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CENTRIFUGAL PUMP MAGNETIC BEARINGS

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Magnetic bearings maintain the rotor of a pump in suspension through the forces of attraction of a magnetic circuit. Thus, although they bear up the weight and hydraulic loads of the impellers and the shaft, they are not really bearings in the traditional sense of the rotating and stationary surfaces bearing on one another. The supporting magnet circuit for each bearing includes stationary magnets in a stator surrounding the shaft, a laminated rotor that fits on the shaft, and the shaft itself. The stator consists of electromagnets in the traditional heteropolar design, and if a homopolar design is employed, permanent magnets can be added. Sensors monitor the position of the shaft and signal a controller to adjust the magnetic loads to keep the shaft to within about 0.001 in (25 μm) of the desired position.

Magnetic bearings are found in small, high-speed turbomachinery such as high-speed, multistage, axial-flow turbomolecular vacuum pumps[1]. They were introduced into large turbomachinery in the early 1980s, mainly in gas compressors and turboexpanders. Their use and acceptance has grown slowly but steadily since then[2]. Pump applications of a significant size have appeared and have confirmed the general position that magnetic bearings can provide a technically sound bearing with maintenance and operating advantages, including zero wear. However, due to the technical complexity of magnetic bearing systems, the economies of scale associated with production quantities are required to make these systems affordable.

Two representative magnetic-bearing-equipped pumps are summarized in Table 1. One is a multistage boiler feedwater pump[3–6] and the other a single-stage double-suction hydrocarbon process pump[7]. The multistage pump was retrofitted with magnetic bearings (as shown in Figure 1) and is shown in Figure 2, together with another identical pump that still contains the oil-lubricated bearings—both installed in an electric generating station. The magnetic-bearing pump is not encumbered with the usual complexity of a bearing lubrication system.

TABLE 1 Example of magnetic-bearing-equipped pumps

Parameter	Multistage Pump	Single-Stage Pump
Power, hp (MW)	610 (0.46)	800 (0.6)
Rated speed, rpm	3,580	1780
Shaft weight, lb (kg)	520 (236)	732 (332)
Radial bearing design load, lb (kN)	800 (3.6)	Thrust end: 930 (4.1) Drive end: 1,415 (6.3)
Thrust bearing design load, lb (kN)	4,000 (17.8)	4,000 (17.8)
Number of stages	8	1

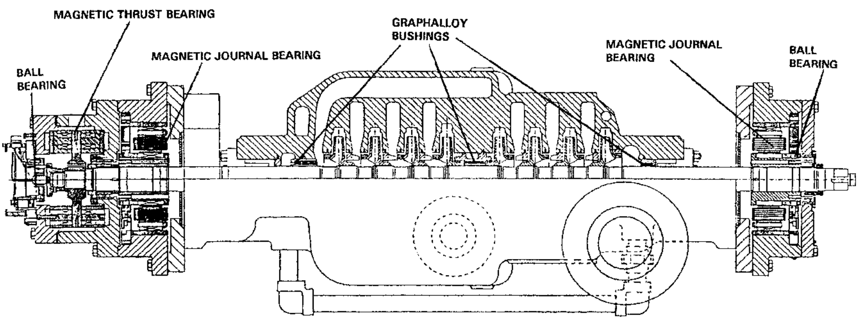


FIGURE 1 Magnetic bearing configuration in multistage pump[6]

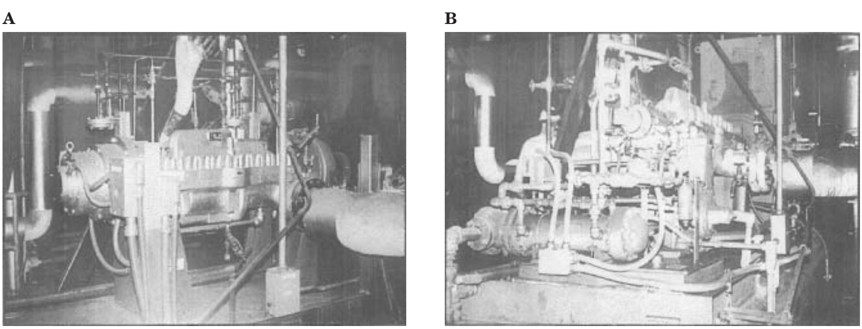


FIGURE 2 Multistage pumps installed in boiler feed service [610 hp (0.46 MW)]: a) pump with magnetic bearings; b) the same pump with oil-lubricated bearings. (Flowserve Corporation)

MAGNETIC BEARING PRINCIPLES

How Magnetic Bearings Work In an active magnetic bearing system, a stator composed of an array of stationary magnets, or electromagnetic coils, interacts with a ferrous rotor (or a ferrous sleeve on a non-ferrous rotor) so as to suspend the shaft in a magnetic field (see Figure 3).

The position of the shaft is maintained dynamically through a continuous feedback system which comprises a position sensor, a controller, and an amplifier system (see Figure 4). Typically there are two radial bearings and one thrust bearing for a complete sys-

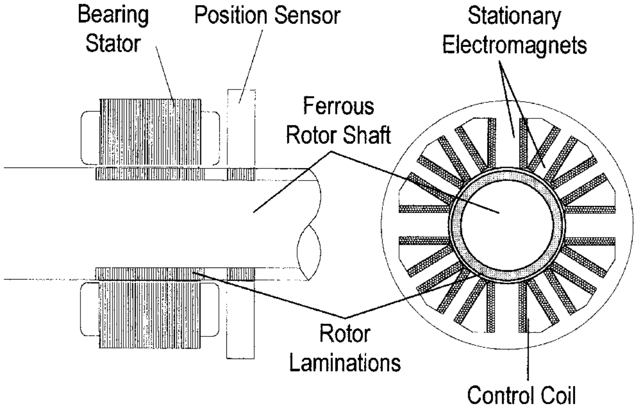


FIGURE 3 Typical radial electromagnetic bearing. (Axial thrust bearings have the stator coils arranged in a disk configuration, a ferrous rotating disk being supported in the resulting magnetic circuit.)

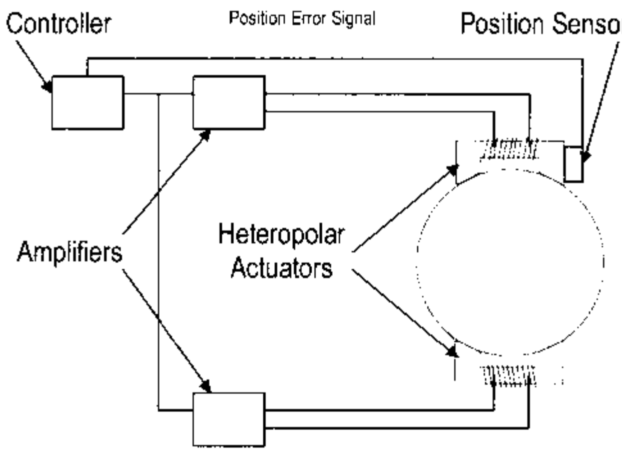
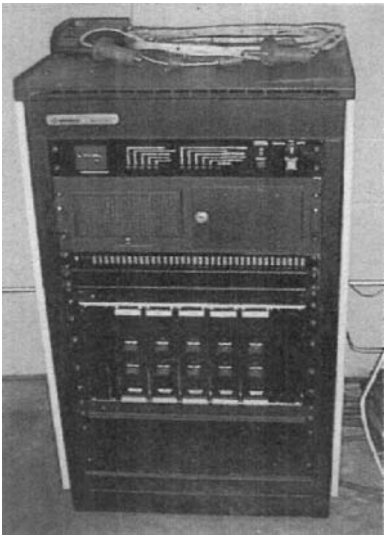


FIGURE 4 Typical system control loop

tem. This system is tuned to the required characteristics of the pump, through a digital or analog controller (Figure 5), with the capability of adjusting the bearing stiffness and damping as a function of pump speed. Alarms and trips can be set at any required rotor offset or bearing load to provide the operator with warnings or to trip the drive unit as necessary. Figure 6 illustrates a typical bearing transfer function, showing a statically stiff bearing, with a dynamic stiffness over the operating speed range designed to meet the rotor dynamics requirements of the unit. The stiffness is then rolled off above the operating range to avoid excitation of higher modes in the rotor or stator.

Controller redundancy can be provided with the control loops switching to backup units upon sensing a failure. Also, backup power supplies should be provided, either through alternate sources or a battery system. Typically the power required is only one or two kW or less.

A catcher (also known as auxiliary, backup, or touchdown) bearing (indicated in Figure 1) is required to protect the rotor stator interface during maintenance and in the event of



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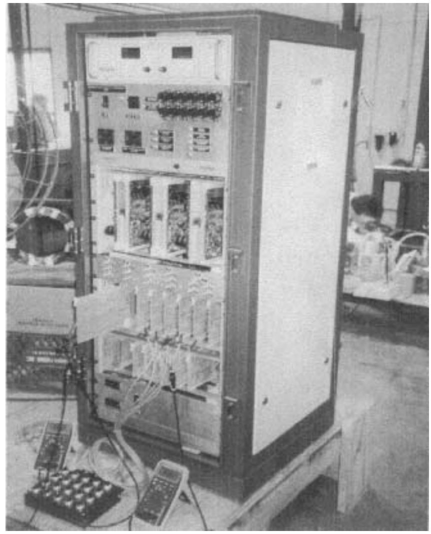

B

FIGURE 5 Controllers for magnetic bearings, containing rectifier and amplifiers: a) digital controller; b) analog controller

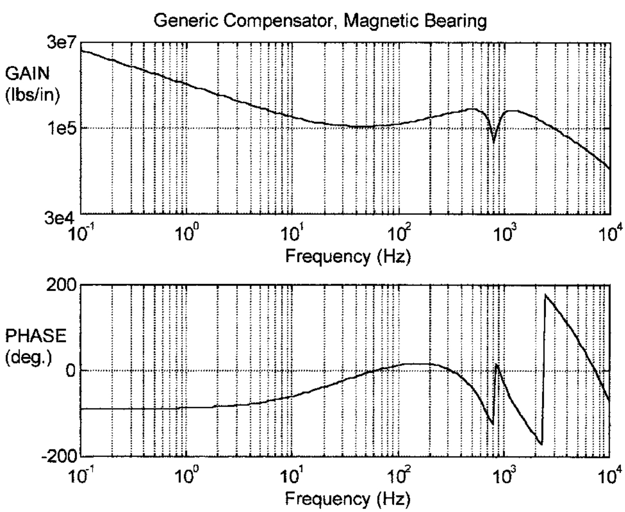

FIGURE 6 Bearing transfer function

loss of power or a severe transient beyond the force capability of the bearing. Typically, the catcher bearing is designed for 5 to 20 lifetime drops from full speed, and will be a readily replaceable rolling element or sleeve bearing. The pump of Figures 1 and 2 has rolling element catcher bearings. The radial clearances G1 (see Figure 7) between the magnetic

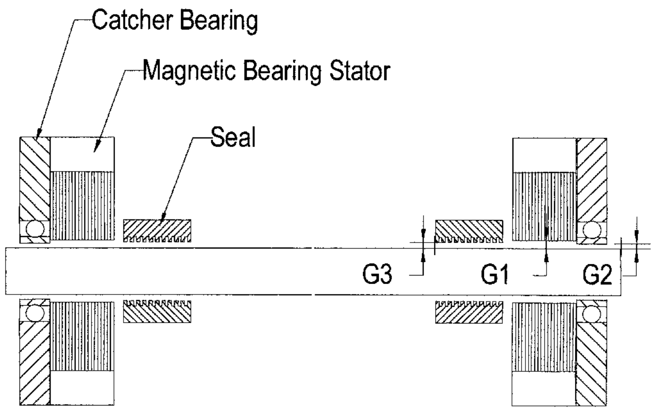

FIGURE 7 Clearance arrangement: Seal ring clearance G3 is greater than catcher-bearing clearance G2, but is less than magnetic-bearing clearance G1.

bearing stator and rotor are of the order of 20 to 40 mil (0.5 to 1 mm), and those in the catcher bearings (G2) are about half of that value.

Reasons for Using Magnetic Bearings There are several reasons to use magnetic bearings in pumps. While any one of these reasons may not be sufficient justification on its own, together they can provide a strong justification.

Reliability is a key incentive. The components of a magnetic bearing are essentially the same components as are found in an electric motor: laminations and coils. Because no wear is involved due to the lack of contact, these components will generally last the life of the equipment involved. Thus maintenance of a magnetic bearing system is transferred from mechanical components inside the pump to the external controller, which has plug-in card replacement maintenance. Pump reliability is therefore improved, whereas repair times and costs are reduced.

Reduced power consumption is a second advantage, with the elimination of all losses associated with fluid film bearings and oil pumping equipment. This is replaced by the smaller power requirements of the bearing controller. Further, if the lifting force is supplied by a permanent magnet, supplemented by an active control circuit, this power requirement can be even smaller.

The ability to submerge the bearing in the pump fluid is a major advantage that allows the outboard mechanical seal to be eliminated, thereby eliminating maintenance and replacement of this seal[8]. [This was not done for the pumps of Table 1 (and Figures 1 and 2), because in both cases magnetic bearings were retrofitted to existing machines.]

More indirect savings are also possible in two other areas. Rotor dynamics can be controlled through the ability to adjust stiffness and damping as a function of pump speed, allowing higher imbalance without the need for shutdown. The diagnostic output inherent in the information provided in the controller can be fed into the overall plant operating system and the short-term and long-term health of the pump and the system can be monitored. This is done by inferring seal wear, transient hydraulic loads, and so on.

The actual figures for the savings possible due to the previous advantages are very pump- and system-specific, and general numbers are not very useful. Reference 9 has developed methodology for considering the economic effect of the types of advantages given.

Main Types of Magnetic Bearings and Their Selection There are two main types of magnetic bearings: passive and active. Passive bearings rely only on permanent magnets in repulsion and provide low stiffness, low damping, and no ability to control either of these parameters. Passive bearings are not applicable to pumps for this reason.

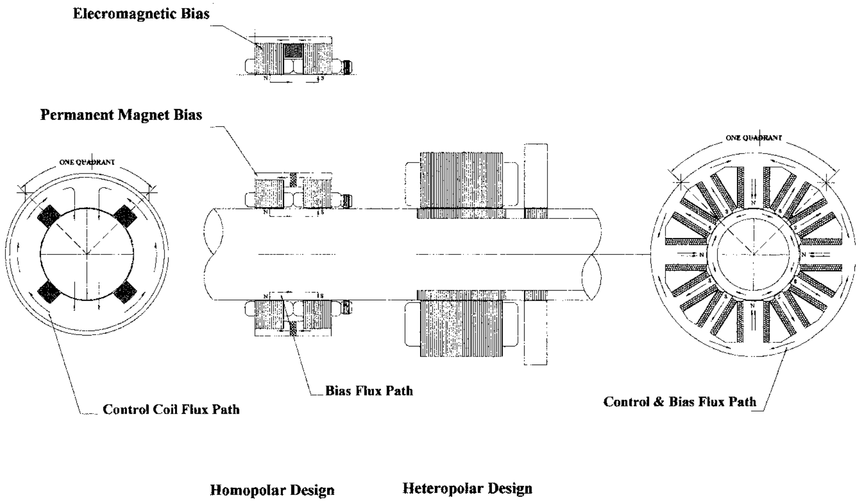

FIGURE 8 Heteropolar and homopolar bearings

Active bearings using the feedback system previously described are essential for pump applications. Within the active bearing systems are the options of heteropolar and homopolar and of electromagnetic and permanent magnet bias. The principles of the heteropolar and homopolar approaches are shown in the Figure 8.

The main difference between the two types is that in the heteropolar design, the bias and control flux flow in the same magnetic circuit radially through the rotor, whereas in the homopolar design the bias flux flows axially along the rotor and only the control flux flows radially through the rotor.

The homopolar design has two options for providing the bias flux for the bearing system[10]. One is to use an electromagnetic effect, and the other is to use a permanent magnet. The permanent magnet generation in the homopolar configuration results in a more linear relationship between force and distance. In a simple magnetic circuit, the attraction force of a magnet on a ferromagnetic target decreases as the square of distance (the target cuts $\frac{1}{4}$ as many flux lines at twice the distance). With the permanent magnet in a homopolar circuit, the effect of the air gap is therefore reduced.

DESIGN CONSIDERATIONS

Design Loads The specification of a magnetic bearing requires a different approach to that required for a conventional bearing system. This section identifies the key areas where these differences occur.

The rotor can move considerably within the magnetic bearing and catcher bearing clearances, typically up to 10 mil (0.25 mm), before any contact is involved. Thus the clearances G3 in the internal ring-seal system (see Figure 7) become the key controlling parameter in setting the bearing clearance design limits and in the degree of motion permitted during transients. Thus, very early in the design process, the magnetic bearing clearances, catcher bearing clearances, and sealing-ring clearances must be optimized with due consideration for manufacturing tolerances and assembly concerns.

Synchronous filters or open-loop control methods can be used to handle imbalance loads so that the degree of imbalance acceptable (based on allowable bearing loads and shaft motion within the catcher bearing clearances) can often be significantly greater than in conventional bearings.

There is an area where careful analysis is needed for each new pump configuration. A conventional bearing, if overloaded, will accept the load with a higher wear rate, but a magnetic bearing has a sharp cut-off at the point where the flux saturation level is reached. At that point, any additional load will be transferred to the catcher bearing. Thus it is important to include all loads in the design specification, with the appropriate margins. All applications to date have generally shown a) that there are loads which were mistakenly considered to be insignificant, or b) that the values of the loads were underestimated due to lack of knowledge.

Magnetic bearings also must be carefully designed to handle transient loads, some of which may occur only once in a lifetime. Two approaches can be taken. One is to overdesign the bearing to handle the load without contacting the catcher bearing; the other is to allow momentary contact with the catcher bearing. Examples of these types of loads are hydraulic loads, seal touchdown loads, system loads such as water hammer, valve operation, pump switchover, pump to driver alignment loads, and seismic loading.

Rotordynamics Considerations Magnetic bearings have the capability to control the rotor dynamics of a pump very effectively. If the pump is running below its first flexible mode, this is usually straightforward. If the pump has to traverse a flexible mode, the position of the bearings and the position sensor must be such that the modes can be recognized by the position sensor, and the bearings can exert a positive restoring force to control the mode. That is, the position sensors and bearings must not be positioned at or close to a node and certainly not positioned on opposite sides of a mode. Thus, rotordynamics considerations should be taken care of very early in the configuration of the system.

The two typical rigid body modes are shown in Figure 9. The location of the bearing and position sensor is not usually an issue for these modes, but the flexible modes require that the position sensor and bearing be positioned in such a way that the rotor deflection can be measured and the bearing can exert the necessary restoring force and damping to control the mode.

Magnetic bearing controllers are programmed with a transfer function designed specifically for the pump. This transfer function, or control algorithm, provides the necessary stiffness and damping at all operating speeds to control the rotor as previously described.

The key requirements of the transfer function are that it

- Provides correct damping and stiffness to handle the rotor rigid body modes
- Provides sufficient force with the appropriate control bandwidth to handle the rotor flexible modes below the maximum operating speed

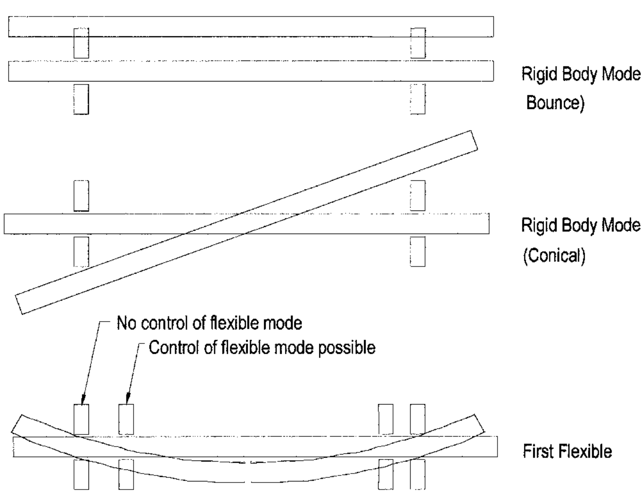

FIGURE 9 Flexible mode design considerations

- Does not excite any rotor modes above the maximum operating speed
- Does not excite any stator vibration modes at any frequency
- Takes into account the stiffness and damping contribution from wear rings and inter-stage annular seals

Auxiliary or Catcher Bearings However reliable magnetic bearings become, a landing surface for maintenance will be required. Further, designing a bearing that will take all transient loads without any possibility of overload will usually result in over-sized and costly bearings. Hence catcher, bearings are expected to be required. The design limits for catcher bearings are usually established by calculating the forces that will result from a drop at operating speed.

For applications running at significant speeds, a non-linear analysis is required to determine the motion and loading on the bearing during such a drop. The key design considerations are impact loads, heat generation during the rundown, and the response to imbalance if, when running on the catcher bearings, any critical speeds have to be traversed.

Rolling element bearings have typically been used as catcher bearings; however, sleeve bearings and bushings have been used in several applications, and are better suited to a submerged application.

MAGNETIC BEARING SIZING

The fundamental magnetic bearing sizing problem is to define the pole area at the bearing air gap that is necessary to achieve the desired force capacity without saturating the selected pole materials. Given the pole area, the minimum stator outside diameter and maximum rotor inside diameter can be determined using simple algorithms to ensure that no other part of the magnetic circuit saturates. The stator geometry must also include sufficient volume for the control and bias coils.

Approximate size and load capability Table 2 contains rough approximations for the dimensions of magnetic bearings. (Refer to Figure 1.) These are based on the experience with the pumps of Table 1. Also included are the unit load capabilities, which are given a) for radial bearings in terms of radial load F_r divided by the projected area DL of the active polar area of the bearing at the air gap, and b) for axial thrust bearings in terms of the axial load F_z divided by the active pole area of the runner disk between the inner diameter D_i and the outer diameter D_o. These approximations provide the designer and user with an idea of the design configuration of a magnetic-bearing-equipped pump. The

TABLE 2 Approximate magnetic bearing sizing relationships

Sizing Parameter	Heteropolar	Homopolar
Radial Bearings		
a) Dimensions, in multiples of the shaft diameter		
Air gap diameter, D	2	1.5
Stator outer diameter	4	3
Overall axial length	2	1.5
Active axial length of poles at air gap, L	0.9	0.8
b) Unit load capability, F_r/DL, lb/in^2 (MPa)	40 (0.28)*	60 (0.41)
Axial Thrust Bearings		
a) Diameter ratio D_i/D_o of active pole area	0.5	0.5
b) Unit load capability, $F_z / [\pi(D_o^2 - D_i^2)]$, lb/in^2 (MPa)	50 (0.34)*	70 (0.48)

*Higher with special lamination material low unit loads of magnetic bearings result in more space being needed for them in comparison to conventional bearings.

Theory of One-Dimensional Sizing To perform more accurate, in-depth sizing, the theory of both heteropolar and homopolar magnetic bearings is applied. Magnetic bearing sizing and geometry programs normally use simple one-dimensional magnetic circuit theory to obtain initial sizing and perform design iterations. This initial sizing is then followed up with design analysis using 2D or 3D magnetic FEA analysis to verify the design. The basis of the classic one-dimensional sizing for a magnetic bearing is discussed next, first for the heteropolar bearing and then for the homopolar bearing.

HETEROPOLAR BEARING

a. *Magnetic circuit*—The basic magnetic circuit equation, derived from Ampere's Loop Law, is

$$MMF = \Phi\mathcal{R} \tag{1}$$

where MMF = magnemotive force
Φ = magnetic flux
$\mathcal{R}$ = path reluctance

A sketch of one quadrant (one electromagnet) of a heteropolar bearing is shown in Figure 10.

Assuming the air gap area and path areas are equal, Eq. 1 becomes

$$2Ni = BA\left(\frac{g}{\mu_o A} + \frac{g}{\mu_o A} + \frac{l_{stat}}{\mu_o\mu_{r,\,stat}A} + \frac{l_{rot}}{\mu_o\mu_{r,\,rot}\mathrm{A}}\right) \tag{2}$$

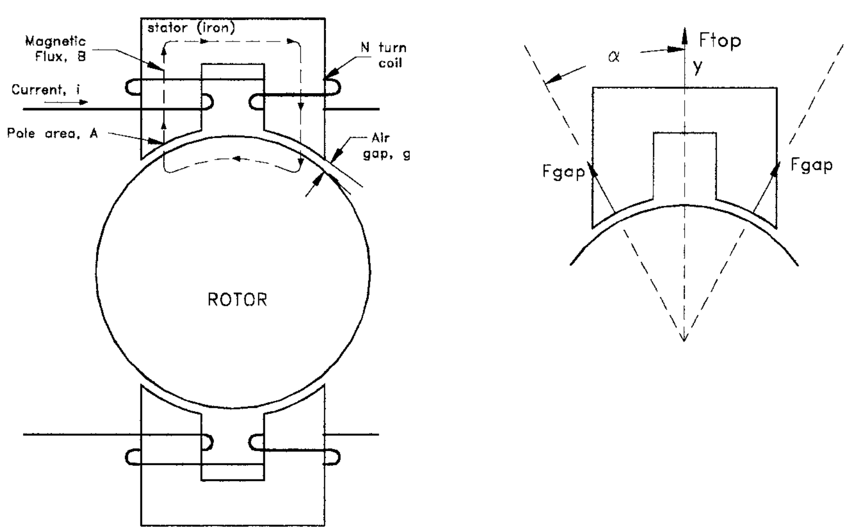

FIGURE 10 Simple electromagnetic circuit for one quadrant of a heteropolar magnetic bearing

Where N = number of turns per pole
i = coil current
B = magnetic flux density
A = pole area
g = air gap
μ_o = permeability of free space
μ_r = relative permeability
l = iron path length

If the iron permeability is high relative to the air gap ($\mu_{r,\,rot}, \mu_{r,\,stat} >> \mu_o$), the iron reluctance terms are insignificant and the following equation can be obtained for the flux:

$$B = \frac{Ni\mu_o}{g} \tag{3}$$

b. *Force calculation*—The basic force equation for an air gap is

$$F_{gap} = \frac{B^2A}{2\mu_o} \tag{4}$$

This equation assumes negligible leakage and fringing and that the flux density is uniform in the air gap. The combined vector force along the center of the bearing for the two air gaps of the top magnet is

$$F_{top} = 2F_{gap} cos\ \alpha \tag{5}$$

Substituting from Eq. 4 gives

$$F_{top} = \cos\alpha\,\frac{B^2A}{\mu_o} \tag{6}$$

If the saturation flux density, B_{sat}, of the iron material is used for B, Eq. 6 defines the load capacity as a function of pole area for a heteropolar magnetic bearing.

c. *Linearization of the force/current characteristic*—Substituting from Eq. 3 gives:

$$F_{top} = \cos\alpha\,\frac{A}{\mu_o}\left(\frac{Ni\mu_o}{g}\right)^2 = \cos\alpha\,AN^2\mu_o\,\frac{i^2}{g^2} = k_1\,\frac{i^2}{g^2} \tag{7}$$

$$k_1 = \cos\alpha\,AN^2\mu_o$$

Thus, the force in a given magnet is proportional to the square of the current, a result that makes the bearing more difficult to control. Additionally, a single electromagnet can only apply a force in one direction (an attractive force). For these two reasons, opposed electromagnets are used together with a bias current (or flux) in each coil.

The current relationship is

$$i = i_{bias} + i_{con}$$

to increase the force,

$$i = i_{bias} - i_{con}$$

to decrease the force.

The rotor may also be off-center in the air gap, described by

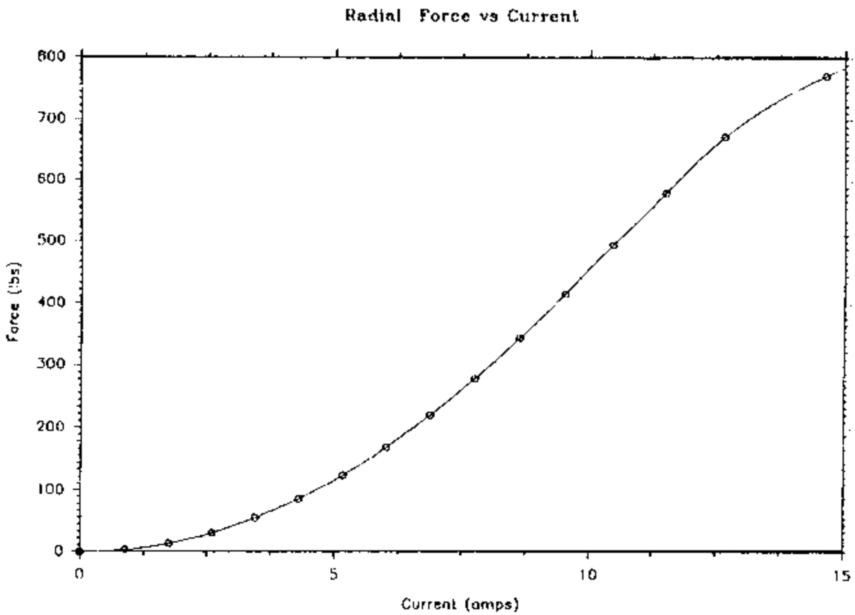

FIGURE 11 Force versus current for radial bearing of multistage pump

$$g = g_0 - y \qquad \text{top air gap}$$

$$g = g_0 + y \qquad \text{bottom air gap}$$

Applying Eq. 7 to both top and bottom electromagnets yields

$$F_y = F_{top} - F_{bot} = k_1\left[\left(\frac{i_{bias} + i_{con}}{g_0 - y}\right)^2_{top} - \left(\frac{i_{bias} - i_{con}}{g_0 + y}\right)^2_{bot}\right] \tag{8}$$

With the rotor centered ($y = 0$), this can be reduced to

$$F_y = 4\frac{k_1}{{g_0}^2} i_{bias} i_{con} \tag{9}$$

Thus the net bearing force is proportional to the control current. Figures 11 and 12 are examples of measured force versus current for radial and axial thrust bearings respectively. These measurements were made on the magnetic bearing of the multistage pump in Table 1 (refer to Figures 1 and 2).

d. *Force constant and negative stiffness*—Eq. 8 can be also be linearized for small motion about the center ($y << g_0$) by differentiating with respect to i_{con} and y, the two quantities in Eq. 8 that can change in normal operation of the bearing. The result is

$$F_y = -k_f\, i_{\,con} - k_n\, y \tag{10}$$

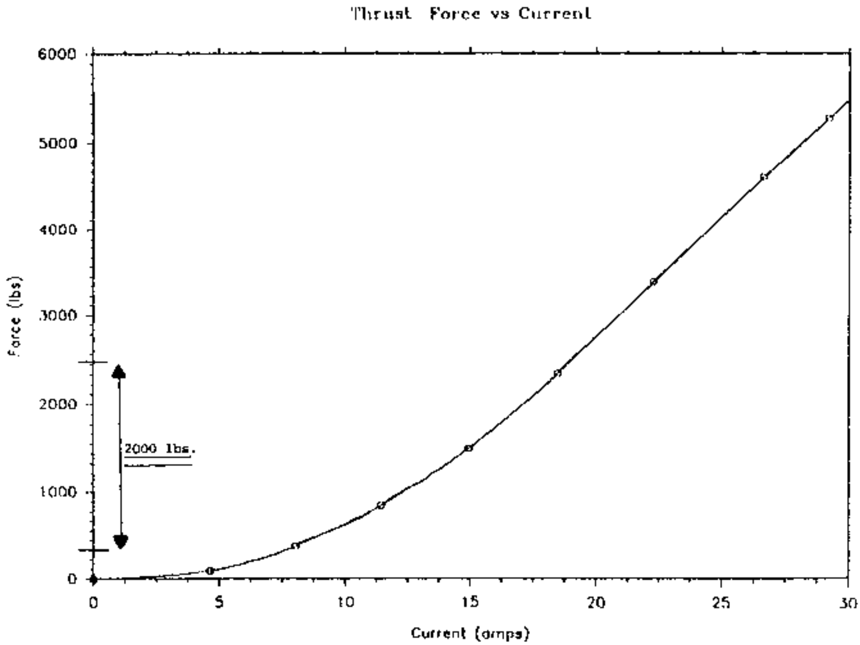

FIGURE 12 Force versus current for axial thrust bearing of multistage pump

where

$$k_f = \frac{-4k_1 i_{bias}}{g_0^2} = \text{force constant or current stiffness} \tag{11}$$

$$k_n = \frac{-4k_1 i_{bias}^2}{g_0^3} = \text{negative stiffness or position stiffness} \tag{12}$$

The position stiffness is the passive stiffness of the bearing with a bias field but with no control current. The position stiffness is always negative, indicating that if the rotor is displaced from center, it will be pulled further from its equilibrium position if no control current is applied. The force constant defines the relationship of the control force to current (lb/amp or N/amp) with the bearing centered. It is also negative, indicating that applying a control current pulls the rotor from its centered position. Many practitioners use positive values for the force constant and the position stiffness as a matter of convenience. In this case, the minus signs in Eq. 10 become plus signs.

The control current is determined by the measured displacement, y, and the characteristics of an adjustable sensor/compensator/amplifier transfer function:

$$G_{con}(s) = \frac{i_{con}}{y} \tag{13}$$

Substituting Eq. 11 into Eq. 10 gives

$$Fy = -k_F\, G_{con}(s)y - k_n\, y \tag{14}$$

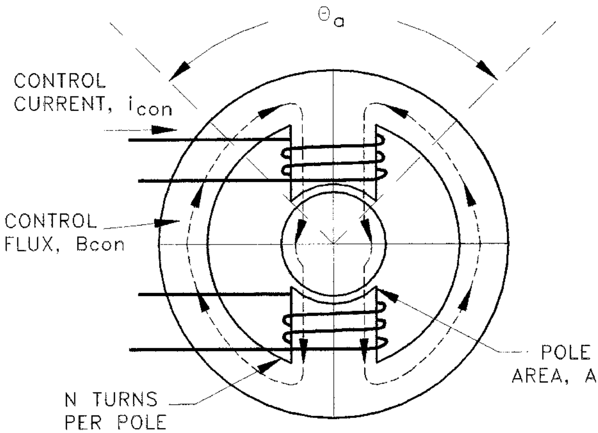

FIGURE 13 Control magnetic circuit for one axis of a homopolar magnetic bearing

The characteristics of the control transfer function $G_{con}(s)$ are determined in order to stabilize the rotor/bearing system.

HOMOPOLAR BEARING In the homopolar bearing, a bias flux is used for linearization just as in the heteropolar bearing; however, the bias flux and control flux follow different paths. Additionally, the bias flux can be generated by either a permanent magnet (most common) or an electromagnetic coil. The use of a permanent magnet for bias reduces power consumption and makes the bearing more linear at large position offsets.

a. *Magnetic control circuit*—The control circuit for one half of the homopolar bearing, shown in Figure 13, is across the top air gap, through the rotor, out the bottom air gap, and around the back iron.

When Eq. 1 is applied to this circuit, the result is similar to Eq. 2:

$$2Ni_{con} = B_{con}A\left(\frac{g}{\mu_o A} + \frac{g}{\mu_o A} + \frac{l_{stat}}{\mu_o \mu_{r,stat} A} + \frac{l_{rot}}{\mu_o \mu_{r,rot} A}\right) \tag{15}$$

Again, if the iron permeability is high relative to the air gap, this can be reduced to the following:

$$B_{con} = \frac{Ni_{con}\mu_o}{g} \tag{16}$$

b. *Magnetic bias circuit*—The magnetic circuit equation for the bias circuit, shown in Figure 14, is

$$\frac{B_r l_m}{\mu_o \mu_{r,\,mag}} = B_{bias}A\left(\frac{g}{\mu_o A} + \frac{g}{\mu_o A} + \frac{l_m}{\mu_o \mu_{r,\,mag} A_m}\right) \tag{17}$$

Where B_r = residual induction of the magnet
l_m = axial length of the magnet
$\mu_{r,\,mag}$ = relative permeability of the magnet (≅1.0–1.05)
A_m = magnet cross-sectional area per quadrant

With the magnet permeability assumed to be 1.0, this can be reduced to

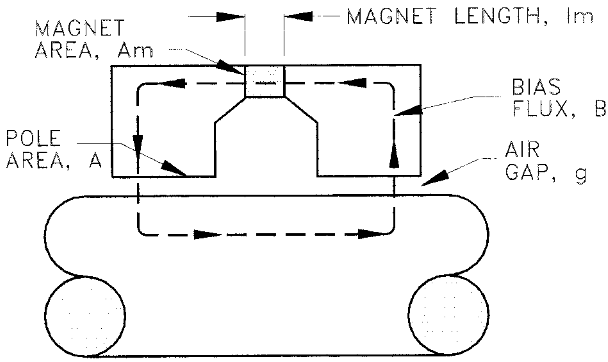

FIGURE 14 Bias magnetic circuit for one quadrant of a homopolar magnetic bearing

$$B_{bias} = \frac{B_r}{\dfrac{2g}{l_m} + \dfrac{A}{A_m}} \tag{18}$$

c. *Force calculation*—The basic force equation for an air gap is given by Eq. 4. In the homopolar bearing, the pole face covers a broader arc of the rotor than in the heteropolar bearing; therefore, the air gap force must be integrated over the surface to obtain the desired vector force, F_{top}. Both the front and back control stacks must also be included. The result is

$$F_{top} = 2C_g F_{gap}$$

Where:

$$C_g = \frac{2\sin(\theta_{a/2})}{\theta_a} \tag{19}$$

This result assumes negligible leakage and fringing and that the flux density is uniform in the air gap. Substituting from Eq. 4 into Eq. 19 gives

$$F_{top} = C_g \frac{B^2 A}{\mu_o} \tag{20}$$

If the saturation flux density, B_{sat}, of the iron material is used for B, Eq. 20 defines the load capacity as a function of pole area for a magnetic bearing.

d. *Linearization of the force/current characteristic*—The air gap flux density in the homopolar bearing is the superposition of the bias and control flux. The control coils and permanent magnet polarity are arranged such that when the control flux adds to the bias in the top air gaps, the control subtracts from the bias in the bottom air gaps. Thus the net vertical force is

$$F_y = F_{top} - F_{bot} = \frac{C_g A}{\mu_o}\left[(B_{bias} + B_{con})^2_{top} - (B_{bias} - B_{con})^2_{bot}\right] \tag{21}$$

This can be reduced in a similar manner as before to produce

$$F_y = \frac{4C_g A}{\mu_o} B_{bias} B_{con} \tag{22}$$

Substituting from Eq. 16

$$F_y = \frac{4C_g AN}{g} B_{bias} i_{con} \tag{23}$$

Thus the control force is proportional to the control current as desired.

e. *Force constant and negative stiffness*—The expression given in Eq. 10 for the heteropolar bearing also applies to the homopolar bearing:

$$Fy = -k_f i_{con} - k_n y \tag{24}$$

The definition of the force constant is

$$k_f = \frac{-4C_g ANB_{bias}}{g_0} = \text{force constant or current stiffness} \tag{25}$$

The expression for negative stiffness is not easily reducible to analytical form due to the complexity of the bias circuit. However, the existence of the permanent magnet in the bias flux path as a fixed and large reluctance improves the linearity of this bearing for off-center operation. Eqs. 13 and 14 for the heteropolar bearing apply to the homopolar bearing as well.

INSTALLATION AND TUNING

Mechanical Installation Magnetic bearings have the advantage that they can be set, by adjusting offsets in the controller, to center the rotor on the magnetic bearing stator, the catcher bearing, the seals, or any other mechanical reference in the pump. The rotor can even be offset vertically to cancel the effect of gravity, thus reducing static power requirements to near zero. Careful consideration is needed during design to decide which is the best approach.

The position sensors in the bearings can also be used to measure the clearances between the rotor and any physical stops such as a sealing ring, without disassembly.

Tuning The rotor dynamics analysis performed during the early design stage will be used to determine the initial controller compensation, which will have a transfer function matched to the pump requirements. During initial testing, this transfer function will have to be adjusted to match the as-built dynamics of the rotor and support structure.

Normally the rotor will be accurately modeled and little change will be needed. If there are shrink fits or bolted joints, there may be some stiffness variation from the theoretical model. This may require on-site controller compensation adjustment. However, the stator is often a complex structure, and adjustments may be needed to avoid the excitation of stator modes.

DIAGNOSTICS AND USER INTERFACE

Diagnostic Capabilities The controller, in order to function, must analyze a continuous stream of information on the shaft location in each of the five control axes, two for each radial bearing and one for the thrust bearing. This information can be accessed for external diagnostic use, as can the corresponding information on bearing current, from which can be inferred the bearing load. This diagnostic information can be a very useful source of information on the health of the pump, its mechanical components, and on the system it is operating in.

Interface Requirements The magnetic bearing system controller can also be interfaced with the plant control system with the following type of logic:

- No drive unit start without levitation
- No delevitation at speed
- Rotor offset and bearing load alarms
- Rotor offset and bearing load driver trips, with possible time delay

RELIABILITY AND MAINTENANCE

Bearing Cartridges As explained earlier, the reliability of the bearing stator and rotor components should be such as to provide lifetime service.

Controller The main life limiting component in the controller is likely to be the amplifier. In a redundant system, online replacement is possible without loss of levitation. In a non-redundant system, a preventative maintenance approach should be used for this component.

OPERATING EXPERIENCE

Multistage Boiler Feed Pump The multistage pump of Table 1 (refer to Figure 1) was installed as one of three otherwise identical pumps (two in parallel, one standby) in an electric utility generating plant (refer to Figure 2). The objective was to show that magnetic bearings would work in a typical field application of a pump of significant power level. This project took the first step of replacing the conventional bearings in this 610 hp (0.46 MW) eight stage centrifugal pump, which were outboard of the pump itself, and replacing them with heteropolar active magnetic bearings without any major design changes[6]. This was seen as the first of two steps, the second being a project where the bearings would be submerged in the operating fluid, allowing one seal system to be replaced[8].

As with many magnetic bearing projects, the main lesson learned was that the transient bearing loads could not always be predicted ahead of time, and that the magnetic bearings gave very precise and important feedback of this information. Table 3 contains the design and field data. In this case, the 2500 lb (11 kN) transient load occurred when the plant underwent a suction pressure transient in which the available *NPSH* became so low that the first stage of this horizontally-opposed staging configuration (refer to Figure 1) apparently lost pressure rise completely[6]. The axial thrust of this stage was accordingly lost, destroying the intended axial hydrodynamic thrust balance of the pump. (See Sections 2.1 and 2.2.1.) Nevertheless, the conservative design of the thrust bearing enabled it to accommodate this load.

Single-Stage Process Pump The 800 hp (0.6 MW) single-stage double-suction process pump of size 8 $\times$ 26 (200mm $\times$ 660mm) of Table 1 was retrofitted with homopolar bearings[7].

Closed-loop testing was conducted in the pump manufacturer's facility (see Figure 15). As indicated in Table 4, the results showed that a substantial operating margin exists for

TABLE 3 Operating experience with multistage pump

	Expected Load, lb (kN)		Design Load, lb (kN)	Actual Load, lb (kN)	
Bearing	Steady	Transient		Steady	Transient
Radial	280 (1.2)	560 (2.5)	800 (3.6)	280 (1.2)	580 (2.6)
Axial	1,000 (4.4)	2,000 (8.9)	4,000 (17.8)	1,100 (4.9)	2,500 (11.1)

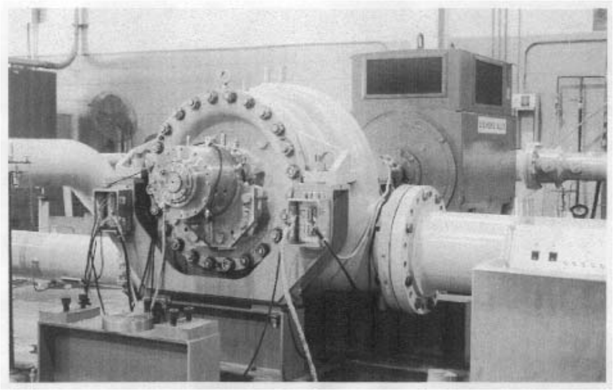

FIGURE 15 Single-stage double-suction 10 × 26-size process pump [800 hp (0.6 MW)] with magnetic bearings. (Reference 7)

TABLE 4 Design and experimental loads for the single-stage pump

Bearing	Expected Load, lb, (kN)	Design Load, lb (kN)	Actual Load, lb (kN)
Radial	953 (4.2)	1,430 (6.4)	1,450 (6.4)
Axial	1,000 (4.4)	4,000 (17.8)	2,100 (9.3)

the axial thrust bearing. Greater design capacity would provide the same margin with respect to radial loads.

The margins evident in Table 3 for the multistage pump may appear excessive, but until more operating knowledge about such pumps is acquired, it would appear that this degree of conservatism in designing magnetic bearings for pumps is merited.

COSTS

Backed by the vision of complete magnetic suspension and the attendant benefits, magnetic bearing technology has been proven and demonstrated in pumping machinery. However, the major deterrent to further application of this technology is cost. Retrofitting the magnetic bearings to the multistage pump described above cost at least twice the price of the pump itself, and this included the analog controller, which accounted for about a third of the retrofit cost. Digital controllers, a later development, are one third the size and cost of analog controllers; this has significantly reduced the overall cost. Much of this overall cost is in the engineering of the magnetic bearings, which includes matching this system to the rotordynamic characteristics of the pump. Use of the same system in quantity production would reduce the cost by up to 80 percent.

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