

MAGNETIC BEARINGS—A PRIMER

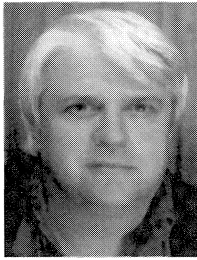
by

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Paul Alves is a Senior Engineer with Nova Chemicals Ltd., in Calgary, Alberta, Canada. His experience spans more than 23 years in the areas of petrochemical industry equipment design, vibrations, and rotating equipment consulting. Since joining Nova in 1990, Mr. Alves has been providing technical support to Nova's extensive fleet of magnetic bearing compressors.

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ABSTRACT

Magnetic bearings have reached the point in their development where they can be considered a mature technology, and as such, should gain wide acceptance by the rotating equipment industry. While current industrial installations have proven durable and reliable, we have just begun to peer into the world of possibilities for advanced applications of this technology to rotating equipment.

Magnetic bearings were developed from the confluence of various disciplines such as physics, mechanical engineering design, feedback controls, electronics, and materials science. It is difficult for the industrial engineer to grasp the entire breadth of the technology to the point where he/she will be comfortable making decisions regarding equipment that include magnetic bearings. The author of this tutorial hopes to promote a greater understanding of this technology.

INTRODUCTION

General

Magnetic bearings can provide immense flexibility to the design of rotating equipment if their potential is well understood. This tutorial presents an overall picture of the technology, highlighting some of the more fundamental building blocks, and leading to an understanding of the general function and operation of the bearings. The focus is on the philosophical understanding of function and system, with emphasis on the more physical and practical aspects of the application.

A section is dedicated to the current state of development, including information from the manufacturers in the newest applications of magnetic bearings. It provides an updated review of users' experiences over the past few years, including up-to-date reliability and availability figures for the current fleet in service worldwide.

Issues referring to the specification and performance tests, are covered at the end of the tutorial. It also presents a review of today's research topics from several universities. Figures 1, 2, and 3 should provide a general overview of the number of magnetic bearing units in operation and their relative sizes.

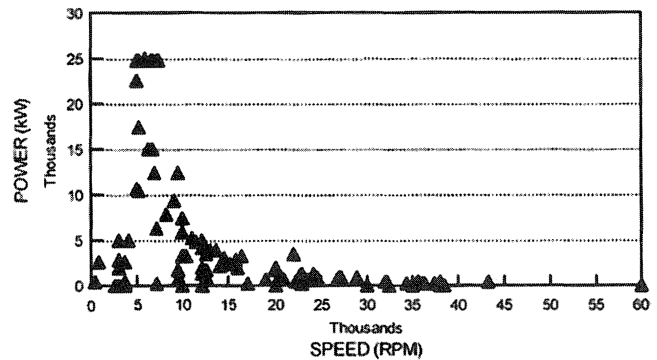


Figure 1. All Turbomachinery by Power. (Courtesy of EPRI)

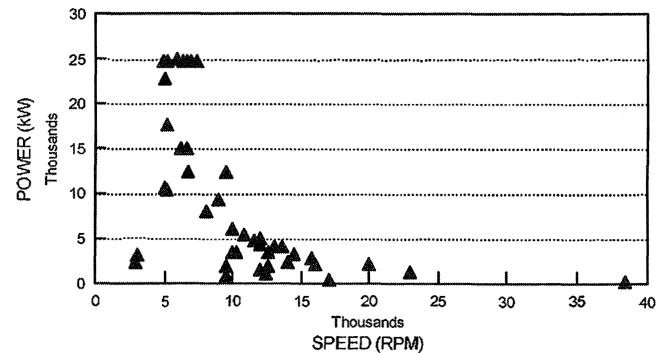


Figure 2. Compressors by Power. (Courtesy of EPRI)

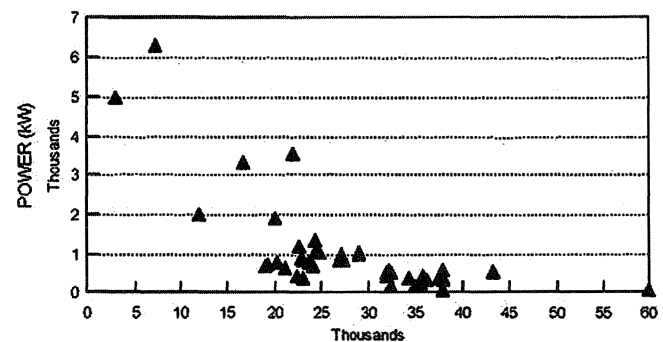


Figure 3. Turboexpanders by Power. (Courtesy of EPRI)

Advantages/Disadvantages of Magnetic Bearings

It seems opportune at this point to mention some of the most important advantages and disadvantages of magnetic bearings as applied to turbomachinery.

Advantages

- One to two percent increase in overall mechanical efficiency of equipment due to reduced friction losses

- Operation of rotating equipment at higher speeds. High DN numbers, up to four million, as compared with two million for high speed rolling element bearings.
- Immersion of the bearings in the process fluid, with the possibility of eliminating seals
- Reduced cost of installation with the elimination of oil
- Equipment can be mounted at floor level.
- Great reduction in required space on the plant floor
- Greater safety of operation with the elimination of lubrication oil
- Use of creative mechanical arrangements with unrestricted number of bearings and bearing locations. This feature will probably be most appealing to manufacturers, since it opens the door to machines rotating at great speeds and with consequently higher efficiencies.
- Manipulation of rotordynamic characteristics of the system
- Creative control schemes with optimized performance, promoting resilience to imbalance and transient forces
- One shot field balancing. The mass elastic characteristics of the rotor and the displacements at more than two locations along the rotor provide all the information necessary to field balance without trial weights and to perform modal balance of flexible shafts (in development).

Disadvantages

- Low pressure capacity. Due to its lesser pressure capacity as compared with oil bearings, magnetic bearings have to be made larger, taking up more space, leading to slightly wider bearing spans.
- The stiffness of magnetic bearings is, in general, considerably lower than equivalent oil bearings.
- Bearings require cooling flow of some kind to remove heat generated in the bearings. Improvements in the technology may soon lead to totally enclosed and sealed bearing cartridges. This cooling, however, is minimal compared with the cooling required for the oil in conventional bearings.
- Machine internal clearances have to be greater than auxiliary bearing gap. This does not constitute a problem for most rotating equipment designs.
- Certain arrangements of bearing laminated rotors require field balancing.
- The cost premium over oil bearings is usually mentioned as a concern, but bearing manufacturers today claim cost advantage over oil bearing installations, including overall initial investment.

General Information

The sizes of magnetic bearings produced to date vary, from tiny bearings measuring 25.4 mm in diameter with capacity of 150 N of force, and up to very large bearings measuring 1,300 mm in diameter capable of carrying 250,000 N. The speeds of operation extend far, with some turbomolecular pumps running in excess of 100,000 rpm. An ultra centrifuge prototype reaches 800,000 rpm. Grinding spindles usually operate above 150,000 rpm. One of the most important parameters determining the size of bearings for a given machine is the pressure capability of the bearing. The maximum static force generated by the bearing is given by the projected area of the bearing, times the maximum pressure provided. Magnetic bearings are generally rated at 200 KPa (30 psi) compared with oil bearings rated at 500 KPa (75 psi), which tend to make magnetic bearings larger, taking up more room than equivalent oil bearings.

Mechanical Configurations

Magnetic bearings have been applied to a wide variety of rotating equipment including compressors, steam and gas turbines, pumps, fans, and centrifuges. A great number of the early applications have been "conversions" from oil bearing operation to run with magnetic bearings. It is expected in the near future, given a wider acceptance of the technology, that rotating equipment will be designed to take full advantage of the versatility of magnetic bearings. Very complex and creative arrangements are to be expected in turbomachinery design with consequent improvements in performance, weight and size reductions, and operational reliability.

Figures 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14, and 15 show some of the more common mechanical arrangements of rotating equipment with magnetic bearings. Also included are some of the more recent advanced applications. Turboexpanders are one of the most numerous applications of magnetic bearings, which have become the standard of this industry. Figure 4 shows the compact bearing arrangement for a turboexpander. Figure 5 shows an integral motor and compressor arrangement for pipeline applications. This compressor uses no mechanical seals.

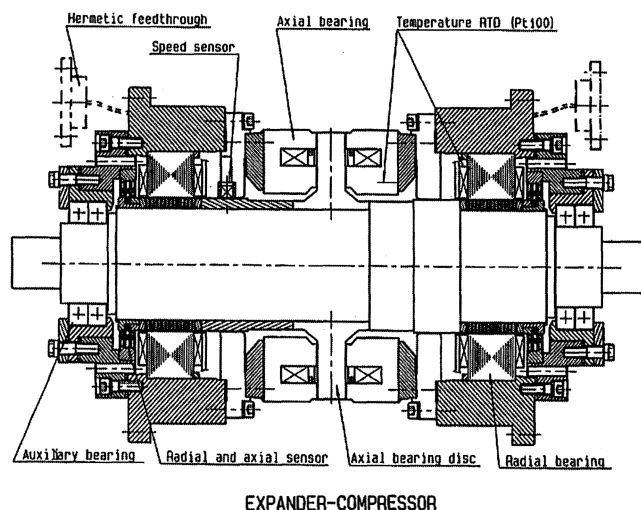


Figure 4. Turboexpander Magnetic Bearing Arrangement. (Courtesy of S2MUSA Magnetic Bearings)

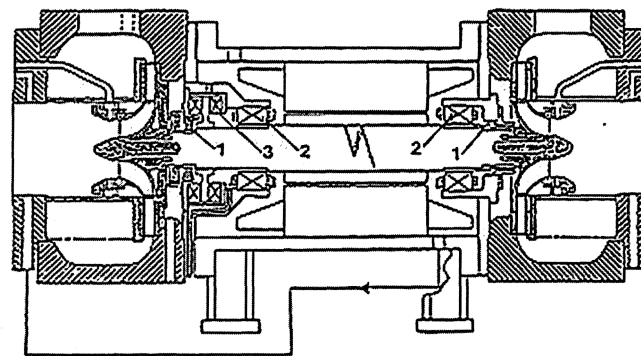


Figure 5. The Mopico Pipeline Compressor. (Courtesy of EPRI)

An overhang pipeline gas compressor is shown in Figure 6. Compressors of this kind produce very high thrust forces when pressurized, constituting serious problems during startup when operating with oil bearings. The magnetic axial bearing design has to account for these forces. Gas power turbine arrangement is shown in Figures 7 and 8. This is the first power turbine to run on magnetic bearings, and it served well to prove the concept. It operated for several years before being decommissioned for operational reasons. Figure 9 shows a gas turbine generator used in aircraft.

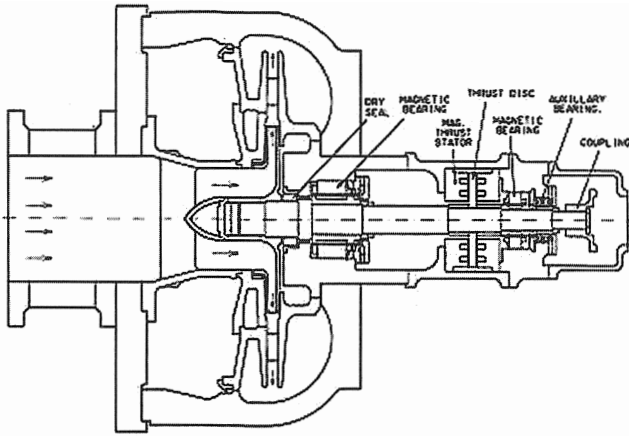


Figure 6. Overhung Pipeline Compressor. (Courtesy of Revolve Technologies)

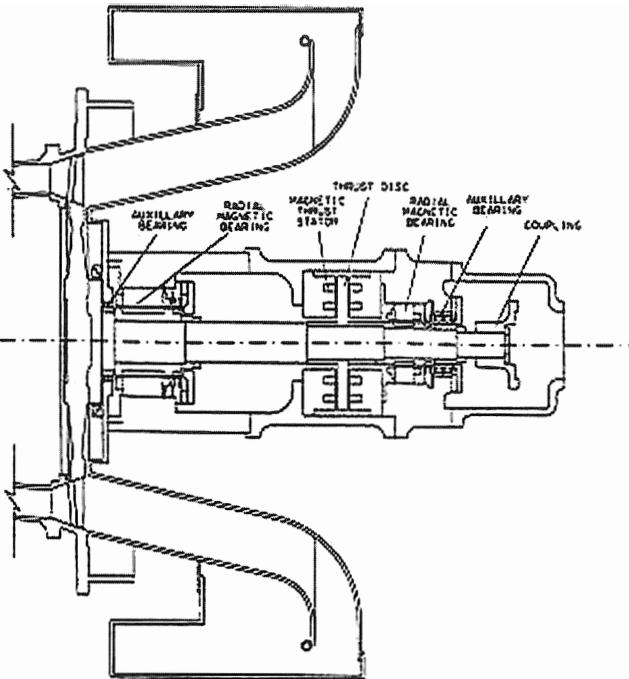


Figure 7. Power Turbine Arrangement. (Courtesy of Revolve Technologies)

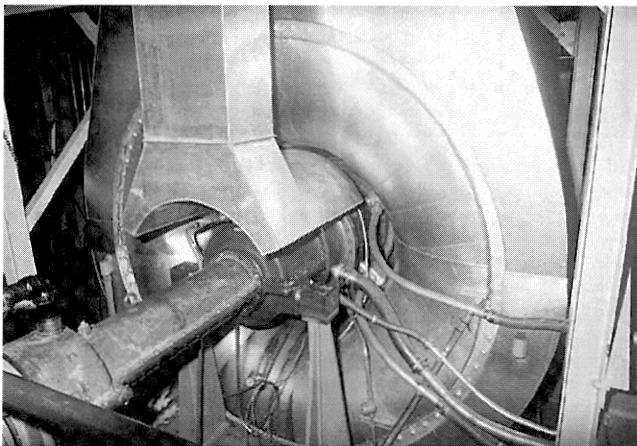


Figure 8. Power Turbine Photo. (Courtesy of Revolve Technologies)

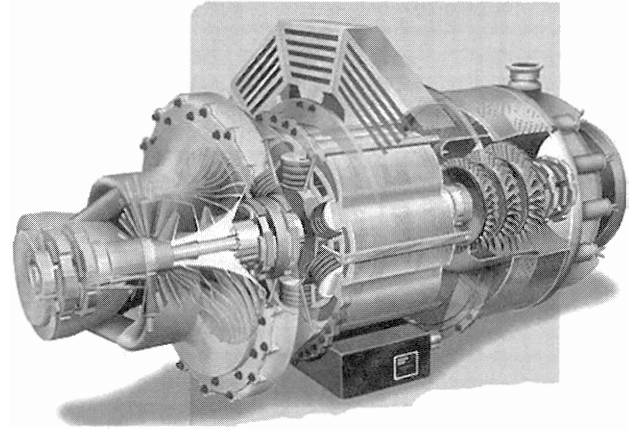


Figure 9. Gas Turbine on Magnetic Bearings. (Courtesy of Avcon Inc.)

The diagram in Figure 10 and photo in Figure 11 show a boiler feedwater pump fitted with magnetic bearings.

8 STAGE BOILER FEED PUMP WITH MAGNETIC BEARINGS

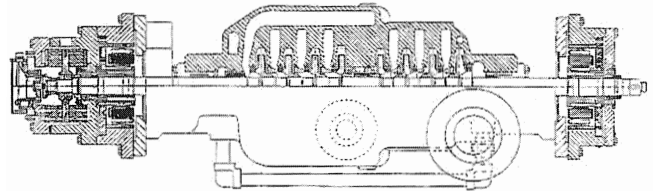


Figure 10. Boiler Feed Pump Arrangement. (Courtesy of EPRI)

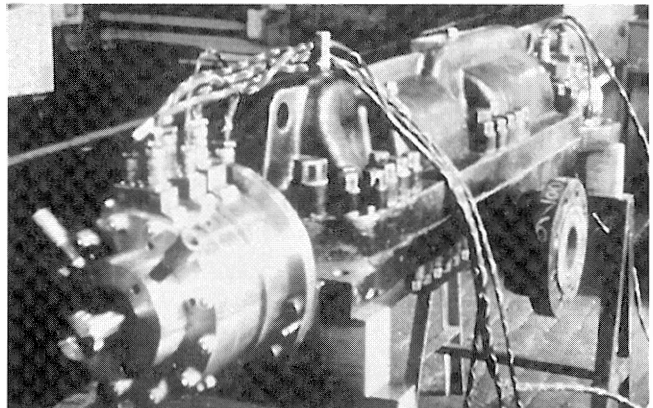


Figure 11. Boiler Feed Pump Photo. (Courtesy of Revolve Technologies)

The shafts in the beam style gas pipeline compressors can weigh in excess of 1300 kg and rotate from 5,000 to 7,000 rpm. There are many units like the one shown in Figures 12 and 13, which have been in service for many years.

Figure 14 shows a magnetic bearing simulator rotor used for demonstration and training. This simulator was designed by the author, utilizing parts from a decommissioned compressor. It has also been used for the training of technical personnel in field balancing techniques.

Figure 15 is a picture of a compressor train powered by electric motor. It consists of a multistage compressor casing and the exciter unit coupled to the motor occupying the center position. This unit is currently in fabrication.

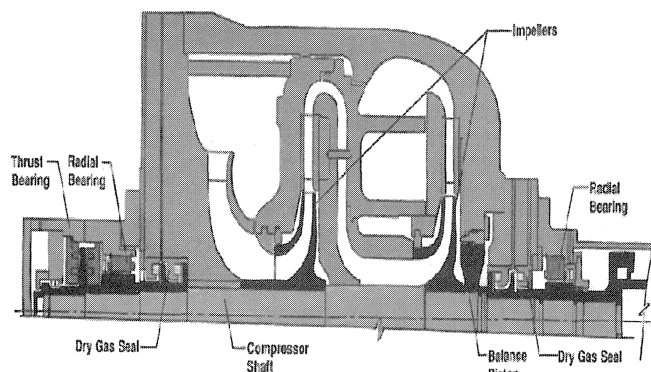


Figure 12. Pipeline Booster. (Courtesy of Revolve Technologies)

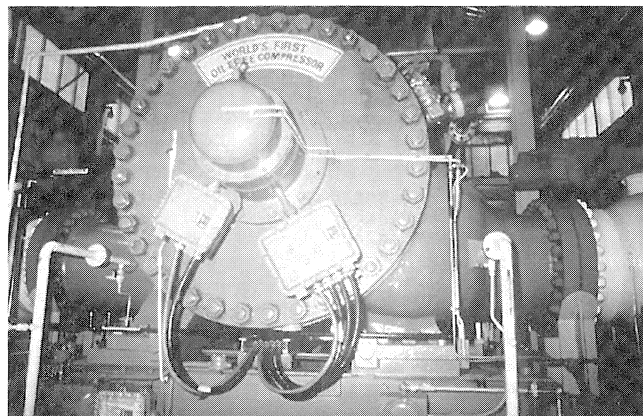


Figure 13. Pipeline Booster Photo. (Courtesy of Revolve Technologies)

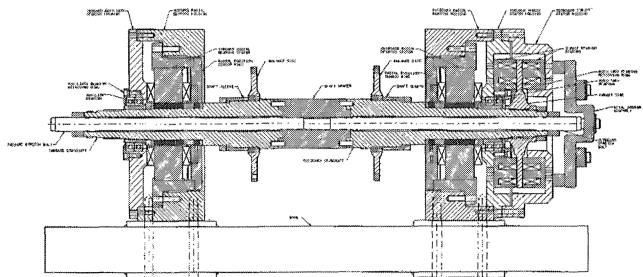


Figure 14. Magnetic Bearing Simulator.

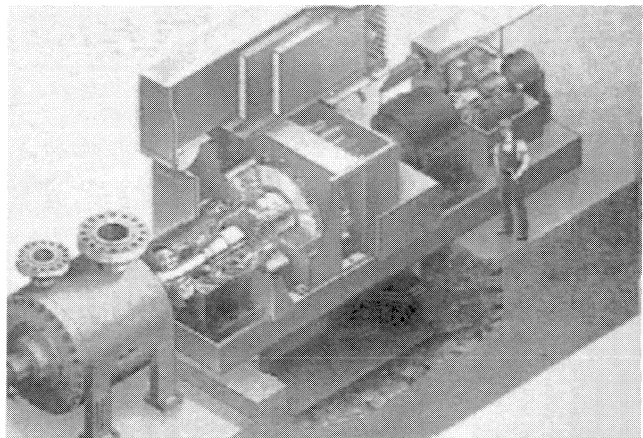


Figure 15. Motor Driven Compressor. (Courtesy of Glacier Magnetic Bearings)

Overview of the System

The magnetic bearing dynamic system is made up of the shaft, the magnetic bearing actuators, the position sensors, the filters, the control unit, and the amplifiers. Figure 16 shows a schematic of the complete dynamic system. Note that in most common configurations for a beam style machine with two bearings, there are five axes of control, four radial axes and one thrust axis. In most systems, the weight of the shaft at each bearing location is carried by two bearing axes, oriented at 45 degrees with the vertical. In this arrangement, the bearings are supporting less weight and can be made smaller. Figure 17 shows a typical five-axis arrangement.

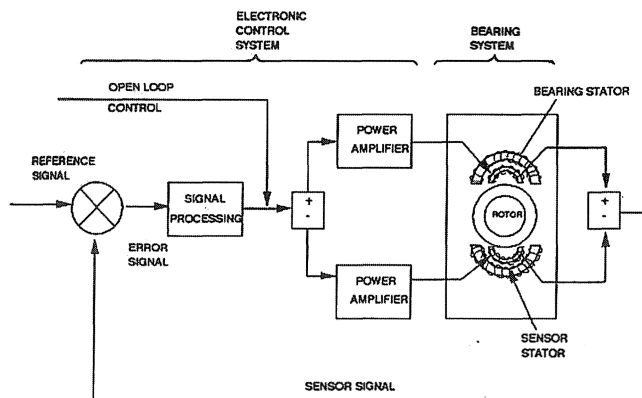


Figure 16. Magnetic Bearing System. (Courtesy of Revolve Technologies)

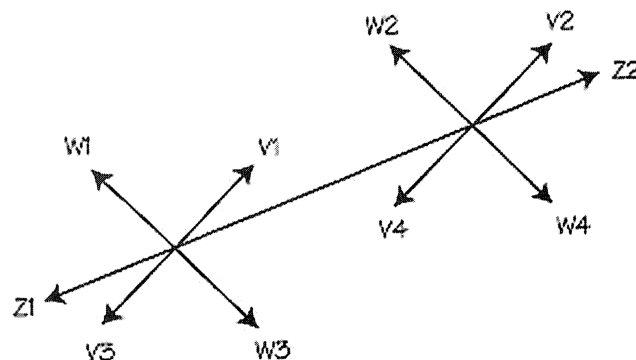


Figure 17. Magnetic Bearing Axis Arrangement. (Courtesy of Revolve Technologies)

Auxiliary Bearings

The auxiliary bearings are an essential part of most machine configurations using magnetic bearings. These are mechanical bearings, usually either rolling element bearings or self-lubricated bushings, that are used to hold the rotor when delevitated and avoid contact between magnetic bearing rotor and stator. The clearance between the rotor and the auxiliary bearings is usually half the gap between the rotor and the stator in the magnetic bearing. Figure 18 shows this general arrangement of clearances.

The auxiliary bearings are also the first line of defense in case of trouble with the magnetic bearing system or if unusual forces coming from the process exceed the capacity of the bearings. Usually some form of damping is provided in the auxiliary bearing connection to the housing. It is important to notice that the shaft, when supported on the auxiliary bearings, constitutes a very different dynamic system from that of the shaft when it is supported by the magnetic bearings. Rotordynamic analysis to determine the behavior of both systems is mandatory in the design of a new machine. Figure 19 shows common auxiliary bearing arrangements.

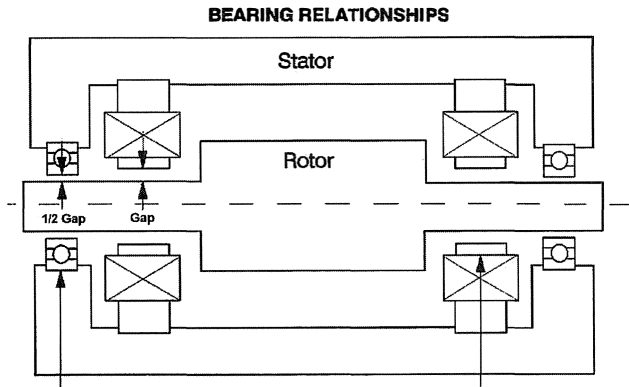


Figure 18. Auxiliary and Magnetic Bearing Relative Clearances. (Courtesy of EPRI)

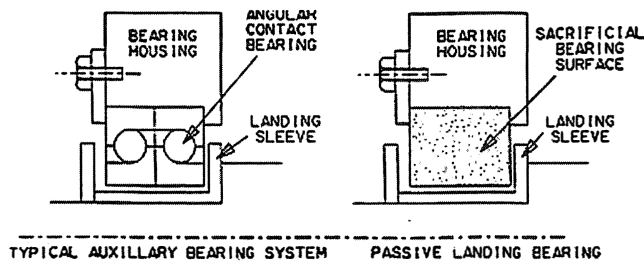


Figure 19. Common Auxiliary Bearing Arrangements. (Courtesy of Revolve Technologies)

MAGNETISM

Actuators

The most fundamental building block of a magnetic bearing system is the magnetic actuator, which produces both the static and dynamic forces necessary to support the shaft. Magnetic bearing actuators can be made of electromagnets (active) or permanent magnets (passive), or a combination of both. A support system where all bearings use only passive magnets is not realizable. Typically, the actuator consists of pairs of magnets working in opposition and acting on the bearing rotor mounted on the shaft. The actuator may have an odd number of magnets around the shaft. It should be mentioned that for the most common arrangements using attracting electromagnets, each bearing is inherently unstable, requiring the use of an active controller for stable operation. Each magnet will consist of arrangements of coils forming pairs of poles.

The magnetic bearing rotor is either mounted on the shaft or, alternatively, is made an integral part of the shaft. The radial magnetic bearing rotors are, in general, made up of ferromagnetic material in the form of laminations mounted on a bushing. Some homopolar bearing designs may act directly on the shaft without the need for laminations. Usually, but not necessarily always, each bearing will consist of two pairs of magnets or two axes. Each actuator or pair of magnets acts as a unit, subject to the actions of a controller. In some arrangements, each axis operates independently, controlling the position of the rotor in the axis direction. In others, combinations of axes, either on the same bearing location or coupled to bearing(s) on the opposite side of the shaft, are subject to multiple controllers. These arrangements promote coupling of the control axis and will be reviewed under the section CONTROLS. Figures 20 and 21 show some typical magnetic bearing designs. An alternative design is the homopolar bearing shown in Figure 22.

Forces/Saturation/Materials

The working force of the magnetic actuator is governed by physical laws and depends primarily on the following parameters:

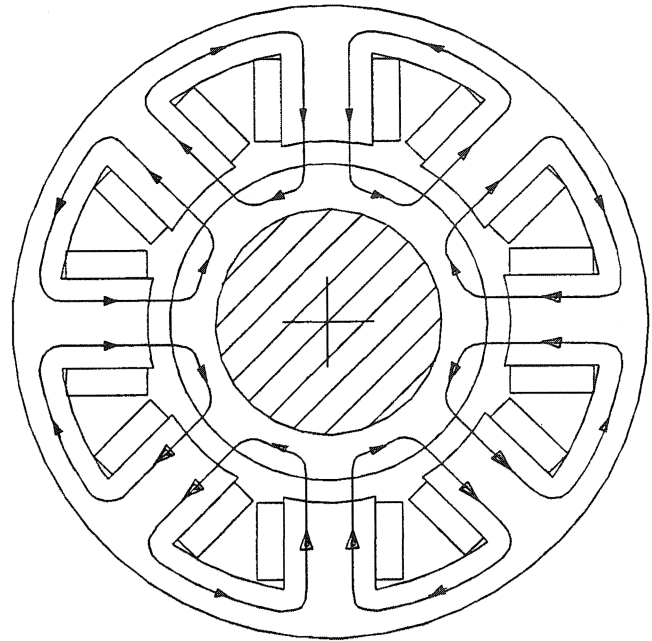


Figure 20. Magnetic Path, Radial Bearing. (Courtesy of S2MUSA Magnetic Bearings)

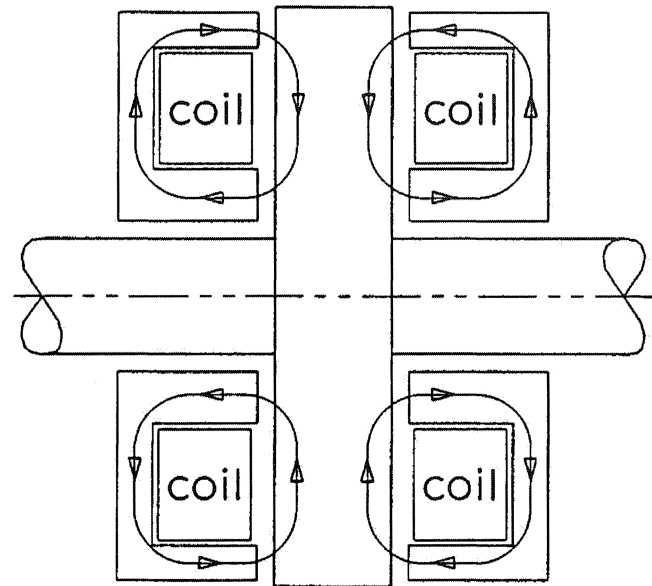


Figure 21. Magnetic Path, Thrust Bearing. (Courtesy of S2MUSA Magnetic Bearings)

- **Material of the magnets**—Each stator and the portion of the rotor immediately in front of it constitute a magnetic circuit. The best magnetic materials will be magnetized to a higher degree and, therefore, will produce more force for the same coil size and current. The limit of magnetization for a material is called the saturation limit. Figure 23 shows a curve of characteristic saturation for various materials. The saturation limit of present technology materials is approximately two tesla. It should be kept in mind that the magnetic material utilized in the rotor also has to be mechanically strong to withstand both the centrifugal forces and the controlling forces of the bearing when in operation.

- **The saturation of the magnet** is the physical limit that defines the static capacity of the bearing. For most common designs, the maximum current corresponding to the saturation of the iron is

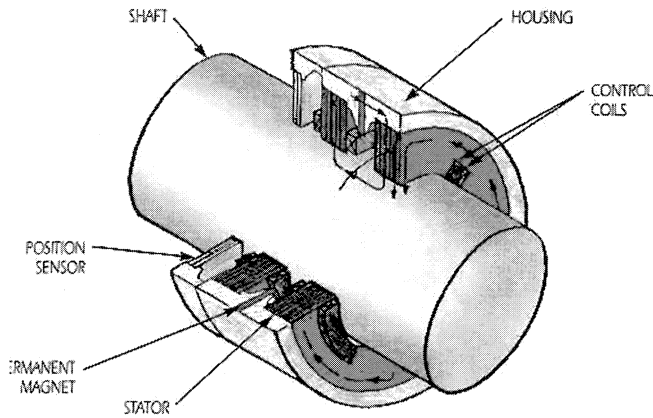


Figure 22. Homopolar Bearing. (Courtesy of Avcon Inc.)

generally the limit current for operation of the bearing. This saturation limit current is generally set to the upper "knee" portion of the saturation curve. Note that some other designs being investigated use currents leading to much higher saturation levels in the iron. It can be seen that the saturation curves do not really flatten out beyond the upper "knee," but continue to increase, although at a much lesser rate.

- The gap between stator and rotor is critical in the design of magnetic bearings, because the force produced by the magnet is in proportion to the square of the gap. Very small gaps are desirable from the point of view of generating high forces, but the mechanical assembly clearances and the type of operation of the units may limit the design. Gaps are optimized in the design phase of the bearings and measure, in general, between 0.127 mm to 0.635 mm (five thousandths and 25 thousandths of an inch).

DC Magnetization Curves for Commonly Used Magnetic Materials

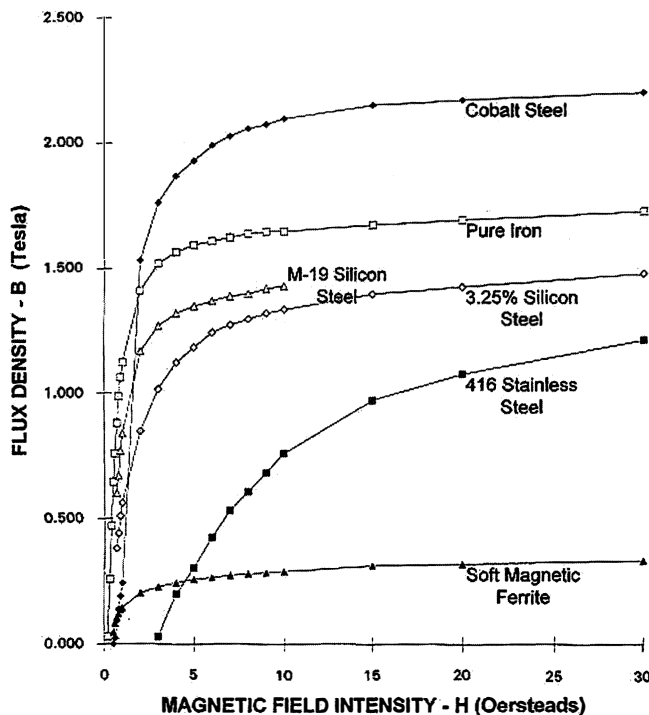


Figure 23. Saturation Curves for Various Materials.

The main parameters in the design of a magnetic actuator are:

- The static and dynamic forces present in the system.
- Bearing diameter and length.
- Bearing materials of rotor and stator.
- The size of the coils, supply voltage, and allowable currents.
- Bearing gap between stator and rotor.

In final analysis, one wants to optimize bearing pressure capacity with the use of materials with higher levels of saturation in order to reduce size, which at the same time leads to reduced windage and electrical losses. Equations (1) and (2) illustrate the relationship between force, current, and gap. Force is proportional to the square of the current, and inversely proportional to the square of the gap.

$$B = (\mu_0 \times N \times I) \div (2 \times g) \quad (1)$$

$$F = (B^2 \times S) \div (2 \times \mu_0) \quad (2)$$

or

$$F = k \times I^2 / g^2$$

Figure 24 shows the force measured against current for an existing pipeline compressor bearing. Figure 25 shows a radial magnetic bearing transversal cross section. Note the arrangement of poles designed to maximize the effective actuator area. Figure 26 is a picture of a bearing stator, 304.8 mm in diameter that produces 88,000 N of static force, to be used in a large process compressor. The bearing cartridges and rotor shown in Figure 27 are used in turboexpanders. Figure 28 shows a homopolar magnetic bearing. Notice the different orientation of the coil windings.

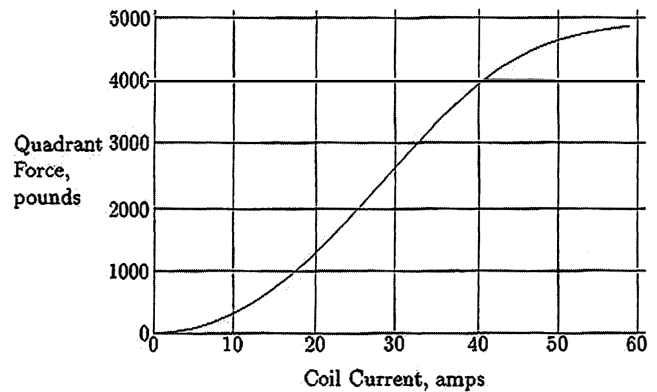


Figure 24. Results of Actuator Force Test.

Losses

Magnetic bearings are remarkable for the small amount of energy they use in operation. This small amount of energy consumed in the bearing is due to four major sources.

Eddy currents generated in the rotor and in the stator are due to the induction effect. Every time the magnetic field changes, eddy currents are generated in the material of the rotor and stator. Also, as the rotor rotates, any point in the rotor traverses many poles each revolution, seeing many reversals in polarity, hence the need for laminated rotors. In the case of homopolar bearings, any point in the shaft does not see polarity reversals but variations in field strength. These eddy currents generate loss in the form of heat in the bearing.

Resistance losses from the electric current flowing in the stator wires, which produce heat in proportion to the resistance of the wire, times the square of the current. Hysteresis losses are due to

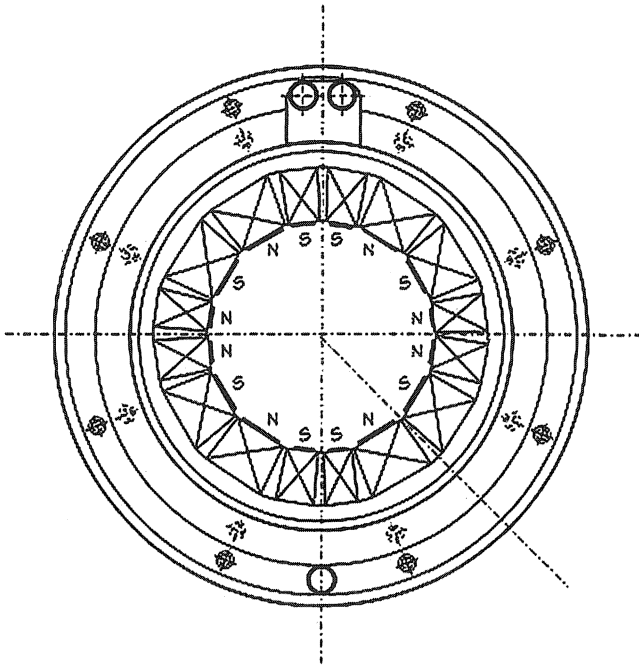


Figure 25. Radial Bearing Arrangement. (Courtesy of S2MUSA Magnetic Bearings)

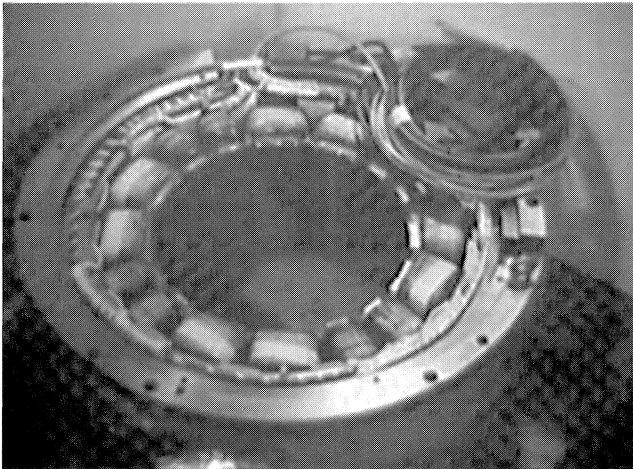


Figure 26. Large Bearing Stator. (Courtesy of Glacier Magnetic Bearings)

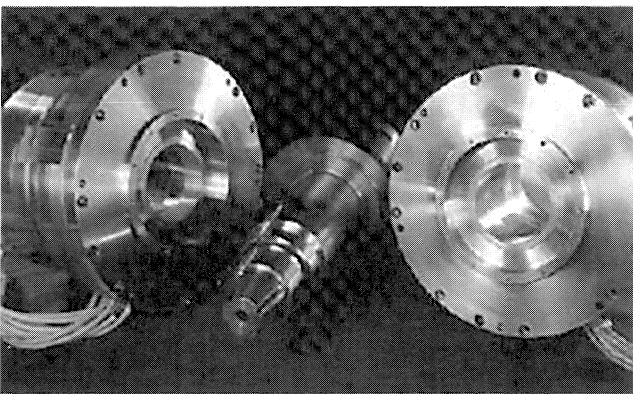


Figure 27. Bearing Stators and Rotor. (Courtesy of Glacier Magnetic Bearings)

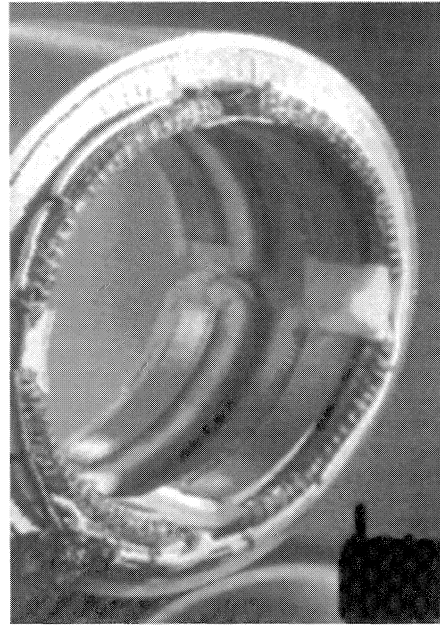


Figure 28. Homopolar Bearing. (Courtesy of Avcon Inc.)

the “memory” characteristic of magnetic materials, which demand extra current swing for each cycle of polarity reversal. The extra current produces heat loss in the wires. Windage occurs due to the shear of the fluid in the narrow gap between stators and rotor. It should be noted that windage does not contribute to the electric power consumption of the bearing, but only to shaft power consumption.

The typical power consumption of magnetic bearings is very low. For a rotor weighing 2500 kg, the total electric power consumption translates into approximately 1000 W. In a comparison with oil bearings, magnetic bearing overall mechanical loss is at least one order of magnitude lower. Savings of up to 2 percent in the equipment power are claimed.

ROTORDYNAMIC ANALYSIS

Purpose of Rotordynamic Analysis

Rotordynamic analysis is a recent development in engineering. It was not until the 1970s, when computers were able to provide the required speed and computational power, that this modelling technique was developed for widespread use. In the not so distant past, rotating equipment was designed based on experience and was somewhat restricted in speed and power.

Rotating shafts will exhibit different patterns of motion at different speeds, from standstill to their operating range. The main objective of the analysis is to predict the displacements of various parts of the shaft and the forces transmitted to the support structures. The main concern motivating the analysis is that the motion of the shaft may come in conflict with the available space inside the machine, and that the forces generated may exceed the capacity or the fatigue limits of some of its structural elements. Precision required in shaft position for some applications, such as machine tools, can be a determining factor in the analysis. Comfort may, in some cases, also be of concern when operating rotating equipment in close proximity to human activity areas.

The rotordynamic analysis tools can be used to either design new rotors and supporting systems, or to audit existing systems in case of need for troubleshooting. The analysis is based on a theoretical model of the rotor and its support system, which is created by dividing the theoretical shaft into many parts or slices (stations). Characteristic mass, stiffness, and damping are attributed to each of these stations. An overall system

mathematical model is built up from the information about the individual stations (mass), and how they interrelate (stiffness). The most common methods used to build the mathematical models are the transfer matrix and the finite element methods. Figure 29 shows a rotordynamic model of a compressor shaft.

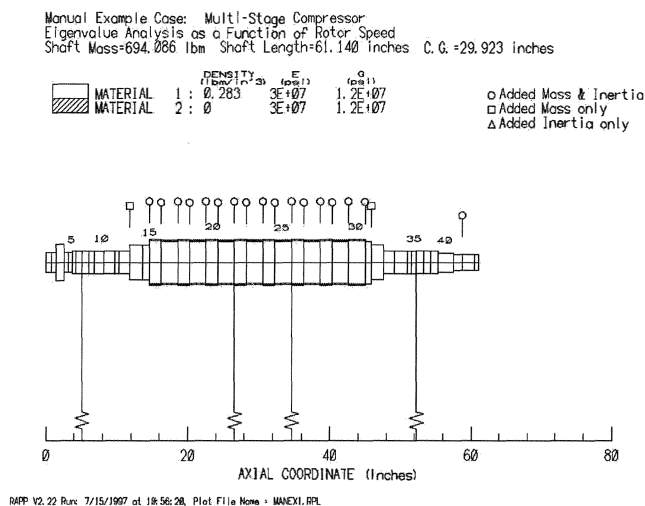


Figure 29. Shaft Model. (Courtesy of Rotordynamic Seal Research)

Natural Frequencies of the Shaft (Critical Speeds)

As an experiment, if we run up the frequency of excitation of a shaker attached to a shaft, we will notice that at certain frequencies, for a constant excitation, the amplitude of the response is greatly amplified. These frequencies are called resonances or natural frequencies of the shaft. When a shaft in operation runs up to speed, being excited by residual imbalance and other random forces, these natural frequencies are also excited. They are the critical speeds for a rotating shaft. In the past, simple calculation schemes were able to estimate the critical speeds for simple shaft designs. Today's computer programs can predict with great accuracy the frequencies of resonance for very complex rotors and support systems. The most common way to show the critical speeds is through a critical speed map, which relates critical speeds to the stiffness of the support and shows the operating speed range. Figure 30 is one example of such a plot.

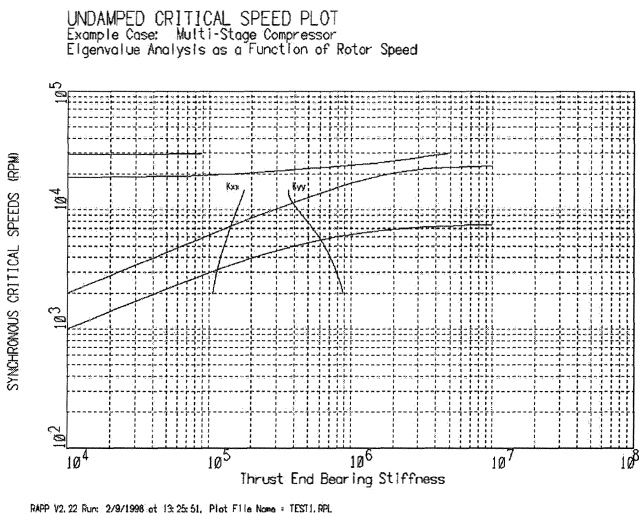


Figure 30. Critical Speed Map. (Courtesy of Rotordynamic Seal Research)

Sometimes the critical speeds are plot against shaft speed. This is called a Campbell diagram, and serves to clearly indicate the coincidences between the running speed range of the machine and its multiples, with the critical speeds. Figure 31 is an example of a Campbell diagram.

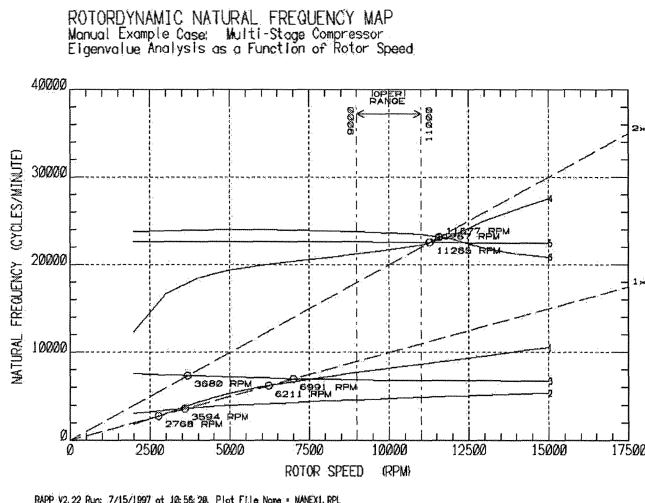


Figure 31. Campbell Diagram. (Courtesy of Rotordynamic Seal Research)

Mode Shapes

At the natural frequencies, the vibration amplitudes along the rotor will be distributed in certain peculiar ways called mode shapes. In essence, the general shape of the relative distribution of vibration along the shaft is practically constant, independent of the strength of the excitation acting on it (normalized). They are important because they tell us how the shaft will bend at the resonance frequencies and whether the location of the bearings (for example) is adequate to provide support to the shaft at these frequencies. There will be node points along the shaft mode shape. If a bearing actuator or a sensor is located at these node points, the bearing can do little to control the shaft motion. For magnetic bearings, another factor, which has to be accounted for by the control system, is called "sensor noncollocation." It happens when a node is located between the sensor and the bearing actuator. Figure 32 shows one example of mode shape for a compressor shaft.

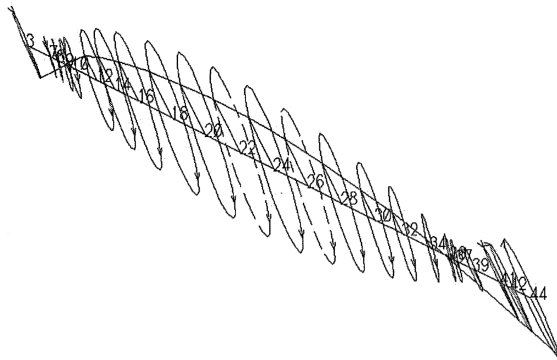
Gyroscopic forces may play an important part in the dynamic behavior of a shaft in rotation. Discs mounted on the shaft will be the major contributors to the amount of gyroscopic effect observed. These gyroscopic forces tend to make the shaft look "stiffer" at higher speeds and increase somewhat the natural frequencies of the system.

Force Response of Rotors

The analysis also permits us to determine the real amount of motion of the rotor at any prescribed station for a given excitation acting at any location along the shaft. This part of the analysis is commonly known as force response analysis and is a very important tool used in the design of rotating machines. Several excitation points can be defined and the resulting motion computed for several stations.

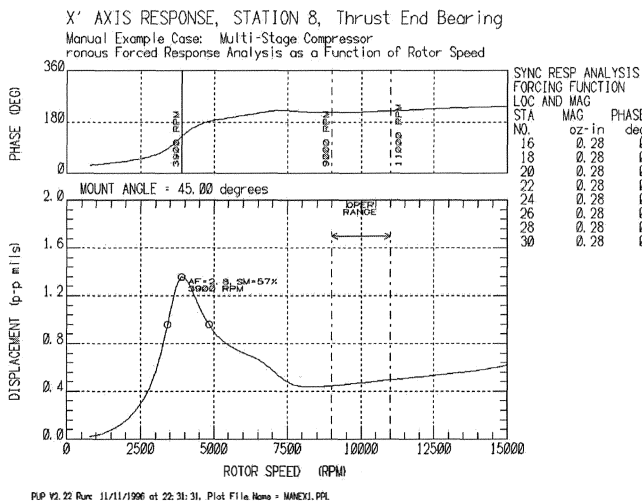
In essence, we can assume a credible amount of imbalance at a point on the rotor where it is likely to occur, such as the impeller(s), and assess the amount of motion happening at important locations where the clearances are critical. The analysis assumes linearity, so the results are scalable with the excitation strength. Figure 33 is one example of force response output.

ROTORDYNAMIC MODE SHAPE PLOT - MODE #3
 Manual Example Case: Multi-Stage Compressor
 Eigenvalue Analysis as a Function of Rotor Speed
 ANALYSIS POINT: ROTOR SPEED = 5000 (RPM)
 NAT FREQ = 7192 cpm, LOG DEC = 0.845, POTENTIAL SYNC CRIT SPEED = 6991 rpm
 STATION 22 ORBIT BACKWARD PRECESSION



RAPP V2.22 Run: 7/15/1997 at 10:56:20, Plot File Name = MANEX1.RPL

Figure 32. Mode Shape for Third Mode. (Courtesy of Rotordynamic Seal Research)



RFP V2.22 Run: 11/11/1996 at 22:31:31, Plot File Name = MANEX1.RPL

Figure 33. Force Response Plot. (Courtesy of Rotordynamic Seal Research)

Stability

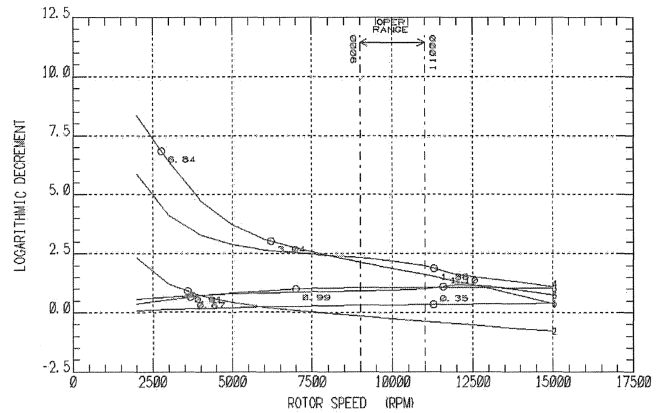
There are certain arrangements of components in turbomachinery that can promote the appearance of destabilizing forces acting on the rotor when in operation. Some of the most common of these mechanisms are annular oil seals, balance piston seals, and impeller seals. They tend to generate forces acting at frequencies different from the running speed of the shaft.

Knowledge of how to include the destabilizing characteristics of these elements in the model allow the prediction of the amount of damping available at these frequencies and the behavior of the rotor when in operation. Figure 34 is one example of a plot of the damping for each critical speed as it varies with speed of the machine.

Magnetic Bearings from a Rotordynamic Perspective

Magnetic bearings, with their ability to tailor the characteristics of the support system, are able to conform to the idiosyncrasies of

ROTORDYNAMIC STABILITY MAP
 Manual Example Case: Multi-Stage Compressor
 Eigenvalue Analysis as a Function of Rotor Speed



RAPP V2.22 Run: 7/15/1997 at 10:56:20, Plot File Name = MANEX1.RPL

Figure 34. Plot of System Damping. (Courtesy of Rotordynamic Seal Research)

the shaft design. Control schemes allow the proper amounts of stiffness and damping to be precisely allocated to different frequencies where they may be needed for stability. Figure 35 is an example of how the characteristic parameters of the bearings can be modified to suit the requirements of the shaft. It is this extraordinary versatility of magnetic bearings that make their application so attractive to novel concept turbomachinery. It gives the designer further confidence in the future stable operation of the equipment.

BEARING STIFFNESS & DAMPING

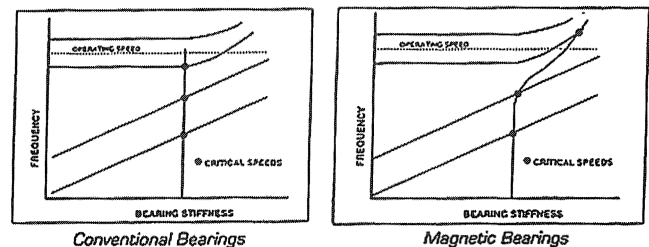


Figure 35. Critical Speed Map Comparison. (Courtesy of EPRI)

CONTROLS

Static Forces and Dynamic Forces

Both static and dynamic forces act on a rotating machine shaft when in operation. The static forces are generally the weight and maybe some added steady state forces coming from the interaction between the working fluid around the shaft and the surrounding casing. The control of static forces in a magnetic bearing machine is accomplished by the imposition of some kind of DC current in the actuator coils. Dynamic forces can be generated by flow related phenomena around the shaft. For most machines, however, the dominant dynamic force acting on the rotating shaft is that due to imbalance. Residual imbalance is always present to some degree in every rotor. The control of the dynamic forces requires some kind of AC current delivered to the coils. This varying current, however, acts against the inductance of the coil in accordance with Equation (3).

$$V = L \times dI / dt \quad (3)$$

The maximum rate at which the current can be varied in the coil in a given time, is limited by the inductance of the coil and the

driving voltage. This maximum rate of current change is also referred to as the dynamic capacity of the bearing. One can see that the existence of dynamic capacity is, in a way, in opposition to the existence of static capacity. In other words, heavy shaft weight requires large coil bearings, which oppose change in current, therefore having lower dynamic capacity.

The following example looks at two extreme situations of force loading on a rotor. One in which the weight of the rotor is quite large, but the rotor rotates at moderate speeds; another in which the weight of the rotor is small, but it rotates at a high rate of speed.

In the first case, the static forces dominate the system, and it is difficult to imagine a level of imbalance that would make the imbalance forces comparable to the weight of the shaft. In other words, the acceleration of the shaft measured by its position sensors would never reach one "g." It can be seen that, in this case, for a pair of magnets acting in opposition, only the top magnet does any work at all. In the latter case, one can see that the dynamic forces dominate the system and can reach levels much above and maybe many times the weight of the shaft. The acceleration of the shaft motion could be measured, perhaps in a few g's. A pair of magnets in opposition will have to take turns acting on the shaft to control its position. At each turn of the shaft, the bottom magnet at some point will have to pull down on the shaft to oppose the imbalance force.

The second case requires work from both magnets in order to deal with the dynamic forces. Figures 2 and 3, which show the installed base of magnetic bearing compressors and turboexpanders, clearly illustrate the general trend to have either heavy shafts running at low speeds, or light shafts running at high speeds. In terms of frequency, control of the shaft at low frequencies is limited by the bearing static capacity, while control at high frequencies is limited by the bearing dynamic capacity. Figure 36 shows the calculated dynamic capacity as it varies with frequency for a pipeline compressor.

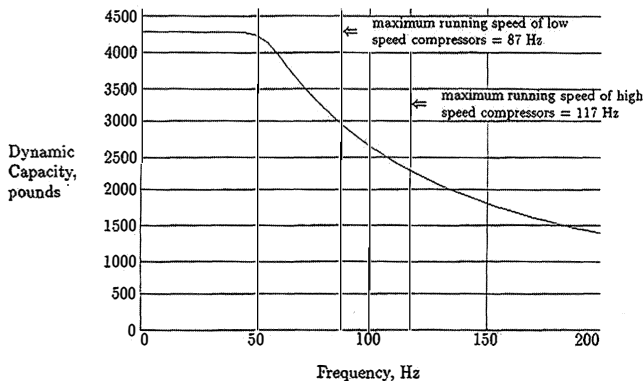


Figure 36. Dynamic Capacity Plot.

Class A and B Control

These two extreme cases, described in the previous section, can also be used to explain the control class A and B used in magnetic bearings. In B class control, the top magnet current is used for levitation and also for modulation of most of the dynamic forces, while the bottom magnet sits dormant at the minimum bias current level (Figure 37). In the first case above, the extreme example of class B operation, the bottom magnets are redundant and could perhaps even be eliminated, since the top magnet does all the control work. Note that this minimum bias current level is set because, at low currents, the bearing generates very little force (Figure 24). It would be a waste of time if one had to increase the current from zero to the point where it could make a difference in the control of the shaft. This waste of time would slow down the reaction time of the bearing and seriously reduce its dynamic capacity.

CLASS "B" CONTROL

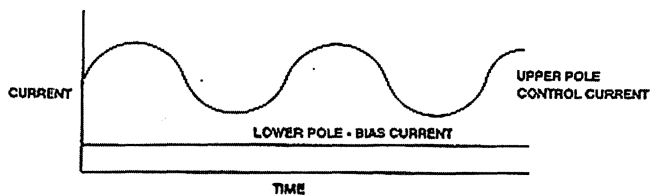


Figure 37. Time Plot of Class B Control Current. (Courtesy of EPRI)

In class A, both magnets are given an initial bias current, "preloading" them one against the other, to about 50 percent of their capacity (Figure 38). The top magnet has, of course, to provide the levitation current in addition to the bias current. In this arrangement, each time a change in force is needed in either direction, one magnet has its current increased by half the current required for the force, while the opposing magnet has its current decreased by half that amount. Each magnet does half the work. In essence in class A, since each magnet is changing only half the current, it is possible to swing double the amount of bearing current in the same amount of time, therefore doubling the dynamic capacity of the original bearing. Note that, in fact, the dynamic capacity more than doubles with class A control, due to the higher initial bias magnetization. It is also clear that class A requires some current headroom before the top magnet is saturated and can only be used when the static forces are not very large. The maximum current limit is to be respected and cannot be exceeded by the peak currents in either of the bearing coils.

CLASS "A" CONTROL

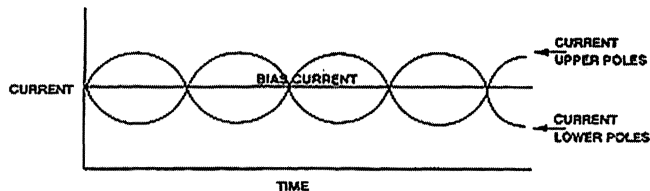


Figure 38. Time Plot of Class A Control Current. (Courtesy of EPRI)

One alternative of design for the case of both large static and large dynamic forces would be to have each bearing split into two. The larger half, taking care of the static load with constant DC current, and the other smaller half, dealing with the dynamic forces.

Imbalance Compensation

Balancing a shaft means trying to make its axis of inertia coincide with its geometric axis. Consider again two extreme situations. One in which a shaft is floating in space in total weightlessness, allowed to rotate about its axis of inertia and the other, in which a shaft is forced by the stiffness of a set of bearings, to rotate very close to its prescribed axis of geometry. In fact, both situations can be realized using magnetic bearings. In the first case, the stiffness of the bearing can be made very low at the running speed, with the use of a tracking filter in the position sensor line. This filter reduces the stiffness of the bearing at running speed only, allowing the shaft to rotate around its axis of inertia. This control scheme is sometimes called "auto balance," or ABS for short. Note: Consider that the controller puts out a controlling signal in some way proportional to the signals it "sees" from the

position sensors (more on this below). In the second case, amplifying the sensor signal at the running speed can make the stiffness of the bearing very high at the running speed, and force the shaft to run more or less true to the desired axis of rotation. In this way, for certain applications, shaft position can be rigorously maintained while operating under severe excitation forces.

In the first case, the shaft can only be allowed to rotate about its axis of inertia if the residual imbalance is small. In essence, the distance between the axis of inertia and the axis of rotation is small enough that internal clearances are respected. In the second case, forcing the shaft to be “pinned” to its axis of rotation can only be done at the expense of bearing coil current, which is a limited resource. Rotating machinery operate, in general, in some condition between the two cases described above.

The latest digital controller schemes have optimization routines to make maximum use of the available clearances and, at the same time, minimize the use of current. The current being saved by such schemes can be used in controlling transient forces that may be present in the process. This can be very useful for some classes of machines. This type of control, sometimes called feed forward adaptive control, can be a powerful tool in providing the rotor system with a much greater degree of tolerance to imbalance. The digital control scheme can be further used to do “modal balance” on the rotating shaft (still in development). Knowledge of the shaft motion in three locations, together with the knowledge of the mass elastic characteristics of the shaft (rotordynamic model), allows for the precise calculation of the motion at any speed and at any station along the shaft (also called, operational deflection shapes). This information can be used in a balancing scheme designed to minimize shaft motion at critical stations throughout the rotating speed range.

Frequency Dependent Support

One of the most important concepts to grasp when looking at magnetic bearing applications is that of frequency dependent support. We are so accustomed to seeing things being supported by structures that are more or less insensitive to the frequency of vibration excitation, that it is difficult to imagine that bearings can be made to have very different stiffness at different frequencies. While, in general, the oil bearings' stiffness is sensitive to speed of rotation, the variations occur progressively over the speed range of a machine, even when considering frequencies many times the running speed. In addition to the use of filters or control schemes to change stiffness at specific frequencies, magnetic bearings can have their general support characteristics (stiffness and damping) shaped to suit the requirements of complex machine and rotor designs. It means that a machine can operate exactly on top of a critical speed without being subject to any vibrations. Figure 39 shows the overall characteristic of a magnetic bearing controller.

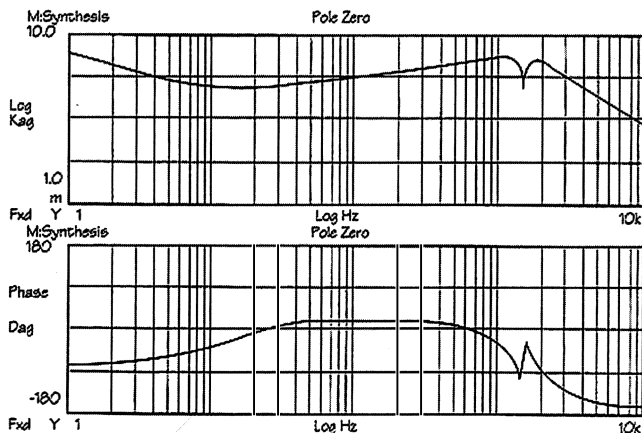


Figure 39. Closed Loop Transfer Function. (Courtesy of S2MUSA Magnetic Bearings)

The closed loop transfer function of the system, seen in Figure 39, shows the gain (or bearing stiffness) variation with frequency. Note in this case how the frequency of the signals reaching the controller is limited by a low pass filter (limits controller bandwidth) and the presence of a notch filter. Positive phase is indicative of the degree of derivative control or damping in the control system.

Sensors

Magnetic bearing systems can make use of several sensors for control of the shaft. The primary sensor is the position sensor that provides the feedback to the controller. These position sensors are usually of the inductive type and operate like tuned oscillators with minimal power consumption. In some bearing system designs, a separate and independent velocity sensor is used in addition to the position sensor. Other sensors commonly used in magnetic bearing systems are the current sensor and, in some cases, the flux feedback sensor. Their function will be reviewed a little later in this text.

It should be noted that the sensor signal is conditioned in a few ways before it reaches the controller. It is first limited in bandwidth by a low pass filter. This filter is usually set to a frequency of a few times the running speed. This low pass filtering also has a smoothing effect on the signal, due to the elimination of the high frequencies. This is very important since these signals will be differentiated in the controller and any high frequency noise is amplified in this process. Also, the controller should not waste current controlling high frequency excitation, since those will not lead to large motion of the shaft anyway. The sensor signal may also be limited to a maximum value by a gain reducing circuit. This circuit prevents the controller from “seeing” large displacements in the unlikely case excessive forces act on the shaft, and allows the controller to retain its ability and resume control once the abnormal excitations have subsided.

PID Controllers

The majority of the controllers used in magnetic bearings installed today are still of the analog kind. The control logic on most of these follows the proportional, integral, and derivative schemes, or PID control for short. Most of the new magnetic bearing systems designed today however, utilize digital controls. A very brief description of the PID control scheme follows.

Proportional control is the most primary and intuitive form of control. The controlling command is generated in direct proportion to the error in the position signal. This type of control, however, is not sufficient to guarantee the appropriate behavior of the shaft under most conditions. In the case of a steady load (such as the weight of the shaft) being applied to the shaft, the proportional controller will put out a signal in proportion to the position error. The controlling signal, however, will balance the steady load and stabilize the shaft at a point away from the desired position at the center of the bearing (provided some damping is present in the system). Another form of control is utilized in addition to the proportional, which puts out a controlling signal that continues to increase (or add over time) as long as there is an error in the position signal. This form of control is called integral or reset, since it serves to reset the shaft to its center position. The error signal used for this part of the control system is obtained from the time integration of the position sensor signal. The third form of control puts out controlling signal in proportion to the speed at which the error is increasing. In a way, this form of control amounts to predicting the future. It is called derivative control. The rate at which the error is changing, or velocity, is usually obtained by time differentiating the position sensor signal. The derivative part of the control signal provides the damping to the bearing. The three controller forms coexist in parallel in a PID controller circuit. The three signals, the shaft position, its derivative, and its integral are fed to three distinct circuits in the controller, each having a

prescribed gain. Their output is summed up and supplied to the amplifiers. These are the classic control schemes used in analog type control.

Digital controllers digitize the signal coming from the sensors, transforming them into a string of numbers. These numbers can then be manipulated by the digital processor algorithm, which may implement a PID control scheme. However, there is much greater freedom to stray from the classic PID format when using digital data. Newer algorithms are being investigated that can optimize the control system, making it more robust. Robust control in this case means that the system will maintain stability even when changes occur in the controlled machine (or plant, in control language). Digital processors have achieved great computational speeds and flexibility in the past few years and today are adequate for control of most machinery. The degree of development of controllers can be seen in Figure 40, a photo of a modern digital controller.

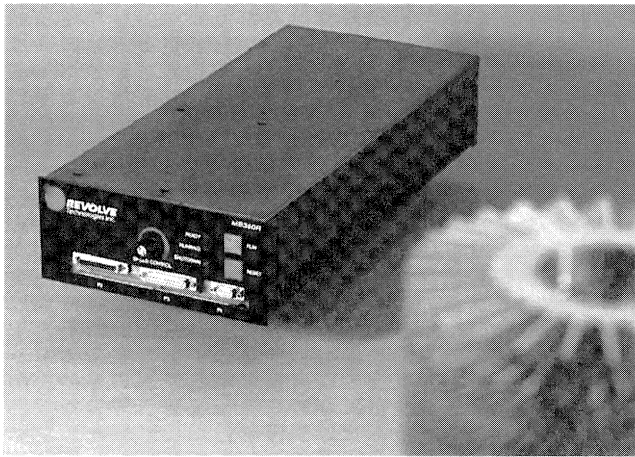


Figure 40. Modern Magnetic Bearing Digital Controller.

Pulse Width Modulation and Switching Amplifiers

This type of amplifier is used in most current installations. The signals coming from the controller are transformed into pulses of different duration, which are used to control the operation of the amplifiers. This is called pulse width modulation and is used in combination with switching amplifiers to provide the required current to the magnets. The pulses drive the switches in the switching amplifiers, making the current increase or decrease in the bearings. The amplifier is switched at very high frequency between its two states by the pulse width modulator (PWM), which controls the duty cycle. One state corresponds to an increase in the current in the coil, the other to a decrease of current. A picture of the PWM and amplifier system is shown in Figure 41.

At 50 percent duty cycle, the average current flowing through the coil is constant. At duty cycles above 50 percent, the average current is increasing, and below 50 percent, the average current is decreasing. With this scheme it is possible to modulate the current flowing through the coils and, therefore, provide a variable controlling force to the bearing. The switching amplifiers work in the principle that energy is stored alternatively in a capacitor and in the bearing coil (inductor). Although the average current flowing through the coil can be quite high, the average current flowing from the supply to the amplifier is comparatively very small. It is this supply current that defines the power consumption of the bearing.

Most amplifiers in use today have two operating states. Switching amplifiers can also take full advantage of the digital output coming from the controller, eliminating the need for a separate PWM circuit. The on and off cycles of the PWM can be easily simulated by a string of bits from the output of the controller.

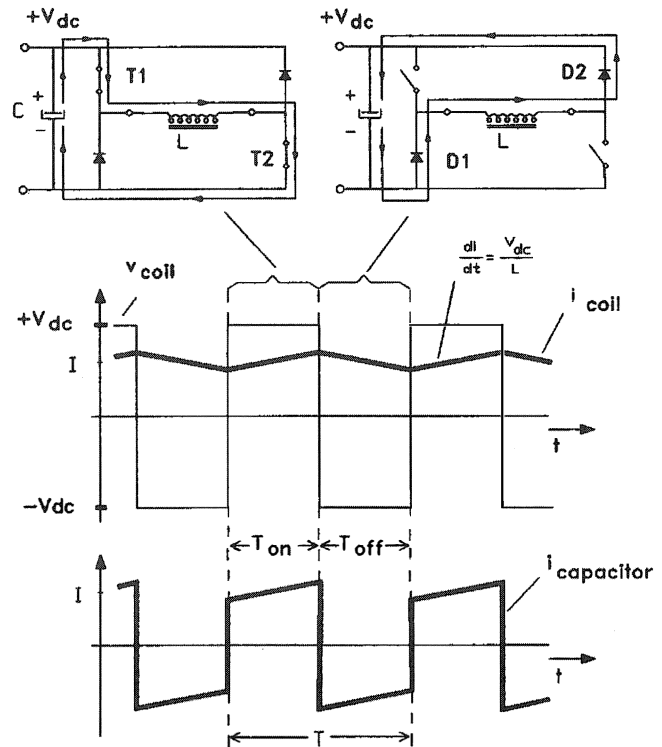


Figure 41. Amplifier Currents and Pulse Width Modulation. (Courtesy of S2MUSA Magnetic Bearings)

Switching amplifiers can also be made to operate with more than two states, which minimizes the current ripple effects due to the switching. Figure 42 illustrates the tri-state switching amplifier. Observing the variation in the duty cycles while the machine is operating, one can tell how much of the dynamic capacity of the bearing is being utilized. Duty cycles of close to 100 percent indicate that the bearing current is being changed at its maximum possible changing rate and that the dynamic capacity is at its limit.

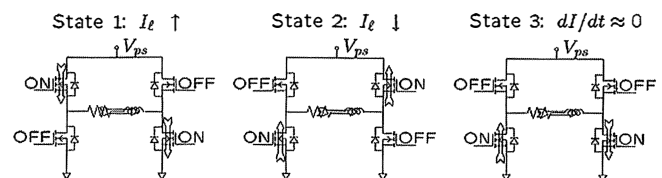


Figure 42. Tri-State Amplifier Scheme.

Minor Control Loops

- **Current feedback loop**—The connection between the orders sent by the controller and the effect they have in terms of bringing the shaft back to set point position are not straightforward and direct. The force generated as a result of a controller command is dependent on the level of current in the magnet and also on the gap at that instant. It should be noted that this dependency is quite strong. According to Equation (2), force is proportional to the square of the current and inversely proportional to the square of the gap. A minor control loop, with a current sensor feeding a signal proportional to bearing current back to the amplifier, is required in most installations to make control of the shaft possible. This minor loop setup is intended to reduce the effects of gap. The control command signal becomes a request for current.

- **Flux feedback loop**—A better system to compensate for the effects of gap and current would be one that measured force directly at the bearing, free from the dependency on current and

gap. The measurement of magnetic flux provides that, since force is directly proportional to the time integral of magnetic flux. This loop connects the amplifier directly with the force it is producing, greatly improving the control. In this setup, the control signal becomes a request for force (force command). Note that at very low frequencies, the measurement of flux breaks down, and is replaced by phasing in a current feedback loop.

Other Control Schemes

Other control schemes include the adaptive feed forward control. In this system, which is in addition to the main controller function, the sensor signal is sampled and provided to a separate digital processor a few times at every rotation. A prediction of shaft position during the next cycle is made from that, and a new control signal is sent, in addition to the main controller output, which can shape the behavior of the rotor during the next few cycles. The system constantly learns during this process, quickly improving its ability to deliver more precise control signals. This control scheme can also be set to optimize the use of current and make the system more robust, or it can produce pinning of the shaft to the center position, or allow it to rotate around its axis of inertial, or anything in between. Such systems can be valuable for machines subject to progressive imbalance due to fouling of the rotating element, by making them more tolerant to imbalance and extending the operation campaign.

Another mode of control is tilt and translation. Consider a beam style machine with magnetic bearings at both ends. If one picks the sensor signals from both ends (in one plane), adds them and then divides them by two, the resulting signal shows the translation motion of a point in the center of the bearing span. Conversely, if one subtracts the two sensor signals and then divides them by two, the resultant signal is in proportion to the tilting motion of the shaft. If the "translate signal" is fed to a PID controller, and its output is sent to the amplifiers at both ends simultaneously, then this controller is controlling solely the translation motion of the shaft in that plane. The same thing can be done with the signal proportional to the tilting motion, so that another controller can control the tilting motion exclusively. The scheme is set up so that one ends up with one tilt and one translation controller for each plane at 45 degrees. This scheme allows the tailoring of the rotordynamic characteristics of the shaft. Increasing the stiffness (or gain) of the translation control affects the first and all odd order critical speeds, while increasing the stiffness of the tilt control affects the second and all even order critical speeds.

Conical bearings produce force both in the radial and axial directions. This magnetic bearing setup eliminates the need for a separate thrust bearing. Figures 43 and 44 illustrate the approach. It can be seen that the force of each bearing has to be split in the axial and radial directions and, therefore, this scheme may be restricted to machines subject to light loads.

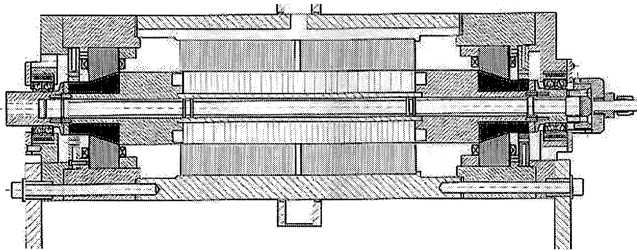


Figure 43. Conical Bearing Arrangement.

Tuning

Tuning is the act of adjusting the parameters in a control system to achieve stability of operation. It is important that these parameters be estimated, based on the characteristics of the combined control system and rotordynamics of the shaft. The

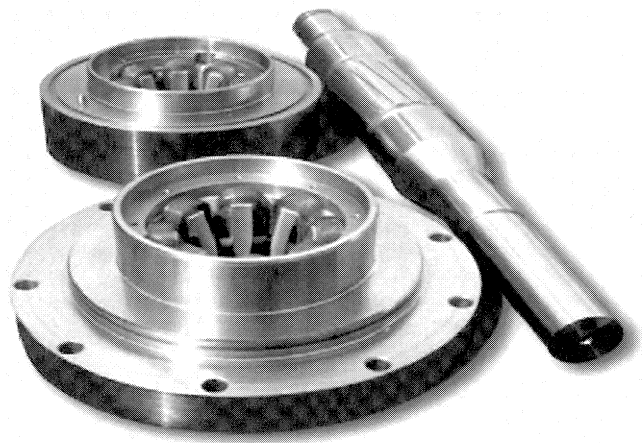


Figure 44. Conical Bearing.

degree or margin of stability of the system is obtained in the tuning process. It is very important that not just a working setting of the controller be obtained by "tweaking" the control parameters. Sufficient stability margin can only be attained from precalculated parameters to guarantee against changes in the system. The tuning process during the commissioning of a machine is directed first at producing levitation of the shaft, with the setting of the oscillator, the sensors, and the controller. While analog controllers have to have their tuning settings changed by soldering components on a board, digital controllers make the tuning process quicker due to the possibility of entering tuning parameters using a keyboard interface. Additional monitoring and transformation of parameters, such as current and rotor position, further allow the results of any tuning changes to be promptly verified. The tuning process with precalculated parameters should not take more than a few hours, and once the parameters are set, it should not be necessary to retune a machine. The only reason for retune of a machine is if there has been a change to the dynamic system, either mechanical or in the controls' settings.

SPECIFICATIONS

Acceptance Tests

New equipment using magnetic bearings should be tested at the factory for operation and performance. Some of these acceptance tests are suggested specifically for the magnetic bearing system.

- **Drop test**—There is a need to test the performance of the auxiliary landing system under realistic conditions of operation. The so-called drop test is done by allowing the rotor to drop on the auxiliary bearings at full speed. The test is carried out preferably at the vendor's facility with the machine stripped of seals, so that internal clearances are increased and potential inadvertent damage is minimized. The expected result of the drop test is the smooth coasting of the rotor, while resting on the bottom portion of the auxiliary bearings. The damping provided by the auxiliary bearing support system plays a key role in preventing high-speed whirl of the shaft in the auxiliary bearing clearance gap. This high-speed shaft whirl could be extremely damaging to the machine if allowed to happen.

- **High amplitude excitation test (imbalance)**—This test is designed to exercise the controller's ability to cope with excessive loads and resume control after the loads are removed.

- **Clearance checks**—The internal assembly clearances can be checked by moving the shaft electrically, using the magnetic bearing. The manufacturer should provide a tolerance range for the internal clearances. In some modern digital control implementations, the clearance checks are done automatically by the controller.

- *Full load amplifier operation test*—This test is performed by subjecting the amplifier to a high load for a period of a few hours, in order to prove its function and heat dissipation capabilities.
- *High imbalance test*—A large amount of imbalance is calculated and added to the shaft for this test. The machine is rotated to its maximum speed to prove the controller's ability to provide stability.
- *Stability margin verification*—The vendor should prove the margin of stability by varying, in turn, several parameters in the control system until the machine becomes unstable. The amount of dynamic capacity being utilized when the machine is in operation is also a good indication of the stability margin.

Applicable Codes

The API and ASME codes can provide limited guidance in the specification of a magnetic bearing machine. New standards were being developed in the early 1990s, which were to be applied specifically to magnetic bearings, but this work has stopped for some time and needs to be restarted. There are remarkable differences between oil-lubricated machines and those running on magnetic bearings. The discussion above should provide an appreciation for these differences and their implications to equipment design and specification. Standards have to be written reflecting those differences and focusing on the user's requirements. For example, code requirements related to rotordynamic behavior, such as critical speed separation margins and classic bearing vibration limits, would constitute a hindrance to the greater freedom of design of control systems allowed to magnetic bearing machines.

Rotordynamic Analysis

In the case of magnetic bearings, there is a requirement for two analyses. One, of the rotor supported by the magnetic bearings, is done in conjunction with the design of the control system strategy. The other, is that of the rotor supported on the auxiliary bearings, and should take into account their specific characteristics. These analyses should be done by the magnetic bearing vendor, and in cooperation with the rotating equipment supplier. The end user should review and discuss the results and interpretation with both parties.

Bearing Loads

The accurate prediction of bearing loads, both static and dynamic, is essential to the proper design of a magnetic bearing system. This prediction should include possible machine upgrades in the future. Transient loads should be taken into account along with potential incremental imbalance during the operation campaign. The bearing manufacturer should demonstrate, at the design stage, the calculated dynamic and static capacity of the bearings suggested for the application.

Maintenance

Magnetic bearings are virtually maintenance free for the end user. Occasional check of major operating parameters, such as cooling flows and bearing currents and vibration, to assess the state of balance is all that is required. It is the experience of the author that even complete re-aeros of large pipeline compressors can be accomplished without any intervention in the magnetic bearing system.

Training

Some amount of training in magnetic bearing technology is required of the vendor of rotating machinery. Apart from a general knowledge of the system, this training would probably concentrate around the requirements for mechanical assembly, fabrication, and fitting of components. The end user of equipment should require

some training in the general system arrangement and minor troubleshooting. In case of need for technical assistance, it should be provided by the magnetic bearing vendor.

Condition Monitoring

The magnetic bearing system provides capability for free and extensive monitoring of the rotating machine. Requirements for condition monitoring of the bearing system itself are minimal; however, knowledge of the static and dynamic forces acting on the rotating shaft can provide enormous insight into the conditions of the equipment and process. In several cases, in the author's experience, the magnetic bearings have given indication of impending problems and saved the machines from severe and extensive damage.

Nova Gas Transmission's Experience

Nova Gas Transmission Ltd. has a large fleet of magnetic bearings in use for many years. Nova's experience with magnetic bearings started in 1986, and today there are 31 pipeline compressors so equipped in the system. These units in operation at Nova have had a virtually perfect reliability record since December 1994. Availability is also very high, on the order of 99 percent or better.

Turboexpander's Experience

There is a large population of turboexpanders running on magnetic bearings. The total number is estimated to be in the order of 90 units worldwide. Turboexpanders have enjoyed great reliability and availability with an enviable track record ever since the first ones were put in service. The nature of the operation of turboexpanders, with significant temperature differences between the two sides of the machine, lends itself well to take maximum advantage of magnetic bearings with the elimination of lubricating oil from the system. Figure 45 shows the constituent parts of a turboexpander bearing setup.

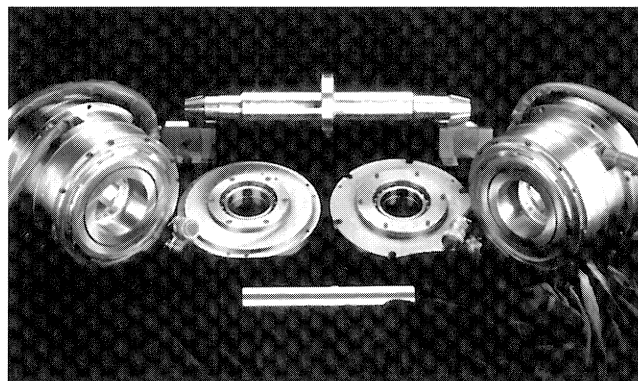


Figure 45. Turboexpander Bearings. (Courtesy of Mafi-Trench Co.)

CURRENT RESEARCH

It has to be stressed here that magnetic bearings are a mature technology, proven in industry, with a record of millions of hours of operation. As with many other fields, there is continuing development also happening with this technology. Here are some of the topics that are currently the focus of research in several universities.

University of Virginia

The following is information provided by Dr. Eric Maslen of the University of Virginia, on some of the current research effort. It can give a good idea and perspective on the overall direction of the development effort.

Major research areas in magnetic bearings:

1. *Control research: sensitivity*

- a. Robust linear, nonlinear, and fuzzy
- b. Self tuning
- c. Adaptive feedback

All methods aim to allow generic controllers to be applied to specific machines with reduced onsite engineering to stabilize and obtain target performance.

2. *Control: performance*

- a. Steady excitation
- b. High performance MIMO (nonlinear)
- c. Gain scheduled or LPV control (linear parameter varying)

Methods are aimed at improving either response to steady loads (static side load, unbalance, and shaft bow) or response to broad-band disturbances like machining loads, shock, and so forth.

3. *Fault tolerance*

Development of systems, which do not fail when single or multiple components fail. This would include actuator components, connectors, sensors, and controller subsystems.

4. *Self sensing*

Reduce cost, component count, and wire count through integration of sensing with actuation.

5. *High temperature*

Develop and demonstrate actuator designs, which can function well in hot gas turbine environments.

6. *High load resolution*

Many applications are being presented where the actuator must also function as a load cell. Such applications are not only research oriented; many process control problems require online monitoring of bearing loads. Signal processing methods and low hysteresis materials appear to be the key directions for improving resolution and stability of load monitoring.

7. *Low power*

Space, biomedical, and flywheel applications are particularly sensitive to power consumption; other high-speed applications are sensitive to rotor losses. Efforts are underway to develop better predictive models for these losses and to incorporate these models into design methods to develop low loss bearings

8. *Auxiliary bearings*

For a long time to come, we will continue to rely on auxiliary bearings as the last line of defense in magnetic bearings.

Current effort focus on:

- a) Better predictive models/characterizations of rotor behavior during a drop.
- b) Better lifetime models for auxiliary bearings in response to transient loads as well as "idle time" degradation.
- c) Better designs meeting complexity, reliability, space, and performance requirements.

9. *The long term outlook:*

Field acceptance of the basic technology is increasing rapidly:

Magnetic bearings are no longer seen as science fiction but rather, as a young but well established technology. Engineers in a very broad range of areas are looking at magnetic bearings as an alternative to existing technologies with very substantial advantages, particularly in system integration. Presently, the primary impediment is cost. What is needed, is to establish some high volume applications that will put some components on the shelf, and allow

vendors to develop the low cost production methods that are attached to high volume production. This will then make development of related volume applications much easier and will encourage development of off the shelf components that will make low volume applications much more cost effective. (Maslen, 1998)

Virginia Polytechnic Institute and State University

The following is a summary of research conducted by Dr. Gordon Kirk and Dr. Mary Kasarda of the Virginia Polytechnic Institute and State University in Blacksburg, Virginia.

- Application of magnetic loading for system identification of fluid-film bearings
- Power amplifiers
- Flywheel applications
- Power losses in bearing actuators
- Auxiliary bearing tests
- Finite element analytical simulation of rotor drop

Dr. Kirk has done extensive work in the area of control systems and dynamics of the auxiliary bearing systems during drop tests. His research facilities include a full-scale magnetic bearing simulator and drop test apparatus.

Texas A&M University

At Texas A&M University, Dr. Reza Langari is doing work on nonlinear control of magnetic bearings that has application in turbomachinery. This is basically a theoretical and experimental study of the use of fuzzy logic control in developing nonlinear control strategies for magnetic bearings.

Dr. Alan Palazzolo is working on high temperature, flywheel, and machine tool application of magnetic bearings. This consists in the development of all aspects of MBs including stator, rotor, radial, and thrust bearing hardware, controllers, power amplifiers, and sensors.

University of New Brunswick

At UNB, Dr. Don Lyon's primary interest in magnetic bearings is in vibration analysis and control. The work focuses on developing a multiplane adaptive active vibration control method that distinctly identifies and cancels both periodic and random vibration, without any prior knowledge of the rotor system. This is based on the new DSP technology he developed for creating distinct periodic signal and random noise models for dynamic signals, about which there is no prior knowledge. Active vibration control is a neat application for this. In general, Dr. Lyon's interest is in stochastic modelling of signals generated by mechanical systems. The research focuses on developing new DSP technologies that relate to mechanical systems. As a result, vibration analysis is a prime area of his work.

NOMENCLATURE

- B = Magnetic field density, tesla, N/(m,A), Wb/m²
- F = Force, N
- G = Acceleration of gravity, m/s²
- I = Current amps
- K = Constant
- S = Surface area, m²
- μ₀ = Magnetic permeability of free space
- V = Volts
- L = Length, m
- t = Time, s

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