DESIGN OF MAGNETICALLY DRIVEN CHEMICAL PUMP TO FIT ANSI B73.1 DIMENSIONS USING A SEMIOPEN IMPELLER

by

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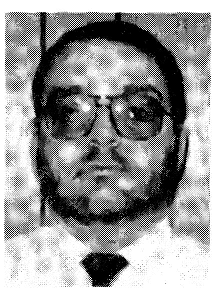

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ABSTRACT

A magnetically driven sealless chemical pump was developed that employs high torque rare-earth magnets. Utilizing a semiopen impeller configuration, the pump was designed to fit into the ANSI B73.1 standard dimensional envelope. This was done to simplify retrofitting the existing installed pump population or, in the case of new installations, the piping design; thus, standard existing NEMA motors, couplings, bedplates, pump impellers, and casings can be used.

Through the use of a nonmetallic containment shell, eddy currents were eliminated, and the pump efficiency was within two points of that of a comparable conventional pump with a mechanical seal. A metallic containment shell was developed for the relatively small number of applications in which the nonmetallic shell is not suitable.

The implementation is discussed of the development program for this sealless pump, including the shell, the thrust bearing needed for the semiopen impeller and the encapsulation with both metallic and nonmetallic materials of the driven magnets.

INTRODUCTION

In recent years, the increasing emphasis on prevention of leakage of process fluids in many areas of application has led to the introduction of sealless pumps. Some of these are canned motor pumps, and others are driven through a magnetic coupling by a standard motor [1]. The latter approach is addressed, and specifically pumps conforming to ANSI Standard B73.1 [2].

The purpose of a sealless pump is to replace a shaft, which penetrates into the atmosphere, and to directly drive the impeller. A magnetically driven sealless pump has a sealed containment shell, and external magnets transmit torque through the shell, thereby driving internal magnets on a shaft that has an impeller attached to it, as shown on Figure 1.

The requirement for this design was to develop a magnetic drive that meets the users' preference for a semiopen impeller in an ANSI dimensional pump envelope. Additional requirements were to utilize existing bedplates, couplings, and a standard NEMA fast start motor. This entire pump and motor combination, which conforms to ANSI Standard B73.1, is shown in Figure 2. Casings and impellers are interchangeable with the manufacturer's existing ANSI pumps to minimize the users' parts inventories.

Because the dimensional envelope is frozen by specification, it was necessary to determine whether commercial magnets producing the required torque would fit into the allocated space of the power end (Figure 3). This can be accomplished for most of the range of power requirements of ANSI pumps through the use of rare-earth permanent magnets. The strongest of these is

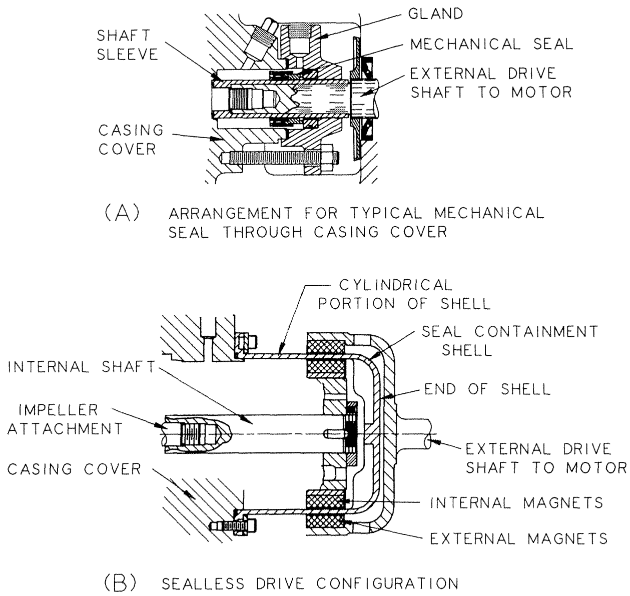


Figure 1. Comparison of Typical Mechanical Sealing Arrangement and Sealless Pump Drive Configuration.

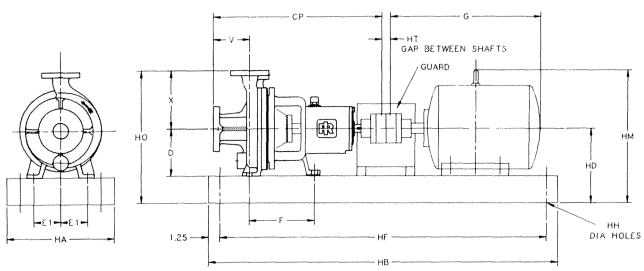


Figure 2. Standard ANSI Pump and Motor Configuration. Dimensions shown are controlled by ANSI Standard B73.1.

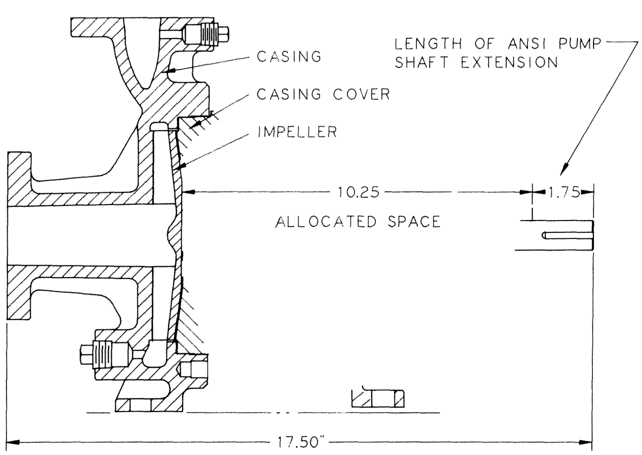


Figure 3. Allocated Space for Sealless Drive Mechanism Within ANSI B73.1 Dimensional Envelope for Size AA and AB Pumps.

the combination known as neodymium iron boron, which has an energy product of 30 megagauss-oersteds, or more.

The implementation of this comparatively recent magnet technology into the design of a standard pump line is described in the following sections.

DESIGN CONSIDERATIONS

Torque Requirement

The torque value of the drive magnets for this application is determined not only by the running torque of the pump, but also by the starting torque. This includes the pump end, which consists of the impeller, liquid in the impeller, and the internal magnets, together with the starting torque of the motor, which can be 130 to 220 percent of the full load torque. The final size of the magnets is usually based on the motor starting torque.

Once the approximate size of the magnets is chosen, the hydraulic axial and radial thrust of the various pump sizes must be tabulated to determine the size of the bearings.

Impeller Type

While most sealless pumps use enclosed impellers, (Figures 4 and 5), the researchers wanted to meet the user preference for the more versatile semiopen design. Since they handle a wider variety of fluids, semiopen impellers have traditionally been the workhorse of the ANSI chemical process pumps.

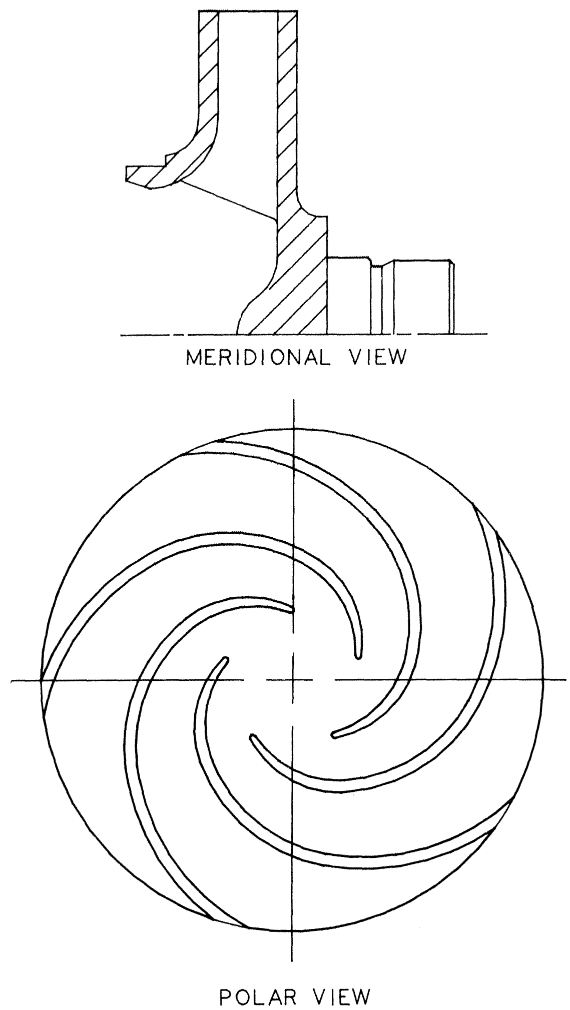


Figure 4. Closed Impeller.

The design challenge was develop a reliable bearing and lubrication system that could handle the higher axial thrust typical of a semiopen impeller. To achieve the reliability and performance objectives, enclosed impellers with wear rings (Figure 5b),

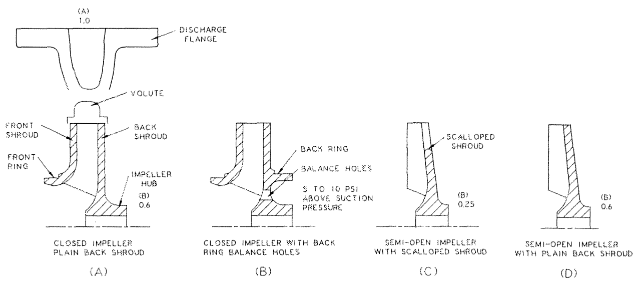


Figure 5. Various Closed and Semiopen Impeller Configurations, Showing Impeller Hub Pressure (b) Compared to the Total Developed Pressure (a). (The numerals indicated the fractions of pump pressure rise existing at locations shown.)

which generally are not suitable for corrosive applications, were eliminated. A problem with front or back rings is that galling to the mating surface of a stainless steel impeller will occur unless the clearances are larger than normal (Figures 5c and 6). The performance and efficiency of a semiopen impeller can be higher than for an enclosed impeller with wear rings that have wide clearances.

The goal was to develop a thrust bearing system to utilize a semiopen impeller. The axial thrust of a semiopen impeller with scalloped back shrouds is greater than a plain shrouded enclosed impeller. In the particular design in question, the back shrouds are not scalloped (Figures 5d and 7). This is because the hydraulic efficiency is three to five points higher with nonscalloped back shrouds. In Figure 5, the relative pressure at the back hub of the impeller is shown for the various back shroud configurations.

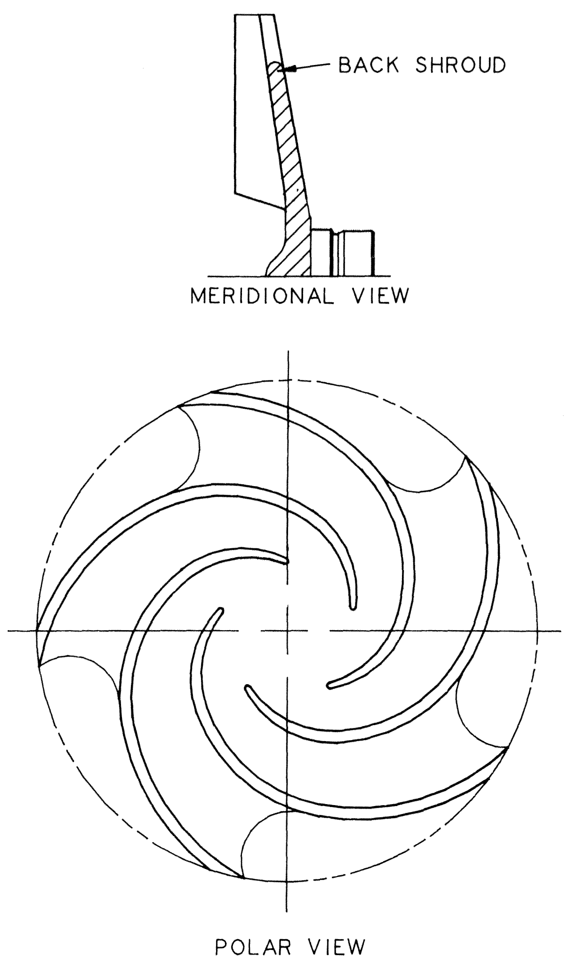


Figure 6. Partially Scalloped Semiopen Impeller.

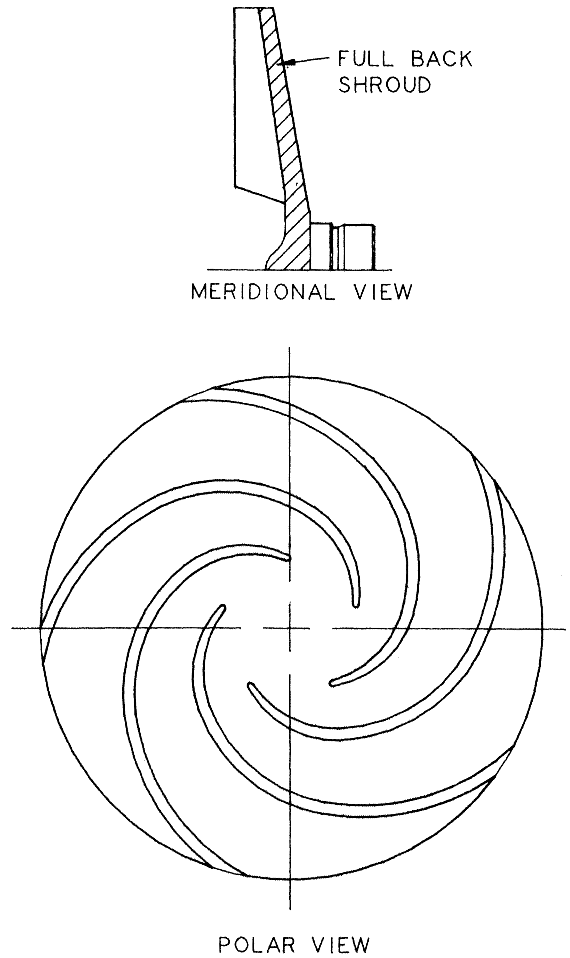


Figure 7. Semiopen Impeller with Full Back Shroud.

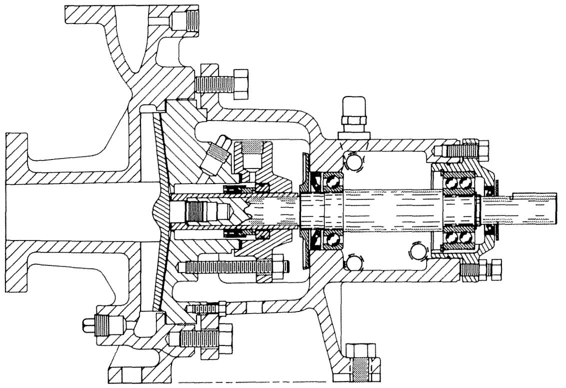
Bearings

The initial concept was to use a semiopen impeller with unscalloped back shrouds and a back ring with balance holes below the back ring. The thrust bearings were rigidly mounted to the shaft. The bearing and mating surface materials were silicon carbide. An initial PV value of 300,000 was used to determine the size of the bearing. P is the load on the bearing in psi, V is the mean velocity in feet per minute. The bearings were sized for a $1.5 \times 1 \times 6$ pump operating at 3550 rpm. PV values of 1.6 million had been obtained by other researchers [3].

After the bearing and magnet sizes were determined, the initial layout was made to determine whether the requirement of fitting the pump into the ANSI envelope could be met. The result is shown in Figure 8.

DEVELOPMENT PROGRAM

Successful deployment and encapsulation of the magnets and the design of the bearings and containment shell required an extensive development program, which is described forthwith.



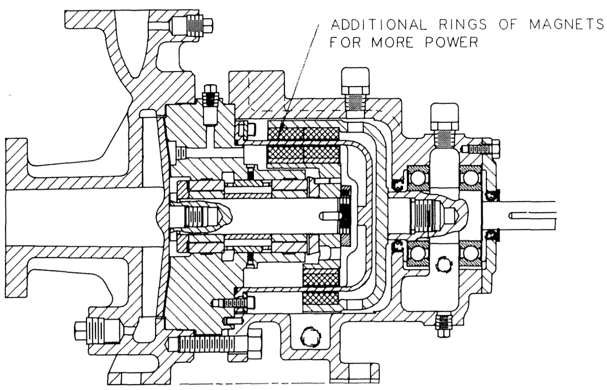


Figure 8. Comparison of ANSI Pump Drive Configuration Having the Same Overall Dimensional Envelopment.

Magnetic Drive Materials

Corrosive resistant materials are required for typical ANSI chemical pump applications. The temperature of most of these applications is under 250°F. Also, the working pressure of most applications is under 200 psig. To reduce eddy current losses and improve efficiency, it was decided to develop composite material components. Both metallic and nonmetallic materials are needed because the composite cannot handle all the corrosive services, nor pressures and temperatures above these limits

Component	*Material*	*Limiting Temperature, F*
Magnet	Neodymium Iron Boron	250
Magnet	Samarium Cobalt	575
Containment Shell	Composite	250
Containment Shell	Hastelloy "C"	1000+
Bearing	Silicon Carbide	1000+
Adhesives	Various Epoxies	250
Magnet Encapsulation	Composite	250
Magnet Encapsulation	Metallic	1000+

The continuous temperature limitation of neodymium iron boron is 250°F; above that, there is a rapid drop in torque capability. The limitation of samarium cobalt is 575°F, however, the torque capability is 85 percent that of neodymium iron boron.

One of the design considerations of the permanent magnets is the axial force that occurs during the pump's assembly and disassembly. The magnets of the 10 horsepower (hp) pump have 100 lb of axial attration; a 75 hp pump will have over 200 lb axial attraction. While there are builtin safety features, strict adherence to warnings and safety practices must be followed by maintenance personnel. Limitation of permanent magnet construction is the size of the forces that maintenance personnel can safely handle.

Metallic components were developed for applications where the current nonmetallic shell is not suitable, however, additional nonmetallic development is already underway which will result in even broader temperature/pressure corrosion limits.

Magnets Characteristics [4]

The magnetic circuit consists of permanent north and south poles and mating conducting rings. The path of the flux in the circuit is shown in Figure 9. The conducting rings are made of carbon steel or ductile iron. Caution has to be taken to ensure that the conducting rings are of the correct size and grade of material to carry the flux density of the magnet.

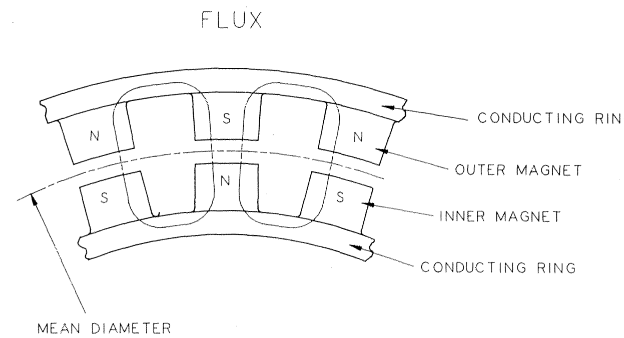


Figure 9. Magnetic Circuit.

The overall torque of the magnets depends on

- The flux density of the magnetic material
- Operating temperature of the magnet
- Overall gap between the inner and outer magnet
- Mean diameter of the magnets (Figure 9)
- Number of magnets
- Length of magnets (Figure 10)

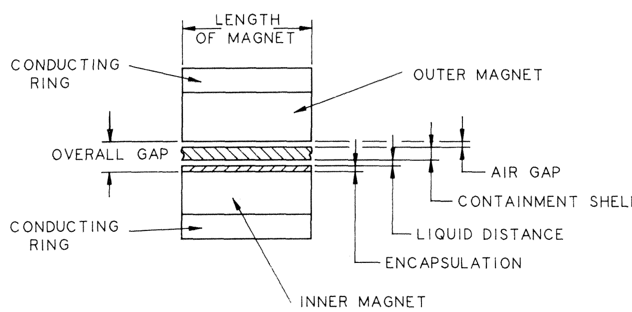


Figure 10. Magnet Configuration.

For a given size, the torque of the magnets changes as the square of the flux density.

The torque produced by a pair of the magnets varies inversely as the square of the overall gap between the magnets (Figure 10). This gap will have the greatest effect on the torque capability of the pump. The width of this gap depends on (Figure 10):

- the amount of encapsulation put on the inner magnet.
- the liquid distance from the outer diameter (OD) of the encapsulation to inner diameter (ID) of the containment shell.
- the thickness of the containment shell.
- the air gap from the OD of the containment shell to ID of the outer magnet.

The thickness of the encapsulation around the inner magnet depends on the process used, the choice of metallic *vs* composite materials and the thermal expansion of the magnetic poles and of the materials between poles.

The liquid distance between the containment shell ID and magnet OD is determined by:

- manufacturing tolerances.
- methods of making the pieces.
- hydraulic drag losses.
- differential pressure from front to back of magnet produced by flow through this gap.

Containment Shell and Magnet Configuration

Containment shell thickness (Figure 10) depends on the working pressure and temperature of the application. Eddy current losses vary with material type and thickness.

To minimize parasitic hydraulic losses, the inner diameter of the shell should be kept as small as possible. As far as practical, the length of the magnets, rather than the diameter, should be increased to achieve a given torque.

Since the torque changes with the cube of available material, the easiest way to change torque is to increase or decrease the number of magnets. When changing the number of magnets, the appropriate north-south pole flux circuits have to be maintained. The other method for increasing torque is to add more rings of magnets, as indicated in Figure 8.

To ensure that they meet specification, each pair of magnets is given a static torque test before being assembled into the pump.

Of the above factors, the two that have the greatest effect on the torque capability are the thickness of the shell, which controls the overall gap, and the material of the shell, which dictates the amount of eddy currents and, therefore, the effective torque transmission.

Nonmetallic Shell

The concern of using nonmetallic material for the containment shell is that it moves under pressure. The four movements that have to be considered are:

- initial movement as pressure is applied.
- creep for a given pressure as related to time.
- movement with increase in temperature.
- movement with increase in pressure and/or temperature and time.

Most movement is axial and takes place at the end of the shell.

The only true test of the strength of a composite is the behavior of the molded piece in actual tests. A fabricated assembly will not have the properties of a molded piece, nor will the results from test coupons be applicable.

A finite element stress analysis was made using the characteristics of molded material. Molded pieces were made of five different materials. The initial failure pressures ranged from 200 to 350 psi. The movement at the end of the shell at 200 psi and 70°F ranged from 0.040 in to 0.050 in, while the cylindrical portion moved from 0.008 in to 0.013 in.

Under hydrostatic test, the initial movement was recorded. Next, the pressure was applied and maintained for 100 hr, and the additional creep movement was then recorded. One of the materials had an initial end movement of 0.027 in and crept another 0.019 in for the next 100 hr. Cylindrical wall movement was initially 0.004 in and crept only another 0.001 in for the next 100 hr.

One composite yielded the following results:

		Movement in thousandths of inch			
		End of Shell		Cylindrical Portion	
PSI	TEMP F	CREEP	THERMAL	CREEP	THERMAL
100	100	6.9	1.4	1.4	2.2
100	180	13.5	5.2	3.0	8.1
100	250	21.0	8.5	4.7	13.0
200	100	14.0	1.4	7.3	7.2
200	180	27.0	5.2	6.0	8.1
200	250	42.0	8.5	9.4	13.2

The final material chosen had 0.030 in end movement and 0.003 in radial at 200°F with 200 psig. At 100°F, the burst pressure was over 500 psig. The limiting working pressure was specified as 200 psig at 100°F and 150 psig at 250°F with a burst pressure of 350 psi at 250°F.

After the pieces had stabilized, they were taken to failure by (a) maintaining pressure and increasing temperature and (b) by maintaining temperature and increasing pressure. This was done to a minimum of three pieces per test run in order to confirm the results.

Encapsulation of the Driven Magnets

One of the most difficult portions of the design of the magnetic drive is the encapsulation of the inner magnet to prevent attack by chemicals being pumped. Once the encapsulation is complete, the challenge is to verify, via nondestructive testing, that the encapsulation is 100 percent effective.

Two kinds of encapsulation were pursued, composite and metallic. Each had its own set of challenges, but the result had to be the same.

With the composite materials these are (Figure 11a).

- The limit of application temperature, usually 250°F.
- The interface seal between composite and metallic parts.
- Thermal expansion.
- The issue of exposing porosity if machined.
- The limit of the process temperatures which can effect adhesives and sintering materials.
- The problems of preventing movement of the magnets due to process pressures.
- Degassing of materials during processing.
- Change in the torque capacity of the magnet due to process temperature.
- Molding thickness.
- Stress rise concentrations.
- Resin rich areas.

The composite encapsulation process can require high pressure for injection of the material and high temperature to keep the composite in a moldable state. The chemical reaction of the curing of the material can produce a high temperature. If the temperature rises too high, it will affect the torque of the magnets. When the pressure for injection is too high, the magnets can be moved from their required positions. If the composite is used to fill in between the poles, its radial thermal expansion in high temperature applications has to be accounted for so that it does not reduce the liquid gap between the OD of the encapsulation and ID of the shell.

The various chemical reactions of the curing process have to be known to determine if degassing will occur. Degassing will cause inclusions or ruptures within the encapsulation. Also, the inclusions can appear if machining of the surface skin is required.

Caution has to be taken at the sharp corners of the poles to avoid stress concentrations and to prevent resin-rich areas from developing during curing.

With metallic encapsulation, the problems are (Figure 11b):

- Type of welding.
  - conventional
  - laser
  - electron beam
- Type of materials that can be welded.
  - wrought to wrought
  - wrought to cast
- Direction of weld.
- Type of premachine.
- Adhesive limits due to temperature.
- Degassing.
- Change in torque capability of magnets.
- Verification of encapsulation effectiveness.

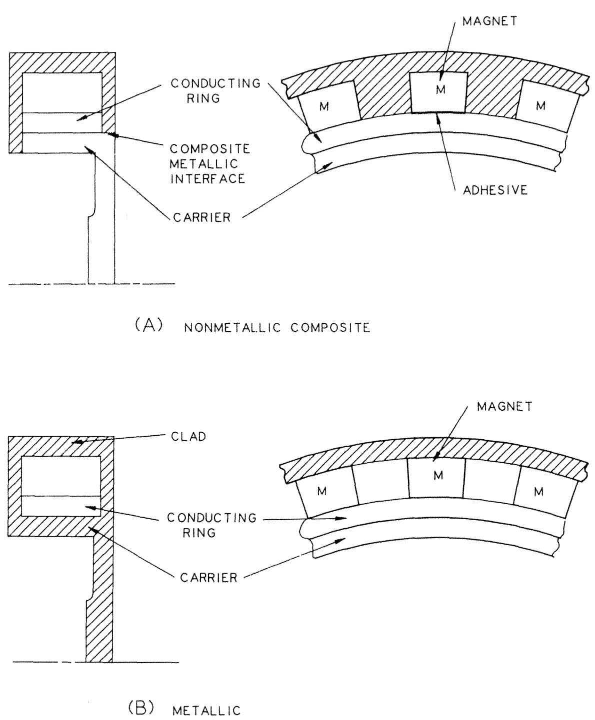


Figure 11. Encapsulation of Magnets Using Nonmetallic Composition and Metallic Clad.

The metallic encapsulation is a clad, which is welded around the magnets. The distance of the magnets to the weld dictate the type of weld. When conventional welding is too close to the magnets, the flux of the magnets pulls the weld arc toward the magnets making it impossible to control. In the design of this pump, there was not enough axial space to use conventional welding. Therefore, the design dictated use of either laser or electron beam welding; in both, heat is concentrated at the joint and does not dissipate enough to effect the torque of the magnets. Selection of materials has to be considered, since cast stainless steel cannot be readily welded to wrought materials.

Metallic Shell

In applications requiring a metallic containment shell, eddy currents will exist and will increase with shell thickness. Therefore, this shell must be as thin as possible within the gap between the magnets. The original 316 stainless steel shell was 0.090 in thick on the cylindrical portion (Figure 1) and 0.160 in at the end (Figure 1). When pressurized to 900 psig, the end moved 0.050 in and cylindrical portion 0.003 in radially. Normal working pressure of an ANSI pump is 350 psig. However, a 1.5 × 1 × 6 pump, which requires 6.0 hp, would not start with a 7.5 hp motor. In fact, the starting heaters promptly opened. It was determined that the eddy current developed by the 0.090 in thick wall was too much for the 7.5 hp motor. The design was finalized at 0.030 in thickness using Hastelloy C. This shell was hydrotested to 1,000 psig. There was permanent deformation, but no failure. The eddy currents are typically 15 to 25 percent of the total drive power.

With metallic shells, the extent of the eddy current losses indicated that a nonmetallic composite shell should be used when possible. The composite shells have no eddy currents. The materials chosen also have an operating limit of 250°F, which is the same as the limit for neodymium iron boron magnets.

Verification of Encapsulation Effectiveness

A nondestructive verification test of the encapsulation was investigated. Of the many methods discussed, the simplest was to put the completed encapsulated magnet in a vessel of water and pressurize it for 15 minutes at 80 to 100 psig, remove and dry it, and then put it in an oven at 220°F. If vapor or a whistle like that of a tea kettle occurred, the encapsulation was leaking. An additional test to see what happens over time is to place the encapsulated magnet in a 75-percent concentration of acetic acid. If the acid turns slightly brown, there is a leak, and the acid is attacking the magnet. This test takes eight to ten days in order to ensure that the acid has not leaked into the encapsulation.

Bearing Design and Lubrication

In order to utilize semiopen impeller construction, a patented self aligning thrust bearing and lubrication had to be employed. This was shown to be necessary by testing with a rigid thrust bearing as now described.

Initially, a 1.5 × 1 × 6 pump was initially tested with a 5.0 in diameter impeller, a composite shell, and one row of magnets each in the driver and driven members. The efficiency was two points less than the equivalent ANSI pump equipped with a mechanical seal. The pump was run for 300 hr without problems. This pump had rigid front and back thrust bearings, as shown in Figure 12. The axial thrust was 300 lb with a PV of 325,000. Commercial tolerances and fits of the rigid thrust bearings resulted in a 50 percent to 60 percent of the interfacial area of the mating pieces being in contact.

When a 6.25 in diameter impeller with 450 lb of axial thrust was used, the rear bearing failed due to heat build up on the ID of the thrust bearing. It was determined that the limiting axial thrust between the rigid thrust bearings was 500 lb. This resulted in a PV value of 488,000.

To reduce axial thrust, a back ring and balance holes were put on the back shroud or the impeller (Figure 13). The axial thrust was 200 lb. The pump ran well, but a back ring was an unacceptable solution, because of possible galling of the mating part and back rings are not suitable for corrosive service. For these reasons, the rigid thrust bearing design was rejected.

A thrust bearing with a self aligning feature was developed and then installed at the back bearing location, as shown in Figure 12. A 6.25 in production impeller was installed in the pump. The pump ran satisfactorily for 1,000 hr without problems. The PV value was 790,000. An eight inch impeller was tested and run

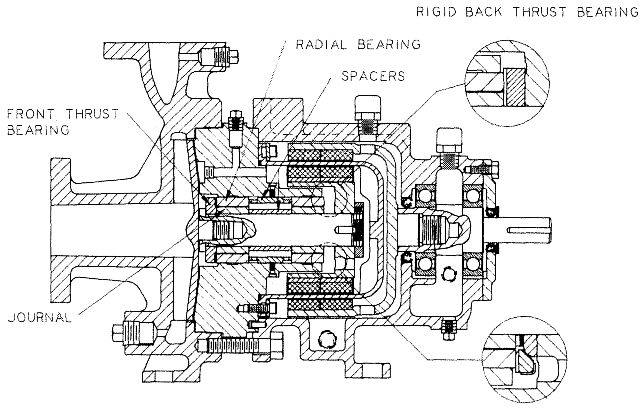


Figure 12. Thrust Bearing Arrangements for Sealless Magnetic Drive Pump.

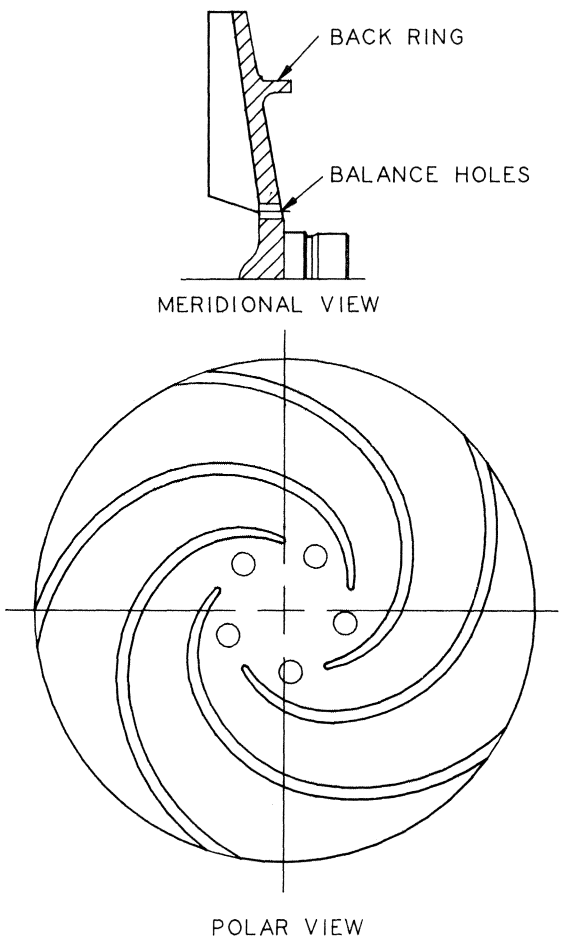


Figure 13. Full Back Shroud Semiopen Impeller with Back Ring and Balance Holes.

for 1,000 hr. In both cases, no wear of the silicon carbide bearings was observed. The resulting PV value of the eight inch pump was 1,300,000. The thrust load at full impeller diameter operating at shutoff is only 80 percent of the bearing design capability. The thrust collar had 100 percent contact and was mounted in a metallic cup to prevent breakage due to suction upset and mishandling. The efficiency of this pump was within two points of the equivalent seal equipped ANSI pump.

Both the thrust and radial bearings are lubricated by the pumped fluid. The path of this lubricating fluid is shown in Figure 14. The amount is from 0.3 to 0.5 gpm, depending on the magnitude of the internal pressure put into the shell. If external circulation is used with a cyclone separator or external filter, the flow goes to 1.0 to 3.0 gpm. By having the circulation go to the periphery of the back thrust collar and out between the impeller shroud and casing cover, rather than into the impeller eye via balance holes or hollow shaft, the $NPSH_R$ characteristics of the impeller are the same as a pump with a mechanical seal. This is especially beneficial when using a metallic shell, where the eddy currents can cause the temperature of the lubricating fluid to increase by 3.0 to 10°F, which can result in the flashing of the pumped fluid and pump damage. The circulation between the impeller and casing cover allows the minimum flow requirements due to thermal rise to be the same as a pump with a mechanical seal. Minimum flow due to shaft deflection is eliminated because there is no shaft overhang.

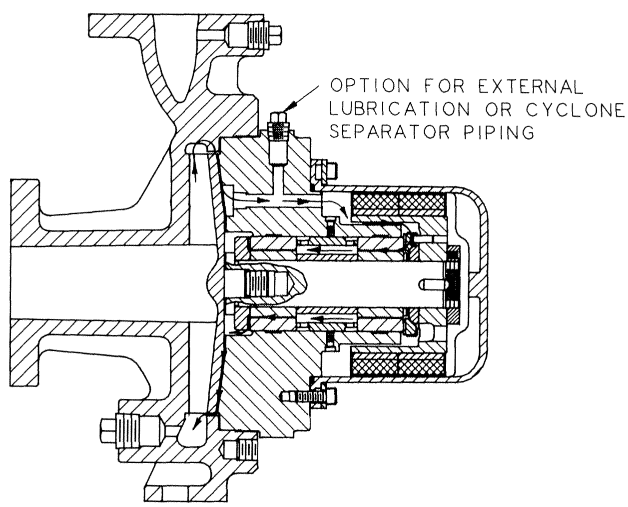


Figure 14. Bearing Lubrication Paths for Sealless Magnetic Drive Pump.

To maintain good bearing life, the size and amount of particles circulating within the lubrication system should be kept below 100 mesh. This can be accomplished by a cyclone separator, external filter or and internal strainer. A 100 mesh internal strainer was designed so the circulating liquid behind the impeller would continuously wash particles off the strainer while allowing clean liquid to circulate through the lubrication system.

Performance

The foregoing developments led to a sealless magnetic drive pump design which has successfully undergone extensive laboratory and field testing in several sizes. A typical performance comparison of one of these sealless pumps is shown in Figure 15, where it is compared to the equivalent conventional ANSI pump with mechanical seals.

As noted above, *best* efficiency point of the sealless pump is within two points of the conventional machine *when* a composite shell is used. After 1000 hours of endurance testing, there was no change in efficiency. Maintaining the efficiency of a pump operating in the field will depend on the corrosiveness of the liquid on the clearances of parts within the pump.

A pump with the internal strainer was installed in a small closed loop. One percent sand, by weight, was introduced into

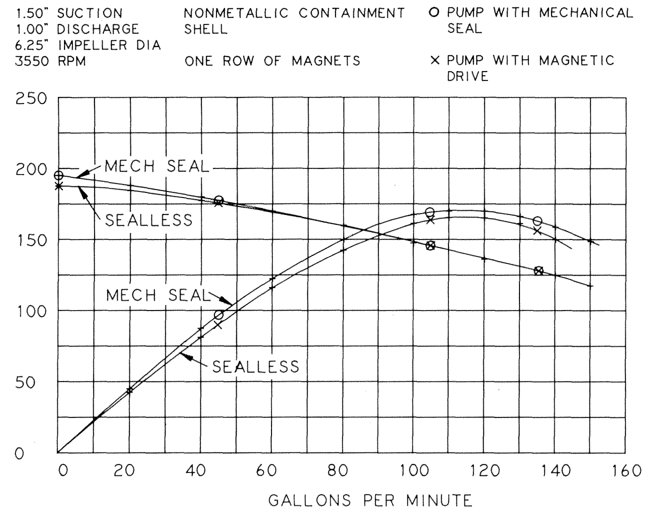


Figure 15. 1.5 × 1.0 × 6 Pump.

the system, and the pump was run for 30 minutes and then inspected. The inspection showed that the casing and impeller had been nicely polished. While were was a small deposit of fine sand at the bottom of the shell, there was no evidence of wear on the silicon carbide bearings.

CONCLUSIONS

Sealless magnetic drive pumps built to ANSI B73.1 dimensions have been built using semiopen impellers. The axial load can be accommodated by a self-aligning thrust bearing arrangement together with a patented system of lubrication via the pumped fluid. Composite or metallic materials are used for the containment shells and for the necessary encapsulation of the driven magnets. This usage depends on the type of fluid, the temperature and the pressure of the application. In the majority of cases, a nonmetallic composite containment shell is used since it is inert to corrosive fluid. This shell is superior because it eliminates the eddy currents produced in a metallic shell and allows the efficiency of this type of pump to come within two points of an equivalent ANSI pump with mechanical seals.

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