DESIGN, CONSTRUCTION AND TEST OF MAGNETIC BEARINGS IN AN INDUSTRIAL CANNED MOTOR PUMP

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ABSTRACT

The replacement of conventional bearings with magnetic bearings in a canned motor pump is discussed. The primary objective was the development of long life bearings for canned motor pumps. The research project was a joint effort between the University of Virginia, Kingsbury, Incorporated, and Goulds Pump Company. The research team designed, constructed and tested the bearings in a 20 hp, 300 gpm, single stage centrifugal canned motor pump running at 3600 rpm. Both magnetic radial and thrust bearings were incorporated in retrofitted bearing housings. Displacement probes sensed the shaft position near the bearings. Control electronics then employed the displacement signal as the feedback signal to control the level of current in the magnets. Proportional-integral-derivative type analog controls were used.

The bearings were successfully installed and tested. Performance and vibration tests were conducted on the canned motor pump with the original bearings and with the magnetic bearings. No significant change was found in the pump performance but some small changes were observed in the pump vibration characteristics. It is believed by the authors that this is the first installation of magnetic bearings in a commercial pump.

INTRODUCTION

Canned motor pumps have not been employed in chemical and petrochemical plants in large numbers in the past. However, it seems likely that they will be increasingly utilized in the future because 1) they do not have mechanical seals, and 2) they have zero leakage during normal operation.

It appears that mechanical seals in conventional industrial pumps have nearly reached their limits of development for increased reliability and lower leakage. One major chemical company has conducted a survey of their conventional pumps, which indicated an average life of approximately 18 months to two years for mechanical seals. The same chemical company has conducted a strong inhouse program to improve mechanical seal life with only marginal improvements.

Mechanical seals always have some level of leakage. Canned motor pumps are the only pumps which have no leakage, because they have no seals. Environmental laws are becoming increasingly restrictive with regard to leakage, particularly in California. Thus, it is expected that the use of canned motor pumps and magnetic drive pumps will increase dramatically over the next decade.

Unfortunately, canned motor pumps require process lubricated bearings. Often these bearings have maintenance problems. Chemicals, petroleum products and other process fluids have low viscosity. This gives inadequate bearing lubrication to support high thrust or other loads on the bearings. Thus, the bearings wear or fail in a relatively short period of time. The average canned motor pump bearing life in one company is less than one year. This sort of performance often requires a company to have standby pumps and leads to high maintenance costs.

Magnetic bearings are a potential solution to these problems in many applications. There is no contact between the shaft and bearing in magnetic bearings. Theoretically, no wear should occur so the life should be infinite from this point of view. Obviously, there are other factors such as electronic component failures and wire insulation breakdown, etc. However, it seems reasonable to expect a life of five to ten years for magnetic bearings in canned motor pumps once they become a production item. Other control systems such as process controls in plants and similar devices such as electric motors have long life expectancies, so similar expectations are reasonable for magnetic bearings.

The concept of magnetic suspension has been around for centuries but only recently has implementation proven feasible. Beams carried out the first work on magnetic bearings for centrifuges and other rotating devices at the University of Virginia in the 1950s and 1960s [1, 2] with vacuum tube technology. It allowed Beams to rotate small bodies up to five million revolutions per second.

Magnetic bearings have been in production for about 20 years. The primary original manufacturer is located in France. Hundreds of magnetic bearings have been installed in a variety of applications [3, 4]. Some applications mentioned in the references include grinding spindles, turbogenerators and turboalternators. A related company in the United States has concentrated on primarily on compressor applications. These applications are well described in the literature [5, 6, 7]. Another related company in Japan has concentrated on precision grinding and molecular turbopumps operating in magnetic bearings. They report over 2,000 molecular turbopumps in operation.

Strong research efforts on magnetic bearings are currently underway in Europe [8, 9] and Japan [10, 11, 12] as evidenced by the papers at the recent First International Symposium on Magnetic Bearings held in Zurich, June 1988. Another Conference on Magnetic Suspension Technology was held by NASA at Langley Field, February 1988. Many papers on American work on magnetic bearings were presented there as well as in Zurich.

Recent work at the University of Virginia on magnetic bearings describes the development of the magnetic bearings for the canned motor pump in much more detail than can be discussed in this paper. Humphris, et al. [13], gave the first description of the control algorithms. Allaire, et al. [14, 15], presented the application of magnetic bearings to flexible laboratory rotors. The application of a magnetic damper to multimass flexible rotors was reported in 1987 and 1988 [16, 17]. Work on thrust bearings related to the canned motor pump was reported in a NASA Conference [18].

Additional related research on nonlinear aspects of magnetic bearing performance, high speed applications and shock loading conditions has been presented [19, 20, 21, 22]. Digital controls have also been implemented in operating flexible rotors [23, 24].

The design, construction and test of a set of magnetic bearings for an industrial canned motor pump are described herein. The objective was to demonstrate the use of magnetic bearings to replace the conventional carbon bearings. It is expected that the magnetic bearings will have long life compared to the original bearings, but this must be demonstrated by field testing.

AN INDUSTRIAL CANNED MOTOR PUMP

The pump chosen for the project was a small canned motor pump employed in a wide range of service. The design flowrate was 300 gpm at 140 ft of head and the rated horsepower was 20. It is expected that magnetic bearings in industrial applications will more likely be employed in somewhat larger pumps than this. However, this size was easily accommodated in the Rotating Machinery and Controls Laboratory at the University of Virginia for prototype testing. Other pump operating parameters are given in Table 1. The pump design is illustrated in Figure 1 and a photograph of the pump is presented in Figure 2.

Table 1. Description of Canned Motor Pump For Magnetic Bearing Research.

Design Flow Rate	=	300 gpm
Design Head	=	140 ft
Pump Impeller Type	=	Single Stage Centrifugal
Motor Type	=	Squirrel Cage Induction (Canned)
Motor Power	=	20 Hp
Overall Length	=	34.4 in
Diameter of Motor	=	8.5 in
Shaft Weight	=	40.3 lb
Rotational Speed	=	3600 rpm

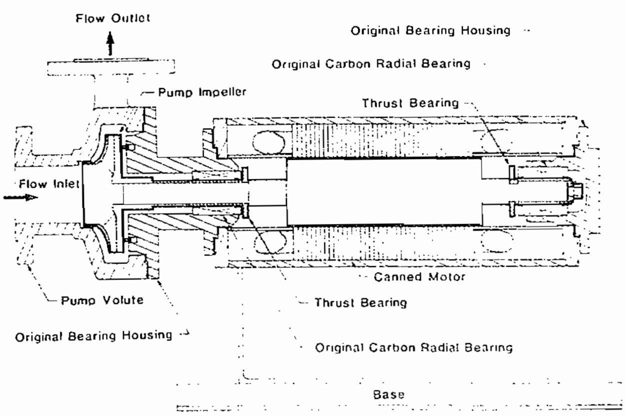

Figure 1. Canned Pump.

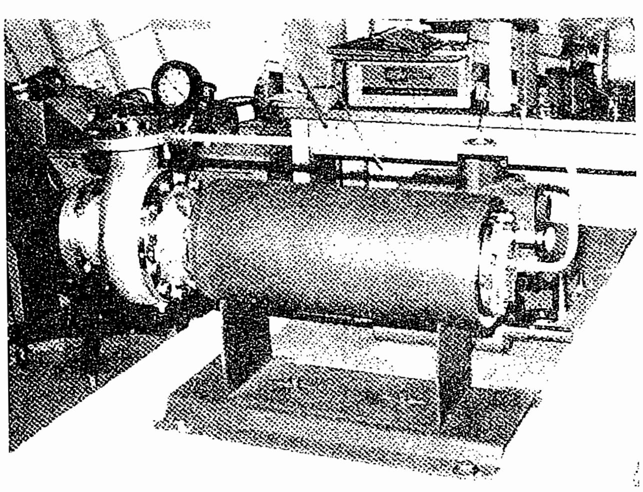

Figure 2. Canned Motor Pump Before Conversion.

The pump consists of a single stage overhung centifugal impeller with a single volute and single discharge. A disassembled view of the pump components is shown in Figure 3. The impeller was a five bladed closed impeller of 6.5 in outer diameter. A 3.0 in diameter suction pipe and 2.5 in diameter discharge pipe were employed in the pump test loop described later.

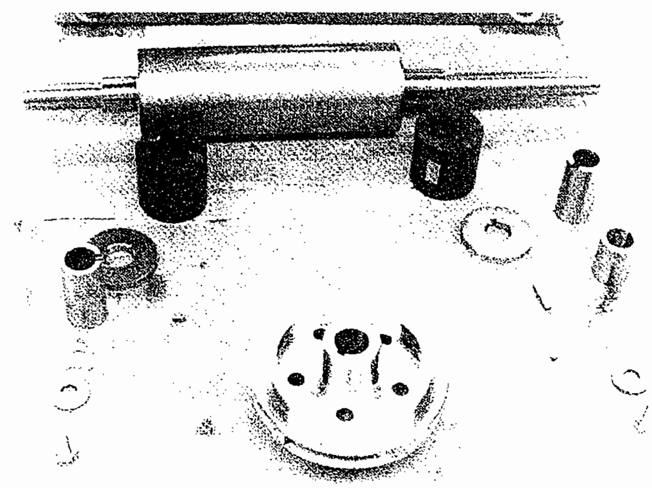

Figure 3. Disassembled Original Pump Rotating Elements.

A canned motor formed the back end of the pump. It is a two pole squirrel cage induction motor of standard type. Both the stator and rotor winding/lamination components were canned to prevent contact between the working fluid and the winding/laminations.

As seen in Figure 1, the shaft weight was supported on two radial bearings. For this pump, the bearings were grooved carbon sleeve bearings. Thrust forces produced by the working fluid around the impeller and the rest of the rotor were taken by the split double acting thrust bearing faces on the carbon bearings. One thrust face was at one end of the canned motor rotor and the other thrust face was at the other end. The bearings were lubricated by the working fluid in the pump.

Cooling required for the canned motor also assisted in cooling the bearings. The cooling flow came through a return line shown in Figure 2. The original cooling flow rate was measured as 5.0 gpm for the design flow of the pump.

MAGNETIC BEARING DESIGN

Magnetic bearings were designed for the canned motor pump just described. The changes made in the pump to accommodate the magnetic bearings are illustrated in Figure 4. This was a retrofit situation in that as little of the pump was to be changed as possible. Basically, the only modifications made to the pump to incorporate the bearings were two new bearing housings, the new magnetic radial and thrust bearings (with associated displacement probes and wiring), magnetic laminations placed along the shaft, and a short extension added to the shaft. The pump casing, impeller, canned motor, seals, etc. were unmodified. The required changes and additions are summarized in Table 2. A photograph of the modified pump with magnetic bearings installed is presented in Figure 5.

Magnetic bearings require feedback controls, a minimum of one sensor for each of the five axes (two horizontal, two vertical and one axial). The new bearings housings were, thus, modified to include four radial probes and one axial probe. Actually, the

Table 2. Modifications to Canned Pump for Magnetic Bearing Retrofit.

New Bearing Housings
Magnetic Radial Bearings
Magnetic Thrust Bearings
Magnetic Shaft Laminations
New Thrust Collars
Short Shaft Extension
Displacement Probes
Bearing Wiring
Control Electronics

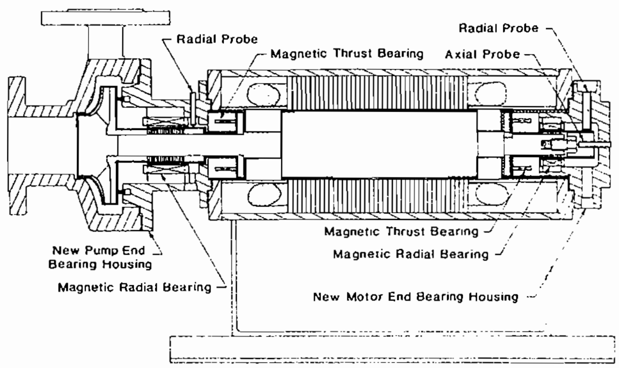

Figure 4. Canned Motor Pump With Magnetic Bearings.

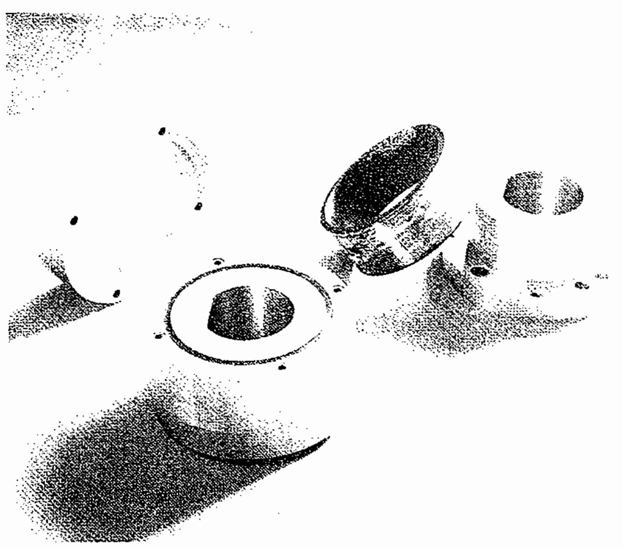

Figure 5. Canned Motor Pump with Magnetic Bearings Installed.

radial probes were placed at 45 degrees from the horizontal as done in many industrial machines, so "probe horizontal" and "probe vertical" were the displacement directions used. Standard industrial high quality inductance probes were employed. Access holes were placed in the bearing housings at both ends for the wires to the electromagnets.

The associated electronic controls were designed for this application. Basically, proportional, integral and derivative controls were employed to center the shaft in each bearing as well as provide the desired stiffness and damping properties. Also, a power amplifier suitable for the controls was employed. These electronic components were mounted in a rack (Figure 6), along with associated meters used for monitoring the vibration levels of the pump and the currents or voltages in the bearings. Tape recordings were made of the operating pump parameters for later analysis.

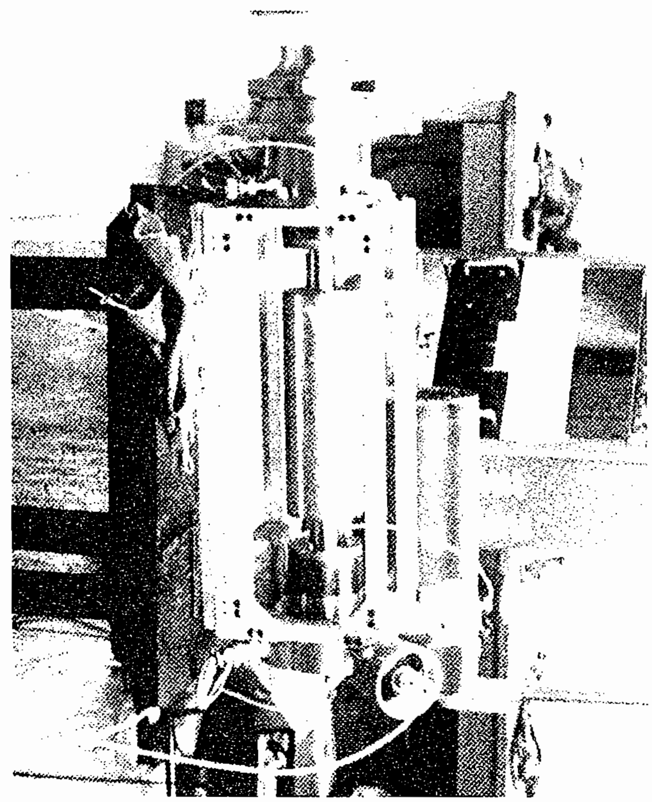

Figure 6. Rotating Element Supported in Magnetic Bearings.

Backup bearings were also incorporated into the bearing housings. These operated to support the shaft when not in operation or take the thrust load in case of an electrical failure. It should be noted that the backup bearings only operate in case of failure of the magnetic bearings; they do not act as startup bearings. The magnetic bearings were energized and the shaft centered before the shaft was rotated. The backup bearings for the radial direction consisted of the inside surface of the thrust bearing stators. The backup thrust bearings were the faces of the thrust bearing stators. A commercially available nickel based wear coating was placed on one surface of each backup bearing area. During the course of initial pump testing, numerous runs (usually inadvertently) were made on the backup bearings with no major problems encountered.

RADIAL BEARING DESIGN

The radial bearings consisted of four electromagnets equally spaced circumferentially about a laminated disk which is attached to the shaft. The geometry of the bearing is illustrated in Figure 7 and more details are given in Figure 8. The geometric and electrical properties of the radial bearings are indicated in Table 3. Each magnet was constructed of a solid piece of magnet iron machined to the proper dimensions. The magnets were attached to the bearing housing with bolts. The coils of wire were wound around thin walled plastic bobbins to avoid contact between the insulation and the metal. The coils were roughly rec-

Table 3. Properties for Radial Magnetic Bearings in Canned Motor Pump.

GEOMETRIC PROPERTIES:		PUMP END	MOTOR END
Outer Diameter	=	4.1 in	3.25 in
Inner Diameter	=	1.02 in	0.815 in
Length	=	2.0 in	1.0 in
Clearance	=	0.015 in	0.015 in
ELECTRICAL PROPERTIES			
Current (Upper Magnets)	=	0.950 amp	0.700 amp
Current (Lower Magnets)	=	0.550 amp	0.600 amp
Number of Turns	=		
Voltage	=		
LOAD CAPACITY, STIFFNESS AND DAMPING			
Load Capacity	=	125 lb	20 lb
Stiffness	=	20,000 lb/in	20,000 lb/in
Damping	=	*** lb-sec/in	** lb/sec/in

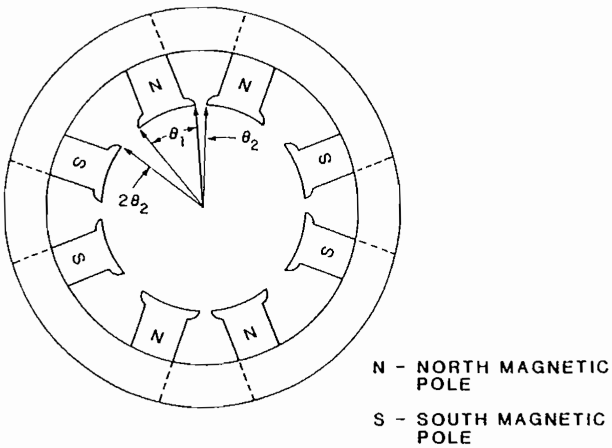

Figure 7. Radial Magnetic Bearing.

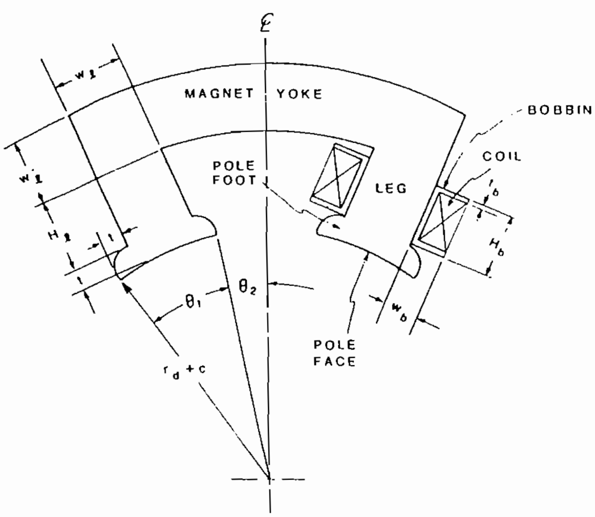

Figure 8. Detailed Drawing of Magnet and Coil.

tangular in cross section. Each coil was coated with an epoxy material to prevent contact with the water. The predicted stiffness value of the bearings was 24,700 and the measured value for the pump end was 22,000 lb/in.

The pump end bearing is shown in Figure 9 partway entered into it's housing, which was later bolted to the pump casing. The eight coils are easily seen. The wiring for the magnets is not shown in this photograph. The motor end and pump end bearings with the coils in place and wires attached is illustrated in Figure 10.

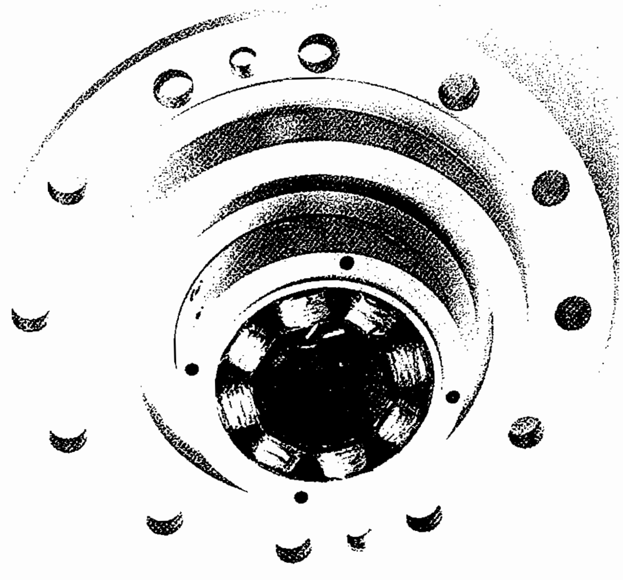

Figure 9. Pump End Magnetic Bearing.

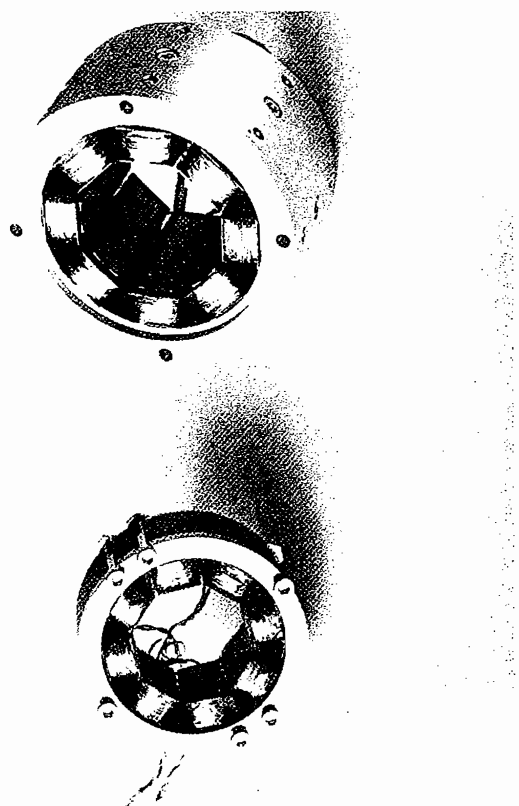

Figure 10. Motor End and Pump End Magnetic Bearing.

Magnetic bearings are open loop unstable [13]. A closed loop feedback control system is required to stabilize them. Several previous publications [13, 17, 18] have described the type of analog controls employed by the bearings. Generally, they consisted of a sensor amplifier, compensator, summer, lead network, and current amplifier.

The design of the radial magnetic bearings was carried out with a design optimization procedure described by Imlach, et al. [18]. The solution parameters, which were chosen to fit either available geometry or available material considerations, were: the axial length of the magnets, the wire size/type, bobbin thickness, lamination radius on shaft, clearance and wire packing factor. An optimization algorithm selected the width and height of magnetic legs, width and height of bobbins, number of turns of wire, steady state currents and required controller gain to produce a specified stiffness.

Laminated disks of magnet iron were placed on the shaft to complete the magnetic circuit with the horseshoe stator magnets. These consisted of 7.0 mil thick disks, insulated on both sides, and cut out using laser machining. They were assembled on to the shaft and held in place by compression, due to the nut on the end of the shaft. Laminations are required on the rotating shaft, because changing flux lines cross the moving disk producing eddy currents. For an eight pole magnet such as this, sixteen flux reversals take place for each shaft rotation, thus generating eddy current losses in the magnet iron. However, with the laminations, the eddy current flux is minimized and posed no problems.

THRUST BEARING DESIGN

The magnetic thrust bearing design for a single acting thrust bearing is illustrated in an exploded view in Figure 11. The design consists of a thrust runner on the rotating shaft and a thrust stator, composed of a base and inner toriod, a coil and an outer toroid. All of the magnetic components are made of magnet iron. A perspective view of the thrust bearing is shown in Figure 12. Properties of the thrust bearing are given in Table 4.

The coil of wire produced the magnetic flux in the bearing. It was wound on a plastic bobbin similar to the radial bearing bobbin. The flux paths followed in the thrust bearing are illustrated in Figure 13.

A solid disk of magnet iron forms the rotating part (runner) of the thrust bearing. Unlike the rotating part of the radial magnetic bearings, the thrust runner does encounter changing magnetic flux lines, except dynamically, so it does not have to be laminated to reduce eddy currents. A photograph of the magnetic thrust bearings is shown in Figure 14. One is disassembled and one is assembled. The thrust runner is not shown in this diagram.

Table 4. Properties of Thrust Magnets in Canned Motor Pump.

GEOMETRIC PROPERTIES	
Outer Diameter of Stator	= 3.625 in
Outer Diameter of Coil Gap	= 2.956 in
Inner Diameter of Coil Gap	= 2.648 in
Inner Diameter of Stator	= 1.620 in
Axial Length	= 1.59 in
Clearance	= 0.020 in
Thrust Collar Length	= 0.42 in
ELECTRICAL PROPERTIES	
Number of Turns	= 576
Design Current	= 1.5 amp
Voltage	= 64 volt
Power Dissipated	= 58 watts
STATIC AND DYNAMIC LOADS	
Maximum Thrust Load	= 295 lb
Stiffness	= 46,000 lb/in
Damping	= 240 lb-sec/in

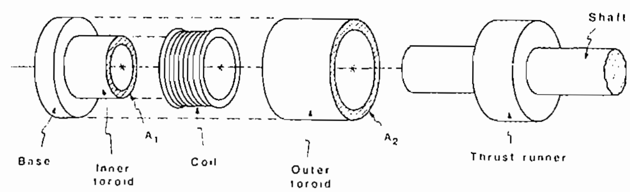

Figure 11. Exploded View of Single Acting Magnetic Thrust Bearing.

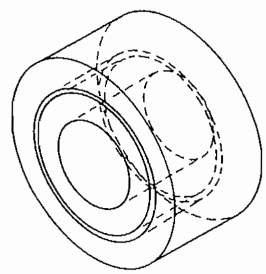

Figure 12. Perspective View of Stator for Magnetic Thrust Bearing.

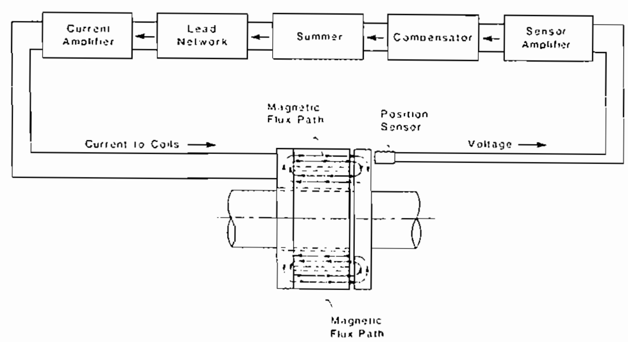

Figure 13. Magnetic Flux Paths and Control Loop in Single Acting Magnetic Thrust Bearing.

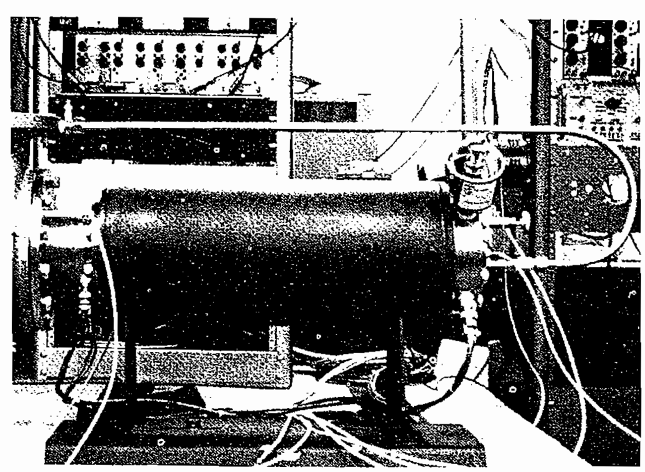

Figure 14. Pump Magnetic Thrust Bearing.

CRITICAL SPEEDS

As with any rotor, it is important that a critical speed of the shaft on magnetic bearings not be at the pump running speed of 3600 rpm. The pump shaft with all components, illustrated in Figure 15, was modelled using standard rotor dynamics methods. A critical speed map was constructed as shown in Figure 16. This indicated that the bearings must have a stiffness over 10,000 lb/in to avoid operation near or above the first critical speed. Thus the bearings were designed to operate at 20,000 lb/in or higher. With this stiffness, the predicted first critical speed is above 6,000 rpm. The calculated first three mode shapes are shown in Figure 17 for the rotor, assuming a bearing stiffness of 10,000 rpm. The mode shapes will not be very different from those with bearing stiffness of 20,000 lb/in. It may be noted that none of these modes, and also none of the higher modes (not shown), had a node point between the bearing centerline and the displacement probe. This indicated that the noncollocation problems encountered in some compressor applications would not be found in this pump.

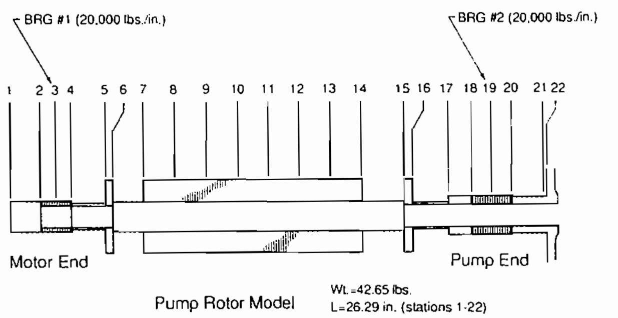

Figure 15. Goulds Pump Rotor Model for Rotordynamic Analysis.

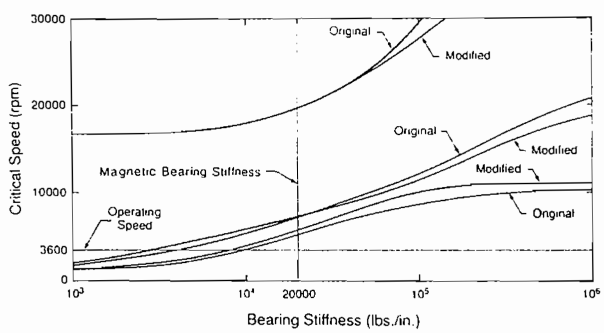

Figure 16. Undampened Critical Speed Map.

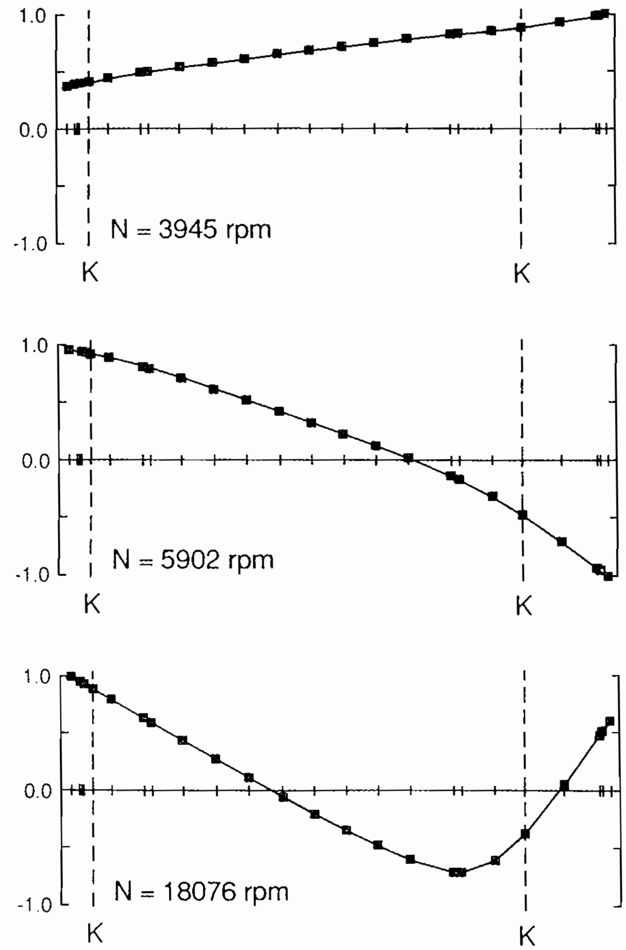

Figure 17. Mode Shapes for Canned Motor Pump.

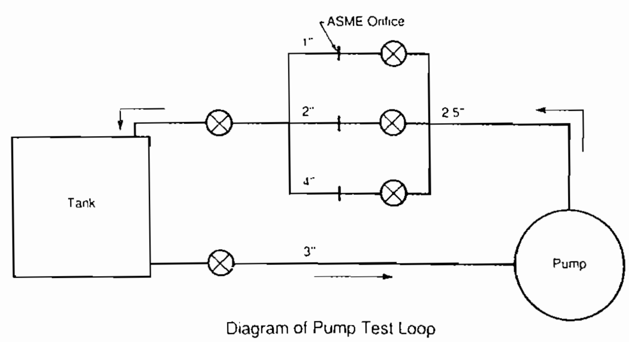

Figure 18. Diagram of Pump Performance Test Loop.

PERFORMANCE AND VIBRATION TESTING

A standard ASME specification pump test loop, sketched in Figure 18, was set up to test the pump before and after modifications for the magnetic bearings. A 550 gallon supply tank was used as the reservoir with a suction supply at the bottom and a discharge outlet from the pump at the top. The suction pipe consisted of a straight pipe of 32 diameters with flow straighteners. Three standard orifices were placed in the discharge side to measure low, medium and high flowrates. Suction and discharge pressure were measured along with the cooling water flowrate. Motor power was also measured by a three phase power meter for each phase.

The pump performance before and after the installation of magnetic bearings is given in Figure 19. Standard head *vs* flow rate data was taken and plotted. Note that all of the data points fall nearly on the average curve indicating that the magnetic bearing modification as expected did not change the pump performance. An efficiency plot is also given in Figure 15. The curves are the same before and after the bearing modification.

It should be noted that the impeller wear ring on the suction side was different between the two cases. Following modification of the pump shaft to include the magnetic bearing components, a misalignment was introduced into the rotor. This created a rub of the rotating component on the impeller casing.

To remove this rub, the simplest method was used. Some 16 mils of material (diametral) was ground from the wear ring seal over the original 50 mil diametral clearance. The curves in Figure 19, following the bearing modification, have been adjusted to take the extra leakage into account.

For the vibration testing, accelerometers were placed in seven locations on the pump casing. These locations are shown in Figure 20. Probes 1-6 measured horizontal and vertical vibrations, probe seven measured axial vibrations. Three major frequency components were detected: the running speed (approximately 58 hz); the blade pass speed (approximately 290 hz); and approximately 4 khz. The latter frequency was determined to be due to the rotating unbalance motor pull.

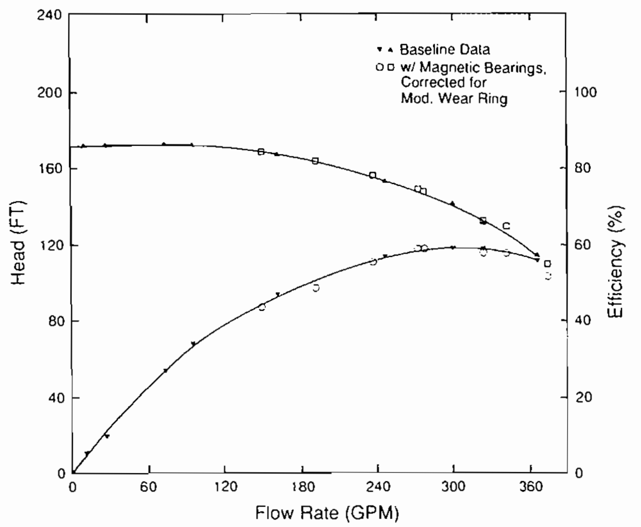

Figure 19. Pump Performance Data Before and After Installation of Magnetic Bearings.

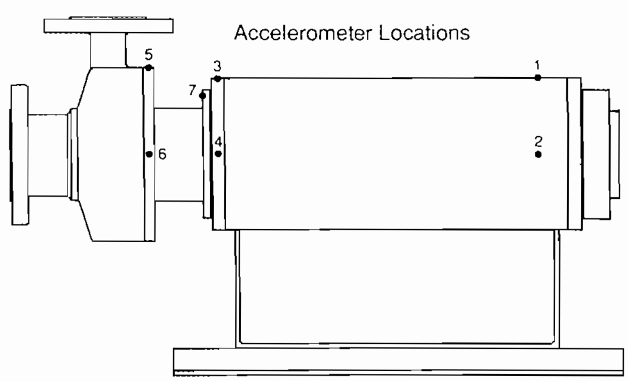

Figure 20. Accelerometer Location for Vibration Measurements.

Vibration data was taken at various flow rates before and after the installation of magnetic bearings. A global comparison of the vibration data before for two flow conditions is given in Figure 21. It can be seen that at running speed the vibration levels were either comparable or slightly higher for the magnetic bearing equipped pump for both flow cases. At the blade pass frequency, this trend is reversed with the magnetic bearing equipped pump having comparable or lower vibration levels, especially at design flow. No measurable axial vibrations at this frequency were noted. The motor unbalance frequency (4 khz) was measurable only at probe locations 1-4. At this frequency the magnetic bearings transmitted less of the vibrations to the motor casing.

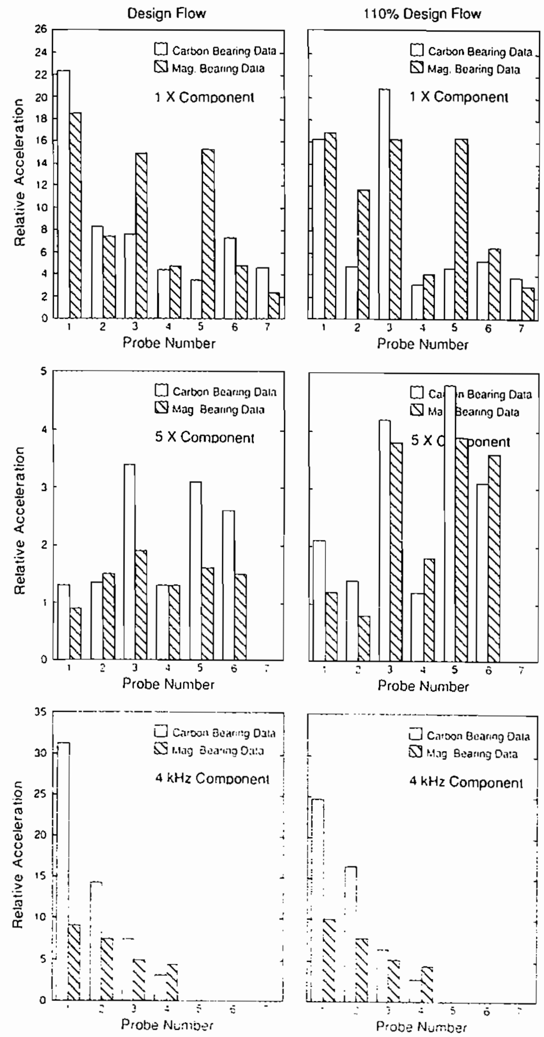

Figure 21. Relative Pump Acceleration Before and After Installation.

This trend can be explained by the frequency dependent nature of the magnetic bearing stiffness. While fluid film bearings are relatively insensitive to the frequency of the disturbance this is not true of magnetic bearings. The magnetic bearings used in this pump have a distinct peak in the stiffness *vs* frequency plot, approximately at the running speed. The stiffness then decreases with increasing frequency until the bandwidth of the system is reached. Above this frequency, the bearing stiffness is zero. At higher disturbance frequencies, therefore, the force transmitted from the rotor to the casing will be small, as is seen from the measurements.

It should be noted that no attempt was made to adjust the bearing properties to minimize the vibration levels. The vibration levels indicated here are well within the tolerances of this pump.

CONCLUSIONS

A standard industrial canned motor pump, one which has experienced bearing wear problems and failures in the field in some applications, was retrofitted with magnetic bearings. This was done in a university laboratory as part of a research project to initiate the use of magnetic bearings in pumps. This paper has covered the design, construction and testing of the bearings.

They were successfully installed in the pump and tested. No adverse effects were found to be introduced by the magnetic bearings except for some additional complexity. The prime objective of the magnetic bearing retrofit was long mean time between failures. Field applications will be required to verify the potential of the magnetic bearings in this area.

The next step in this project is the design, construction and installation of magnetic bearings in a similar pump. Improvements in the design will be made as a result of the experience gained with the first pump. The next pump will be placed in a chemical plant for testing in a particular application pumping hexane. It will be extensively instrumented and monitored for several years to determine its performance.

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