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# APPLICATION OF MAGNETIC BEARINGS IN A MULTISTAGE BOILER FEED PUMP

Gordon McGinnis and Paul Cooper
Ingersoll-Rand Company (IR)

Gerald Janik
New York State Electric and Gas (NYSEG)

Graham Jones
Gas-Cooled Reactor Associates (GCRA)

Richard Shultz
Magnetic Bearings, Inc. (MBI)

Abstract

Magnetic bearings have been fitted to an eight-stage, 600 hp. boiler feed pump in an effort to bring to the electric utility industry the attendant benefits of longer bearing life, lower maintenance costs and better operability through control of the rotodynamics.

This pump generates 2605 ft of head at 680 gpm and 3560 rpm. It will be subjected to varied and severe operating environment of a power plant, which results in loads that can now be measured at the bearings. The pump will be monitored and evaluated against a similar pump containing conventional bearings for a period of one year in order to establish the feasibility of magnetic bearings for boiler feed pumps.

## INTRODUCTION

Magnetic bearing usage in pumps is a natural development when one considers the successes of these bearings in large centrifugal compressors *(1). For pumps of significant power level, the elimination of lubrication systems that would otherwise be needed for conventional bearings and the attendant reduction of maintenance are benefits that should be claimed.

Recently, magnetic bearings were applied to a 20-horsepower, canned-motor, single-stage, centrifugal pump, wherein the conventional bearings were replaced by submerged, canned, magnetic radial and thrust bearings (2).

Application to multistage pumps with orders of magnitude greater power is the next step and is a further test of the feasibility of magnetic bearings, because such pumps often operate with close internal clearances and flexible shafts. Typical of such pumps are those used for boiler feedwater in the electric utility industry. In studies of magnetic bearings for that industry, financial viability was established for applying these bearings in boiler feed pumps (3,4). Increased availability and reduction in operating and maintenance costs were projected, leading to a favorable evaluation (5).

The design of a multistage boiler feed pump is a compromise between allowing sufficient clearance for shaft flexure and thermal distortion and minimizing clearances to increase the efficiency of the impeller ring and the shaft sealing systems. Ideally the bearings would be mounted closer together than is current practice, with a closer tolerance on shaft movement. If submerged magnetic bearings were used, then this could be accomplished by eliminating the outboard (non-drive end) seal and putting the inboard bearing between the first-stage impeller and the inboard seals (6).

Phenomena unique to pumps that could affect magnetic bearing performance are:

(a) The Lomakin Effect. This is the bearing-like stabilizing action that occurs as high-pressure fluid from the discharge of each impeller leaks through the tight annular clearance of the neck ring "seal" back to the impeller suction eye. Shown in Figure 1, this

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* Numbers in parenthese denote references listed at the end of the paper.

effect is not always stable - being influenced negatively by swirling of the fluid entering the clearance space. These phenomena can have a confusing influence on otherwise straightforward and understood rotordynamic behavior (7).

(b) Increases in Clearance. Wear of the impeller neck rings or adjacent casing rings leads to loss of stabilization and therefore to adverse changes in rotodynamical behavior.

(c) Loss of Liquid. A further complication would be the loss of any stabilization arising from the Lomakin effect should liquid be momentarily absent from one or more stages of the pump.

(d) Stall Effects. These are the non-uniform flow interruptions or reversals that occur in a pump impeller and diffuser when they operate at reduced flow rates in comparison to that for which they were designed. These effects introduce unsteady, random, radial and axial loads (8).

The mechanical responses to these phenomena become more evident and critical in pumps of high energy level; i.e., those of more concentrated and higher power. For this reason it would be prudent to avoid the extensive and fundamental engineering design effort associated with magnetic bearings for high-power pumps until more is known about their behavior in such pumps in general, their ability to handle these otherwise detrimental situations, and the actual instantaneous loads that might be encountered.

This understanding would appear to be more readily obtainable by first building an intermediate-power pump in the 500 to 1000 hp range with external magnetic bearings running in air and replacing the existing externally mounted conventional bearings.

Since the current in a magnetic bearing is related to the load, readings of current vs. time, speed and pump flow rate should provide the needed insight into the stall effects. Scaling of the loads involved is fairly reliable and should enable one to make a more informed estimate of the loads for a still-higher-power submerged magnetic bearing pump. Also, the ability of the magnetic bearings to cope with the rotordynamic uncertainties mentioned above would be revealed and could be analyzed for the future submerged-bearing application.

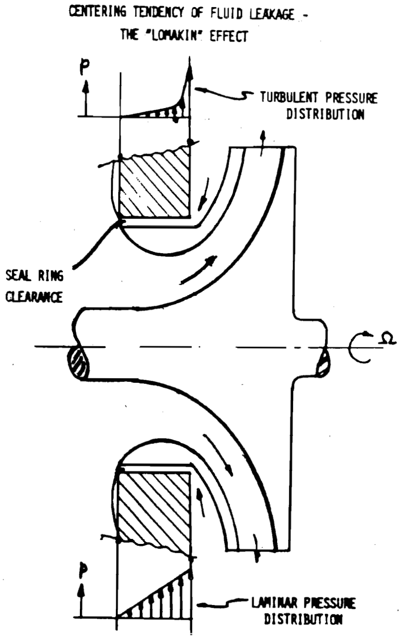


Figure 1. The Lomakin effect.

## THE BOILER FEED PUMP APPLICATION

GCRA, part of a team under contract to the Empire State Electric Energy Research Corporation (ESEERCO) initiated a magnetic bearings application study in 1988 (4). The purpose of this study was to identify potential application(s) for magnetic bearing technology in large rotating machinery in a utility application. As a result of this generic study, many potential utility applications/equipment were identified, one of which was boiler feed pumps.

A continuation of this study in 1989 was to identify specific candidate pieces of equipment and a host site to demonstrate this evolving technology in a fossil-fired power plant application.

A boiler feed pump application is felt to be an ideal test/demonstration candidate as it is an energy intensive piece of equipment subject to many operational constraints (upsets). In addition, standard (traditional) boiler feed pump bearings are a big industry problem accounting for more than 11,000 MW-hrs of lost generation (9).

Magnetic bearing technology in a boiler feed pump application offers many potential benefits including improved maintenance techniques, reliability and efficiency. Reduction in failures due to shaft breakage and other mechanical problems caused by vibration, wobbling or thrusting, as well as reduction in forced outages or load loss situations are further benefits.

In mid-1989, New York State Electric & Gas Corporation (NYSEG), an ESEERCO member, became aware of the magnetic bearing technology and its potential application to boiler feed pumps, Since ESEERCO was looking for a host fossil site and NYSEG was planning to replace the boiler feed pumps on Unit No. 3 at its Greenidge Station in the Spring of 1990, NYSEG offered this unit as a host site for this boiler feed pump magnetic bearing test/demonstration.

Greenidge Unit No. 3 is a 58MW (gross) plant with an in-service date of May 1950. This Greenidge Unit No. 3 site was felt to be attractive since the three outmoded existing 50 percent boiler feed pumps were in the process of being replaced, the timing was right for getting a boiler feed pump equipped with a magnetic bearing and the unit is equipped with a spare boiler feed pump. The 700 HP size for this first time electric utility application was ideal. A magnetic bearing installed on a single pump at this location can be tested with a reduced amount of risk to power plant operation since there will also be two 50 percent pumps equipped with conventional bearings and the application will provide a direct comparison of magnetic with conventional bearings.

It was agreed that ESEERCO would fund the fabrication and installation of the magnetic bearing on a single pump and NYSEG would fund the one year test program at its Greenidge Unit No. 3; a two boiler, two feed pump operation.

The one year test program is scheduled to begin in August 1990. Testing will take place over a one year period during normal/usual operation of this base-loaded unit. During the test program, unusual events will be recorded and diagnosed. Upon completion of the test program, performance as well as malfunctions/outages due to pumps equipped with magnetic bearing versus conventional bearing will be determined. From this, it is expected that projections of future pump performance can be made.

PUMP DESIGN

The boiler feed pump is designed to pump 680 gpm (42.9 l/s) of 254 F (123 C) feedwater to a head of 2605 ft. (794 M). Shaft power consumption at this condition is 553 hp (413 kW). a conventional 700 hp (522 kW) induction motor drives the pump at 3560 rpm. The pump is designed as an eight-stage, volute-type, opposed-impeller machine; with four impellers facing the drive end and four facing the non-drive end, as shown in Figure 2.

A high degree of axial thrust balance exists, producing low resultant loads on the axial thrust bearing. Similarly a low resultant radial thrust unbalance results from the use of opposed dual volutes and stage crossovers in the horizontally split pump casing.

Hardened stainless steel rotating and stationary wearing rings have a 0.009 inch (0.23mm) radial clearance. These impeller and casing rings are fitted to each stage of the pump. Ring contact during normal operation is prevented by the self-centering magnetic bearings in the pump so equipped, or by the conventional oil lubricated journal bearings in the other two pumps.

A loss of power to the magnetic bearings when in operation would allow the rotor to drop. To guard against ring contact in this circumstance, three metal filled carbon-graphite auxiliary bushings have been fitted to the magnetic bearing pump as depicted in Figure 2. The two bushings located adjacent to the mechanical seal fitted boxes as well as the center bushings have a radial clearance of 0.005 in. (0.13mm). Metal filled carbon-graphite is successfully used for bearings and bushings in a wide variety of liquid and gaseous applications.

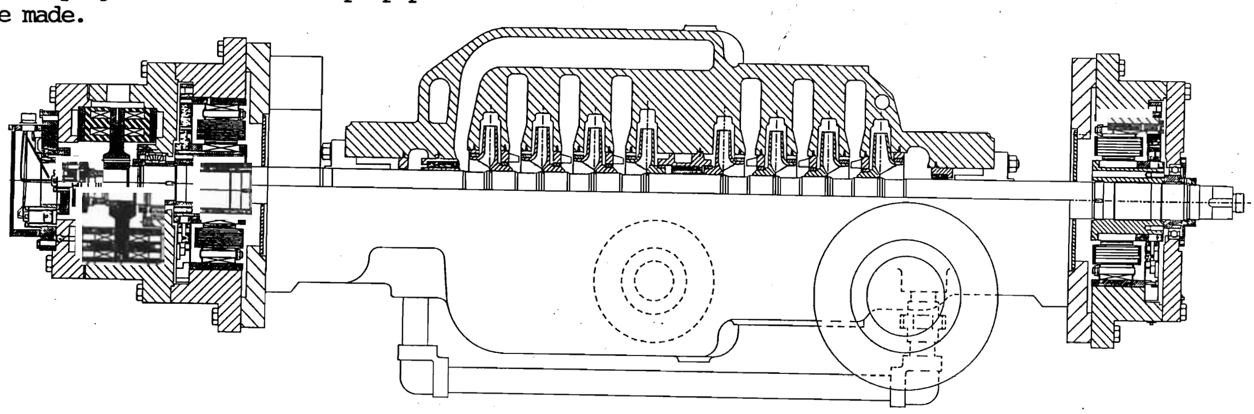

Figure 2. Multistage volute pump fitted with magnetic bearings.

MAGNETIC AND AUXILIARY BEARINGS

The load capacity requirements for each of the magnetic radial bearings are 280 lb. (127 kg) steady-state and an additional 280 lb. (127 kg) for transient conditions. The requirements for the magnetic thrust bearing are 1,000 lb. (454 kg) steady-state and an additional 1,000 lbs. (454 kg) for transient conditions.

In the design of magnetic bearing systems, it is prudent to include load capacity margins to accommodate process induced surge loads and dynamic loads such as abnormal imbalance loads and hydrodynamic loads. Standard magnetic bearing designs were used which provided additional margins over conventional practice. The magnetic radial bearings are designed to provide a maximum steady-state load capacity of 600 lb. (272 kg) and to modulate a force of 400 lb. (181 kg) at a frequency of 60 Hz. The magnetic thrust bearing is designed to provide a maximum steady-state load capacity of 4,000 lb. (1814 kg) and to modulate a force of 1,000 lb. (454 kg.) at a frequency of 60 Hz.

The active magnetic bearing system assemblies are shown on Figures 3 and 4. Each magnetic radial bearing consists of a laminated silicon steel rotor attached to the shaft and a wound stator with a silicon steel lamination stack. The radial bearings have a 6 inch (152.4 mm) diameter bore and a lamination stack length of 2.5 inches (63.5 mm). The magnetic thrust bearing consists of a 11.25-inch (285.8 mm) diameter thrust disk and a laminated wound stator with a 6 inch (152.4 mm) inside diameter and a 11.25-inch (285.8 mm) outside diameter.

All magnet wire insulation and coatings are rated for continuous operation at 200 C. The bearing windings are equipped with RTD's which can be used to monitor operating temperatures.

The electronic control cabinet contains all transformers, control cards, amplifiers, and batteries necessary for the active magnetic bearing system. The cabinet power supply requirement is 208V, 60Hz, 3-phase, 2KVA. The overall cabinet dimensions are 52 inches (1321 mm) high, 21 inches (533 mm) wide, and 20 inches (508 mm) deep and the cabinet is air-cooled. The magnetic radial bearings are powered with eight 120V/15A power amplifiers and the magnetic thrust bearing is powered with two 120V/30A power amplifiers.

The cabinet is equipped with a battery backup power supply. The 208V AC external supply is transformed and rectified to a 93V DC supply. If the 93V DC supply decreases by 15 percent, the external supply is switched out of the circuit and the batteries provide the power for the cabinet. The batteries will power the bearing system for 10 minutes. After 10 minutes the system will maintain operation, but bearing load capacity may be diminished.

The auxiliary bearing at the non-drive end of the machine is a duplexed pair of angular-contact ball bearings. The bearing provides radial and axial load carrying capability if the pump were ever required to coast to a stop upon loss of both primary and backup power to the magnetic bearings. The auxiliary bearing at the drive end of the machine is a Conrad type single radial bearing.

The radial air gap for the magnetic radial bearing is .020 inch (0.51 mm) and the radial air gap for the adjacent auxiliary bearing is .010 inch (0.25 mm). The magnetic axial thrust bearing air gap are 0.028 inch (0.71 mm) and the auxiliary thrust bearing airgaps are 0.014 inch (0.36 mm).

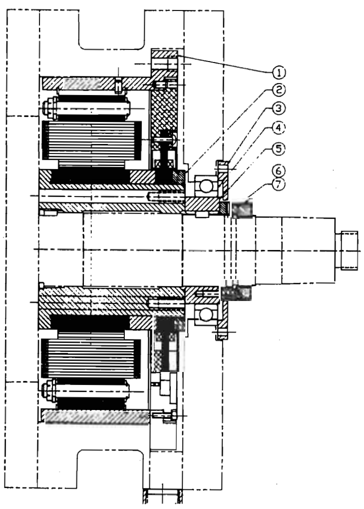


Figure 3. Radial magnetic bearing on drive end of pump.

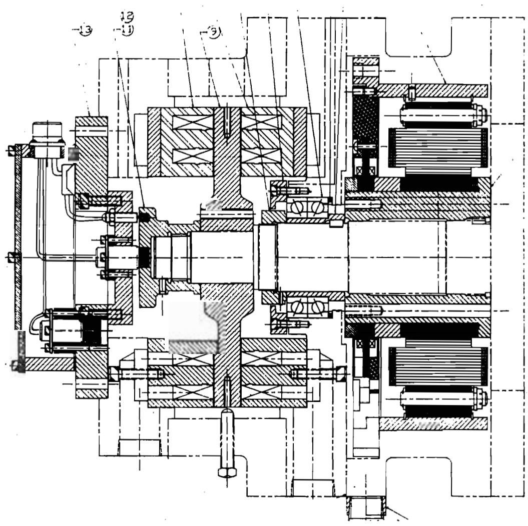

Figure 4. Radial and axial magnetic bearings on non-drive end of pump.

The active magnetic bearing system includes two inductive radial position sensors and one inductive axial position sensor. The radial position sensors are located adjacent to the radial bearing lamination stacks and the axial position sensor is located at the end of the shaft on the non-drive end of the machine. After signal processing, the position sensor signal has a sensitivity of approximately 625 mV per mil. The radial position sensors are designed to reject the third and even number harmonics of any rotor imperfections which would cause position sensor signal noise.

OPERATION

The magnetic bearings, when activated, maintain the rotor position within the clearances mentioned above. Should the power to these bearings be lost - including the reserve power provided by the battery back up system - the shaft will drop. As the pump coasts down, the shaft will then contact the carbon-graphite auxiliary bushings because they have a smaller radial clearance than the rolling element auxiliary bearings.

At first glance it would thus appear that these rolling element auxiliary bearings are not needed. However, if sustained operation on the carbon-graphite bushings should cause them to wear significantly, the rolling-element auxiliary bearings will provide the ultimate protection against magnetic bearing rotor-to-stator contact. Further, the impeller rings will also be protected, because shaft deflection outboard of the carbon-graphite bushings will assure contact with the rolling element auxiliary bearings prior to impeller ring contact with the adjacent casing rings. Protection is assured against sustained loss of magnetic levitation as well as against any upset condition that might occur as a consequence of pump operation in the power plant environment.

SCHEDULE

The magnetic bearing equipped multi-stage boiler feed pump will be assembled and factory tested at IR in June/July 1990. After the radial and axial magnetic bearings and the electronic control cabinet are tuned to the feed pump characteristics by MBI, an operational test will ensue. Using the same 700 hp motor that will drive the pump at Greenidge, the unit will be run at varying flows, and rapidly changing flows, to simulate plant load changes and a load rejection.

Primary power will be shut-off to assure that the back-up battery system functions as intended to maintain the pump rotor levitation. A feed pump trip, following a cessation of all power to the magnetic bearings, will be conducted to demonstrate satisfactory pump coast-down with the pump rotor supported by the auxiliary support system as described above.

Following shipment of the magnetic bearing fitted boiler feed pump, driver and baseplate to Greenidge, the equipment will be installed, and under the direction of IR and MBI any needed additional fine tuning of the electronic/electrical performed.

IR and MBI will train NYSEG operations and maintenance personnel in the performance and care of the magnetic bearing equipped feed pump. At this point the equipment will be released for commercial operation and a one year monitoring program, described in the next section, will commence.

MONITORING

The one-year monitoring program will be undertaken in order to make a reasonably accurate assessment of this overall magnetic bearing demonstration project. The monitoring program will determine,

a) The net benefit to the operating utility relative to conventional bearings - in terms of economics, staffing levels, skill required, and plant availability.
b) Whether operational feed back from this first pump will allow future pumps to be designed that will be more efficient and able to withstand potentially damaging transients,
c) Whether the enhanced diagnostics that include the display or other evidence of shaft loads will increase the operator's ability to protect equipment during transients through a better understanding of these loads and the attendant rotodynamics.

To make the economic assessment, data will be obtained for both the magnetic bearing pump and one of the other conventional-bearing pumps for one year. Demonstration of an expected availability of the bearing system of 40,000 hours will require that data acquisition continue for several more years as well.

Data will be obtained at a greater frequency and in more detail in the first month of operation. This will provide the feedback needed for load analysis and establishment of magnetic bearing design requirements for future higher power pumps. At the power level and configuration of the current machine, there is no reason to expect excessive loads; the low-energy volute-type design with axially opposed stages limiting the radial and axial imbalance. These pumps are thus very suitable for this demonstration as operators will be able to display actual bearing loads under running conditions - thereby allowing the pump designer to determine these loads more precisely for future designs.

Control of pump operation and monitoring will be accomplished by interfacing with the plant computer. This computer has a maximum sampling speed of 1/10 second, which will be adequate except in the first month of operation when the more detailed bearing load data will be obtained on strip charts and viewed on an oscilloscope. Appropriate alarms will be provided from limits on loads, temperature, etc., which can then be acted upon a coordinated manner.

## CONCLUSIONS

Boiler feed pumps in an electric utility can benefit from the life and low maintenance advantages of magnetic bearings. Such pumps can provide a comprehensive platform for evaluating the feasibility of magnet bearing technology under a variety of severe operating conditions that can impose strong and random loads on the rotating elements.

By conducting an in-plant demonstration of a 600 hp magnetic-bearing equipped eight stage volute-type boiler feed pump, it is believed possible to more precisely determine the load requirements for such bearings in pumps with an order of magnitude greater power input.

By monitoring the operation of the 600 hp pump, which has the magnetic bearings mounted external to the pump casing and running in air, it should be possible to gain the rotordynamical insights to design improved, higher-power machines with these bearings submerged in fluid within the interior of the casing.

Fabrication of the demonstration pump is underway, with a one-year test, monitoring and benefits evaluation program on two side-by-side new pumps - one with magnetic bearings; the other, with conventional bearings - to begin in the Summer of 1990. If successful, this project could provide an alternate technology to the utility industry resulting in higher availability and more efficient operations.

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