DEVELOPMENT OF A MAGNETIC BEARING API PROCESS PUMP WITH A CANNED MOTOR

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ABSTRACT

A new class of pumps has been developed where both end face seals and contacting bearings are eliminated. The objective is to provide a sealless pump with long life bearings meeting the duty requirements of API Standard 610 for centrifugal pumps. The pump combines hydraulic components, impellers and volute, cases, from existing product lines, with a derivative canned motor and an active magnetic bearing system capable of supporting the pump loads under all operating conditions.

A prototype pump has been built and tested. The tests were successful in proving the concept and after correction of minor manufacturing problems, the pump is expected to enter field demonstration service in early 1992.

Besides providing a pump with no mechanically contacting surfaces, the addition of magnetic bearings provides useful diagnostic output of pump loads and rotordynamics generally not available in prior designs. These new capabilities should prove very useful in the evaluation and planned maintenance of magnetic bearing equipped pumps.

INTRODUCTION

Recent federal and state clean air legislation will require users of centrifugal pumps handling volatile liquids to monitor their emissions and retrofit units that do not comply with fugitive emission limits. Retrofits will include new technology such as tandem or double seals or sealless pumps. These modifications are generally less reliable and will add increased maintenance costs to the burden of the monitoring program costs. One solution to this is to add active magnetic bearings to a canned motor pump to produce a highly reliable pump with no mechanically contacting parts during its complete operation cycle, including startup and shutdown.

Sealless pumps, including those fitted with magnetic drive and traditional canned motor pumps, have depended on product lubricated bearings to support the pump rotor. These bearings have several problems, especially when applied to the varied applications typical of API 610 pump duties. The pumped liquid often acts as a poor lubricant, due to its low density and viscosity. The pumpage often is close also to its boiling point and can flash in the bearings causing premature failure. One popular bearing materials, carbon graphite, is subject to rapid wear in the presence of catalyst fines or coke particles common in refinery streams. The other most common type of bearing material, silicon carbide, has poor running capability when dry or in flashing conditions. One final problem with these arrangements is the difficulty of assessing the condition of the bearings, since the rotor is not directly accessible and the good vibration transmissibility found in ball bearings systems is lacking.

Magnetic bearings address all these issues. They do not depend upon the pumped product for lubrication and can even run dry. In addition, the clearances are several times larger than product lubricated bearings and therefore, are able to pass larger solid particles. Finally, the control system provides continuous monitoring capability of the rotor vibration and bearing loads. Magnetic bearing systems do introduce new reliability considerations. First, they require reliable power, since the rotor loses all support if the electric supply fails to the controller. One consequence of this is that the pump drive must always be interlocked to trip when the bearing power fails. Auxiliary power can be added in the case of critical services, to allow orderly shutdown of pumps and bearings. The control system also adds many electronic components that traditionally are not part of a process pump package. However, refinery and petrochemical plants make extensive use of electronic controls and with conservative design practices these systems should have long life expectancies. Bearing control systems themselves are typically modular and can often be fixed in minutes by card replacement.

At least two applications of magnetic bearings to canned motor pumps have been undertaken as demonstration projects over the last few years. The most directly applicable work was a research project developing a 20 hp chemical plant pump done by joint effort between the University of Virginia, Kingsbury, Incorporated, and Goulds Pump Company. That work was reported in 1989 by Allaire, et al. [1]. A similar 10 horsepower version of that pump has been placed in chemical plant service and is operating successfully. A second similar project was reported by Marscher, et al. [2]. This pump was also a chemical pump design. The pump was rated at 50 HP and was unique in that a hydraulic balancing device was used without an active thrust bearing.

An extension is discussed herein of these research project pumps to the design and manufacture of the prototype pump shown in Figure 1 and Figure 2. This pump is expected to lead to the development of a commercial design aimed at the larger horsepowers and pressure levels typical of API-610 pumps in refinery and commodity petrochemical plants.

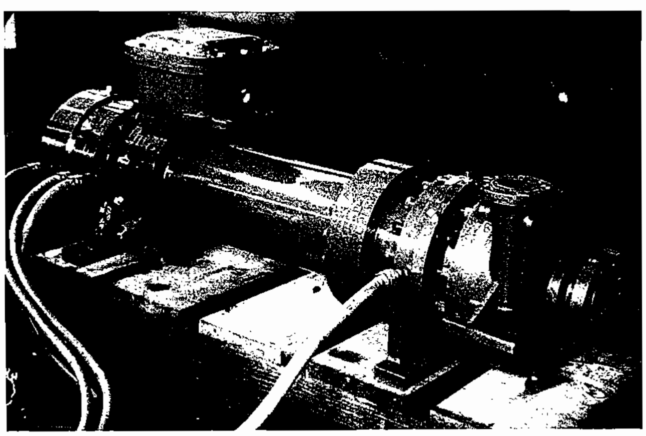

Figure 1. Magnetic Bearing Equipped Canned Motor Pump.

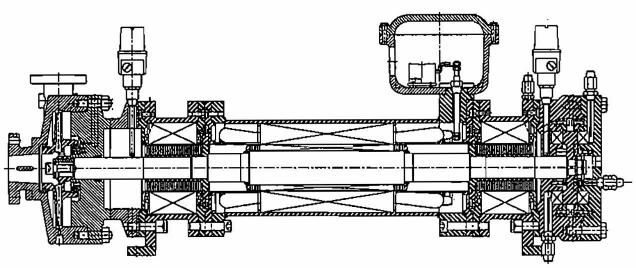

Figure 2. Magnetic Bearing Equipped Canned Motor Pump Cross Section.

PUMP DESIGN

The prototype pump was based on hydraulic components from an existing product line originally developed over 30 years ago. The particular size, a Byron Jackson 2 × 3 × 13 ¼ H SJA was selected to match a motor and bearing system that would fit a significant fraction of the installed pumps. The operating conditions of the pump are given in Table 1. Another reason for selecting this specific size is that its larger diameter impeller and single volute construction imposes higher loads than other hydraulic selections of the same horsepower and, therefore, provides an excellent test of the bearings. Predicted loads used in the design of the bearings are also given in Table 1.

Table 1. Pump Operating Conditions/Predicted Bearing Loads.

Pump Operating Conditions	
Design Flow Rate	375 gpm
Design Head	600 ft
Motor Power	125 hp (nominal)
Predicted Bearing Loads	
Impeller Reactions	280 lb static + 280 lbs dynamic
Motor Magnetic Pull	480 lb
Rotor Weight	290 lb
Inboard Bearing Max	1160 lb
Outboard Bearing Max	550 lb

MOTOR DESIGN

The motor was adapted from an oil filled design used to power vertical submersible pumps. Such motors tend to have smaller diameter rotors and, consequently, reduced friction losses when run with a wet rotor. The original submersible motor is designed to be oil filled and cans have to be added to the rotor and stator, since the intention is to circulate liquids through the motor that are not compatible with the motor laminations or windings. The cans are made from Inconel 600 (ASTM B168 alloy NO600). This material has a high electrical resistance and consequently lower eddy current losses. It is also compatible with almost all common applications.

The second major motor concern is removal of heat from the stator windings. The stator uses form wound wiring because it offers superior heat transfer and insulation integrity. The temperature at the end windings was calculated using a pure conduction model and estimated to reach 100.4°F above the can surface. Six resistance temperature detectors (RTDs) are installed in the stator to measure end winding and slot winding temperatures.

The outer shell of the motor is designed to withstand the full pressure rating and be capable of the full 1000 psig hydrotest pressure of the pump. Electrical feedthroughs for the power and RTDs are similarly rated.

BEARING DESIGN

The magnetic bearing system consists of four major components: rotors, stators, sensors, and control electronics. These elements are adapted into the pump design as individually replaceable components allowing field replacement of each subassembly, should a failure occur. Back up bearings are installed between the magnetic bearing and motor rotors as simple close clearance bushings. They are made from AISI type 410 stainless steel, heat treated to a differential hardness similar to that required for wear rings in API-610. The radial bearing rotors consist of electrical steel laminations shrunk onto stainless steel carriers. End plates and stainless steel cans (ASTM A240 Grade 316L) are welded around the laminations to form a sealed assembly. The rotors were then shrunk onto the motor rotor shaft assembly to prevent looseness in the bearing. The radial bearing stator laminations and coils are installed in two identical bearing housings, and sealed using bolted end plates and welded stainless steel cans on the inside diameter. This construction is expensive, but allows for easier rework and modification of the parts. This was considered important in a prototype test article. The bearing housings are designed to the full pump rating the same as the motor. These bearing housings are bolted to the motor assembly with register fits, since the rotor must be concentric to the stator to provide balanced bearing currents and avoid motor side pull.

The thrust bearing construction is somewhat different. A thrust disk of fully annealed AISI Type 410 stainless steel is attached to the shaft with a nut. A full anneal job is necessary to provide maximum flux densities. The stator consists of two large coils on either side of the thrust disk that form large electromagnets that act on the thrust disk. No laminations are used, since the magnetic field is essentially static and no significant eddy currents will develop.

Five sensors, one for each bearing degree of freedom, are provided to detect the position of the rotor. In this prototype pump, inductive proximity probes similar to those used in the 20 hp chemical pump mentioned before are used. These function satisfactorily for this experimental pump, but will be replaced by ring sensors built into the bearing housings and canned with the bearing stators for commercial applications. The signal from these sensors is conditioned and compensated to provide input of position and velocity that is fed into an analog PID controller. Adjustable controller gains allow the effective stiffness and damping of the bearing to be adjusted to provide appropriate characteristics for different types of rotors, or different pumpages, should they affect the rotor significantly. The signal conditioner, controller, amplifiers, and power supplies are all built into a large, 48 in × 36 in × 18 in, electrical box that must be connected to the pump with wiring. The distance from the pump to the controller is not critical except that the sensor drivers should be located near the pump, similar to the arrangement required for proximity type vibration probes.

ROTORDYNAMICS

The rotordynamics of modern API 610 overhung pumps have not been of major concern. The specification's restriction on shaft deflection at the seal and the rigidity of the typical ball bearing installation has forced both rigid body and bending mode critical speeds well above operating speed. By comparison, canned motor pumps run a much greater risk of resonance problems, but have generally been small pumps on fairly stiff bearings, so the number of problems caused by rotordynamics has probably been small. Adding the relatively soft magnetic bearings to a canned motor pump increases the risk of critical speeds occurring at running speed. Another peculiar problem of canned motor pumps is the effective increase in the modal mass for the first critical speed, probably due to the liquid in the motor gap. This phenomenon was reported in a University of Virginia thesis report on the 20 hp pump by Blair, et al. [3]. The same phenomenon was observed in this pump, where the first resonance dropped from 32.4 to 23.6 Hz when the pump was filled with water. This may not be a problem. No pump excited resonances were found during running tests, but a better theoretical understanding is required that explains the effect on both a nonrunning pump along with the further changes in these resonances when the pump is running. One significant advantage of magnetic bearings is their ability to excite the rotor with nonsynchronous vibrations for analysis purposes and then be tuned to eliminate resonances when necessary. As a result, resonant systems can be identified, analyzed, and corrected while pumps are in the OEM's test lab or during initial plant startup.

TESTING

The manufacturing and assembly of the development prototype was completed in May 1991, and the unit was installed in a closed loop test stand shown schematically in Figure 3. The test setup included a 3500 gal stainless steel test tank capable of being pressurized to 60 psig and 3.0 in stainless steel pipes connecting to both the inlet and outlet pump flanges. The pump also had local piping, so that varying amounts of cooling flow could be introduced into the motor gap and either internally recirculated or bypassed back to the pump suction. Instrumentation was provided to measure the following parameters necessary to determine pump performance:

- Pump flow
- Pump suction pressure
- Pump discharge pressure
- Input electrical power
- Pump suction temperature
- Pump discharge temperature

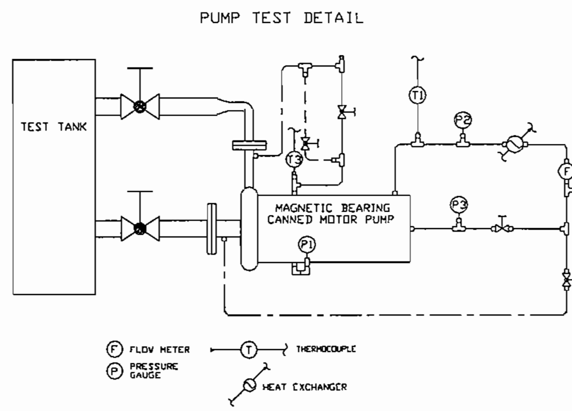

Figure 3. Test Stand Schematic.

Additional instrumentation was available to monitor performance of the bearings and motors:

- Cooling flow
- Motor adapter cavity pressure
- Cooling flow outlet pressure
- Cooling flow recirculation pressure
- Motor adapter cavity temperature
- Cooling flow outlet temperature
- Six embedded RTDs to measure motor end winding and slot winding temperatures
- Embedded thermistors in each bearing to measure coil temperatures

The control system provides a plug connector with signals proportional to the shaft position at each sensor and the currents through each leg of the bearings. These were connected to a remote analysis facility, approximately 300 feet away, where tape recordings are made and offline analysis performed.

A nominal 75 hp variable frequency drive (VFD) is also available that allows slow startup and running at speeds up to approximately 3200 rpm with a full 13 in diameter impeller, or to full speed with a reduced 10 in diameter impeller.

PERFORMANCE RESULTS

The pump performance curve is shown as Figure 4. Using the results of a performance test conducted with the same hydraulic components configured as a ball bearing mechanical seal pump, it was possible to compare the relative performance of the magnetic bearing pump to a conventional sealed unit. In Figure 4, the ratio of hp absorbed for a conventional pump to that for the magnetic bearing is shown as a function of flow rate. It can be seen that near the design point of the pump, about 10 percent additional power consumption results with the magnetic bearing pump.

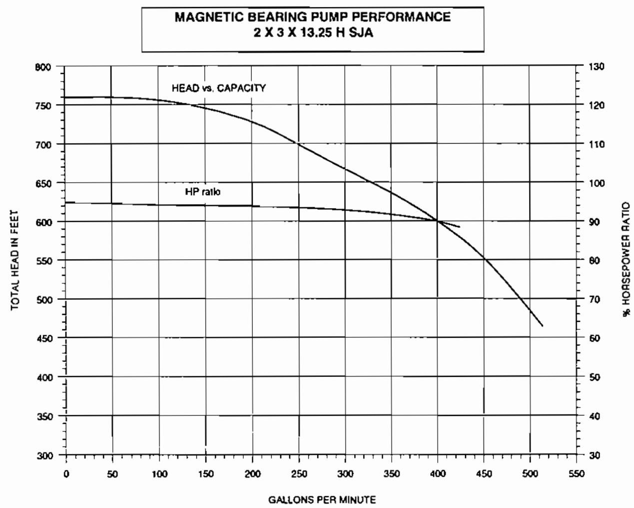

Figure 4. Pump Performance Curve.

VIBRATION AND ROTORDYNAMICS

The basic vibration results of the running pump showed the rotor to be vibrating synchronously, approximately three to four mils peak-to-peak in response to unbalance and runout. The relatively high vibration is due to the soft nature of the bearings. The level of vibration can be reduced by adjusting the bearing bias currents, but this will increase the bearing losses and winding temperatures. For comparison, the minimum diameter internal clearance of 20 mils occurs at the backup bearings and is more than five times larger than the peak vibration. The external vibration measured on the bearing housings was 0.5 to 0.7 mils vertically, and 0.2 to 0.4 mils horizontally.

Rotordynamics testing was done using the capability of the bearing control loop to be perturbed by an external signal. The signal appears to the controller as a displacement of the shaft under the sensor. To compensate the controller causes the magnetic bearing to push on the shaft. As an analysis tool, different signals can be introduced and the response of the rotor measured. To detect resonances, the authors experimented with both a white noise signal and a sinusoidal output from a sweep signal generator. The output from the position sensors was then analyzed and averaged. The results from both methods were about the same as far as frequencies were concerned. The plots shown in Figure 5 were made from data collected using the sweep signal generator method.

The plots give data for four different sweeps. In the bottom plot, the rotor was supported in air with the controller set to give an effective stiffness of approximately 70,000 lb/in in the motor end radial bearings. The system is heavily damped but the first, second, and third resonances are believed to occur at 32.4, 47.5, and 117.4 Hz. The second plot shows the effect of merely adding water to the pump. As is shown, all three resonances drop due to this added mass effect mentioned above. The third plot shows the

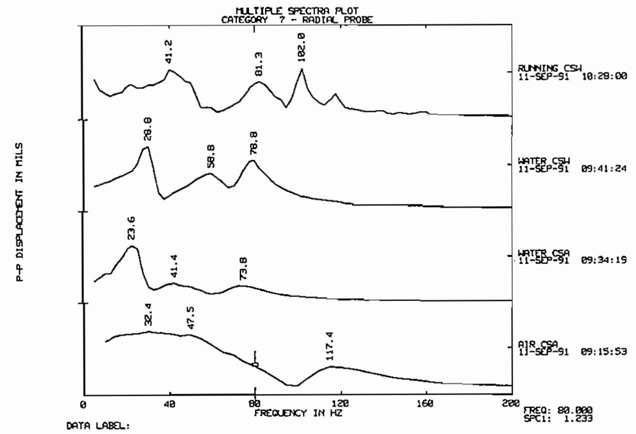

Figure 5. Resonance Plots Using Sweep Generator.

effect of changing the effective stiffness of the bearing to 143,000 lb/in by increasing the gain in the controller. The fourth plot shows the results of running the pump. A notch filter was used in this case to eliminate the effect of the synchronous vibration. Interpretation of this data is that running the pump effectively stiffens the rotor to increase the first two rigid body modes by 43 percent and 38 percent, and the first bending mode by 29 percent. The little peak just under 120 Hz is from the 2× vibration that is not affected by the notch filter. A conventional analysis plot of the pump vibration is shown in Figure 6 without external perturbation. Notice that the bending mode vibration at just over 100 Hz shows up. This plot was taken with the pump running at shutoff. A similar plot (Figure 7) with the pump running at high flow (453 gpm) does not show this, but does show a higher 2× signal. From a practical application point of view, this data may appear academic, but it does show the capability of the magnetic bearing system to act as a diagnostic device using modest tools such as a sweep signal generator and a spectrum analyzer. A cascade plot is shown in Figure 8 with the pump run using the VFD. This shows that the system does not respond significantly even as we go through the resonance found by the shake tests.

MOTOR AND BEARING COOLING

The motor produces significant heat in the electrical windings that must be conducted into the stator cans, where it can be

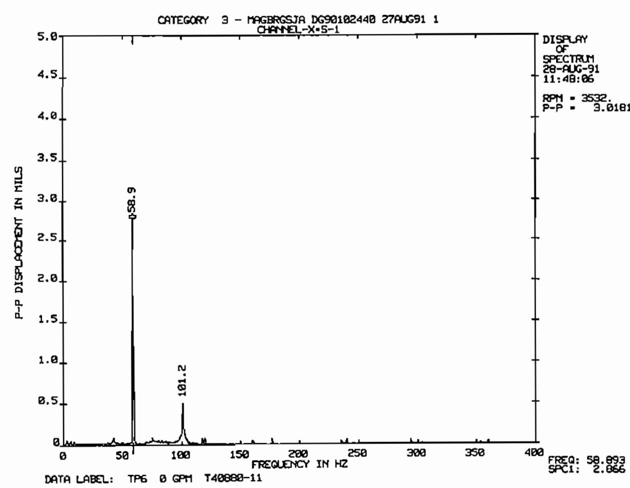

Figure 6. Pump Vibration at Shutoff Flow.

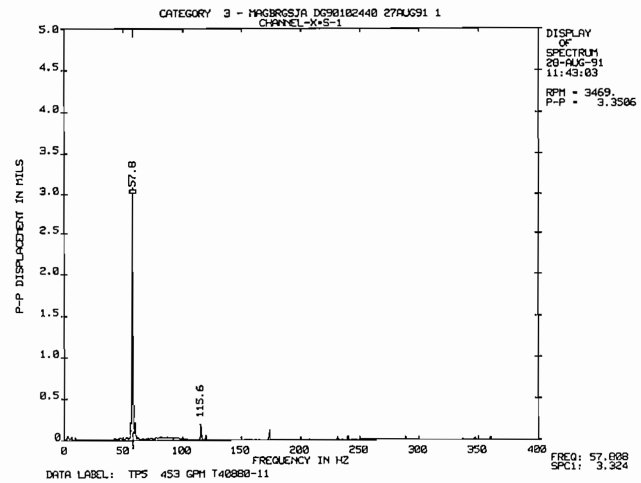

Figure 7. Pump Vibration at High Flow.

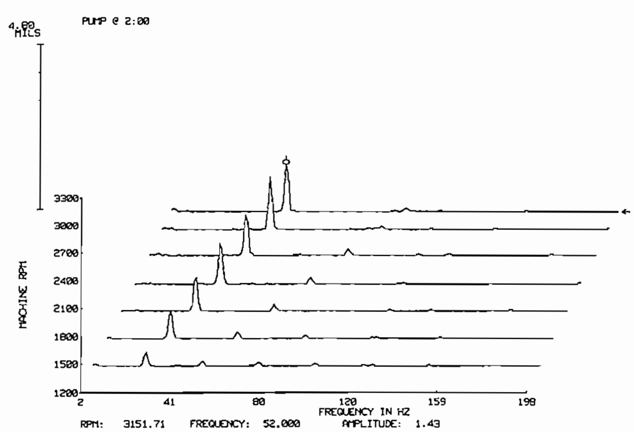

Figure 8. Cascade Plot with Pump Run Using VFD.

convected into the fluid flowing down the motor gap. A test was run where the motor was run at a constant load with the expected results of a final stabilization temperature near 200°F to 250°F, based on the original analysis. The results of this run are shown in Figure 9. However, the temperature increased beyond the design value and was approaching the insulation limit. The cause of this was isolated to poor thermal contact between the motor laminations and the can. The prior experience for canning motors was based on a random wound motor in which the laminated poles covered a much larger arc along the stator inside diameter. In addition, the can was permanently deformed into the gaps of the earlier design, due to the high hydrostatic test pressures used. The prototype pump was built using a formed wound stator, which normally would produce cooler winding temperatures, except that the slots between the lamination poles are larger, resulting in less contact between the lamination and the can. Remedial plans to correct this condition have been developed including vacuum potting the motor and pressure forming the can against the lamination. Commercial versions of the motor will use a modified lamination design to provide good thermal contact.

A similar situation existed in the bearing, except the thermal resistance is believed to result from gaps between the coils and the laminations. Vacuum potting of the bearings is expected to fix this problem.

The plot of temperature rise includes data for two flows. The results confirm the relative insensitivity of the cooling to the flow

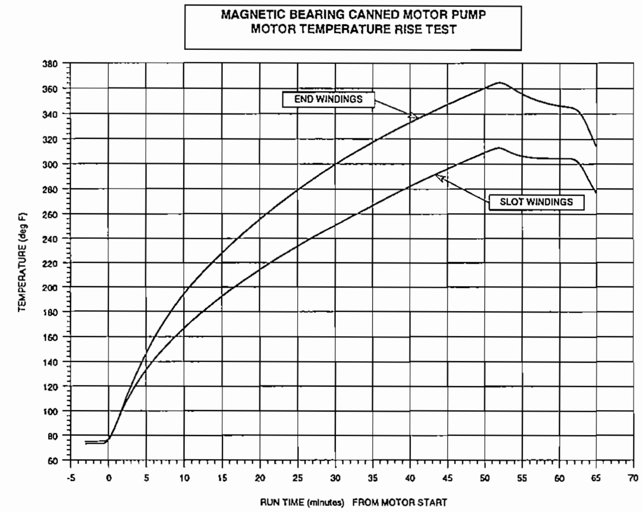

Figure 9. Motor Temperature Rise.

in the gap. As anticipated during the design phase of this project, the heat transfer from the motor can to the liquid is very effective. However, application of these pumps must consider this cooling effect carefully. Many petroleum products have low specific heats and steep vapor pressure curves, so care must be exercised to avoid flashing, if the product is introduced back into the suction of the pump. Many schemes exist to avoid these problems including external heat exchangers, returning to a higher pressure area in the pump, or returning the cooling stream to the suction source. These are expected to be used in API 610 pump applications.

For field testing, it is expected that an application will be found where the pump case and suction and discharge piping can remain as is, and the motor and bearing system can be installed in place of the existing pump cover, bearing housing, coupling, and motor.

CONCLUSION

The prototype for a new class of centrifugal pumps has been built and successfully tested. Some minor modifications are required to allow operation of the motor and bearings at full load. When this is done and followed by some field application experience, a basis will exist for introducing a complete line of wear free, sealless pumps. Also, the introduction of magnetic bearings provides useful diagnostic capability in that pump loads and rotordynamics performance can be directly measured. These should prove very useful in the evaluation and maintenance of magnetic bearing equipped pumps. Although the canned motor magnetic bearing is expected to be higher in first cost than conventional sealed pumps, the authors believe that the value added in terms of reduced maintenance costs and improved reliability should justify the additional original equipment cost.

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