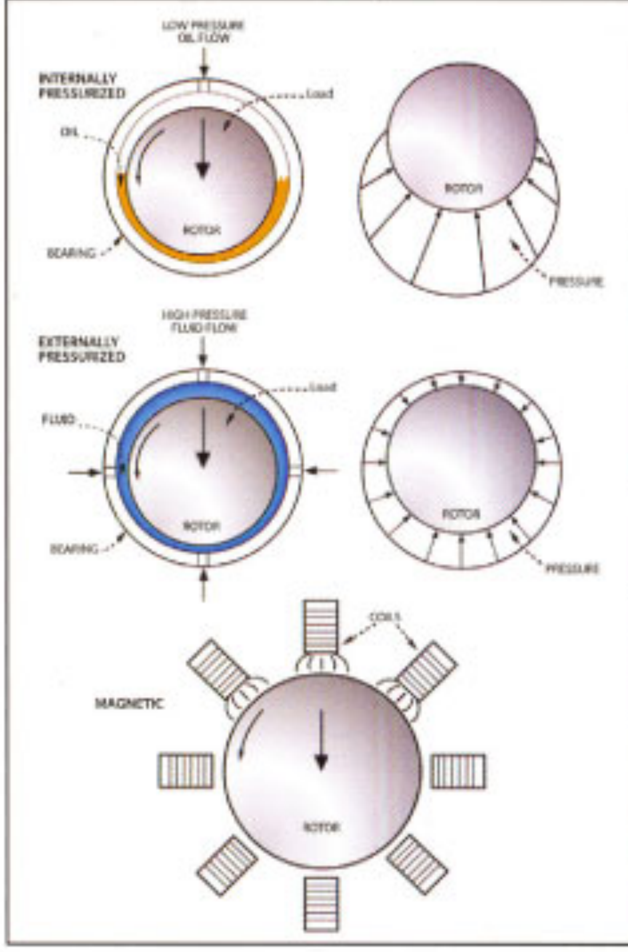
Shaft levitation made simple

EXTERNALLY PRESSURIZED BEARINGS OFFER COST AND ROTORDYNAMIC ADVANTAGES OVER MAGNETIC BEARINGS

DONALD BENTLY AND CHARLES HATCH
BENTLY PRESSURIZED BEARING COMPANY



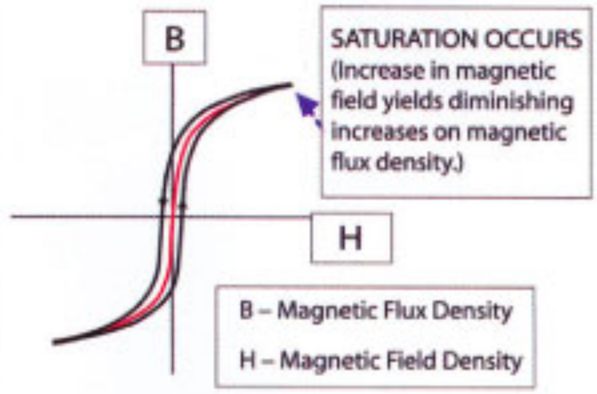


***Figure 1:** A given magnetic material has limited ability to develop flux (above), which limits the load-carrying ability of magnetic bearings*

***Figure 2:** To the left are internally pressurized bearings (top), externally pressurized bearings (middle), and magnetic bearings (bottom). Complex control systems are required in magnetic bearing systems*

Conventional oil-lubricated bearings have been used in turbomachinery for over eighty years and they constitute a mature technology. While modern oil-lubricated bearing designs are quite reliable, they suffer from significant limitations. These include poor low-speed performance (possibility of boundary lubrication), viscous drag, heating in high-speed machines, and vulnerability to fluid-induced instability in some designs. Further, there is a danger of process fluid contamination by the lubricant and a requirement for external lubricant supply systems.

Two bearing technologies have emerged that attempt to address these limitations: externally pressurized bearings and magnetic bearings. Externally pressurized bearings have a simplicity that leads automatically to high reliability. These bearings are inherently stable and can run passively with little maintenance. When the process can supply the pressurized fluid, contamination is avoided and there is no additional power or equipment requirement. Because of their simplicity, the initial and long-term cost of these bearings is relatively low (typically 30% of that of magnetic bearings).

By comparison, magnetic bearings are complex and expensive. They are inherently unstable and require an expensive control system and a set of back-up catcher bearings. Magnetic bearings typically have long lead times and must be carefully tuned during the commissioning process. While day-to-day operating costs of magnetic bearings may be low, long-term maintenance may involve the expensive replacement of obsolete electronics (see p. 16 for an article by a magnetic bearing manufacturer).

Flooded bearings

Since we will be discussing externally pressurized bearings, it would be good to start with a quick definition of internally pressurized bearings. Internally pressurized bearings (Figure 2, top) are also known as hydrodynamic, or sleeve bearings. In these bearings, the lubricant fluid — usually oil — is injected into the bearing at relatively low pressure.

Internally pressurized bearings that constitute the vast majority of oil-lubricated bearings in use today are almost always partially lubricated — that is, the clearance between the journal and the bearing is only partially filled with the lubricant. The forces that support the rotating shaft are produced by the rotation of the journal, which, because of the viscosity of the fluid, causes the lubricant to be dragged into a region of decreasing clearance. As the flow area decreases, the local pressure increases, creating a hydrodynamic pressure force (fluid wedge) that supports the journal. Because the support forces are completely dependent on journal rotation, these bearings have poor load-carrying capacity at low speed.

Externally pressurized bearings (sometimes called hydrostatic bearings) differ from internally pressurized bearings in that the fluid is injected at relatively high pressure (typically a few hundred psi, but higher pressures are also used) through several circumferentially distributed radial ports (Figure 2, middle). The inlet ports may be equipped with flow-control orifices. In gas bearings, the entrance region, where the inlet port meets the internal bearing clearance, acts as a restrictor in series with an optional inlet orifice restriction. In liquid bearings, an orifice is almost always used, and an inlet pocket is used to distribute the pressure over the journal area near the inlet.

In either case, the inlet system produces a pressure drop, and the pressure in the bearing clearance near the inlet is typically about half of the supply pressure. Downstream from the inlet or pocket, the bearing clearance acts as an additional flow restrictor that produces an axial pressure gradient from the inlet region to the bearing exit. Because all of the inlet ports are fed with fluid, externally pressurized bearings are always fully flooded by design.

Using process fluids

In an externally pressurized bearing, when the journal (which can be rotating or non-rotating) is located at the center of the bearing clearance, the circumferential pressure distribution around the journal produces no net force on the shaft. However, when the shaft is displaced away from the center of the bearing, tighter clearance on one side causes the pressure to increase on that side and to simultaneously decrease on the opposite side. This results in a circumferential pressure distribution that causes a net force to appear, which acts opposite to the displacement — a stable behavior.

Changes in the internal pressure distribution in an externally pressurized bearing produce the stiffness that provides load-carrying capacity. Externally pressurized bearings can be primarily hydrostatic in behavior at low eccentricity ratios (typically 0.1 to 0.2, the term defines the ratio of the offset of shaft centerline from the bearing center to bearing clearance) and they can include hydrodynamic effects at higher eccentricity ratios.

The pressure distribution in an externally pressurized bearing is essentially hydrostatic in nature; thus, these bearings can provide full load-carrying capacity at low speed or even zero speed. This makes them a deserving candidate for low-speed or reversing machinery. Because there is no possibility of metal-to-metal contact between the journal and bearing at low speed, boundary lubrication is not an issue and any suitable fluid can be used in the bearing. This means that, for many applications, such as in the petrochemical industry, externally pressurized bearings can use the process fluid, whether liquid or gas.

Contamination of the process fluid by the lubricant is avoided and, when the machine is embedded in a process that has high-pressure fluid available, the bearings can be operated without supporting pressurization equipment. This makes for a simple, nearly maintenance-free installation. The inherently low friction of a gas bearing film (due to lower viscosities of gases compared to liquids) makes externally pressurized gas bearings an ideal choice for high-speed machinery, such as turboexpanders. For instance, a recent application of radial and thrust externally pressurized bearings in a turboexpander (84,000 rpm) uses pipeline natural gas. The supply pressure is about 450 psi discharging into 60 psi.

Gas bearings have load-carrying and stiffness characteristics comparable to magnetic bearings, while liquid externally pressurized bearings have much higher load-carrying capacity and stiffness (typically 10 times greater than gas bearings). In general, the inherent damping of externally pressur-

The magnetic bearing is always unstable and completely dependent on active control to maintain shaft position

when imbedded in a process that supplies the pressurized fluid, need little maintenance. Because of the use of inlet orifices, filtering of the fluid is usually a requirement, and periodic filter replacement is the only maintenance item. For applications where the fluid must be pressurized, an external, fluid-supply system is required. This can be as simple as a booster pump-filter system or as complicated as a fully circulating fluid delivery system with fluid cooling.

Magnetic bearings

Magnetic bearings consist of an array of electromagnetic coils that are circumferentially distributed around the shaft (Figure 2, bottom). When an electric current is applied to a coil, the magnetic flux creates an attraction force between the coil and the rotor that acts in the direction of the coil. The oscillating magnetic fields can produce eddy currents in the rotor that can cause losses and heating. To suppress these currents, the rotor area in the magnetic bearing is usually constructed of thin laminations, similar to the construction of transformers.

For a given current, the magnetic force is a strong nonlinear function — an inverse square — of the clearance between the rotor surface and the coil. As the clearance decreases, the magnetic force increases strongly, and vice versa. Thus, (for constant current) when the rotor is displaced away from equilibrium in a magnetic bearing, the magnetic forces change in such a way as to strongly increase forces in the same direction as the displacement. This is an inherently unstable arrangement that, if uncontrolled, would cause the rotor to impact and adhere to the bearing.

To address this problem, magnetic bearings must use an active control system that operates in a closed loop. Displacement sensors measure the instantaneous position of the rotor in the bearing clearance.

In its simplest form, the control system incorporates a Proportional Integral Derivative (PID) controller to which the signals are fed. The controller continuously compares the position of the rotor to a predetermined setpoint and rapidly adjusts the current in individual magnetic bearing coils to change the forces to move the rotor back toward its setpoint position.

are aerodynamically unstable and rely on active control to maintain flyability.

Unlike other types of bearings, magnetic bearings do not have inherent stiffness and damping; instead, these two properties are synthesized by the control system. The Proportional-Derivative gain settings in the controller cause the system to behave as if it had stiffness and damping. The proportional gain causes the force to increase as a function of displacement; this simulates stiffness. The derivative gain causes the force to increase as a function of the rate of change of displacement (velocity); this simulates damping.

Because of the inherent instability of magnetic bearings, failure of the control system will result in a hard and destructive crash of the rotor against the bearing. This is a real possibility that must be protected against, and magnetic bearings are equipped with rolling element catcher bearings that are designed to "catch" the rotor and prevent major damage in the event of a control failure. However, contact of a spinning rotor with the catcher bearing can also cause damage to the catcher bearing and the rotor. Catcher bearing design has improved over the years, but this is still a vulnerable area of magnetic bearing design.

Limitations in loading

Catcher bearings complicate the rotordynamics of the rotor-bearing system. Catcher bearings are located at different axial positions from the magnetic bearings, and their stiffness and damping properties are different. If a control system failure occurs at full speed, the rotor must be able to shut down — perhaps through a resonance — without a rotor-stator rub. If not carefully designed, the catcher bearings, by changing the rotor modal frequencies and mode shapes during the shutdown, could allow a damaging rub to occur at locations away from the bearings. This is of particular concern for flexible rotors where vibration antinodes (locations of maximum vibration) typically occur at midspan locations.

Magnetic bearings are limited in the maximum amount of magnetic force — and hence load-carrying capacity — that can be generated. This is because the rotor material begins to magnetically saturate as coil current is increased beyond a point, reaching a maximum value (saturation of the flux density vs field curve, Figure 1). This produces an additional nonlinear factor that must be accounted ized bearings is high. For instance, recent tests of several air bearing designs of 1.76 inch diameter revealed damping of 30 lb-sec/inch to 120 lb-sec/inch.

Externally pressurized bearings,

Thus, the magnetic bearing is always unstable and completely dependent on active control to maintain shaft position in the bearing. This is similar in concept to some modern military fighter aircraft that

for in the control system. The hysteresis effect (delay in response) associated with the curve also causes heating of the rotor.

The control system of a magnetic bearing has a limited frequency response, yet

the rotor — especially a flexible rotor — is capable of producing many modes of vibration over a wide range of frequencies. The magnetic bearing control system responds to measured vibration by adjusting the signals to the magnetic coils in real time. As long as the measured vibration frequency is below the frequency limit of the control system, the correction forces will have the phase relationship necessary to correct the rotor position.

If rotor vibration frequencies exist that are above the frequency limit of the controller, then correction signals may have an incorrect phase relationship that can actually excite one or more of those frequencies and destabilize the rotor system. This effect would manifest itself as super-synchronous vibration.

A related problem occurs when transducers measuring rotor displacement are located on the opposite side of a nodal point (location of zero vibration amplitude) from the bearing. If this were to happen, the displacement signal would have the wrong phase relationship and cause the controller to send a correction force that would destabilize the rotor.

Nodal points are often located near bearings, and they shift position with rotor speed as the rotor changes from one mode of vibration to another. Thus, the location of nodal points are frequency dependent; the rotor mode shape will be different for different frequencies, and the nodal points will have different locations. When sources of broadband excitation exist — for example, due to aerodynamic forces in compressor wheels — and if one of these modes produces a nodal point located between the transducer and the bearing, then these higher modes could be excited by the control system. The presence of these higher modes complicates the design of the control system.

Because of many of the technical issues we have discussed here, magnetic bearing systems usually require "tuning" during commissioning. The systems often remain a "black box" to the end-user afterwards.

Facing obsolescence

Magnetic bearings systems can also have long-term maintenance issues. Electronic components in the control system will age or fail and will have to be replaced over time. With the rapid rate of change in electronics and microprocessor technology, it is highly likely that control system components will become obsolete, making replacement difficult and expensive. This is a continual problem that faces all electronic and computer system manufacturers, where product lines are discontinued when the product-sustaining effort becomes too expensive. The complexity of control systems used in magnetic bearings suggests that long-term maintenance of these systems will be difficult and expensive.

Magnetic bearings do have one significant advantage over other types of non-contacting bearings. They can be used in a vacuum, making them an excellent alternative to rolling element bearings for space applications. TI

Authors

Donald Bently is the Chairman and CEO of Bently Pressurized Bearing Company.

Charles Hatch is Engineering Manager of Bently Pressurized Company. Reach him at charles.hatch@bpb-co.com

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