ANALYSIS AND TESTING OF ALTERNATIVE BEARINGS MATERIALS FOR APPLICATIONS IN SEALLESS PUMPS

by

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ABSTRACT

As the usage of sealless pumps increases, so do the demands for journal bearing materials that can operate successfully through the thermal upsets that can take place in these pumps. Material specification literature is only a start at making a knowledgeable choice. A test facility and procedure was developed to evaluate different journal bearing materials for use in sealless pumps. The facility simulates an actual American Petroleum Institute (API) sealless pump bearing installation. Fifteen different material combinations were tested. Tabulated and graphed test results are presented. A 30 percent carbon fiber polyetheretherketone (PEEK) and a 15 percent graphite filled polyimide material operated 40 minutes in a no flush water condition. Silicon carbide and most of the carbon graphite based materials had no wear after the no flush water tests were terminated. Polymer based materials had the most wear and operated the shortest amount of time in the no flush water condition.

INTRODUCTION

The increased demand for the production and use of sealless pumps has produced a number of technical challenges. One of those challenges is the proper material selection and design of the journal bearings (which are known to fail catastrophically under upset conditions) used to support the driven rotating assembly (driven magnet carrier, shaft and impeller) in these pumps. The lubrication for these journal bearings is the pumpage. Dry running capabilities of these sealless pump bearings is a key issue for successful and reliable operation during transient process conditions that alter the flow of fluid through the pump. During these transients. many bearing materials experience thermal shock, leading to failure. Since these bearings are running inside the pump, and not readily accessible, it is difficult to determine the exact nature of their failure. It is known that bearing failure may trigger a series of events that can result in puncture of the isolation shell, thereby allowing fugitive emissions.

It is desirable to evaluate the performance of these bearings, for different material combinations and conditions, by separate testing outside the actual pump, to properly observe their failure mode. Such focused testing ensures a known and controlled bearing environment, such as applied load, stationary to rotating bearing alignment, consumed power, temperature and flowrate. The nature of the failure can be clearly determined, with quantifiable results, to compare dry running capabilities of various materials such as silicon carbide, carbon and polymers. This systematic evaluation of these different materials can then be related to a "real world" pump environment. The test facility, test procedure, test data, and results of such an effort are presented.

BEARING TEST FACILITY

A test facility was constructed specifically to test the different journal bearing materials (Figures 1 and 2). A 40 hp, 3560 rpm motor, driven by a adjustable speed controller, was used. The geometry of the shaft and stationary bearing housing were identical to an API magnetic drive sealless pump. Therefore, the journal bearings used for the API sealless pump could be tested. This journal bearing test facility also has the flexibility to accommodate and test, by simple alterations, other bearing geometries, such as the bearings used in the ANSI sealless pumps or other similar journal bearings.

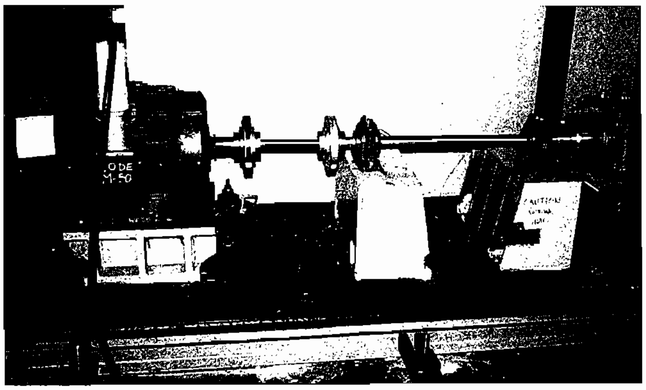


Figure 1. Overall View of Bearing Test Facility.

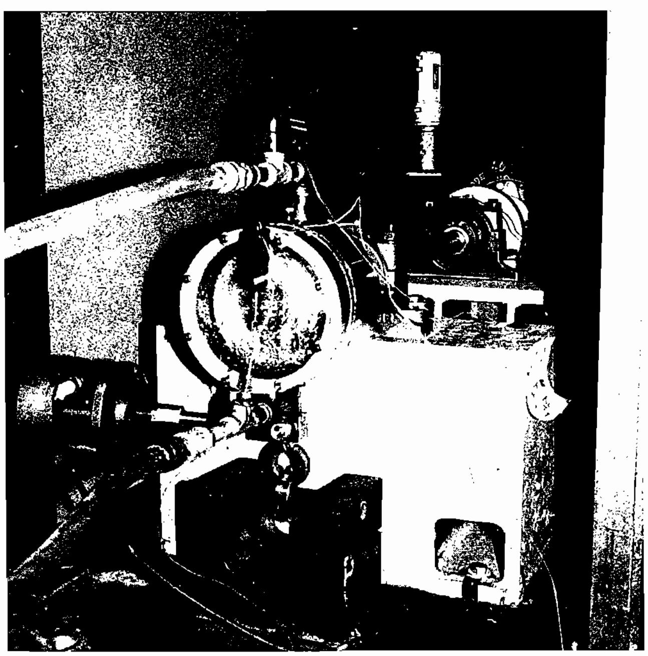


Figure 2. Closeup Front View of Test Rig and Instrumentation.

A sectional drawing of the journal bearing test rig is shown in Figure 3. Flush water was introduced in the top, back side of the stationary bearing housing and the water drain was located in the bottom, front. The front plate of the stationary bearing housing was constructed of clear plexiglass so the flow through the bearings could be observed. Flush water inlet and water drain temperatures along with flow were measured. The stationary bearing housing was restrained from rotating by a load cell rated for 200 lb (Figure 2).

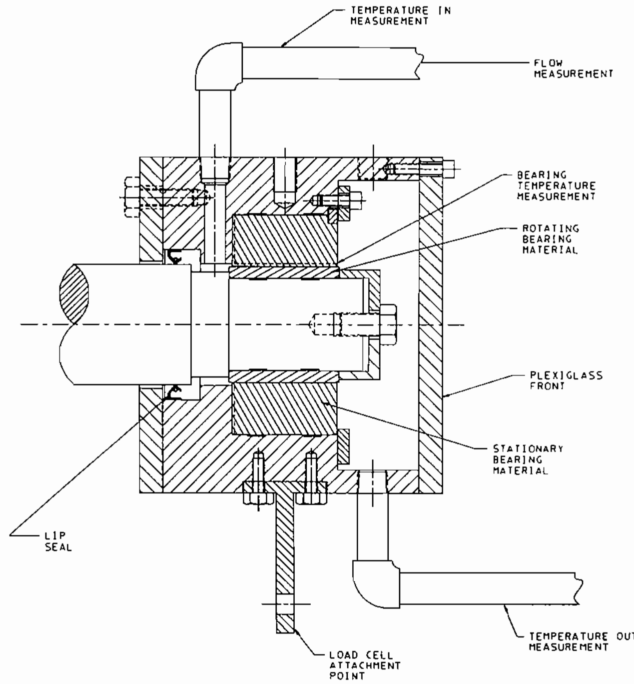


Figure 3. Sectional Drawing of Test Rig and Instrumentation.

The total weight of the stationary bearing housing along with a block weight beneath it was 92.5 lb. This approximated the total weight of the driven rotor assembly from the API pump that these bearings were designed to fit. The scope of this research project focused on the pump suddenly operating without any fluid (empty tank or suction valve shutoff). Therefore, there would be no hydraulic radial and axial forces acting on the bearings. It is recognized that this may not always be true in which case the bearings would be subjected to both the weight and unbalance forces of the driven rotor assembly and the impeller hydraulic forces.

The API magnetic drive designed journal bearings have a nominal diameter of 2.5 in and are 2.25 in long. This gives a projected bearing surface area of 5.625 square inches. Material combinations along with their configuration (test) number are listed in Table 1.

TEST PROCEDURE

An important specification throughout the bearing materials testing was to subject each bearing material combination to the same test conditions. A test procedure was created that duplicated, as close as possible, actual pump operating conditions.

All the rotating journals were mounted on the shaft with tolerance rings. The stationary journals were either pressed into the bearing housing, mounted into the bearing housing using tolerance rings, or pressed into an adapter that was mounted into the bearing housing with tolerance rings. After each bearing combination was inserted on the shaft and in the bearing housing, it was allowed to "creep" for 24 hr, to stabilize any possible deformations in the bearing's shape. The outside and inside diameters were measured using micrometers in four locations on both the rotating journal and stationary sleeve, respectively.

Once the test rig was assembled and all instrumentation was connected and operational, the flush water was turned on and the motor accelerated to 3560 rpm. During the next half hour, the flush inlet and drain temperatures, flush water flowrate, load cell measurement, and temperature near the bearing interface surface

Table 1. Combinations of Journal Bearing Materials Tested.

	CONFIG #	STATIONARY JOURNAL	ROTATING JOURNAL
CARBON	1	CARBON-GRAPHITE	316 STAINLESS STEEL
	2	CARBON-GRAPHITE	NiCrSiB COATED 316 STAINLESS STEEL
	4	BRONZE IMPREGNATED GRAPHITE	NiCrSiB COATED 316 STAINLESS STEEL
	12	CARBON-GRAPHITE	ALPHA SINTERED SILICON CARBIDE
POLYMERS	3	POLYIMIDE 15% GRAPHITE FILLER	NiCrSiB COATED 316 STAINLESS STEEL
	5	POLYETHERETHERKETONE PTFE/CARBON/GRAPHITE	NiCrSiB COATED 316 STAINLESS STEEL
	6	PTFE COMPOUND	NiCrSiB COATED 316 STAINLESS STEEL
	7	55% RYTON, 30% GRAPHITE & 15% TEFLON	NiCrSiB COATED 316 STAINLESS STEEL
	13	POLYETHERETHERKETONE ALIGNED CARBON FIBER	NiCrSiB COATED 316 STAINLESS STEEL
	14	POLYETHERETHERKETONE 30% CARBON FIBER	NiCrSiB COATED 316 STAINLESS STEEL
SILICON	8	6% GRAPHITE IMPREGNATED SILICON CARBIDE	6% GRAPHITE IMPREGNATED SILICON CARBIDE
	9	50 MICRON POROSITY SILICON CARBIDE	50 MICRON POROSITY SILICON CARBIDE
	10	SILICONIZED GRAPHITE WITH ANTIMONY	REACTION BONDED SILICON CARBIDE
	11	ALPHA SINTERED SILICON CARBIDE	ALPHA SINTERED SILICON CARBIDE
	15	12% GRAPHITE IMPREGNATED SILICON CARBIDE	12% GRAPHITE IMPREGNATED SILICON CARBIDE

(Figure 3) on the stationary sleeve, were recorded to establish a baseline. Because of excessive flooding in the bearing housing, flush water flow was limited to 60 gal/hr. All measured parameters reached steady state within this 30 minute run time.

After this half hour of run time, the flush water was turned off. A timer was started at this point. Any one of the following criteria was used to define a bearing failure and terminate the test:

- Motor horsepower exceeded 10 hp.
- Stationary sleeve temperature exceeded 250°F.
- Abnormal noises from the bearing under test.

If none of the above criteria existed, the test was terminated forty minutes after the flush water was turned off. Upon completion of the run test, the bearings were allowed to cool 24 hr, after which the outside and inside diameter measurements were taken at the same locations. The bearings were then disassembled and inspected.

CALCULATIONS

The coefficient of journal friction (f) was calculated as follows:

$$f = \left[\frac{M}{W \times r} \right]$$

where

M = moment of journal friction, inch pounds
W = total load on journal, pounds
r = nominal radius of journal, inches

The journal bearing pressure velocity value (PV) in psi ft/min was calculated as follows:

$$PV = \left[\frac{W}{2 \times r \times L} \right] \times \left[k1 \times r \times N \right]$$

where

L = length of the bearing, inches
N = rotational speed of bearing, rpm
k1 = 0.5236, units conversion constant

Operating these API type bearings in this test facility at 3560 rpm with a load of 92.5 lbs results in a PV equal to 38,315 psi ft/min.

The heat power loss (BHPh), when the bearings were operated with water lubrication, was calculated as follows:

$$BHPh = k2 \times Q \times (To - Ti)$$

where

Q = flush water rate, gal/hr
To = water drain temperature, °F
Ti = water flush temperature, °F
k2 = 0.00327, units conversion constant

The friction power loss (BHPf), when the bearings were operated with water lubrication, was calculated a follows:

$$BHPF = \left[\frac{M \times N}{k3} \right]$$

where

k3 = 63025, units conversion constant

RESULTS

The 15 combinations of bearing materials tested were divided into three classes: carbon graphite, polymer and silicon carbide. Results for each class of bearings materials are presented.

The changes in the coefficient of friction after the flush water was shutoff for each bearing material tested is graphically presented in Figures 4, 5, and 6. The value for the Coefficient of friction greater than 1.0 were not included in Figures 4, 5, and 6 for simplicity (graphed values of 1.0 indicate that the actual values were larger).

The changes in the stationary bearing temperature after the flush water was shutoff for each bearing material tested is graphically presented in Figures 7, 8, and 9.

The original diametrical clearance and the maximum measured wear for each configuration tested is presented in Table 2. A tabulation of the water inlet and outlet temperatures, temperature difference, flush flowrate, bearing torque values with and without flush water, and the time to failure for each bearing material tested are in Table 3.

DISCUSSION OF RESULTS

Coefficient of Friction

Referring to Table 3, the coefficient of friction for all bearing materials tested with flush water is low, ranging from 0.026 to 0.078. Once the flush water was shut off, the expectation was that

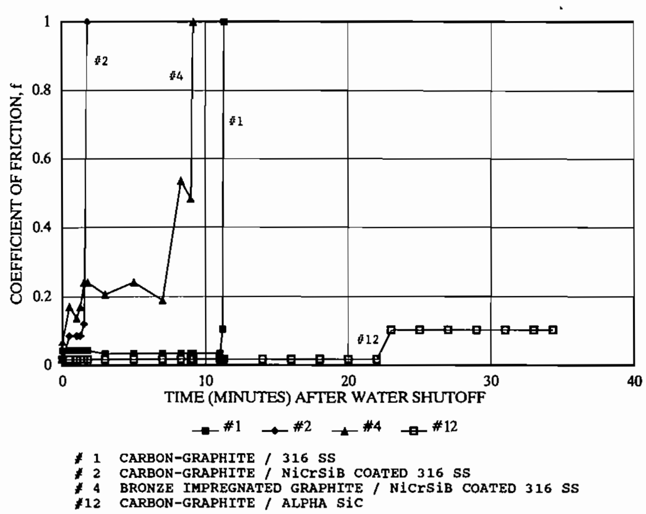
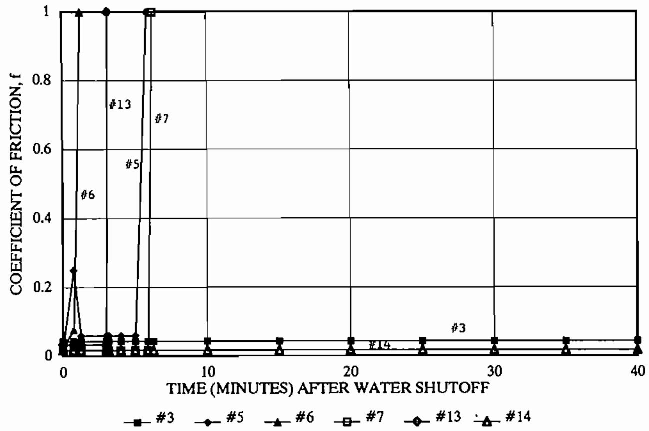
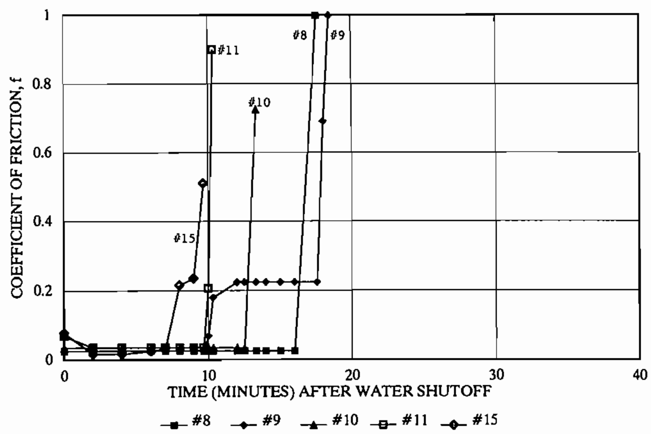


Figure 4. Coefficients of Friction for Carbon Graphite Bearings.



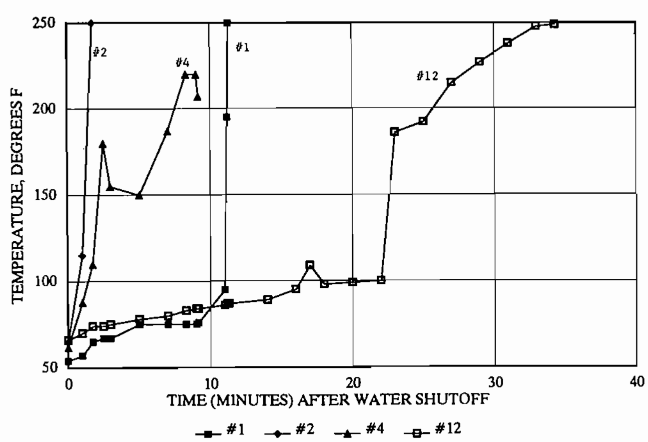
\# 3 POLYIMIDE 15% GRAPHITE / NiCrSiB COATED 316 SS
\# 5 PEEK (PTFE-CARBON-GRAPHTIE) / NiCrSiB COATED 316 SS
\# 6 PTFE COMPOUND / NiCrSiB COATED 316 SS
\# 7 RYTON, GRAPHITE, TEFLON / NiCrSiB COATED 316 SS
\#13 PEEK (ALIGNED CARBON FIBER) / NiCrSiB COATED 316 SS
\#14 PEEK (30% CARBON FIBER) / NiCrSiB COATED 316 SS

Figure 5. Coefficients of Friction for Polymer Bearings.



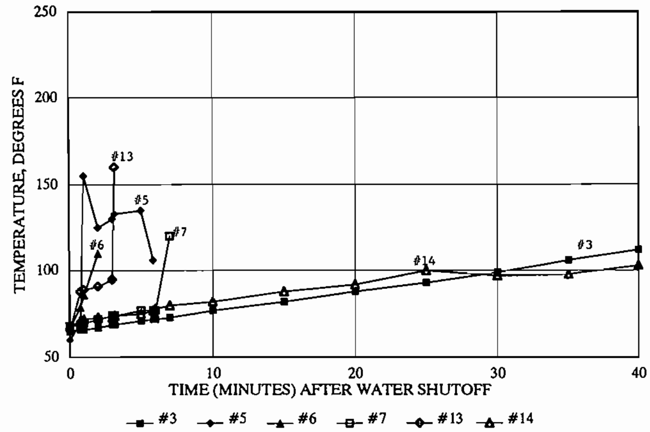
\# 8 6% GRAPHITE SiC / 6% GRAPHITE SiC
\# 9 50 MICRON POROSITY SiC / 50 MICRON POROSITY SiC
\#10 Si GRAPHITE WITH ANTIMONY / REACTION BONDED SiC
\#11 ALPHA SINTERED SiC / ALPHA SINTERED SiC
\#15 12% GRAPHITE SiC / 12% GRAPHITE SiC

Figure 6. Coefficients of Friction for Silicon Carbide Bearings.



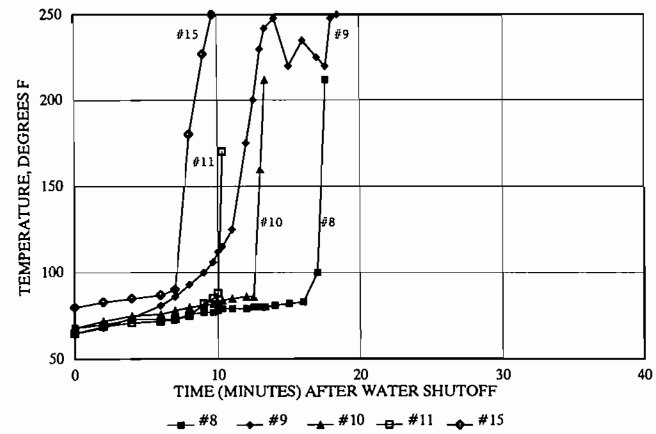
\# 1 CARBON-GRAPHITE / 316 SS
\# 2 CARBON-GRAPHITE / NiCrSiB COATED 316 SS
\# 4 BRONZE IMPREGNATED GRAPHITE / NiCrSiB COATED 316 SS
\#12 CARBON-GRAPHITE / ALPHA SiC

Figure 7. Stationary Bearing Temperatures for Carbon Graphite Bearings.



\# 3 POLYIMIDE 15% GRAPHITE / NiCrSiB COATED 316 SS
\# 5 PEEK (PTFE-CARBON-GRAPHTIE) / NiCrSiB COATED 316 SS
\# 6 PTFE COMPOUND / NiCrSiB COATED 316 SS
\# 7 RYTON, GRAPHITE, TEFLON / NiCrSiB COATED 316 SS
\#13 PEEK (ALIGNED CARBON FIBER) / NiCrSiB COATED 316 SS
\#14 PEEK (30% CARBON FIBER) / NiCrSiB COATED 316 SS

Figure 8. Stationary Bearing Temperatures for Polymer Bearings.



\# 8 6% GRAPHITE SiC / 6% GRAPHITE SiC
\# 9 50 MICRON POROSITY SiC / 50 MICRON POROSITY SiC
\#10 Si GRAPHITE WITH ANTIMONY / REACTION BONDED SiC
\#11 ALPHA SINTERED SiC / ALPHA SINTERED SiC
\#15 12% GRAPHITE SiC / 12% GRAPHITE SiC

Figure 9. Stationary Bearing Temperatures for Silicon Carbide Bearings.

the coefficient of friction would increase. Except for tests #4 and #13, the opposite was true, the coefficient of friction for each bearing tested initially decreased! The presence of the lip seal in the stationary bearing housing (Figure 3) could account for the extra friction when flushed with water.

Stationary Bearing Temperature

The graphs in Figures 7, 8 and 9 have an unexpected result. Generally, the temperature of the stationary bearings slowly increase when the flush water is first shut off. When the coefficient of friction increases rapidly, the temperature likewise increases. But, test #5 (a polymer bearing) had a rapid increase in temperature followed by a reduction. This corresponded with a rapid increase in the coefficient of friction followed by a reduction until failure. It is possible that the higher friction generates heat, which melts the contacting layer of polymer. This reduces the coefficient of friction by running the hard rotating bearing over the melting polymer stationary piece.

Test #9 also had a decrease in temperature. This material is a porous silicon carbide. The fluid is entrapped in the pores when

Table 2. Diametrical Clearances and Maximum Measured Wear of the Journal Bearing Materials.

	CONFIG #	DIAMETRICAL CLEARANCE INCHES	STATIONARY JOURNAL WEAR (INCHES)	ROTATING JOURNAL WEAR (INCHES)
CARBON	1	.003"	NONE	.004"
	2	.004"	NONE	NONE
	4	.003"	.035"	NONE
	12	.005"	NONE	NONE
POLYMERS	3	.009"	.002"	NONE
	5	.013"	.011"	NONE
	6	.008"	.013"	NONE
	7	.006	.009"	NONE
	13	.007"	.003"	NONE
	14	.011"	NONE	NONE
SILICON	8	.003	NOT AVAILABLE	NOT AVAILABLE
	9	.005"	NONE	NONE
	10	.005"	NONE	NONE
	11	.005"	NONE	NONE
	15	.004"	NONE	NONE

Table 3. Journal Bearing Test Data and Calculated Results.

CONFIG #	TIME TO FAILURE MINUTES	WATER TEMP. OUT To DEG F	WATER TEMP IN Ti DEG F	WATER DELTA TEMP To−Ti DEG F	WATER FLOW Q, GPH	HEAT POWER LOSS BHPh
1	11.28	49.3	47.1	2.2	42.0	0.303
2	1.62	62.1	58.0	4.1	25.0	0.336
4	9.17	59.2	53.0	6.2	18.0	0.365
12	34.33	64.1	62.3	1.8	48.0	0.283
3	40.00	74.7	73.3	1.4	* 60.0	0.275
5	5.83	59.1	57.6	1.5	* 60.0	0.295
6	1.25	64.0	61.0	3.0	42.0	0.413
7	6.25	68.1	64.8	3.3	* 60.0	0.648
13	3.19	63.8	61.1	2.7	30.0	0.265
14	40.00	66.0	65.1	0.9	* 60.0	0.177
8	17.58	64.2	61.0	3.2	50.0	0.524
9	18.45	65.2	62.4	2.8	30.0	0.275
10	13.33	66.0	65.3	0.7	* 60.0	0.138
11	10.33	63.6	62.4	1.2	* 60.0	0.236
15	9.67	77.9	71.0	6.9	24.0	0.542

* MAXIMUM FLOW−SET

CONFIG #	TORQUE RUNNING LUBRICATED IN.*LBS	TORQUE RUNNING DRY IN.*LBS.	COEF. OF FRICTION LUBRICATED	COEF. OF FRICTION DRY	FRICTION POWER LOSS BHPf
1	7.0	5.0	0.061	0.043	0.394
2	5.0	2.0	0.043	0.017	0.282
4	8.0	24.0	0.069	0.208	0.451
12	4.0	2.0	0.035	0.017	0.225
3	7.0	5.0	0.061	0.043	0.394
5	3.0	7.0	0.026	0.061	0.169
6	6.0	3.0	0.052	0.026	0.338
7	3.0	2.0	0.026	0.017	0.169
13	3.0	6.0	0.026	0.052	0.169
14	6.0	2.0	0.052	0.017	0.338
8	8.0	3.0	0.069	0.026	0.451
9	8.0	3.0	0.069	0.026	0.451
10	6.0	4.0	0.052	0.035	0.338
11	5.0	4.0	0.043	0.035	0.282
15	9.0	2.0	0.078	0.017	0.507

operating in the lubricated state. Once flush is shutoff the trapped fluid expands out of the pores as the bearing heats and lubricates the bearing. As the temperature of the bearing exceeded 212°F, the fluid vaporized, providing both cooling and lubrication. Eventually all the fluid is vaporized from the bearing leading to increased temperature, coefficient of friction, and failure.

Bearing Wear

Except for the 316 stainless steel rotating journal, none of the rotating journals exhibited any wear (Table 2). Configuration #8 was a silicon carbide bearing that shattered because the test was allowed to continue after the bearing noise first started, therefore no measurements could be made after the test. In most tests, the wear occurred on the stationary journal material. The bronze impregnated graphite bearing had the most wear, 0.035 in. This material is not designed to handle the high dry running PV levels this test was conducted at and was expected to wear. None of the other carbon graphite or silicon carbide stationary bearings had any measurable wear. Except for the material used in test #14, all the polymer bearings exhibited wear.

Bearing Clearance

From the pump design standpoint, the bearings need to have as small as possible bearing clearance to keep the driven rotating assembly in alignment and prevent the driven magnetic carrier, impeller wear rings, or impeller from rubbing against the stationary parts of the pump. Also, relatively small bearing clearances are normally desired from a rotordynamics standpoint. On the other hand, the bearings have to be designed with adequate clearances to allow sufficient flush through them to remove the generated heat and allow for the expansion of the bearing materials with changes in pump and pumpage temperatures, especially during thermal transients.

Presently, typical product lubricated bearing diametrical clearance is in the range of 0.003 in to 0.006 in. All the carbon graphite and silicon carbide bearings were manufactured to these tolerances. Their mode of failure was usually an expansion of the bearing materials, due to heat generated by friction until the clearance was eliminated, causing the bearing to bind. If some of these bearing materials were tested with larger clearances there, time to failure would increase. Therefore, a proper compromise between the rotor contacting, rotordynamics, and thermal transients must be carefully investigated for a successful journal bearing design.

Because polymer materials have a higher coefficient of thermal expansion, their diametrical clearances range is from 0.006 in to 0.011 in. Except for bearing #5, all the polymer bearings had clearance within this range. As the temperature of these materials increase, in addition to expanding and binding, they also can melt, deform and then bind up. The polymer in test #14 was tested with two clearances 0.016 in and 0.011 in. During both tests, the bearing's coefficient of friction never increased; and its temperature increased gradually. After 40 minutes of operation, the tests were terminated to inspect the bearing for wear. The polymer in test #3 was tested with two clearances 0.003 in and 0.009 in. It failed in 45 secs with a clearance of 0.003 in. At 0.009 in clearance the bearing had only a gradual increase in temperature with no change in coefficient of friction. It did have some wear after operating for 40 minutes. These two bearing materials (configurations #3 and #14) never bound up, because they were operated with sufficient clearance, and high temperatures were never reached to cause sufficient expansion or any melting.

Bearing Alignment/Mounting

The same installation, measurement, instrumentation, and test operating procedure was used on each bearing material combination. Inspection of the stationary bearing materials used in tests #6 and #13 revealed wear patterns on only one end of the bearing. In the other bearing tests, the wear patterns were on the inner diameter across the entire length of the top of the bearing. Even though differences with bearing alignment or mounting design may be two possible reasons, there is no clear explanation for this difference in wear patterns.

Power Loss

Both the heat power loss and friction power loss calculation results are in Table 3. These calculations were computed in an effort to determine the power consumed by the journal bearings. The correlation between the two methods is not acceptable. The reason being the lack of sensitivity of both the torque and temperature measurements. However, the measurements do indicate that the journal bearings consume on the average about 0.25 hp per bearing.

Actual Pump Operating Experience

The results obtained from the bearing testing program have been applied to commercial magnetic drive sealless pumps, both ANSI and API specifications. For ANSI pumps, silicon carbide bearings have been the standard applied for many years at chemical and petrochemical plants. An API magnetic drive sealless pump has been successfully tested with graphite, silicon carbide and polymer journal bearings. None of the bearing materials had measurable wear after the pump was operated properly throughout its flow range. Silicon carbide bearings have also been tested in this pump at pumpage temperatures to 400°F with no failures (Figure 10).

CONCLUSION AND SUMMARY OF RESULTS

There are several conclusions in regard to the proper material selection and design for journal bearings:

- The bronze impregnated graphite was the only carbon graphite bearing of the four tested that had wear.
- Except for the 30 percent carbon fiber filled PEEK material (configuration #14) all other polymers tested had wear.
- None of the silicon carbide bearings had any measurable wear.
- Proper clearance for the bearing material selected is critical for successful dry running operation.
- From the 15 bearing materials tested, the 30 percent carbon fiber PEEK and the 15 percent graphite filled polyimide (configurations #3 and #14) bearings operated the longest in the no flush water condition.

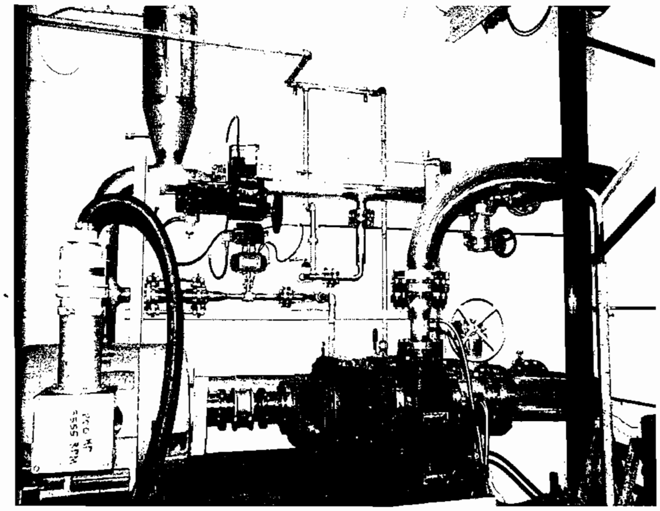


Figure 10. API Sealless Magnetic Drive Pump Operating at 400°F, in Hot Loop Test Facility.

This effort is but a start at understanding the capability of different bearing materials for use as journal bearings in pumps. As more materials become available, they will be subjected to the same test to determine their initial promise. Further research testing is being planned. The chemical resistance of the materials tested needs to be reviewed before specifying any of these materials for pump bearings.