either from discharge to discharge pressure, utilizing an internal pumping device, or from discharge to suction pressure, where, because of the heat acquired by the circulating flow, problems with vaporization in the magnet area or in the impeller inlet may occur (Figure 5). A third possibility exists, however, and that is circulation from discharge to an intermediate pressure, which inhibits vaporization and eliminates the need for a power absorbing pumping device to force the internal circulation.

Developmental testing has confirmed the effectiveness of this discharge-to-intermediate pressure circulation system to circulate a significant volume of liquid (Figure 6). The flow path is designed

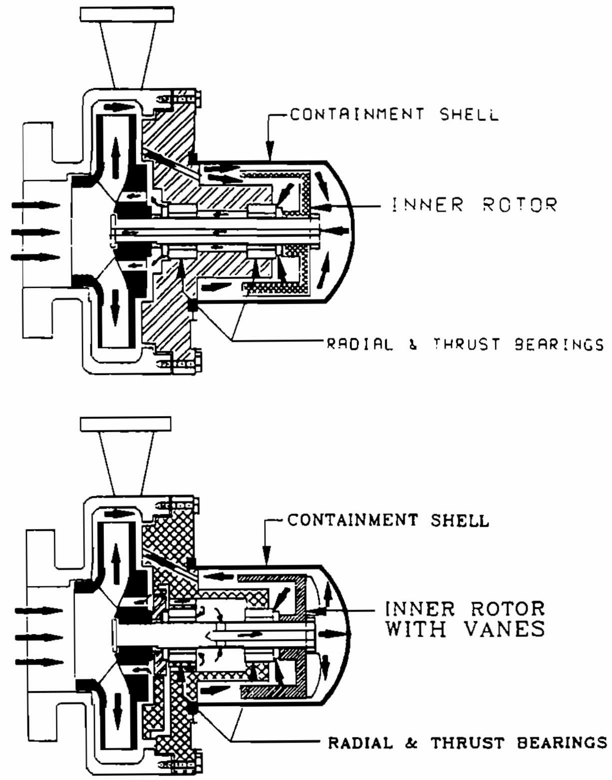


Figure 5. Other Circulation Systems.

to provide proper, positive lubrication to the product bearings, while, at the same time, providing sufficient flow volume around the drive magnets to eliminate heat buildup in the containment shell without a significant rise in circulation fluid temperature [1]. Additionally, the circulation flow is reintroduced to the main flow stream through the pump casing and impeller in a manner which results in negligible temperature rise at the pump suction, thereby having no effect on required NPSH margins and minimum continuous flow values.

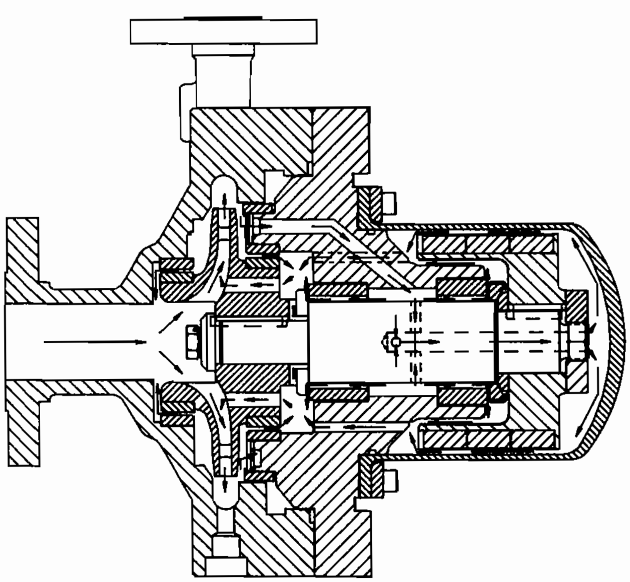


Figure 6. Discharge to Intermediate Pressure Circulation System.

To confirm and qualify the correctness of the chosen circulation system, testing was performed using a $3 \times 4 \times 10$ pump, with a Hastelloy C containment shell, operating at 3550 rpm and 88 bhp on a closed water test loop. By cross drilling passages in the casing cover, it was possible to access key areas inside the wet end of the pump to collect actual temperature and static pressure data during pump operation (Figure 7). In addition to testing with internal circulation, a series of tests were run while injecting pumpage from the pump discharge (from a tap in the side of the discharge nozzle) into the circulation passages through a digital flow meter (Table 1) mounted in the injection line. Using data from previous tests, it was possible to simulate internal circulation with the measured flow from the flow meter by matching the pressure distribution inside the pump. These data confirmed the actual magnitude and the direction of the flow path.

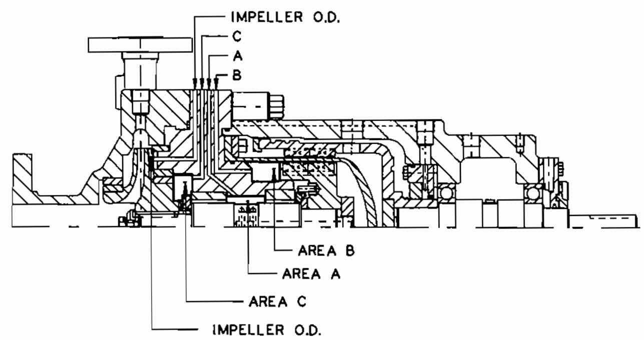


Figure 7. Circulation System Location Designation.

Table 1. Test Equipment.

TEST EQUIPMENT	MANUFACTURER	ACCURACY
DISCHARGE PRESSURE TRANSDUCER	VIATRAN	+/- 0.04%
SUCTION PRESSURE TRANSDUCER	VIATRAN	+/- 0.04%
FLOW CIRCULATION METER	FISHER	+/- .01 GAL
TEMPERATURE - RTA'S	THERMO ELECTRIC	+/- 2 DEG

During pump operation, flow is circulated to the bearings and magnets from a high pressure area near the outside diameter of the impeller. The pressure at this location in the pump is normally 75 to 80 percent of the total head (TDH) generated by the pump. After the flow passes through a self-cleaning strainer, it enters a cavity between the two product lubricated pump bearings. The flow is then split into three parallel paths, each of which will be described in detail. Note that each bearing is lubricated with cool fluid; the fluid has not previously been used to cool the magnet area, thereby minimizing the danger of creating a flashing situation in the pump bearing area.

The Flow Paths

The three parallel flow paths of the discharge-to-intermediate pressure circulation system are described as follows:

Parallel Path (1). Approximately one-quarter of the total circulation flow is directed through the front bearing clearance to the cavity below the wear ring diameter at the rear of the impeller (Figure 7, area C). This flow is used for the lubrication of both the radial and axial bearing faces. It also removes the relatively small amount of heat generated by the loaded bearing surfaces. The substantial pressure differential between the two areas creates a significant volume of flow through the bearing set.

Parallel Path (2). The second parallel path starts out in a manner very similar to the first, in that it is used to lubricate and cool the rear bearing set. This flow volume is also approximately one-quarter of the total volume of circulation flow. After it passes through the bearings, however, this flow mixes with the flow from Path (3) in the containment shell (Figure 7, area B) before entering a passage leading to area C, where the combined flow mixes with flow from Path (1) and wear ring leakage.

Parallel Path (3). The remaining circulation flow volume (about one-half of the total flow) is passed through a drilled shaft passage which leads to the rear of the containment shell. Here it picks up the heat created by the eddy-current losses in the shell (if any) before mixing with the flow from Path (2) and entering the passage leading to area C. The temperatures of the fluid in areas B and C are shown in Figure 8.

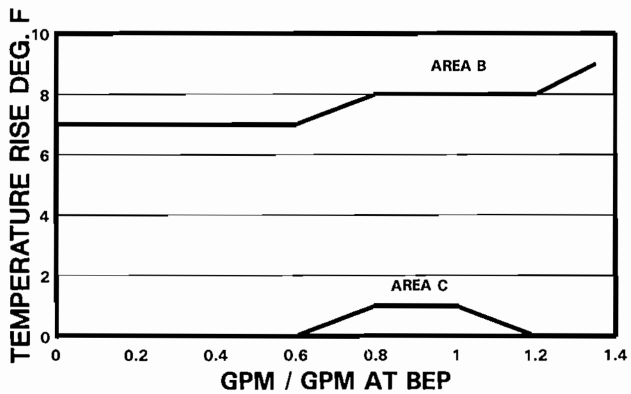


Figure 8. Temperature Rise vs Pump Discharge Flow.

Effects on NPSH

The temperature rise in area C (Figure 7) represents the key to the success of the discharge-to-intermediate pressure circulation system described. The entire circulation flow volume terminates in area C, after providing lubrication and cooling in the containment shell area. A common perception is that high temperature fluid is flowing to suction pressure. Obviously, if this were to occur, the pump NPSHR would have to increase to suppress flashing of the higher temperature liquid; in severe temperature rise instances, even the pump minimum flow might have to be increased to avoid cavitation in the impeller inlet.

With the arrangement described here, the assumption that high temperature fluid is being returned to suction pressure is not correct, for two very important reasons. First, the balance holes in the impeller are located in the impeller passage at a diameter greater than the diameter of the inlet vane tips. This means that some head, or pressure, has already been generated by the impeller, and the pressure in area C is actually higher than suction pressure (pressure at the pump inlet flange) by approximately 12 percent of the TDH (Figure 9).

Secondly, there is a substantial volume of flow through the back impeller wear ring clearance. This leakage flow is typically an order of magnitude greater than the total circulation flow volume through the pump bearing and magnet area, and, therefore, it is the dominant factor in determining the temperature in area C. From the tests performed on the $3 \times 4 \times 10$ test pump, it was determined that the actual temperature of the mixture in area C was only about one degree (F) above the injection temperature (Figure 8). Therefore, the pressure and temperature distribution in this area of the described magnetically coupled process pump is essentially equal to that found in a conventional process pump, and the NPSH margin is unaffected by the lubrication and cooling in the containment shell area.

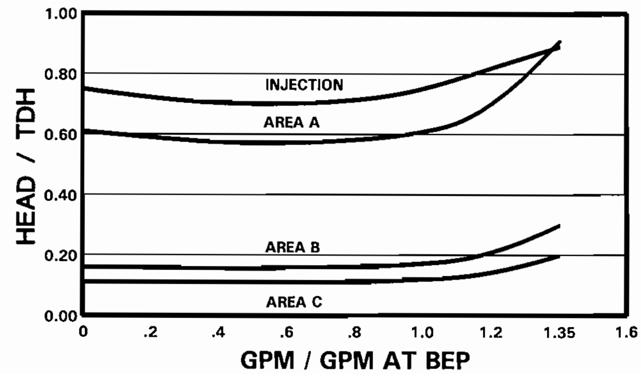


Figure 9. Static Head vs Pump Discharge Flow.

Minimum Flow Criteria

API Standard 610 defines minimum continuous flow in two ways:

- minimum continuous stable flow
- minimum continuous thermal flow

Stable flow limitations are largely a function of pump hydraulic design and are not related to whether or not a pump is magnetically coupled [9].

Thermal limitations are defined as "...the lowest flow at which the pump can operate without its operation being impaired by the temperature rise of the pumped liquid." In conventional process pumps, operation is typically impaired in this regard when the liquid in the pump suction vaporizes. Magnetically coupled process pumps fitted with nonmetallic containment shells have the same minimum flow requirements as corresponding conventional pumps.

Although substantial material developments have been made, and will continue to be made, in the nonmetallic shell area, some pump users will continue to require metallic containment shells [2, 10]. Therefore, pump engineers must design circulation systems which 1) have substantial, positive flow through the magnet area to minimize temperature rise and 2) maintain back pressure on the heated liquid until it can be mixed with the cooler pump main flow stream in a controlled manner.

PUMPED LIQUID LUBRICATED BEARINGS

A very important consideration in magnetically coupled pump design is the choice of materials for the pumped liquid lubricated bearings which support the wet end rotor assembly.

Comparison of Chemical and Physical Philosophy

To date, the majority of magnetically coupled pumps have been designed for and installed in chemical pump applications. The application of this technology to API Standard 610 process pumps has been limited, and, in general, most initial process pump designs have simply utilized chemical pump bearing materials and configurations. It is important to evaluate the differences in these services and the effect of these differences on the bearing requirements. This discussion will concentrate on corrosion properties and entrained solids, and the effect of these on the bearing material selection.

Since chemical pump applications are typically very corrosive, high alloy stainless steels and other relatively exotic materials are often required. In addition, many of these applications contain entrained solids consisting of "stringy" materials, which have a tendency to clog enclosed impeller passages. Semiopen impellers are less susceptible to clogging than enclosed impellers with close clearance wear rings, and they handle "stringy" solids more effectively; therefore, they have become the standard for chemical pumps.

In contrast, API 610 services very rarely require materials with corrosion resistance beyond that of 300 series stainless steels, although in many cases the liquids are not as "clean" as those in chemical services. Many applications include entrained solids in the form of particulate, which can be very abrasive. API Standard 610, 7th Edition requires the use of enclosed impellers with wear rings. When enclosed impeller with front and back wear rings are used, balance holes reduce the pressure on the back of the impellers, resulting in axial thrust loads significantly lower than those typically experienced in comparable chemical pumps with semiopen impellers.

Application Properties Affecting Bearing Material Selection

Property	*Chemical Pump*	*Process Pump*
Corrosiveness	Highly	Mildly
Entrained solids	"Stringy" materials	Particulates
Optimum Impeller type	Semiopen	Enclosed with balance holes
Axial thrust loads	High	Nominal

Process Pump Performance

Silicon carbide has been the material of choice for magnetically coupled pump bearings. It is a ceramic material with excellent corrosion resistance, wear properties, and load carrying capabilities. Its major drawback is that proper lubrication and exceptionally smooth surface finishes are absolutely essential for operation without failure. Additionally, the parallelism of the bearing faces is critical to ensure proper operation of the thrust bearing system. In short, silicon carbide bearings tend to be very "sensitive" to their operating condition.

Many magnetically coupled sealless pump failures have been caused by lack of flow to the silicon carbide pumped liquid lubricated bearings. Since the bearing lubrication is normally diverted from the main pump stream, this "run-dry" condition can occur for a variety of reasons, including incorrectly positioned valves (closed instead of open) or loss of suction pressure.

Bearing materials which are much more durable in these "run-dry" conditions, and thus eliminate a very common cause of failure, are available. Carbon and carbon graphite materials, impregnated with an appropriate metal, have self lubrication properties that allow them to withstand run-dry situations for relatively short periods of time without sustaining damage. They have proven to be very successful in a variety of applications, including wear rings and pumped fluid lubricated bushings, in other API process pumps.

The corrosive nature of the services in the chemical industry has been the biggest factor in limiting the use of these impregnated bearing materials in magnetically coupled pumps. This very large obstacle does not exist in most refinery services, which leaves two areas of concern to address, namely, axial thrust loads and entrained particles.

Axial Thrust

A study was conducted to compare the axial thrust of the described magnetically coupled process pump design, which uses an hydraulically balanced enclosed impeller and wear rings, to that of an equivalent chemical pump design, which uses a semiopen impeller with no balance holes or wear rings. The components of axial thrust are 1) impeller thrust and 2) thrust from pressure on the inner magnet carrier caused by circulation of fluid in the containment shell.

The results of the study show the axial thrust loads for magnetically coupled process pumps are typically less than 50 percent of those for equivalent chemical pumps. Additionally, the use of the self-aligning thrust bearing system previously developed and proven in the chemical pump (Figure 10) assures parallelism of the thrust faces [2].

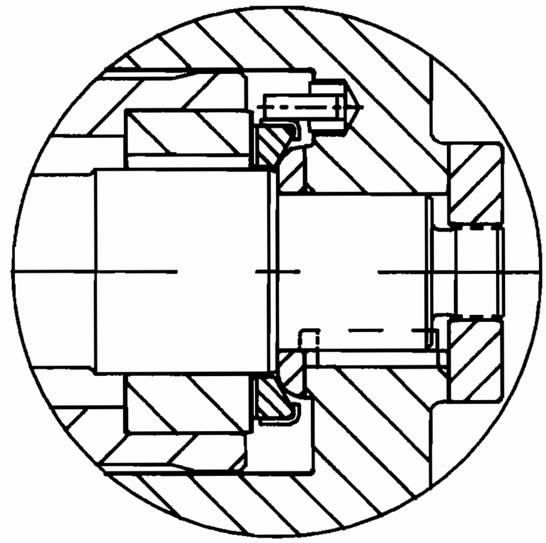


Figure 10. Self Aligning Thrust Bearing System.

Successful testing has been completed using the magnetically coupled chemical pump with maximum thrust loads and impregnated carbon graphite bearings. During qualification testing, the pump ran for several hundred hours with no measurable wear of the bearings, just a polishing of the thrust surface. These results, teamed with the lower thrust loads of the process pump, provide a conservative service factor for the bearing system loading.

Entrained Particles

Entrained particulates can increase bearing wear of the softer impregnated materials, especially when the particles are similar in size to the bearing radial clearances. To ensure that these particles are eliminated from the lubricating fluid, a non-clogging strainer should be used to filter the circulating fluid. This accomplished by incorporating a wash flow strainer into the standard process pump.

The Wash Flow Strainer Feature

One of the special features incorporated in the standard process pump configuration is the "wash flow" strainer (patent applied for by Ingersoll-Rand Company). The concept of the strainer, located near the periphery of the pump impeller (Figure 11), is that the natural flow of the pumpage in the passage between the casing cover and the back shroud of the impeller "cleans" the strainer and prevents the buildup of foreign material on the surface of the screen, as happens in a conventional screen strainer configuration. The purpose of the strainer is to prevent the passage of relative large particles of solids into the bearing and magnet areas within the containment shell section of the pump.

Wash Flow Strainer Testing

The wash flow strainer concept was tested for effectiveness on a closed loop water and sand test rig. The loop consisted of a tank and a simple system of pipe from the pump discharge, through the tank and back to the pump suction, with provision for the addition of sand to the suction of the pump while the rig was operating. The

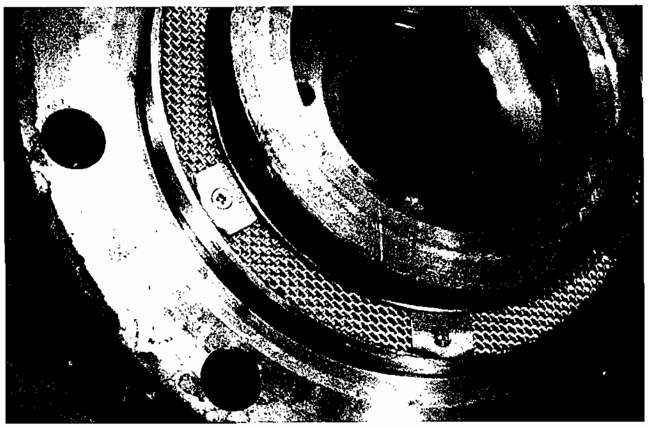


Figure 11. Wash Flow Strainer.

strainer is a 100 mesh matrix of triangular cross section wire (approximately 0.006 in openings), sized to be consistent with internal bearing clearances of 0.003 - 0.005 in dimetral. The sand had a size range from less than 270 mesh to greater than 40 mesh, with 80 percent in the 100 to 200 mesh size range. The testing consisted of operating the pump near its design flow rate (approximately 50 gpm at 3550 rpm for a 1 × 1-½ × 6 pump) for approximately three hours while introducing approximately 15 lbs of sand directly into the test loop at the suction to the pump. Although this does not give accurate concentration data, it does represent a concentration of solids in the liquid (water, not a liquid such as a hydrocarbon that might contribute positive lubricating properties to the mixture) that are greater than expected to occur in actual operation in a refinery or process plant. It more realistically represents a "shock treatment," just to find out "what would happen if. . . "

Results of the test were encouraging. After three hours of operation, the pump was disassembled and inspected. It was found that the hydraulic passages of the pump impeller and volute were highly polished (the pump was constructed of austenitic stainless steel). There was no evidence of this polishing in the bearing and magnet areas of the containment shell; the bearings themselves (made of silicon carbide) showed no signs of polishing or wear of any kind. There was a significant amount of "mud" in the passages of the strainer itself. This "mud" appeared to consist of sand particles nearly the size of the strainer mesh. It was concluded, since there was no measurable effects on pump performance during or after the introduction of the sand, that the strainer very effectively eliminated the larger sand particles from the containment shell area. It was most likely that the finer particles (less than 100 mesh) washed through the containment area almost completely, not accumulating and not even polishing the surfaces which were in contact with the liquid-sand mixture. The only concern was with the high concentration of particles which were of the same magnitude (size) as the openings in the strainer; these tend to clog the strainer and reduce its effectiveness. This, of course, could even result in blocking the flow to the containment area if allowed to continue without inspection.

Guidelines have been established for the application of the integral wash flow strainer concept to address the possibility of clogging in the event of either known (or suspected) high concentrations of solids comparable in size to the strainer openings, or even for applications where the pumped liquid viscosity is unusually high (in excess of 1500 SSU) or the liquid could polymerize. When such situations are present, it is recommended that the wash flow strainer be used only for startup protection. For particle sizes other than those nearly equal to the strainer mesh size, however, the test confirmed the wash flow concept to be extremely effective.

SPECIAL INSTRUMENTATION CONSIDERATIONS

Instrumentation to monitor vibration, bearing temperature, mechanical seal leakage and other pump performance characteristics is becoming commonplace on conventional pumping equipment. Monitoring key performance factors has proven beneficial to pump reliability and availability, and such practices are rapidly becoming an integral part of plant operating systems.

Magnetically Coupled Pump Instrumentation

General instrumentation requirements for magnetically coupled pumps are well outlined in previous papers [4, 5]. Particular importance is placed on pump protection from a dry-run condition either as a result of loss of suction or heat buildup from low flow operation, or from magnet drive decoupling. At least one pump manufacturer recommends temperature monitoring of the circulation system with direct control of the motor circuit to facilitate pump shutdown in the event that vaporization occurs in the magnetic area [5]. In addition, monitoring of pump horsepower is highly recommended. Loss of pump horsepower is the quickest, most direct indicator of loss of suction and decoupling. Motor current sensors are used for this purpose.

Obviously, from a refinery operations viewpoint, pump protection based on pump shutdown is not desirable. It is, therefore, incumbent on pump manufacturers to develop technologies which will improve the dry-run capabilities of magnetically coupled pumps.

API Instrumentation

API Standard 610 does not specifically address instrumentation requirements for magnetically coupled pumps. However, by applying the analogy between conventional mechanical seals and the secondary containment concept discussed earlier, it does provide guidelines for leakage detection instrumentation.

Secondary seal piping systems, such as API Plan 52, are specifically defined in API Standard 610 (Figure 12) and further enhanced by pump users' specifications. These systems provide lubrication to the faces of conventional seals and are instrumented to monitor seal leakage. Similar instrumentation can be provided between the primary containment shell and the secondary containment vessel to detect leakage from the shell (Figure 13).

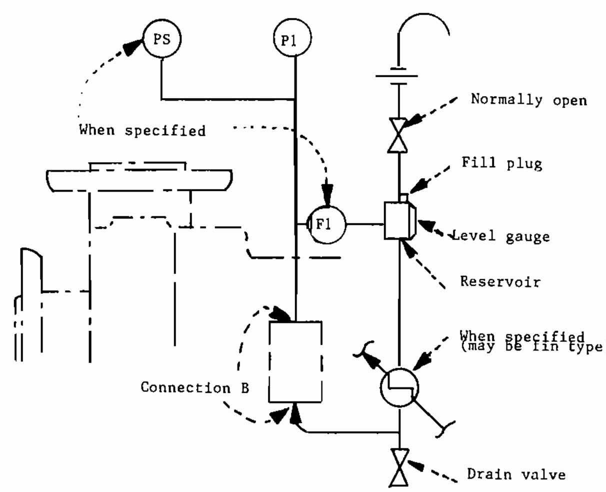


Figure 12. API Plan 52 Secondary Seal Piping System.

Leakage detection on magnetically coupled pumps is simpler than conventional seals with typical API Plan 52 seal piping systems for two reasons. First, seal tanks and interconnecting

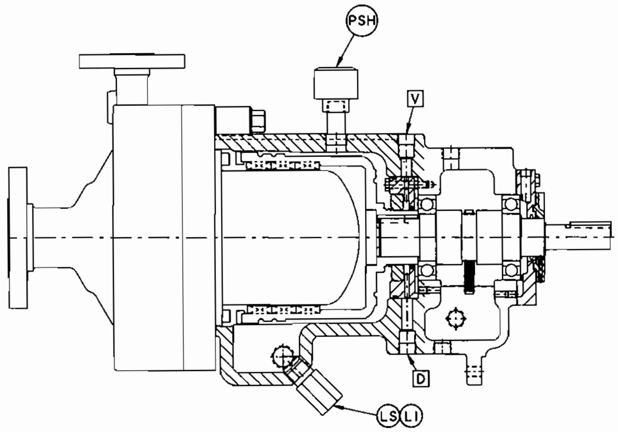


Figure 13. Secondary Pressure Containment Instrumentation.

piping are eliminated because there is no barrier fluid for lubrication. Therefore, the instrumentation is installed directly on the pump's secondary containment vessel. Second, the integrity of the primary containment shell as a pressure containing device far exceeds that of any conventional seal; the shell has zero leakage. Therefore, routine maintenance of a magnetically coupled pump leakage detection system is not required. Nevertheless, leakage detection is important because magnetically coupled pumps are not designed to operate with pumpage in the secondary containment vessel in the unlikely event of a shell failure.

The principle leakage detection device recommended is an optical moisture sensor (labeled LS/LI in Figure 13). The moisture sensor is located in a collect location area at the bottom of the secondary containment vessel. This device functions similarly to the level indicator and level switch in API Plan 52 systems. Unlike a switch, the moisture sensor provides early detection of liquid traces collecting in the normally dry secondary containment area.

Not all pumpage in a refinery is liquid at atmospheric pressure. Therefore, a pressure switch (labeled PSH in Figure 13) can be provided to detect leakage in either gas or liquid form. The backup sealing device (sealing bushing or optional mechanical gas seal) creates back pressure to trip the pressure switch in the same manner as the orifice in API Plan 52. The pressure switch is typically set to activate at 10 psi.

Treatment of the Vent and Drain Connections

The vent and drain connections after the backup sealing device, which are labeled V and D, are similar to the vent and drain connections in conventional seal gland. As shown in Figure D-1 of API 610 (Figure 2, herein), these connections are followed by a throttle bushing to direct leakage out the vent or drain. Again, API conformance of this magnetically coupled pump is maintained by providing a throttle bushing between the connections and the bearing housing (Figure 13).

Treatment of the vent and drain connections should be consistent with a given refinery's practice on conventional seals. However, the secondary containment area should be maintained at atmospheric pressure. Note that the vent connections afford access for so-called "sniffer" leakage detections devices.

HIGH TEMPERATURE SERVICE CONSIDERATIONS

Pumps designed to API Standard 610 are classified as heavy duty and are usually associated with high temperature service. Conventional API overhung process pumps are typically designed to operate to 800°F maximum allowable temperature, although it is common to use this class of equipment to pump flammable and toxic fluids regardless of temperature.

High temperature considerations for magnetically coupled process pumps must address the following areas:

- Materials
- Differential thermal expansion
- Cooling requirements

Material Selection

Materials of construction are selected for maximum allowable design temperature without consideration for external cooling effects. The selection process is simplified by categorizing applications as either low or high temperature. Because of the inherent temperature limitations of the rare earth permanent magnet materials for process pumps in a hot standby mode, maximum temperatures of 300°F and 500°F typically are selected to differentiate between high and low temperature applications. The corresponding materials of construction are outlined as follows:

	Maximum Allowable Temperature	
	Low (< 300° F)	*High (< 500° F)*
Magnets	Neodymium Iron Boron	Samarium Cobalt
Magnet Adhesives	Epoxy	High Temp. Epoxy
Magnet Encapsulation	316 S.S.	316 S.S.
Primary Containment Shell	Peek	Ceramic/ Hastelloy
Containment Shell Gasket	Viton	Viton/Kalrez
Wet-end Shaft	316 S.S.	Titanium

Magnet material performance is a function of temperature (Figure 14). Maximum allowable temperatures are based on the temperatures at which irreversible losses in torque capacity begin to occur; specifically, 338°F for neodymium iron boron and 536°F for samarium cobalt. Adhesives are used to bond the magnets to the inner and outer magnet carriers. The selected epoxies are applied with a minimum safety margin of 30 percent under the manufacturer's recommended maximum temperature.

Magnet encapsulation is necessary to prevent chemical attack of the magnets on the inner carrier. The magnets are encapsulated with a 316 S.S. cladding, which is electron beam welded to the carrier. Although composite encapsulation is possible at low

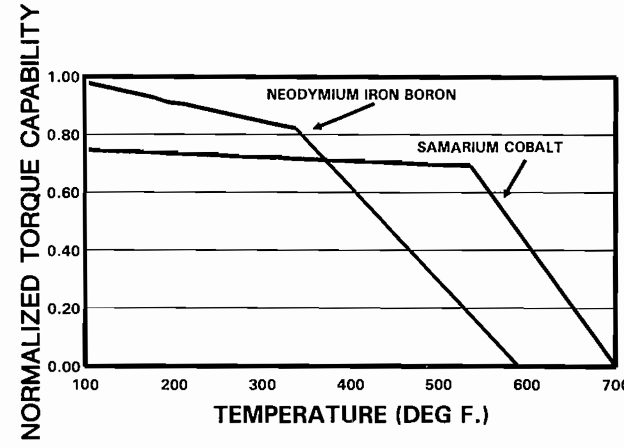


Figure 14. Magnetic Drive System Torque vs Temperature.

temperatures, it is not usually pursued because of processing difficulties [2].

Nonmetallic primary containment shells have been developed and tested in polyetheretherketone (PEEK) and zirconium oxide (ceramic). Hydrostatic proof testing was performed at elevated temperatures (Figure 15). As described previously, proof testing results establish maximum allowable working pressures equal to one-half the minimum proof test pressure at maximum pumping temperature.

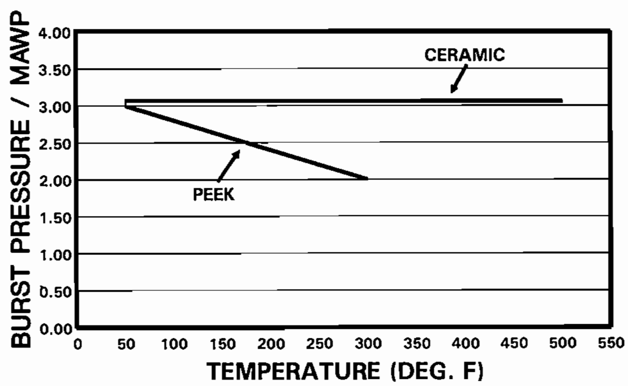


Figure 15. Hydrostatic Proof Test Pressure vs Temperature.

An optional metallic shell of Hastelloy material has also been developed. Its maximum allowable working pressure of 500 psig is based on ASME Code allowable stresses at 500°F [7].

The main casing parting flange gasket is an austenitic stainless steel, spiral wound gasket. The primary containment shell gasket is a viton O-ring (fluoroelastomer) arranged to minimize stress in the shell. Both of these arrangements conform with the API Standard 610 requirement for ". . confined controlled-compression gaskets." Kalrez O-ring material (ASTM D 1418 FFKM elastomer) is used for application above 400°F.

Differential Thermal Expansion

The material selection for the wet-end shaft, that is, the shaft on which the impeller and inner magnet carrier are mounted, is dependent on temperature because of differential thermal expansion considerations. Before elaboration on this point, some background information is necessary.

Silicon carbide (SiC) journal sleeves, mounted on the shaft, have been widely used to provide hard surfaces under the pumped liquid lubricated bearings. This arrangement has presented a significant design problem to pump engineers, in that the fit of the SiC journal sleeves on the shaft should be tight in order to control runout of the bearing surfaces, but radial stresses associated with tight fits can cause SiC sleeves to crack.

The problem is compounded by the fact that the differential thermal expansion between SiC journal sleeves and shaft materials is substantial (for example, stainless steel expands at nearly five times the rate of SiC). The change in fit between the journal sleeves and shaft as pump temperature rises must be compensated for in order to avoid radial stress in the hard and brittle SiC rotating journal sleeves. One manufacturer even provides ". . . a metallic holder to prevent the pieces from destroying the pump internals, should the silicon carbide fracture" [4].

As another author put it:

> When designing SiC bearings units, the special attributes of this material must be considered. For example, the different thermal expansion coefficients of SiC and the shaft material cause special problems in designing the connection between SiC shaft sleeve and pump shaft. A proper run requires an absolute concentricity, i.e., radial stress in the SiC shaft sleeve through thermal expansion of the shaft is inadmissible [5].

Working designs that address the differential expansion problem are being produced today. The design concept involves providing adequate clearance between the shaft and journal sleeves, and fitting expansion rings in the clearances. The expansion rings act as springs, compressing as the pump temperature increases and expanding as it decreases.

The Motivation

Misgivings regarding the application of similar expansion ring design concepts to high temperature API Standard 610 process pumps include:

- The difficulty in securely locking the journal sleeves to the shaft without restricting the expansion rings.
- The effects of the expansion ring spring rates on rotordynamics.
- The general sensitivity of the rotating SiC journal sleeves is inconsistent with the philosophy of the heavy duty API Standard 610 pump classification.

These misgivings became motivation for a research and development effort to find or invent technology which would replace rotating SiC journal sleeves.

The Technology

To eliminate the problems associated with mounting SiC journal sleeves on magnetically coupled pump shafts, shaft treatment processes which employ laser technology were developed. The processes, consolidated surface coating of 316 stainless steel and surface alloying of titanium, produce shaft surfaces which perform in a manner similar to SiC. Therefore, journal sleeves are not required (Figures 16 and 17).

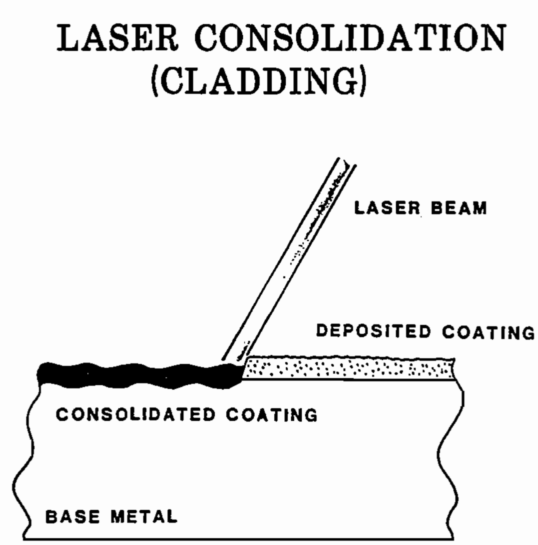


Figure 16. Diagram of Laser Consolidation Process.

Laser Consolidation

The laser consolidation process produces a homogeneous weld bonded surface coating of engineered components. This process uses a concentrated laser beam to melt a preplaced powder for enhanced local wear or corrosion resistance of a material. Coating

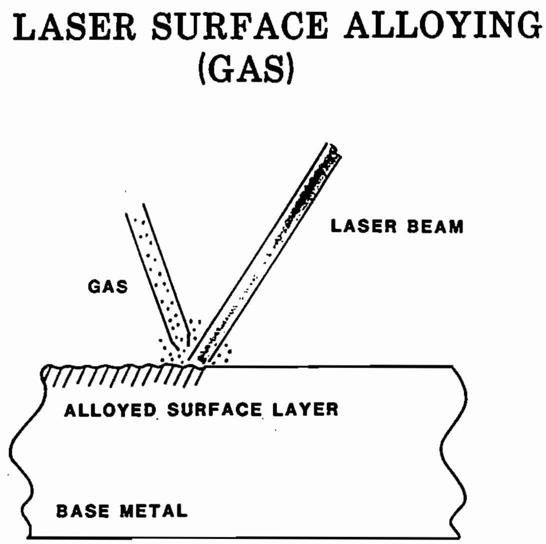


Figure 17. Diagram of Laser Surface Alloying Process.

depths of 0.025 in to 0.030 in can be achieved using this process. Standard wear resistant hardfacing powders, such as the stellites and colmonoys, nickel chromium boron, can be consolidated using this process.

Advantages over conventionally applied coatings are:

- a true metallurgical bond with the substrate.
- a fine dendritic structure which is beneficial in the distribution of carbides for enhanced wear resistance.
- low heat input which minimizes distortion.
- rapid local melting which minimizes the susceptibility to sensitization of the substrate.

In addition to consolidation of standard coating alloys, this process allows for the introduction of various alloying elements and/or gases to further enhance the performance of a particular coating. In this way, the surface enhancement can be engineered for a particular need.

Laser Surface Alloying

Surface modification can also be achieved through a local smelting process with the laser. In this case, specific properties of the substrate can be enhanced by melting the material while introducing a gas to produce a new alloy or ceramic. A controlled process has been developed to produce a titanium nitride coating to a depth of 0.025 in to 0.030 in on the surface of commercially pure titanium (Grade CP2). Titanium nitride is currently used for enhanced wear resistance of cutting tools and dies. In conventional processing, this ceramic coating is produced by either chemical vapor deposition (CVD) or physical vapor deposition (PVD). These two processes deposit titanium nitride, one atom at a time, to a maximum depth of less than 0.001 in. The laser titanium nitride coating process produces a composite of titanium with needles of titanium nitride for the full depth of the melt zone. The bulk hardness of the laser titanium nitride coating is above 50 Rc, however, the individual titanium nitride needles have hardness in excess of 60 Rc, compared to less that 20 Rc hardness for CP2 titanium.

Shaft Material Options

As stated previously, the material selection for the wet-end shaft is dependent on temperature, because of differential thermal expansion considerations. However, these considerations no longer involve the potential cracking of rotating SiC journal sleeves, since these components have been eliminated. They simply involve maintaining optimum bearing clearance at the pumped liquid lubricated bearings throughout the operating temperature range for the application.

Bearing diametral clearances are calculated as a function of operating temperature (Figure 18). The following coefficients of thermal expansion are used:

Bearing Materials	
SiC	0.0000022 in/in –° F
Graphalloy	0.0000038 in/in –° F
Shaft Materials	
316 S.S.	0.0000096 in/in –° F
Titanium	0.0000049 in/in –° F

Based on this design study, the 316 S.S. shaft is suitable for low temperature applications (≤300°F) and the titanium shaft is suitable for low and high temperature applications (≤500°F).

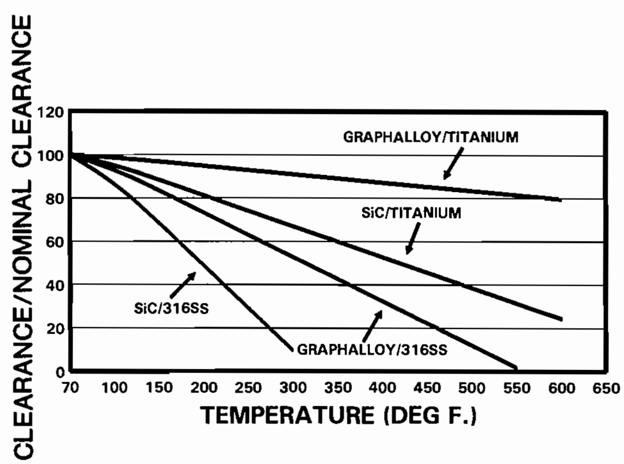


Figure 18. Pumped Liquid Lubricated Bearing vs Temperature.

Cooling Issues

In conventional process pumps, cooling is addressed in two areas: bearings and mechanical seals. Mechanical seals may require flush liquid to carry away heat generated at the seal faces when the pump is operating. However, seals must be capable of containing static pressure at maximum allowable temperature without flush liquid or cooling in a standby mode. Even when thermal siphon effects from a barrier fluid system may provide some degree of cooling to the mechanical seal area, this lower temperature should not be considered in determining the seal's ability to contain static pressure.

The primary containment shell in a magnetically coupled process pump always works in a static mode, even when the pump is operating. The only cooling that takes place in the shell area usually comes from the pumped fluid. Alternatively, cool flush may be provided from an outside source, but this should not be considered when rating the shell material to contain static pressure. It follows, therefore, that cooling cannot be used to increase the maximum allowable working pressure of the shell material.

API Standard 610 specifies (hydrocarbon) oil lubrication for bearings. Typically, lubrication for the antifriction bearings supporting the drive-end shaft for the outer magnet carrier is provided by an oil ring. A fan type flinger provides forced air cooling, which is directed through fins on the bearing housing by the coupling guard (Figure 19). Water cooling of the oil sump is provided with a finned tube cooler for high temperature applications (300F). As recommended by API Standard 610, provisions are made for purge or pure oil mist lubrication of the antifriction bearings (Figure 20).

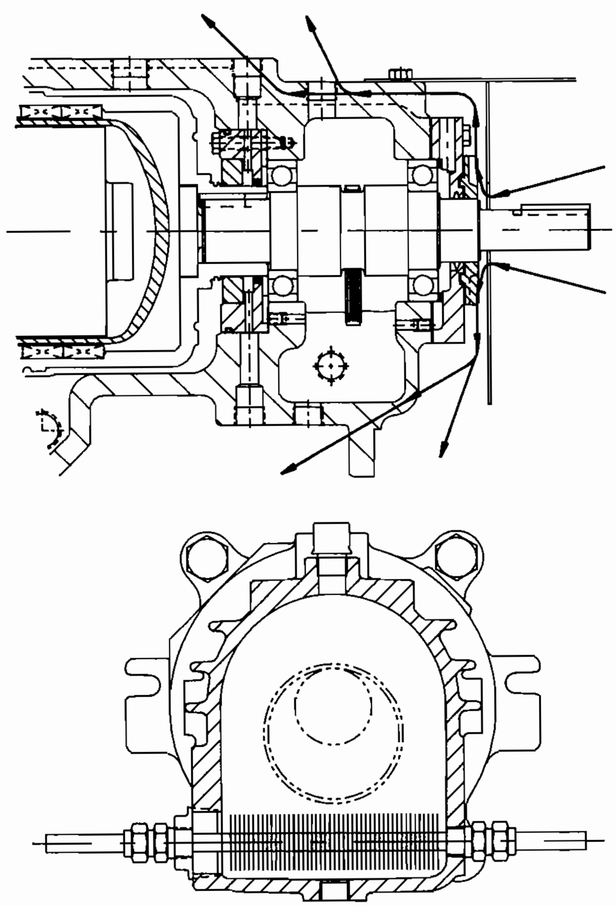


Figure 19. API Bearing Housing Cooling.

Nonsparking Feature

API Standard 610 requires that bearing housing seals and deflectors be made of nonsparking materials (reference paragraph 2.9.2.7). In magnetically coupled pumps for refinery service, it is important that this requirement be properly applied to the area between the rotating outer magnet carrier and the secondary containment housing. In order to do this, a nonsparking ring is located such that if the bearings in the outer bearing housing should fail, causing the outer magnet carrier to orbit, contact would first be made with the nonsparking wear ring, and not the steel bearing housing pressure boundary. This feature ensures compliance with the safety requirements of the API Standard.

SUMMARY

It is possible to design and produce a magnetically coupled process pump to comply with the requirements and intent of API Standard 610, including stress levels given in Section VIII, Division 1, of the ASME Code. Construction details which have caused problems in the past, such as severe differential thermal expansion properties of mating materials, ineffective or power absorbing product circulation within the bearing and magnet areas, and excessive heating of the circulating fluid from eddy current losses

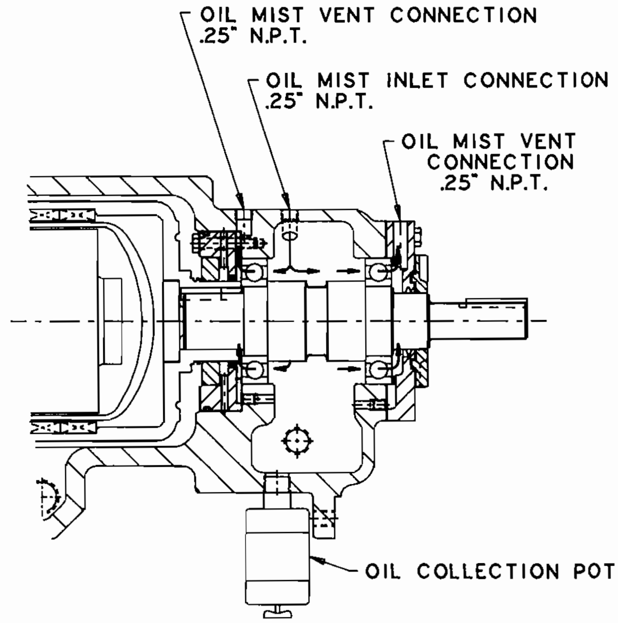


Figure 20. Pure Oil Mist Lubrication.

in the shell (even the vaporization of this heated fluid as it reenters the main pump flow stream) can be resolved by innovative applications of new and emerging technologies and ideas that focus on these problems and experiences. The design approaches explained here represent only some of the ways these problems can be effectively resolved.

In the authors' opinion, qualification testing of API magnetically coupled process pumps should include two sequential phases: laboratory testing of prototype units and field testing of actual production units in actual refinery services. By testing production pumps in a variety of applications, the final design and manufacturing processes are thoroughly proven prior to full market release. At the time of this writing, production pumps are being prepared for field testing in major U.S. refineries.

Also, at the time of this writing, the Mechanical Equipment Subcommittee of the American Petroleum Institute (API) has authorized and formed a Task Force with the objective of creating a new API Standard for Sealless Pumps (API Standard 685). The scope of this new standard will be to establish the minimum requirements for sealless centrifugal pumps used in the hydrocarbon processing industry where fluid containment is required. It is expected that this new standard will follow the existing API Standard 610 very closely. In fact, it is possible that the next issue of the 610 Standard (the 8th edition) will be structured such that the new sealless pump standard can become a subsection, without repeating the many basic sections that are common to all pumps intended for general refinery service. It is expected that the design features and philosophies outlined herein will comply with the requirements established in the new API Standard 685, or the revised API Standard 610, 8th Edition, whichever the future may bring.

REFERENCES

1. Mayes, J. D., "Magnetically Driven Centrifugal Pumps—Eliminating Seal Problems in Refinery and Chemical Processing Plant Equipment," *Proceedings of the Seventh International Pump Symposium*, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1990).

2. Buse, F. W., and Stoughton, C. D., "Design of Magnetically Driven Chemical Pump to fit ANSI B73.1 Dimensions Using a Semiopen Impeller," *Proceedings of the Seventh International Pump Users Symposium*, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1990).
3. "Centrifugal Pumps for General Refinery Service," API Standard 610, Seventh Edition, Refining Department, American Petroleum Institute (1989).
4. Hernandez, T., "A User's Engineering Review of Sealless Pump Design Limitations and Features," *Proceedings of the Eighth International Pump Users Symposium*, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1991).
5. Schommer, H. J., and Johnson, T., "Design, Construction, and Applications of Magnetically Coupled Centrifugal Pumps," *Proceedings of the Seventh International Pump Users Symposium*, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1990).
6. Smith, J. H., and Oliver, J. B., "Techniques for Eddy Current Reduction in Magnetic Drives Using Metallic Containment Barriers," *Proceedings of the Eighth International Pump Users Symposium*, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1991).
7. ASME "Boiler and Pressure Vessel Code," Section VIII, "Rules for Construction of Pressure Vessels," American Society of Mechanical Engineers.
8. Larson J., "Material Selection," portion of Short Course on Centrifugal Pump Fundamentals, Third International Pump Users Symposium, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A & M University, College Station, Texas (1986).
9. Heald, C. C., and Palgrave, R., TECHNOLOGY, "Backflow Control Improves Pump Performance," Oil & Gas Journal (February 25, 1985).
10. Rix, P., and McLean, R., "Dual Containment Shells—Added Safety for Magnetically Driven Pumps," *Proceedings of the Eighth International Pump Users Symposium*, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1991).

ACKNOWLEDGMENT

The authors wish to thank the management of Ingersoll-Rand Company for permission to publish this paper. They also extend special thanks to Paul Cooper, Ron Miller, Fred Buse, and others for their contributions and assistance.