DRY RUNNING TESTS UTILIZING SILICON CARBIDE BEARINGS AND POLYMER LUBRICATING STRIPS WITH CONDUCTIVE AND NONCONDUCTIVE CONTAINMENT SHELLS IN AN ANSI MAGNETIC DRIVE PUMP

by

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ABSTRACT

As users continue to turn to sealless pumps to solve their leakage (emission) problems they are demanding robust designs that are more tolerant of system upsets and have the ability to survive short periods of dry running. Silicon carbide bearings are known to be the least forgiving during dry running, but are still the best universal choice. The purpose of this study is to demonstrate means to extend the life of silicon carbide bearings in a dry running environment.

A test rig was designed to simulate broken prime and drained pump dry run conditions. The test pump was a $1^1/_2 \times 1 \times 8$ ANSI magnetic drive pump driven by a 15 hp 3600 rpm motor. Both sintered silicon carbide and 70 micron controlled porosity silicon carbide with Teflon lubricating strips were tested in conductive and nonconductive containment shells. Damage assessment was made after each test.

Results of water tests show that robust pump design and proper bearing alignment is essential. Under broken prime conditions with a nonconductive containment shell, controlled porosity silicon carbide with Teflon lubricating strips ran for $4^1/_2$ hours without overheating, failure or damage. The broken prime dry run time for conductive containment shells is partly dependent upon the magnitude of eddy current heat generated and the time required to boil off the product lubricant; for the models tested, this time was approximately 32 to 35 minutes. Drained casing tests with nonconductive containment shells show a doubling of dry run time with controlled porosity silicon carbide and Teflon strips over that of sintered silicon carbide.

A drained casing test with conductive containment shell configured with controlled porosity silicon carbide and Teflon lubricating strips had a dry run time of 36 minutes and underwent much less bearing damage than a broken prime test with sintered silicon carbide and conductive containment shell.

The addition of controlled porosity silicon carbide and Teflon lubricating strips can add substantially to dry run life and help reduce damage to the product lubricated bearings.

INTRODUCTION

With users turning more and more to sealless pump designs to solve their emission problems and to extend mean time between failures (MTBF), the design weakest link has been transferred from the mechanical seals to the product lubricated bearings. Bearing environment, loading and alignment are all critical factors in maintaining long trouble free operation. One of the perceived weaknesses of sealless pumps has been the inability of the product lubricated bearings to survive short periods of dry running. While it is not endorsed that silicon carbide bearings be deliberately run dry; it is recognized that it does occur in the field and some provision for extending dry running life would be desirable. Dry running, defined here in the broadest sense, can be attributed to both misapplication and misoperation as a result of any of the following conditions in a magnetic drive pump:

- Starting the pump without liquid such as would occur if the suction valve is closed.

- Loss of liquid to the pump suction as may occur during tank unloading operations. This is typically one of the most common types of dry run conditions encountered in operation.

- Blocked lubrication/circulation paths in the product lubricated bearings or containment shell area that may be the result of excessive solids accumulation or polymerization/ crystallization of the pumped liquid.

- Cavitation at the impeller inlet due to insufficient $NPSH_A$ or system related pressure-temperature transients that can result in liquid flashing through the drive cooling/lubrication paths.

- Liquids pumped close to their boiling points are particularly sensitive, due to temperature rise caused by eddy current losses in metal containment shells.

- Entrained air or gasses that may cause vapor pockets to form and block cooling flow.

Of the various materials available for product lubricated bearings, silicon carbide is considered the least forgiving under dry running conditions. Despite this shortcoming sintered silicon carbide is the best all around choice when considering wear resistance, load carrying ability, temperature stability, and chemical inertness. Other materials which have been commonly used such as carbon-graphite and various polymer materials have limitations in three or more of these areas. These materials simply cannot be used in as broad an application range as sintered silicon carbide.

DESIGN CONSIDERATIONS

In addition to good bearing design, there are several design features that will enhance the dry running ability of a magnetic drive pump.

Nonconductive Containment Shell

Aside from the bearing design, a zero eddy current loss containment shell is the biggest contributor to dry running capability; especially under broken prime conditions. Under this common type of dry running, with a nonconductive containment shell, heat input to the pumped liquid would be relatively low and attributable only to the frictional power loss in the product lubricated bearings and viscous shearing of the liquid by other rotating parts. In conductive shells of Hastelloy C material, eddy current losses are typically 10 to 20 percent of the magnet drive rating for a pump operating at 3600 rpm. For example, a magnet drive rated at 50 hp at 3600 rpm can have 10 hp of additional heat input due to eddy current losses. Since bearing failure essentially takes place when any remaining liquid in the pump boils off and boundary lubrication is lost, the benefits of nonconductive containment shells become clear. Another benefit is that the lower heat generation with nonmetallic shells would also serve to extend dry running time without permanent loss of magnetism to the magnetic rotors. Typical nonconductive containment shell materials are available in zirconium oxide and various reinforced thermoplastic and thermosetting materials.

There are; however, some tradeoffs when selecting between nonconductive and conductive containment shells. Reinforced thermoplastic and thermosetting materials are generally limited to 250°F and have dimensional stability and mechanical properties that degrade with increasing temperature. Chemical resistance of these materials range from good to excellent; however, they are not suitable for abrasive applications.

Ceramics on the other hand, have excellent dimensional stability and abrasive resistance and possess high mechanical strength at elevated temperatures. Chemical resistance is also very good.

Ceramics, however, tend to be notch sensitive and may have application limitations based on thermal shock resistance. Proper design, manufacture, and quality control therefore are essential.

Nonmetallic containment shells are not covered by ASME Code; however, some design aspects are covered by the Hydraulic Institute Sealless Standards. Thicker wall sections are also required in nonmetallic shells resulting in increased magnet volume.

Resilient Bearing Mounts

This provision is essential for high temperature operation and dry running due to the adverse differential expansion effects between silicon carbide and the pump shaft and stationary bearing carrier material. For 316 stainless steel, the variation in expansion coefficient is approximately 4.5:1 in favor of 316 stainless steel. Thus, some means are required to prevent the growth in shaft diameter from cracking the rotating silicon carbide journal. Additionally, means are required to prevent the stationary journal from becoming loose in its mounting, as the bearing carrier expands faster than the bearing.

One additional benefit of resiliently mounted bearings is that they can provide a limited amount of self-aligning capability, thereby helping to reduce the risk of edge loading which will quickly destroy a silicon carbide bearing. Typical types of resilient mounts used are tolerance rings and O-rings.

Auxiliary Circulation Vanes

Under broken prime conditions, when there is still a considerable amount of liquid in the pump, these vanes usually located at the rear of the inner magnet assembly (that normally provide pressurized cooling), help to circulate the product lubricant. This provides for more uniform heating of the liquid, thereby reducing the risk of localized hot spots and accompanying fluid flashing.

Impeller balance holes may also be beneficial for dry running under broken prime conditions with open impellers. At the instant prime is broken, the balance holes would provide a low pressure path for relieving high pressure behind the impeller thereby reducing the instantaneous high axial thrust.

TEST MODELS AND PROCEDURE

Previous research [1] has identified the relative performance between various bearing materials and combinations in a special test rig. However, the true test is to evaluate the dry running performance of any product lubricated bearing material while operating under the dynamics and heat transfer conditions of an actual pump. This study was conducted to evaluate sintered silicon carbide and 70 micron controlled porosity silicon carbide with the addition of Teflon lubricating strips under two types of dry run conditions in an operating pump. The tested dry run modes were with broken prime condition, which is a common field occurrence during tank unloading operations, and a drained pump condition, a much more severe test, whereby only wetness remains within the liquid containing parts. The test liquid was water at ambient temperature.

The test pump, a $1^1/_2 \times 1 \times 8$ ANSI end suction magnetic drive pump, shown in Figure 1, is coupled to a 15 hp 3600 rpm motor. As shown, the containment shell area and product lubricated bearings are fully drainable through the pump casing.

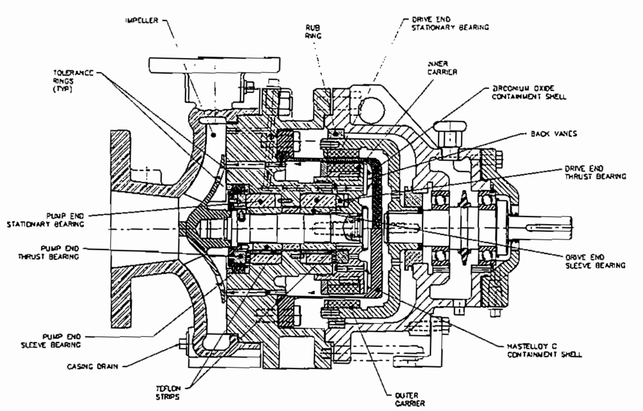

Figure 1. Test Model $1^1/_2 \times 1 \times 8$ ANSI Magnetic Drive 15HP/ 3600 RPM.

The test loop shown in Figure 2 was built to simulate dry running conditions as would occur when pumping a tank dry. After normal pumping is established, the dry run cycle is begun by closing the suction valve upstream of the standpipe. Since the standpipe is open to atmosphere; liquid is drawn down until prime is broken. The test pump was fitted with a type J thermocouple positioned near the outer diameter (OD) of the pump end stationary bearing as shown in Figure 3. The drive end bearing was inaccessible and could not be monitored. The recorded bearing temperatures should be considered as relative values rather than absolute.

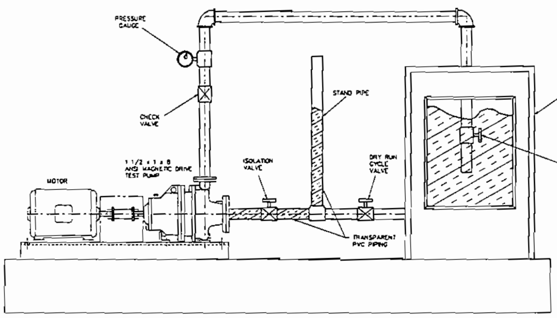

Figure 2. Test Rig Layout.

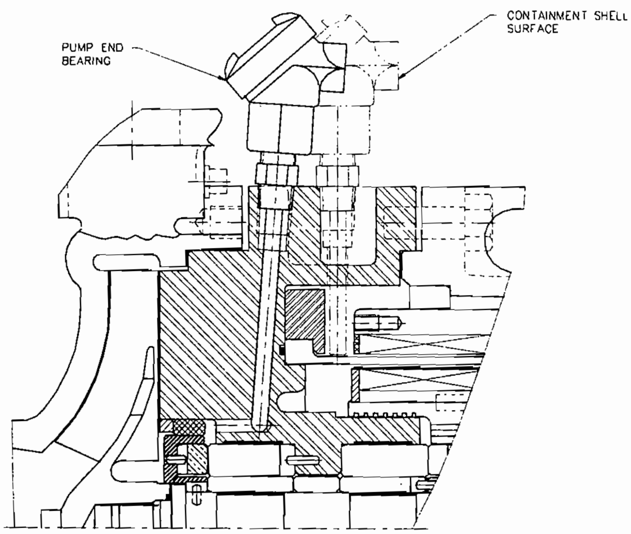

Figure 3. Thermocouple Layout.

The dry running tests under drained pump conditions were conducted after 30 minutes of normal pump operation. The pump was then shut down and drained as fully as possible through the casing drain connection. The suction and discharge piping was then removed.

The tested configurations are shown in Table 1. For each test configuration the rotating journal, stationary journal and thrust bearings were of the same material. The test configurations using Teflon lubricating strips were arranged with three strips equally spaced. Each strip was fitted into grooves within the stationary journal bearing as shown in Figure 4. The strips were L-shaped in order to provide lubrication to both radial and thrust faces of the silicon carbide bearings. Under normal pumping operation the lubricating strips would be inactive; however under dry running conditions the additional heat generated would force the lubricating strip to expand outward, due to its greater thermal expansion coefficient, and wipe the radial and thrust surfaces of the bearing. The 70 micron controlled porosity silicon carbide provides enhanced lubrication by virtue of its spherical micropores which act as tiny reservoirs to trap pumped liquid and Teflon between running surfaces. Once normal lubrication is reestablished bearing temperatures would decrease and the lubricating strip would retreat back into its groove where it

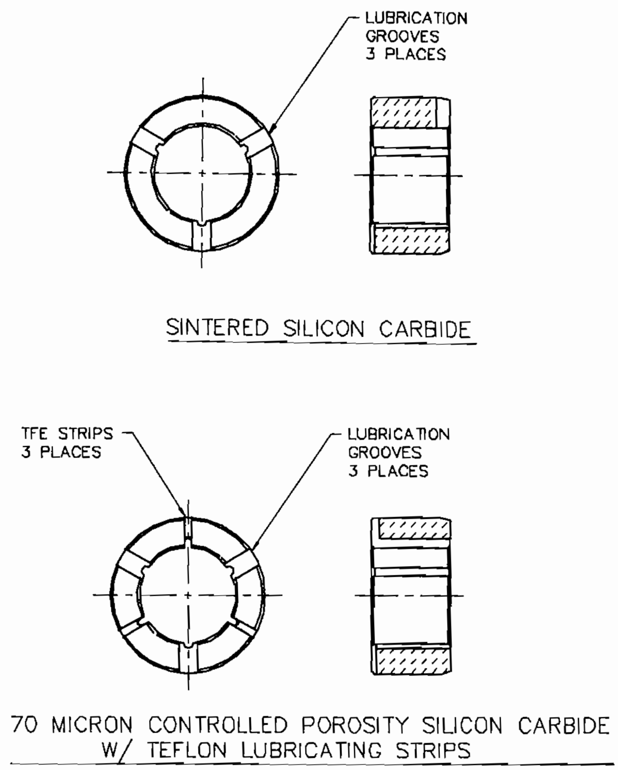

Figure 4. Bearing Test Configurations.

Table 1. Test Configurations.

Test #	Magnet Material	Containment Shell Material	Type Test	Stationary Bearing	Rotating Bearing & Thrust Bearing	Diametral Bearing Clearance (ins.) Pump End	Drive End
1A	NdFeB	Zirconium Oxide	Broken Prime	70 Micron Porosity SiC w/Teflon Strips	70 Micron Porosity SiC	.0025	.002
2A	NdFeB	Zirconium Oxide	Drained Casing	70 Micron Porosity SiC w/Teflon Strips	70 Micron Porosity SiC	.0025	.002
3A	NdFeB	Ziurconium Oxide	Broken Prime	Sintered SiC	Sintered SiC	.0025	.003
4A	NdFeB	Zirconium Oxide	Drained Casing	Sintered SiC	Sintered SiC	.0025	.003
1B	SmCo	Hastelloy C	Broken Prime	70 Micron Porosity SiC w/Teflon Strips	70 Micron Porosity SiC	.003	.003
2B	SmCo	Hastelloy C	Drained Casing	70 Micron Porosity SiC w/Teflon Strips	70 Micron Porosity SiC	.003	.0035
3B	SmCo	Hastelloy C	Broken Prime	Sintered SiC	Sintered SiC	.0025	.0025

would remain inactive. Although some wear should be expected to occur in the Teflon strips when pumping abrasive solids, the concentration of these solids at the product lubricated bearings is typically limited to two to five percent. If prolonged wear were to occur in the Teflon strips, the added benefit of the controlled porosity silicon carbide would still remain.

Tests were performed with both a zirconium oxide containment shell and a Hastelloy C containment shell. The benefit of testing with nonconductive and conductive containment shells is to isolate the dry running performance with and without the additional heat attributed to eddy current losses. The eddy current loss in the test pump fitted with a Hastelloy C containment shell was 2.5 hp at the 3550 rpm test speed.

RESULTS AND DISCUSSION

Results showed that during broken prime tests, as shown in Figure 5 Test IA, the controlled porosity silicon carbide with Teflon lubricating strips and zirconium oxide containment shell ran for $4^1/_2$ hours without overheating or failure. The test was discontinued due to time constraints. After dismantling the unit, it was found that no damage had occurred, and it was expected that this unit would have continued to run in these conditions successfully for many more hours. Sintered silicon carbide as shown in Test 3A exhibited a similar trend on a test of shorter duration, however; the dismantling inspection did show minor damage to the drive end thrust bearing.

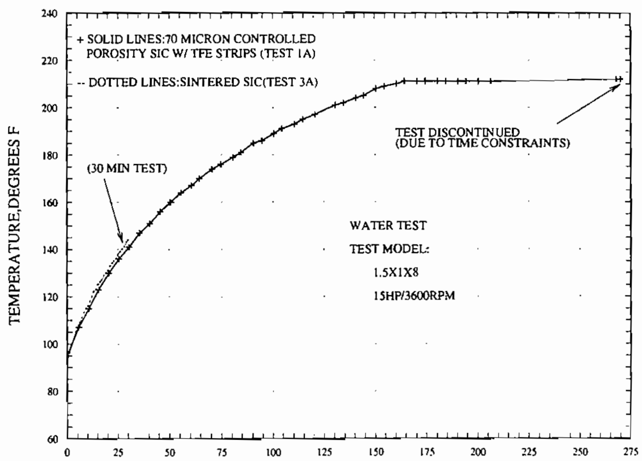

Figure 5. Broken Prime Test—Zirconium Oxide Containment Shell.

Drained pump Tests 2A and 4A shown in Figure 6 with zirconium oxide containment shell show a doubling of dry run time with the controlled porosity silicon carbide and Teflon strips when compared to sintered silicon carbide. It is interesting to note that during drained pump tests 2A Figure 6 and 2B Figure 7 the configuration with controlled porosity silicon carbide and Teflon strips was at least as durable as the magnet material which experienced a permanent loss of magnetism as indicated in Table 2. The tests do show that with proper pump design and the absence of edge loading, silicon carbide is more tolerant of short term dry running than typically recognized, especially when used with nonmetallic containment shell designs. The effects of some deliberate minor misalignment are dramatic as shown in Figure 6 Test 5A. This test configuration with controlled porosity silicon carbide without Teflon strips reduced dry run time to failure over a properly aligned bearing to 40

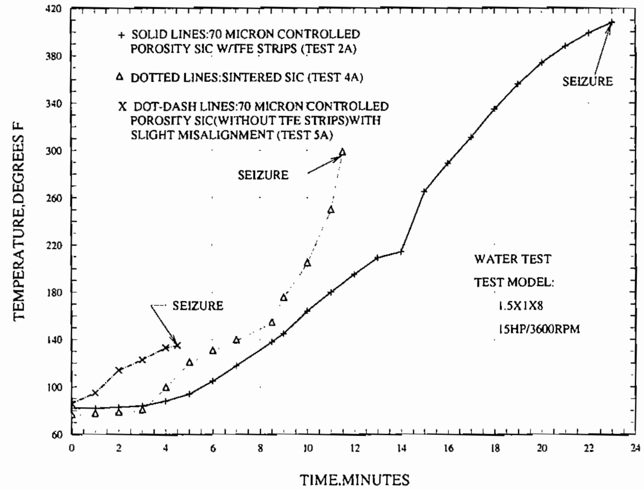

Figure 6. Drained Pump Test—Zirconium Oxide Containment Shell.

percent that of sintered silicon carbide and 20 percent that of controlled porosity silicon carbide with Teflon strips. The amount of misalignment which was introduced into the drive end stationary bearing was 0.003 in in perpendicularity, with respect to the thrust face. This perpendicularity is normally specified as 0.001 in or less.

The drained pump tests with controlled porosity silicon carbide and Teflon strips showed an unexpected result when tested alternatively with conductive and nonconductive containment shells. It was found that dry run time to failure was 50 percent greater with the Hastelloy C shell (Figure 7 2B) as compared to the zirconium oxide shell (Figure 6 2A). It is believed that the increased running time may be largely attributed to the high concentration of eddy current heat at the containment shell area. This may have helped to maintain bearing clearances by transmitting more of this heat to the stationary bearing due to its proximity to the containment shell, thereby expanding it to a greater degree than the rotating journal. It should also be noted that initial bearing clearances in the Hastelloy C shell test was on the average 0.001 in larger than in the zirconium oxide shell test.

Tests 1B and 3B, shown in Figure 7, were broken prime tests using the Hastelloy C shell. In these tests there was a period

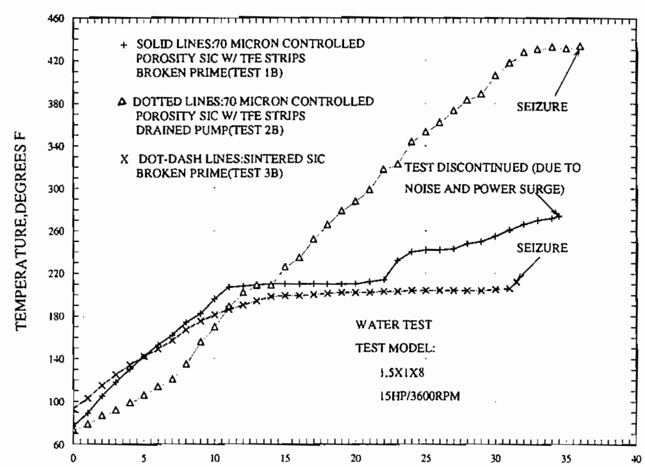

Figure 7. Broken Prime and Drained Pump Tests Hastelloy C Containment Shell.

Table 2. Damage Assessment - Major Components.

	Test #'s						
Component	1A	2A	3A	4A	1B	2B	3B
Impeller:							
rubbing	No	No	No	No	No	No	No
Containment Shell:							
rubbing contact	No	Slight	No	No	No	Moderate	No
breakage	No	No	No	No	No	No	No
distortion	No	No	No	No	No	No	No
Outer Carrier:							
bumper ring rub	No	No	No	No	No	No	No
containment shell rub	No	No	No	No	No	Slight*	No
loss of magnetism	No	No	No	No	No	Slight Loss	No
distortion	No	No	No	No	No	Yes*	No
Inner Carrier:							
bearing carrier rub	No	No	No	No	No	No	No
containment shell rub	No	Yes	No	No	No	Yes	No
distortion	No	Yes	No	No	No	Yes	No
loss of magnetism	No	Complete	No	No	No	Nearly Complete	No
Product Lubricated Bearings:							
Pump End Sleeve Bearing:							
scoring	No	Slight	No	Heavy	No	Slight	Slight
chipping	No	No	No	Yes	No	No	Yes
breakage	No	No	No	Yes	No	No	No
Drive End Sleeve Bearing:							
scoring	No	Heavy	No	Heavy	Slight	Heavy	Heavy
chipping	No	Yes	No	Yes	Yes	Yes	Yes
breakage	No	Yes	No	Yes	No	Yes	Yes
Pump End Stationary Bearing:							
scoring	No	Slight	No	Slight	No	Slight	Slight
chipping	No	No	No	Yes	No	No	No
breakage	No	No	No	No	No	No	No
Drive End Stationary Bearing:							
scoring	No	Heavy	No	Slight	Slight	Slight	Slight
chipping	No	Yes	No	Yes	No	No	Yes
breakage	No	Yes	No	Yes	No	No	No
Pump End Thrust Bearing:							
scoring	No	No	No	Moderate	No	No	No
chipping	No	No	No	No	No	No	No
breakage	No	No	No	No	No	No	No
Drive End Thrust Bearing:							
scoring	No	Heavy	No	No	Slight	Slight	Slight
chipping	No	Yes	Yes	No	Yes	Yes	Yes
breakage	No	Yes	No	No	No	Yes	Yes
Teflon Strips:							
recessed (due to use)	Slight	Yes	N/A	N/A	Yes	Yes	N/A
melted	No	Yes**	N/A	N/A	No	Yes**	N/A
Test Minutes:	$4^1/_2$ hrs.	23 min.	30 min.	$11^1/_2$ min.	$34^1/_2$ min.	36 min.	$31^1/_2$ min.
Failed:	No	Yes	No	Yes	No	Yes	Yes
SiC Bearing Damage Factor:	0	.5	.12	.88	.28	.28	.67

* Epoxy potting between magnets swelled and rubbed against O.D. of containment shell.
**Drive end Teflon strips only.

where the water took time to boil off as is evident from the flat regions of these curves. As noted, test 1B, with controlled porosity silicon carbide and Teflon strips, did not seize but was shutdown after $34^1/_2$ minutes, due to noise and a power surge. After a minute or two of cooling by readmitting ambient temperature water to the pump, the shaft turned freely and the unit was restarted and pumped successfully for 30 minutes without incident prior to disassembly. Subsequent inspection showed that the unit had comparatively minor damage as opposed to the sintered silicon carbide material (Test 3B) as shown in Table 2.

Damage Assessment

The damage to the major components for each test configuration is shown in Table 2. The failure mode in each case was due to a loss of clearance between the stationary and rotating journals. This caused the inner rotating assembly to seize, and subsequently shatter, the rotating journal bearing. In all tests where the pump seized, the drive end rotating journal bearing shattered. In test configurations that did not have the additional benefit of the Teflon lubricating strips and controlled porosity silicon carbide, bearing damage was more severe and is reflected in a higher bearing damage factor, as shown in Table 2.

$$\text{bearing damage factor} = \frac{\text{cost of failed bearings}}{\text{cost of complete bearing set}}$$

The overall part replacement cost may be higher if total pump damage consequences is taken into account.

Bearing clearances should not arbitrarily be increased to improve dry running performance, since these clearances must be small enough to maintain proper alignment with respect to the inner and outer magnet carriers, and to prevent contact between rotating and stationary parts. Bearing clearances however, must also be adequate to allow for sufficient cooling and lubrication. Typical diametral bearing clearances are 0.002 in to 0.006 in. Good design practice is to have a radial lubrication groove(s) on the thrust face of the stationary journal and an axial lubrication groove(s) on the radial surface of the stationary journal. This will improve lubrication flow and allow for passage of small solids, if any.

It should be noted that no detectable wear could be measured on any of the surviving bearings other than scoring where noted. Such scoring generally occurred in both rotating and stationary journals. Also noted is that in no case did a failure occur due to inner magnet carrier contact with the containment shell. In drained pump tests 2A and 2B, heat was high enough, however, to cause swelling of the inner magnet carrier and subsequent rubbing contact with the containment shell. There was a complete loss of inner carrier magnetism in test 2A, which was constructed with neodymium magnets, and a nearly complete loss of magnetism in test 2B, which used samarium cobalt magnets. As a reference the Curie temperature for neodymium magnets is 590°F and for samarium cobalt magnets it is 1472°F. The Curie temperature is the transition temperature above which the magnetic material permanently loses all magnetic properties.

CONCLUSION

Test results based on water show that, with robust pump design and proper alignment, silicon carbide ran dry successfully for short periods with a near lack of product lubricant. Silicon carbide under broken prime conditions with nonconductive containment shells, ran successfully for longer periods. The addition of controlled porosity silicon carbide and Teflon lubricating strips can add substantially to dry run life and help reduce damage to the product lubricated bearings.

The test model with nonconductive containment shell and 70 micron controlled porosity silicon carbide bearings with Teflon lubricating strips ran for $4\frac{1}{2}$ hours under broken prime conditions without overheating, failure or damage. Upon examination of the unit, it was expected that the pump could have continued to run under these conditions successfully for many more hours.

The test results are based on a 15 hp ANSI magnetic drive pump operating at 3550 rpm using ambient temperature water as a product lubricant. Liquids having lower specific heat, less lubricity and higher volatility than water would be expected to have a more rapid temperature rise and, consequently, shorter dry running times. Additionally, pumps with conductive containment shells having smaller magnetic drives and lower eddy current losses would have longer dry running times than larger magnetic drives with higher eddy current losses. This is, at least partly, a function of the amount of heat input, due to eddy current losses and the time required to boil off the remaining liquid in the pump.

REFERENCE

1. Lorenc, A. J. and Nelik L., "Analysis and Testing of Alternative Bearing Materials for Applications in Sealless Pumps," *Proceedings of the Eleventh International Pump Users Symposium,* Turbomachinery Labortory, Texas A&M University, pp. 51-56 (1994).