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Testing of a Magnetic Bearing Equipped Canned Motor Pump for Installation in the Field

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Abstract

Magnetic bearings have been installed in an industrial canned motor pump. The rotor of the pump is fully supported by active magnetic bearings. The bearings have been canned in a manner similar to that used to isolate the motor components from the working fluid. Thus, the magnetic fields of the bearings act through a non-magnetic, conducting can.

Several diagnostic procedures using magnetic bearings, reported in an earlier paper by the authors, have been applied to this pump. These diagnostics were performed prior to the start-up of the motor but with the machinery in an assembled configuration. This allowed the pre-determination of optimum control settings prior to delivery. It also provided a baseline indication of the system characteristics for use in future studies and for maintenance purposes.

1. Introduction

1.1 Sealless Pumps. Recently a trend towards the use of sealless pumps has developed in the chemical process pump industry [1]. The driving forces behind this trend are both economic and environmental . The economic benefits associated with the use of sealless pumps are mainly due to the elimination of shaft seals which are the dominant root cause of failure in this pump application. These failures add significantly to the life cycle cost of these pumps. The environmental factors driving the change to sealless pumps include a reduced societal tolerance of chemical emissions of all forms. One source of these emissions is chemical process pumps whose seals leak during normal operations. In the case of a shaft seal failure the leakage becomes much larger. Sealless pumps, when they are made to be reliable, will eliminate these types of leakage.

The present state of the art for sealless pumps, which include both canned motor pumps and magnetic drive pumps, use process fluid lubricated bearings, typically formed from a carbon sleeve. The reliability of these bearing is low, with at least one reported application having an average bearing life span of six months. Thus the maintenance costs associated with these bearings can be high. Additionally, when the bearings fail, the liners are often damaged. This generally results in the loss of the pump as well as potential process fluid leaks. For these reasons a more reliable sealless pump must be developed.

1.2 Magnetic Bearing Application. Magnetic bearings are very well suited to this application. With the magnetic components separated from the working fluid in the same manner as the motor of a canned motor pump, the seal boundary of the pump can be maintained. Because there is no contact between the rotating and stationary components under normal conditions there is no wear and no maintenance associated with the bearings. It is projected that properly designed magnetic bearings may have a life expectancy similar to the 19.7 to 76.3 year average life of electric motors currently operating in this environment [2]. This wide span of life expectancy is due to differences in designs and construction details.

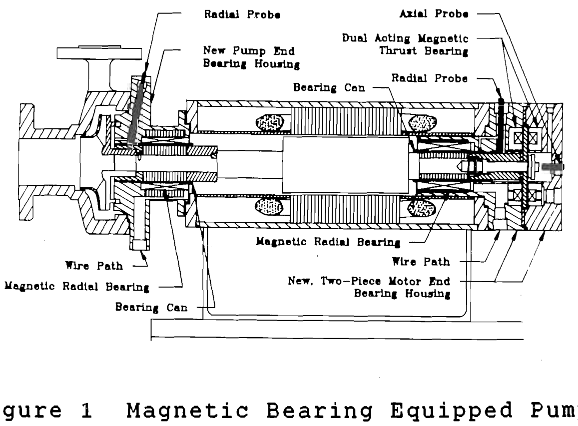

Figure 1 Magnetic Bearing Equipped Pump

1.3 Current Application. Magnetic bearings have been retro-fit into an industrial canned motor pump (figure 1). This pump will be placed in an industrial environment to prove the compatibility of the technology with the application. The installation of the pump in the field represents the conclusion of an ongoing project reported in [4] and [5]. In brief, the project involved two phases: a proof of concept phase in which magnetic bearings were retrofit into an industrial canned motor pump for laboratory use; and a demon-stration phase in which modifications were made to the design to allow a second pump to be configured with magnetic bearings for an industrial application.

The basics of the design procedure used in the development of these bearings is given in [4], while some results from the proof of concept phase were reported in [5]. In the demonstration phase a 7.5 kW (10 hp) pump was selected for the retrofit. The pump specifics are listed in table 1. Figure 1 is a cross sectional drawing of the pump equipped with magnetic bearings. The bearing stators were separated from the working fluid by a .254mm (.010 inch) thick stainless steel can of 316 Stainless Steel (Figure 2).

Motor Rating	7.5 kW	(10 hp)
BEP Flow	1.11×10^{-2} m^3/s	(174 gpm)
Max. Head	5.17×10^5 Pa	(173 ft)

Table 1 Pump Specifications

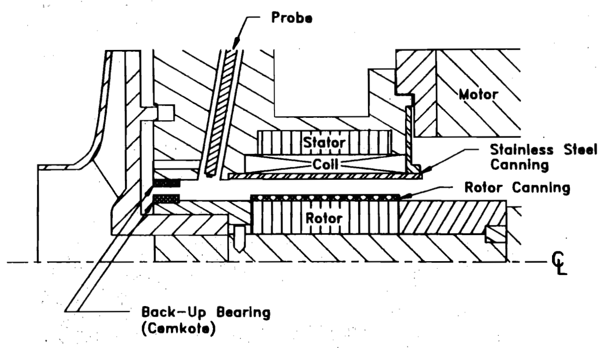

Figure 2 Example of Bearing Canning

The bearings are constructed from .18 mm (.007 inch) thick silicon iron laminations. The bearings were designed to provide the maximum load capacity consistent with the geometric constraints of the retrofit. This general design specification of the load capacity of the bearing resulted from the uncertainty of the end use of the pump. The major load component on the bearings is from the impeller fluid forces which are dependant on the working fluid as well as the hydraulic conditions. Each of these vary with the pump application. For reference, table 2 compares the design loads with the maximum predicted loads, assuming water as a working fluid. It is seen that this design strategy results in a margin of safety regarding bearing load capacity.

Bearing	Design load	Max. Load
PE Radial	1110 N	565 N
ME Radial	445 N	95 N
Thrust	2225 N	555 N

Table 2 Design and Max. Predicted Loads

Prior to the installation of the pump in the field several diagnostic tests were performed on the magnetic bearing equipped pump in order to further demonstrate the capabilities of magnetic bearings reported by Humphris et. al. [3].

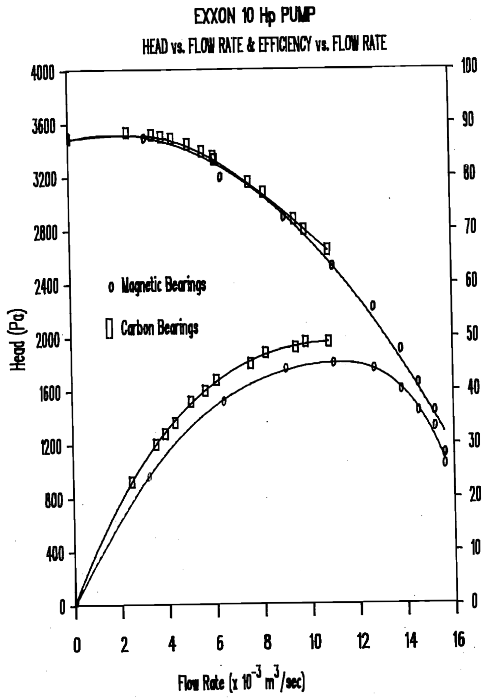

Figure 3 Head and Efficiency Comparison

2. Flow Testing

One of the objectives of this project was to prove that canned magnetic bearings have no adverse effect on pump characteristics. To accomplish this, head and efficiency vs. flow rate data was collected on the "as received" pump. This data was collected again after the magnetic bearings were installed. These sets of data are compared in figure 3. The slight decrease in pump performance in the magnetic bearing equipped pump is attributable to an increase in the by-pass flow recommended by the pump supplier to cool the motor.

3. Diagnostics

In order to investigate the system characteristics prior to operating the machine, the magnetic bearings were used as an investigative tool [3]. This allowed the system to be analyzed and the controller characteristics to be set before the pump was ever run. The diagnostic functions performed in this investigation include: centering the magnetic offset position within the available back-up bearing clearance; measuring the system critical speeds and appropriately adjusting the bearing stiffness, and; measuring the system damping ratio and appropriately adjusting the bearing damping.

3.1 Centering the Bearing There are two possible configurations in which the bearing can be considered centered. The first is when the rotor is centered within the available magnetic clearance. The second, when it is centered within the geometric clearance. Ideally these two will coincide. Practically, however, they will almost always be different by a small amount. Because the magnetic bearing can provide support anywhere within the magnetic clearance it was decided to center the bearing geometrically. To do this for the radial bearings a sinusoidal perturbation signal was injected into the controller of each horizontal axis. Simultaneously, the same signal with a 90 degree phase shift was injected into each vertical axis. This causes the shaft to orbit without rotating about its centerline. The resulting orbits can be observed on the output of the bearing position sensors.

Using this procedure the amplitude of the disturbance can be increased until the backup bearings are contacted by the shaft. The zero position of the shaft can then be adjusted to eliminate the contact. This can be repeated until the bearing is geometrically centered within its available clearance. Figures 4 - 5 show the pump end bearing orbits just prior to contact and with the back-up bearing in full contact. Figures 4 and 5 represent the bearings before and after canning respectively. The gain of the phase shifting portion of the electronics varied slightly from unity resulting in the elliptical shapes of the orbits. Additionally it is noted that the 'before canning' shaft is slightly (approx. .02 mm) more centered than the 'after canning' shaft. This could have been corrected if necessary, but the demonstrated clearances were more than sufficient for the machinery involved.

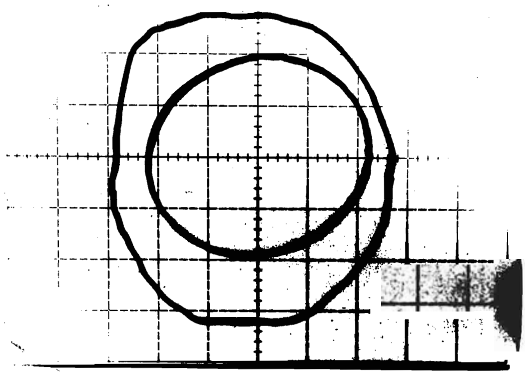

Figure 4 Bearing Orbits Prior to Canning

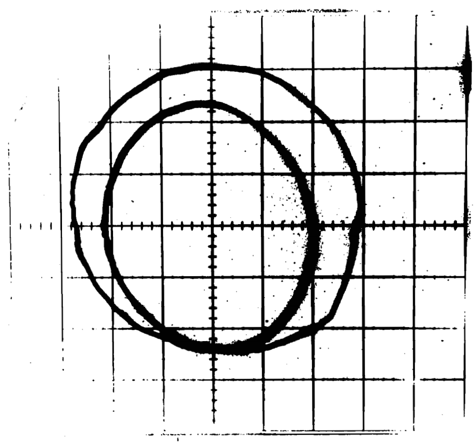

Figure 5 Bearing Orbits after Canning

It is most important to note that the outer orbit, representing full contact with the back-up bearing, is similar in size for the two cases. This shows that the canning of the bearings does not infringe on the back-up bearing clearance at any point. A similar method was used to center the thrust bearing and verify its clearance.

<u>3.2 Measuring System Critical Speeds.</u> The system critical speeds can be investigated in two ways. The first and simplest way is to inject white noise into the perturbation input of the bearing controller and examine the averaged frequency spectrum that results. This is the electrical equivalent of multiple rap tests of the system. The result of this will generally give a broad spectrum peak at each system resonant frequency, whether it is a structural or a rotor critical. No phase information is obtained.

The second method was presented by Humphris et. al. [3] and is based on the orbiting perturbations described in the previous section. To obtain critical speed information, the frequency of the rotating field is varied over the frequency range of interest. The output can be analyzed in the same manner as a machine run up or coast down. This method is preferable to the one described above in that the output is more representative of the actual machinery response and phase information is available.

Figure 6 illustrates some testing results and demonstrates the procedure for selecting an appropriate controller proportional gain (bearing stiffness) for this pump. The first curve represents a stiffness setting of 1.25 on the controller. A comparatively large peak occurs at 43 Hz (2600 rpm) and the phase information verifies that this is the

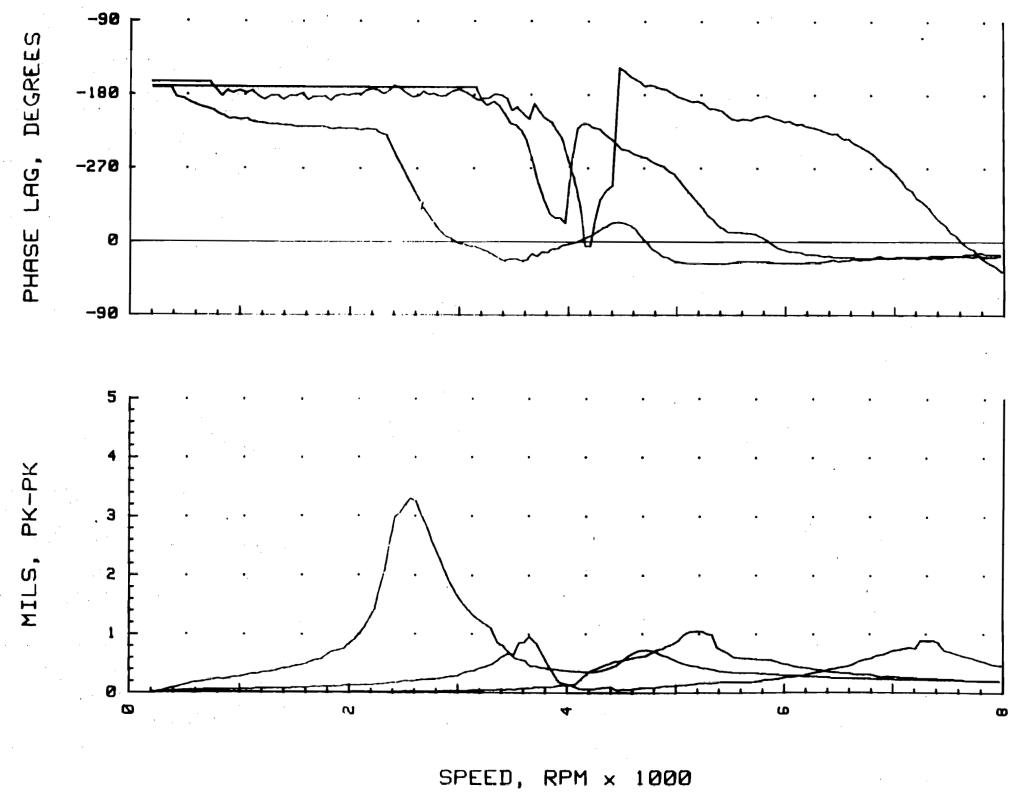

Figure 6 Setting the Bearing Stiffness

first rotor critical. The second curve represents a doubling of the controller gain and a corresponding increase in the first critical to 60 hz (3600 rpm). This is the running speed for this pump, so a further increase in stiffness is desired. The third curve represents another doubling of controller proportional gain. For this case the first critical (from the phase information) is at 67 Hz (4000 rpm) and appears very well damped compared to the previous peaks. This setting is consequently a good candidate for an operating stiffness. This procedure can easily be performed on each axis of support to determine the overall stiffness requirements.

Figures 7 and 8 represent additional examples of this diagnostic method. The figures are representations of identical controller stiffness settings prior to and after canning. The damping differs between the two figures, resulting in the different magnitudes. The curves show the same characteristic shape. The foundation natural frequency (3000 rpm) in the vertical axis is quite evident in both figures. The first rotor critical at approximately 4000 rpm is also apparent in both figures. The curves maintain similar shapes until 5600 rpm when the curves in figure 8 level off and become ragged. This probably indicates a minor contact between the rotor and

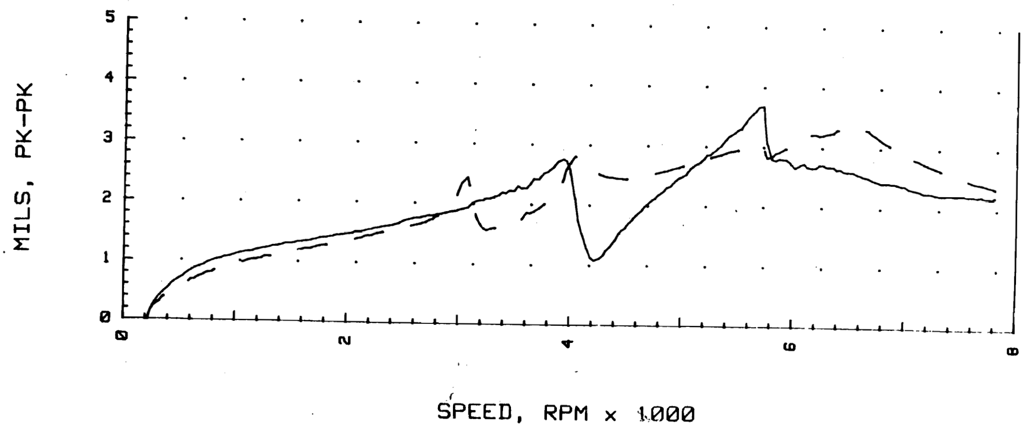

Figure 7 System Criticals for Un-Canned Bearings

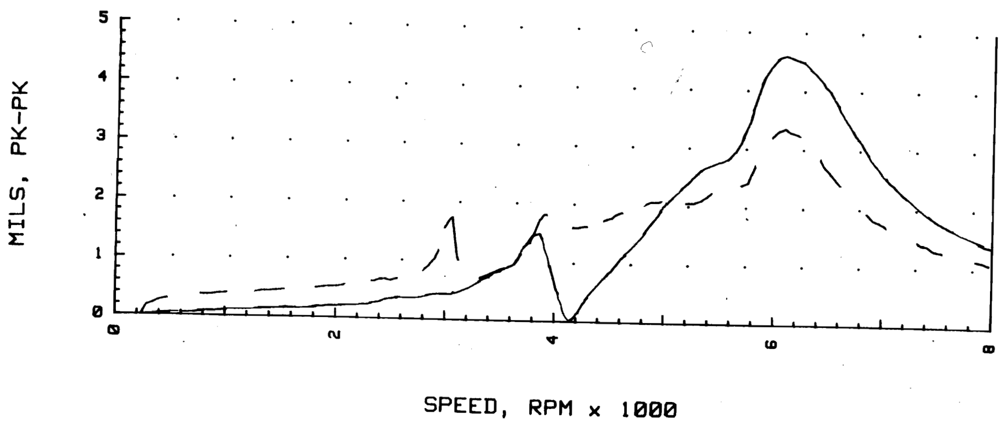

Figure 8 System Criticals for Canned Bearings

stator in the horizontal axis. In general, however, this method is seen to give very repeatable results even though the bearing damping may have changed. Additionally, the comparison between before and after canning shows no significant differences.

3.3 Determining System Damping. The damping of the system can be investigated by inputing a step perturbation and examining the resulting shaft displacement vs time. Damping ratios can be calculated by measuring consecutive peak heights. Usually, however, it is sufficient to adjust the damping such that critical damping is approached. Figures 9 and 10 illustrate this process for the present pump prior to and after canning, respectively. The first plot in each figure corresponds to an initial, poorly damped, setting. The second plot is characteristic of the same system with increased damping. The controller derivative (bearing damping) gains are the same for the two figures. Damping settings similar to the higher ones shown were chosen as the operating points for all of the support axis. When the pump is filled with operating fluid, these derivative settings correspond to over damped system response.

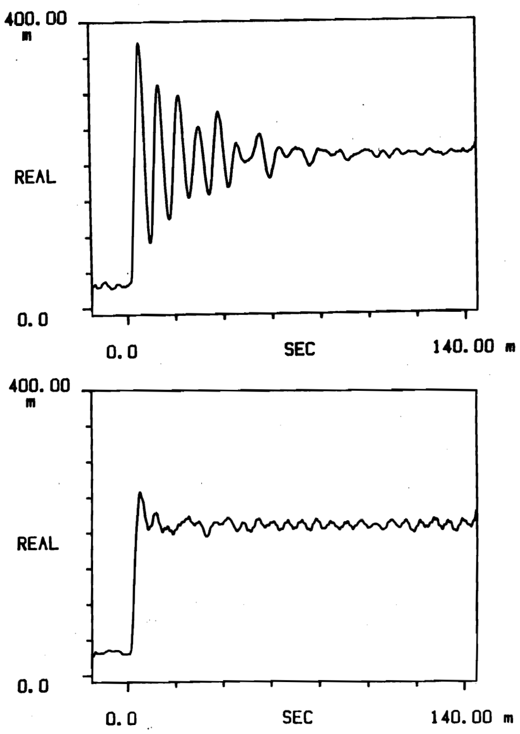

Figure 9 Un-Canned Bearing Damping Characteristics

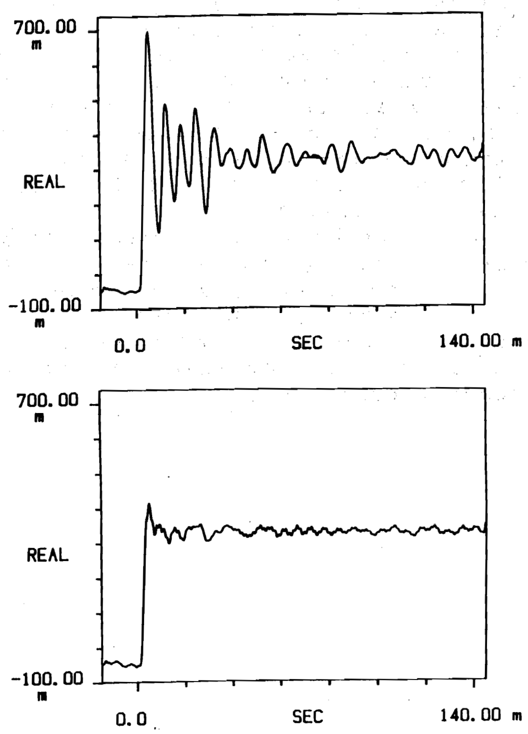

Figure 10 Canned Bearing Damping Characteristics

4. Discussion and Conclusions.

After the above procedures were used to determine the operating point the pump was run. No problems were encountered with the rotordynamic operation of the pump in magnetic bearings. Flow testing was repeated and no significant changes were noted as a result of the magnetic bearing retrofit.

The comparison of physical clearances and bearing damping characteristics before and after canning of the bearings indicate minimal, if any, adverse effects. This was also demonstrated by the pump performance at running speed (60 Hz). The expectation is that canning may effect higher frequency disturbances, but this was not investigated in the current work.

The diagnostic capabilities of the magnetic bearings have been shown to be a valuable tool in setting bearing parameters prior to machine start-up. These same diagnostic methods and capabilities can be used for machine characteristic trending as part of a preventative maintenance program and for diagnostics in the event of machine problems. Additionally, because these diagnostics can be performed without rotating the shaft, there is less risk of damaging equipment as compared to multiple start-up and run-down tests.

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