13

ADVANCED SEALING AND BEARING SYSTEMS*

In the 1970s, a forward-looking Canadian company, Nova, initiated a program to identify, research, and implement technologically and economically sound means of eliminating the many problems associated with oil systems on their centrifugal compressors. The purpose of this program was to improve safety and reduce maintenance and operating costs.

In the years that followed, Nova worked closely with mechanical dry gas seal (dry seal) and magnetic bearing manufacturers on the development, application, and installation of these technologies for centrifugal compressors. By 1978, a fully functional dry seal had been installed successfully. By 1985, the first oil-free compressor using both dry seal and magnetic bearing technologies was operational.

Development of the technologies continues, with recent advancements in thrust-reducing seals and magnetic bearing control system enhancements. Other manufacturers have become active in the field, and by the late 1980s dry seal technology had advanced to the stage where the industry began actively discussing specification standards.

13.1 BACKGROUND

Internal studies conducted by Nova's Alberta Gas Transmission Division in the early 1970s indicated that a significant portion of the downtime on their 79 centrifugal compressors related to problems with the seal or lube oil systems. The problems ranged from failures in auxiliary systems, such as motors and pumps, to leaks and failures in pressure piping resulting from vibration.

* Originally developed and contributed (except as noted) by Stan Uptigrove, Paul Eakins, and T.J. Al-Himyary of Revolve Technologies, Calgary, Alberta, Canada. Revolve was spun out of Alberta, Canada–based Nova Corporation, and its technology is now owned by SKF.

A Practical Guide to Compressor Technology, Second Edition, By Heinz P. Bloch

In 1978, Nova and a seal vendor developed a dry seal that reliably replaced the conventional seal oil system in centrifugal compressors. Dry seal development with other vendors continued, with the recent application of alternative face materials, groove patterns, and thrust-reducing configurations—all with successful results.

In 1985, an active magnetic bearing system was installed in one of the centrifugal compressors that had been retrofitted earlier with dry seals. As of 1994, this compressor, an Ingersoll-Rand (IR) CDP-230 driven by a General Electric LM 1500 gas generator and an IR GT-51 power turbine rated at 10.7 MW (14,500 hp), had accumulated nine years of operation since the bearing retrofit. It also served as a test bed for several other pilot installations of new developments in this technology.

In 1988, Nova retrofitted magnetic bearings onto the power turbine of this package. Operating history since then indicates that this higher-temperature environment poses no limitations to magnetic bearing technology. This makes it possible to eliminate the lube oil systems commonly serving the gas compressor and the power turbine, a feature found frequently in aircraft derivative gas turbine compressor packages.

A system expansion undertaken by Nova in the late 1980s has seen the procurement of over 40 new compressors equipped with one or both of these technologies, along with the increasing involvement of equipment vendors in the development of these technologies. Other user companies, among them Alberta Natural Gas, Marathon Oil, Shell Canada, and Exxon Chemicals, have moved in the same direction.

13.2 DRY SEALS

13.2.1 Operating Principles

Dry seal design is based on the gas film technology used successfully in other applications, such as air bearings in high-precision machining and measurement equipment. The heart of the sealing mechanism is comprised of two seal rings (Fig. 13.1). The mating ring has a groove pattern etched into a hard face and rotates with the shaft. The primary ring has a softer face and is restrained from movement except along the axis of the shaft.

Springs are located to axially force the primary and mating ring faces toward one another. When the compressor is shut down and depressurized, the spring forces result in contact of the faces. As the compressor is pressurized, the balance of static pressure forces on the seal mechanism allows a minute volume of gas to leak past the faces.

When the compressor is running, the combination of process gas pressure (hydrostatic forces) and the pumping pressure provided by the spiral grooves (hydrodynamic forces) results in noncontact seal face equilibrium. Increased clearance reduces gas film pressure, and hydrostatic pressure behind the faces tends to reduce clearance. The noncontacting nature of the gas seal film indicates that there is virtually no mechanical wear.

Depending on the process application, the seal mechanism can be used by itself in a single (Fig. 13.2), double, or tandem arrangement. The tandem arrangement (Fig. 13.3) is most common in natural gas pipeline applications. Each stage is capable of sealing against full process gas pressure, up to approximately 10,350 kPa gauge (1500 psig).

In normal operation, the primary stage seals against full process gas pressure and the second stage is unloaded. The secondary stage sees process gas pressure only in the event of a failure of the first-stage seal. This provides backup for safe shutdown. For ease of installation, the entire seal assembly is encapsulated so that it can be installed and removed as a complete unit (Fig. 13.4).

FIGURE 13.1 Dry seal rings. *(Revolve Technologies, Calgary, Alberta, Canada)*

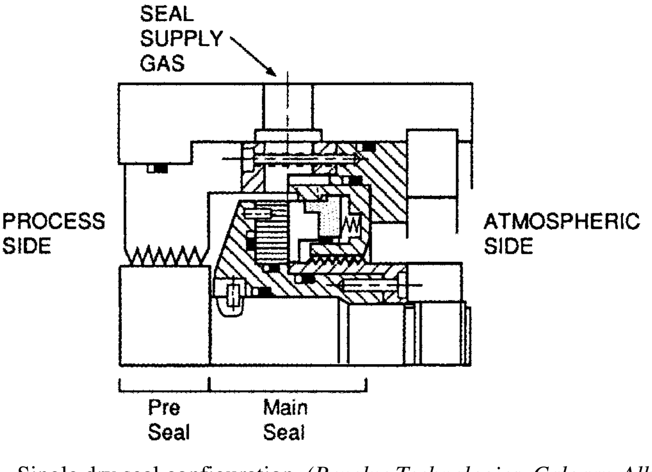


FIGURE 13.2 Single dry seal configuration. *(Revolve Technologies, Calgary, Alberta, Canada)*

A monitoring and control system ensures that the seals are provided with a clean gas supply to prevent potentially dirty process gas from entering the seal (Fig. 13.5). Although other sources can be used, seal gas supply is typically taken from the compressor discharge piping and filtered. A coalescing 0.1-μm filter is used in case liquids are present.

Seal gas is monitored through a small flowmeter and sent to the seal gas supply, where the majority reenters the process cavity across a labyrinth seal. Only the volume of leakage gas pumped by the grooves passes across the seal faces; then it drops across the smooth dam area of the mating ring until it reaches the pressure in the first-stage leakage port.

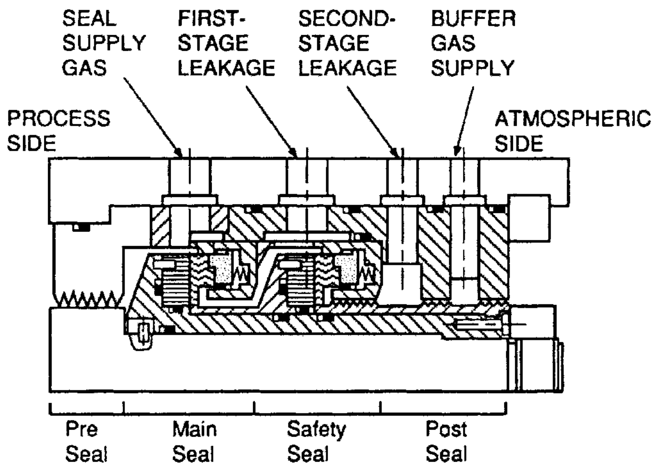


FIGURE 13.3 Tandem dry seal configuration. *(Revolve Technologies, Calgary, Alberta, Canada)*

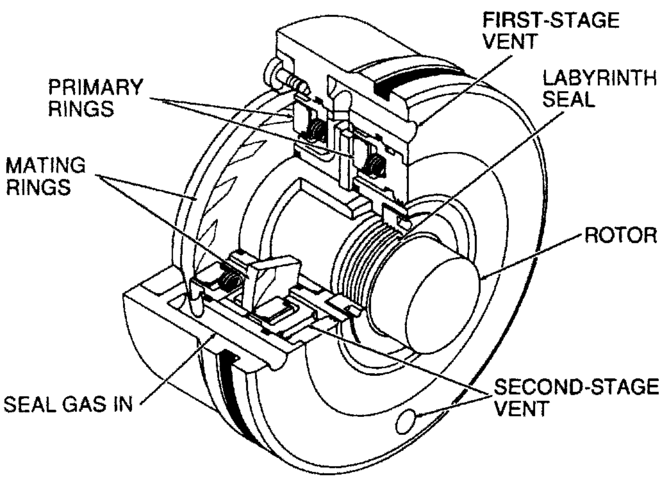


FIGURE 13.4 Dry seal cartridge. *(Revolve Technologies, Calgary, Alberta, Canada)*

Leakage volume is measured at this point because it provides a diagnostic measure of seal operation. An increase or decrease in leakage volume beyond predetermined levels results in an alarm or shutdown of the compressor. A second flowmeter gives a visual indication of leakage volume, enabling operators to gauge the cleanliness of the gas leaving the seal. Leakage volumes vary with size and speed but are typically below 80 L/min (3 scfm).

The seal can also be contaminated by oil from the adjacent bearing cavity. Nitrogen or instrument air is provided as a buffer to prevent oil contamination.

13.2.2 Operating Experience

Nova is one of the largest users of dry seals. From the initial pilot installation, the dry seal retrofit program has grown to include over 30 units, ranging in shaft size from 45 to 255 mm (1.75 to 10 in.), with pressures up to 10,880 kPa gauge (1600 psig) and speeds up to 27,060 rpm.

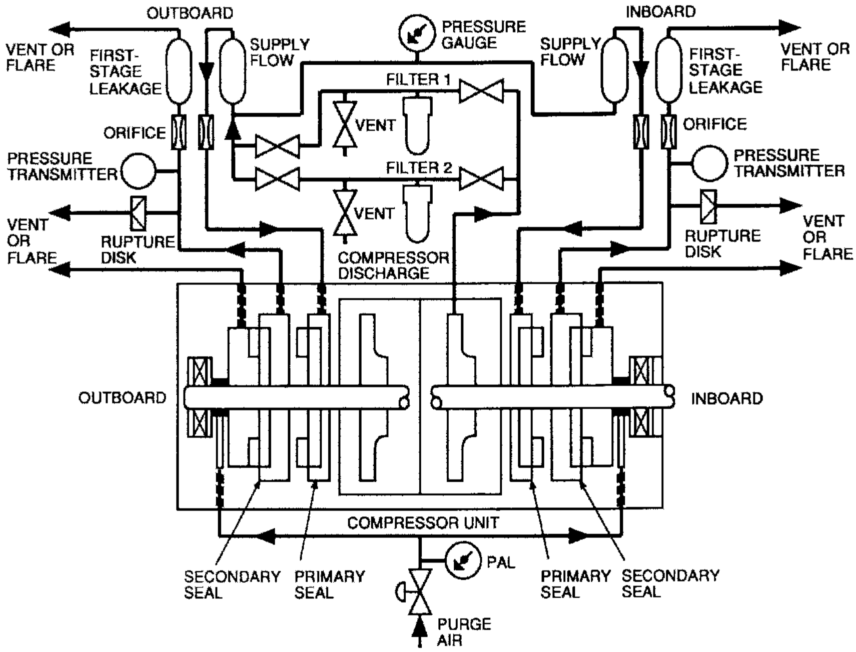


FIGURE 13.5 Dry seal monitor and control system. *(Revolve Technologies, Calgary, Alberta, Canada)*

The program was expanded to include the specification of dry seals on newly procured gas compressors. As of 2005, the total exceeded 100 units with dry seals. Nova's lead has been followed by numerous other companies, and dry gas seals should be considered for the majority of centrifugal compressors. With dry seals, gas compressor safety has increased, and operating and maintenance costs have decreased. Reliability is almost always better than that of oil systems. Dry seals are now an accepted design standard at many cost-conscious and reliability-minded user companies.

13.2.3 Problems and Solutions

Several technical problems were encountered over the course of the program. One problem that continues to occur intermittently is static pressurization of the compressor casing at the beginning of the startup cycle. Working with the seal vendors, users have minimized problems by changing the spring rate of the primary ring carrier springs and by improved quality control of dynamic O-ring and retainer.

In the past, seal gas contamination caused problems because of the extreme flatness tolerances required on the seal faces. Even minute particulates can damage the soft primary ring enough to disrupt seal film stability. Filtration system improvements, including a second coalescing filter upstream of the original, have resolved most of these problems. When a new compressor package is being commissioned, a separate source of seal gas, usually nitrogen, should be provided. This protects the seals from contamination from debris that may be present in recently constructed compressor systems or mainline pipe. Normally, such debris does not cause a problem beyond the first several hours of operation.

Another problem concerned the explosive decompression of O-rings. During operation at pipeline pressures of 4500 to 11,000 kPa gauge (650 to 1600 psig), gas becomes entrapped

FIGURE 13.6 Bidirectional dry seal faces. *(Revolve Technologies, Calgary, Alberta, Canada)*

in the elastomeric O-ring materials. Upon shutdown and compressor casing depressurization, this gas would blister or burst the O-rings as the gas attempted to escape. Using higher-density elastomers and changing the compressor control logic to depressurize only in the event of an emergency have largely corrected this problem.

13.2.4 Dry Seal Upgrade Developments

As early as 1986, Nova began working with a seal vendor on the design of a dual hard-face dry seal. Hard faces enable a face geometry to be devised that allows complete static separation of the faces, taking the noncontact concept one step further. Use of a hard silicon carbide primary ring provides this capability. It also enables dry seal technology to be used in applications as high as approximately 20,000 kPa gauge (2900 psig). User companies have been applying this type of seal since 1988 with very good results.

More recently, additional groove patterns have been used successfully. In 1991, the first bidirectional T-groove seals (Fig. 13.6) were installed. Unlike the spiral groove and other one-directional mating ring designs, the bidirectional seal can be installed on either end of the compressor, which reduces spares inventory requirements. Also, if the compressor rotates in reverse, the seal is not damaged. Operationally, this seal has proven to be as effective as spiral groove seals and offers substantially lower leakage (Fig. 13.7). Since most dry seals in natural gas pipeline applications vent their seal leakage to the atmosphere, this may be of interest in light of increasing attention to environmental emissions. As of 1994, Nova had over 30 compressors operating with bidirectional seals. By 2005 there were dozens more.

13.2.5 Dry Gas Seal Failures Avoided by Gas Conditioning*

Compressor seal statistics are important barometers of equipment reliability. Analysis of modern dry gas compressor seals received from the field for refurbishment validates that 90% of all seal failures result from a lack of clean and dry buffer gas. This high percentage

* Based on an article in *Hydrocarbon Processing*, courtesy of Joe Delrahim, John Crane Company, Morton Grove, Ill.

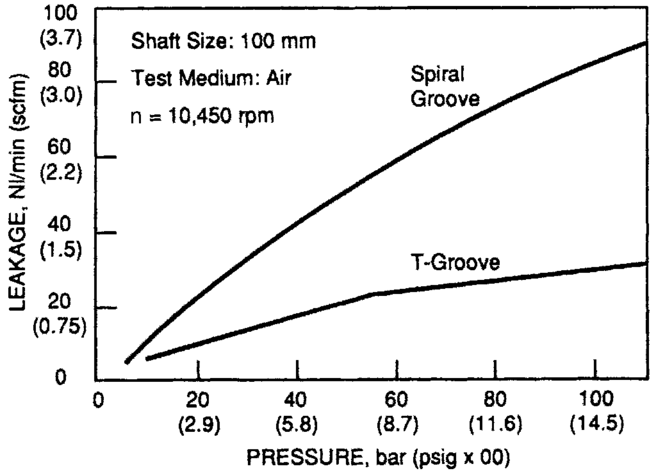


FIGURE 13.7 Groove shape and seal leakage rates. *(Revolve Technologies, Calgary, Alberta, Canada)*

is significant because most companies in the industry invest vast dollars and energy to monitor or record gas seal leakage flow, yet insufficient emphasis is generally placed on reliability assurance and prevention of problems in the first place. More often than not, seals fail because they are deprived of clean and dry buffer gas.

Experience shows that a continuous supply of clean and dry buffer gas is one of the most important requirements for trouble-free operation and long seal life. Unfortunately, this key requirement is often ignored throughout the various planning, commissioning, and operating processes. This shortcoming, particularly during the commissioning period, is almost guaranteed to result in multiple seal failures that cause operational loss and delays in plant startup.

As mentioned earlier, common control system designs for gas seals consist of filtration, regulation, and monitoring. However, although these control systems typically offer elaborate monitoring and regulation features, the filtration issue is often overlooked. In most cases, users and contractors initially choose standard filtration on virtually every application, regardless of gas composition and/or the presence of liquid or of condensation occurring in certain gas mixtures. One competent seal manufacturer's documented project histories show this shortcoming to be the cause of the majority of seal failures, particularly during the commissioning phase, but also during periods of normal operation.

Recognizing that most control systems feature inadequate, indeed elementary, filtration systems, the only obvious advantage of incorporating sophisticated monitoring devices is to indicate when a seal has failed or is about to fail. At best then, these monitoring systems serve to point to the cause of seal failures, yet do little, if anything, to prevent them.

Although in no way implying that dry gas seals are a bad choice for the overwhelming number of process compressors, getting to the heart of the problem should always begin with the proper analysis of mechanical failure. This analysis involves a review of gas composition, commissioning procedures, and control system design, as well as of the various interfaces between the seal, compressor, and seal control system. Further, it is essential to understand how the control system piping and instrument layout interact and function; this requires close examination of the associated flow schematic, which may (or may not) be similar to Fig. 13.5. Of equal importance is control-related and typically preestablished software, which must allow proper logic input to ensure the safe operation of the unit.

A commonly overlooked factor is determination of when condensation will result from a drop in pressure or temperature. For example, a Malaysian petroleum facility operated by a multinational oil producing and processing company experienced frequent seal failures. Upon investigation, personnel from maintenance and from control systems engineering, as well as other plant representatives, concluded that the malfunction was the result of condensate contamination. This condensate formed when buffer gas was being depressurized across the pressure regulator valve, which was located downstream of the filters. With this knowledge, the maintenance team decided to relocate the pressure regulator upstream of the filter. When they also decided to add a heater and insulated the entire line, the problem was solved permanently.

Based on this experience and on numerous requests from the field, engineering resources at John Crane Company in the early 2000s began to be dedicated to the development of gas conditioning units (GCUs). Designed with the intent of consistent delivery of clean, dry, properly pressurized gas to seals, well-engineered GCUs have greatly enhanced the reliability performance of dry gas seals by solving critically important gas supply issues. Unlike conventional gas panels that incorporate only coalescing filters, a modern GCU features a knock-out filter/coalescer vessel that removes solid particulates as well as free liquids and aerosols. A heater-controller also monitors and maintains gas temperature. Maintaining gas temperature above its dew point prevents condensation of aerosols in the process gas stream. Therefore, the collective features of successful GCUs must effectively manage liquids to ensure that the cleanest possible gas supply is always available.

In startup, slow-roll, and settling-out conditions, a thoroughly engineered GCU will maintain adequate gas flow using a seal gas pressure intensifier or similar device. Typically, a flow switch signals the intensifier control, which activates and deactivates the intensifier automatically as needed. The intensifier then provides sufficient seal gas flow to prevent unfiltered process gas from working its way back to the seal faces across the inboard labyrinth. Experienced machinery engineers and failure analysts know that clogging the seal face grooves will cause failures. With minimal customer interface connections and self-checking and self-regulating functions, modern GCUs meet the difficult sealing challenges faced by many industrial facilities.

Need for Training Let's face it: Another common source of problems is lack of training. For example, many maintenance technicians in plants with dry gas seals have never received essential training in seal operation and maintenance. Maintenance technicians familiar with conventional or "wet" seals are used to seeing flooded seal cavities with no consequences. In contrast, dry gas seals not only require no lubrication, but their support systems must be configured to keep liquids, including lube oil, away from seal faces.

A seal training program should not be triggered by mechanical failure, the delivery of new equipment, or a seal replacement occasion. These would also be the wrong times for operating and maintenance technicians to acquaint themselves with the compressor or seal operating manual. Rather, training programs should be arranged as part of supplier selection. The compressor and mechanical seal manufacturer, as well as the design engineering contractor, should offer training as part of their respective services. Also, it is essential to choose seal manufacturers capable of providing on-site technicians to help owners improve equipment reliability, mean time between change or repair, and overall plant productivity.

Case histories show that planning and training pay dividends. The staff of the PT Arun Natural Gas Liquefaction Plant in northern Sumatra, Indonesia, have experienced firsthand the benefits of component analysis and getting to the root of the problem. More than a decade ago, oil leakage into the process gas stream was adversely affecting heat exchanger performance and contaminated the liquefied natural gas (LNG).

The wet seals that PT Arun was using allowed seal oil to enter the process compressors, which disturbed the main heat exchangers used to liquefy feed gas that becomes LNG. PT Arun's engineering forces recognized that dry gas seals were the logical solution. Converting 10 compressor casings to dry gas seals at locations throughout the facility eliminated seal oil migration into the compressed process gas. At this facility, a number of dry gas seals have been in operation for more than a decade without problems. As part of a scheduled compressor inspection, a 10-year-old seal was found to show little wear and tear. This success was attributed to extensive planning and training of personnel prior to implementing the sealing solution.

Minimizing the Risk of Sealing Problems In summary, the key to minimizing seal failure is a thorough review of the components of the plant and their impact on the total system. In order of importance, the following factors should be considered in examining dry seal support systems for centrifugal compressors:

- *Gas composition.* Understanding the actual gas composition and true operating condition is essential, yet often overlooked. For example, it is necessary to understand when and where phase changes and condensation will result in the sealing fluid.
- *Commissioning procedures.* Is clean and dry buffer gas available? Is the seal protected from bearing oil? How is the compressor pressurized or depressurized? How is the machine brought up to operating speed? Are all personnel fully familiar with the compressor maintenance and operating manual? Is the full control system included and adequately described in these write-ups?
- *Control system design.* is clean and dry buffer gas available at all times? Key elements of the system design include buffer gas conditioning, filtration, regulation (flow vs. pressure), and monitoring. It is important not only to review the control system to ensure that problems don't arise, but also to gain an understanding of the system's design philosophy in order to gain an appreciation for the manner in which the logic is addressed. Further, it is crucial to review the buffer gas conditions. Is a heater required? What is the temperature setting for the heater? It is necessary to do some homework to find out the connecting piping structure, size, material, and type, to avoid liquid entrapment. Also, what components and systems are most suitable to withstand harsh outside environments?
- *Interface between the compressor, seals, and control system.* Review the startup and shutdown sequence, liquid removal if applicable, alarm settings, shutdown settings, and flow measurement units. Be sure to recognize signs of problems within the startup and shutdown points. Also, in terms of the logic and the interface, what control setting do you want? Remember that you cannot design one control system to fit every scenario.
- *Plant specifications, including tubing vs. piping, pipe sizing, logic system, and wiring diagrams.* Sometimes, the plant specification is totally different from a supplier's recommendation, but for good reason. For example, a plant may specify tubing instead of piping, or a different type of welding procedure; or, a supplier may recommend a shutdown on a specific setting, but the plant may opt for coordinated shutdown to avoid process upset.

13.3 MAGNETIC BEARINGS

13.3.1 Operating Principles

Magnetic bearings for gas compressor applications are used in both radial and axial configurations, performing the same tasks as their hydrodynamic counterparts. Each bearing consists of a rotor and stator, position sensors, and an electronic control system (Fig. 13.8).

The rotor of a radial magnetic bearing consists of a stack of circular laminations pressed onto a sleeve that can be fitted to the compressor shaft. Used to reduce eddy-current losses, these laminations are selected from a material with high magnetic permeability for higher magnetic flux conductance. The radial magnetic bearing stator is similar to that of an electric motor, with a stack of slotted laminations about which coils of wire are wound (Fig. 13.9). The stator is divided equally into four distinct electromagnetic quadrants, each with pairs of north and south poles. In horizontal rotor applications, quadrant centerlines are oriented at 45°

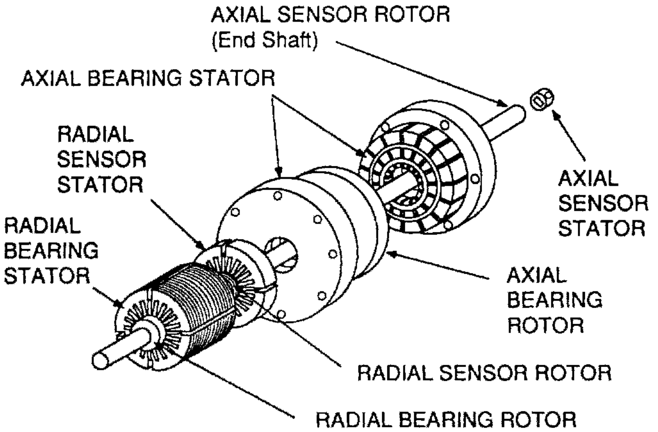


FIGURE 13.8 Magnetic bearing construction. *(Revolve Technologies, Calgary, Alberta, Canada)*

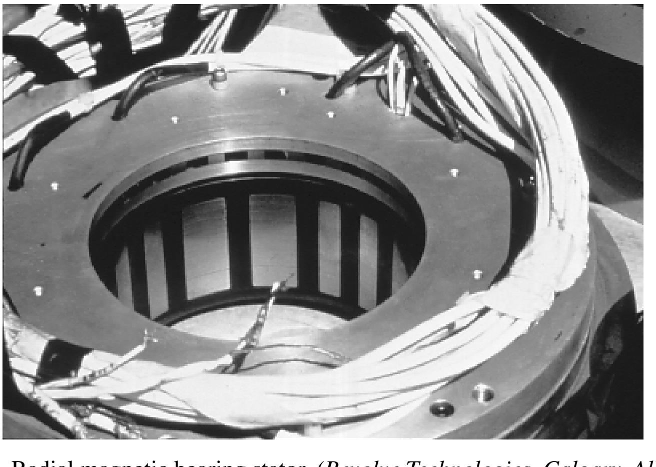


FIGURE 13.9 Radial magnetic bearing stator. *(Revolve Technologies, Calgary, Alberta, Canada)*

to the vertical, so that forces due to gravity are reacted by the upper two adjoining quadrants. This increases load capacity and stability.

The rotor of an axial magnetic bearing is a solid ferromagnetic disk secured to the compressor shaft. The axial bearing stator (Fig. 13.10) is made from solid steel wedges within which coils are wound in annular grooves to form electromagnetic windings. Laminations of highly permeable material are placed between the wedges to decrease eddy-current losses. By positioning a stator on both sides of the rotor disk, a double-acting thrust bearing is created.

The position sensors provide feedback to the control electronics on the exact position of the rotor. Among the well-proven sensor types, we find those that form an inductive bridge. As the air gap increases or decreases, the inductance varies. When the bearing rotor is centered, the position error signal is zero. A shift in the shaft location results in a corresponding change in inductance that alters the position error signal.

The position signal from the sensors is sent to the control electronics and compared to a reference signal that indicates where the compressor shaft should be. Any difference between these two signals generates an error signal. This error signal is processed by the control system. The output of the controls is used to vary the current supplied to the appropriate electromagnet via power amplifiers. The dc voltage for the amplifiers is stepped down for use in the control logic circuitry (Fig. 13.11) by dc–dc converters.

Also located in the control electronics is a monitoring and security system that can initiate alarms and shutdowns to protect the unit from damage. A battery backup system is provided to maintain operation in the event of electrical power failure.

The bearing rotor and stator surfaces are ground smooth to minimize mechanical runout and variation in forces. For this reason, it is important that these surfaces do not come into contact during operation or any other time. To prevent this, an auxiliary landing system is provided, consisting of rolling element bearings located in a removable bearing holder. The clearance between the shaft and auxiliary bearings is normally half the clearance between the magnet rotor and stator surfaces. When the system is deenergized (either in motion or at rest), the shaft coasts down on or remains at rest supported by the auxiliary bearings. Use of a removable sleeve fitted to the shaft, with predetermined radial and axial dimensions, provides a sacrificial means of maintaining the desired clearances. Auxiliary bearings are

FIGURE 13.10 Axial magnetic bearing stator. *(Revolve Technologies, Calgary, Alberta, Canada)*

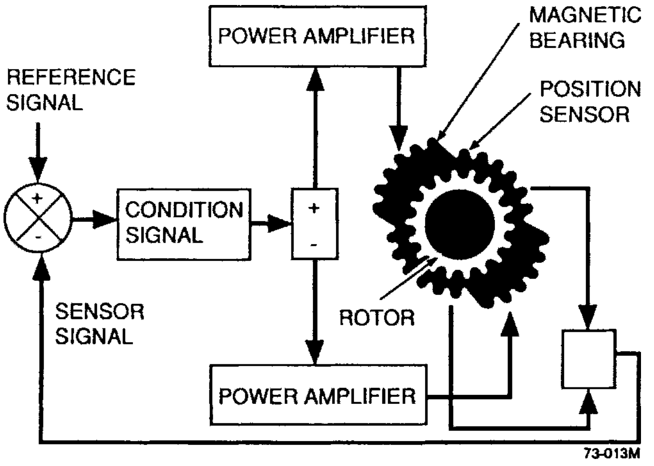


FIGURE 13.11 Control loop for magnetic bearings. *(Revolve Technologies, Calgary, Alberta, Canada)*

typically rated for three emergency coastdowns from full load and speed. Finally, purge air is supplied to the bearing cavities to meet electrical code requirements. Since instrument air is already available at virtually every compressor location, this requirement is satisfied easily.

Compressor startup begins with the supply of air to the bearing cavities, followed by static levitation of the shaft. The casing is then pressurized, and the unit valves that isolate the compressor from the piping system are opened. The driver start sequence then begins. Normal shutdowns involve closing of the unit valves upon detection of no compressor shaft rotation. The casing is left pressurized, but the shaft is delevitated after a predetermined period of time.

13.3.2 Operating Experience and Benefits

Magnetic bearings have been applied by a number of users, and their acceptance is clearly on the increase as run hours are accumulated. At the Alberta Gas Transmission Division of Nova, retrofit programs included three compressors that have the shaft supported by bearings on either side of the impeller (beam-type), one compressor that has the shaft supported by bearings on one side of the impeller (overhung-type), and one overhung power turbine. These installations include a range of shaft sizes from 100 to 175 mm (4 to 7 in.) at the journal diameter, weights from 320 to 1360 kg (700 to 3000 lb), and rotational speeds from 5000 to 11,500 rpm. Newly procured gas compressors were specified by Nova with magnetic bearings. As of 1994, more than 30 of these had been installed at that company. Other turbomachinery installations elsewhere exceeded a total of 200 machines.

The benefits of magnetic bearing systems include increased efficiency due to elimination of the parasitic shear losses associated with the oil system. Power consumed by the magnetic bearing system averages 3.5 kW (5 hp). This power represents losses in the bearing coil windings and control and amplifier electronics as the energy is transferred back and forth between the two. No other power is consumed. On a compressor or power turbine package with two magnetic bearing systems, the energy savings amount typically to just under 3% of package output power.

Another benefit is the increase in safety of the installation due to elimination of the oil system. Insurance company statistics indicate that the rotating equipment user industries, 80% of all equipment fires are oil-related.

The signals from the magnetic bearing system (current and position) provide a source of diagnostic information useful for machine condition and system operation monitoring. This is helpful in detecting negative trends and taking preventive action. Assignment of alarm and shutdown set points to these signals further improves the operating safety and risk of damage to a machine.

To date, conditions detected by magnetic bearing systems include improperly installed or damaged inlet and exit guide vanes, balance line blockage, and incorrectly sized balance pistons as well as buildup of debris on the rotating parts of the aero-assembly, causing unbalance. Many of these situations would remain hidden with hydrodynamic bearings, resulting in higher loads and shorter bearing life spans and, ultimately, would manifest themselves as bearing failures.

When evaluating the application of magnetic bearings, substantial credit may be given to significant reductions in weight and space. This may be of special importance in offshore installations using turboexpanders and canned motor centrifugal pumps. The latter would eliminate mechanical seals as well.

13.3.3 Problems and Solutions

Several technical problems encountered with magnetic bearings are more closely related to design and methods of manufacture than to the technology itself. One bearing rotor lamination failure caused damage to the radial bearings. This failure was traced to manufacturing procedures and material selection. After making appropriate modifications, there were no other failures associated with the bearing mechanical hardware itself.

Failures have occurred with commercially procured subcomponents in the control and power systems. These include problems with axial position sensors, dc–dc converters, and power amplifiers. After the vendor replaced the sensors and converters with improved designs, no failures occurred. Most amplifier problems are related to loose connections within the amplifier, and preventive tightening of connections corrects these.

Establishment of stable control loops spanning the range of disturbance frequencies encountered is achieved by means of physically changing electronic components on printed circuit boards. The time required to complete this activity cannot be predicted with great precision. Depending on operational constraints, this can be construed as a problem. Experienced companies that manufacture magnetic bearings, such as Revolve, have focused on this area for further development. The use of digital technology has made significant improvements possible.

13.4 DEVELOPMENT EFFORTS

Some of the areas where further development would improve industry acceptance of magnetic bearing technology include auxiliary landing systems, higher permeability of materials, sensor developments, standardized electronic hardware, and software tuning capabilities. The auxiliary landing systems used by Nova have typically included only rolling element bearings. However, these bearings are not specifically designed for this application and have only a limited life span in such demanding service. Alternative systems, developed without rolling elements, rely on the passive friction between the rotating element and the stationary element. Materials that have low coefficients of friction and that are sized to allow dissipation of the energy removed from the shaft in the form of heat have yielded encouraging results. To date, they have not been used on shafts within the heavier range of major compressors. Nevertheless, with careful attention applied throughout the design phase, such a landing system could be implemented successfully.

Materials with higher magnetic permeability would enable bearings to be made of smaller physical size. This has advantages not only in terms of overall physical size but also in minimizing the colocation distance between inductive sensors and the center of the load-bearing portion of the magnetic bearing. These considerations can point to certain rotor-dynamic constraints that make good designs more difficult to achieve.

The phenomenon of colocation involves physical separation of the point on the shaft where position is sensed and where the centerline of the reactive force is applied to the shaft. Depending on the rotor-dynamic characteristics of the shaft, the displacement of the shaft at the bearing and sensor locations may be opposite. A signal to correct the rotor position could actually cause the shaft to move farther away from the desired position rather than closer. As one of the major users of magnetic bearings, Nova has no evidence that this has been a problem in any of its magnetic bearing installations. The use of different types of sensors that are colocatable with the centerline of the load-bearing portion of the bearing is a recent development that will ensure that this does not become a cause for concern.

Standard electronic hardware would eliminate the need for spares specific to each installation. Currently, this is not the case with many of the systems in use, which are tuned by changing out hardware components on printed circuit boards.

Software-tuning capabilities would greatly reduce the time taken for hardware tuning of bearing systems. It would also make them more user-friendly and thereby effect greater industry acceptance of the technology. This feature involves the development of digital control systems that are responsive and robust enough to handle the envelope of loads imposed during compressor operation. Their introduction has been delayed by limitations in hardware speed, combined with the complexity of the necessary control algorithms.

A number of users, including Nova, have recently put into operation an enhanced digital magnetic bearing control system that not only incorporates standard electronic hardware and a more user-friendly interface but also takes better advantage of the diagnostic abilities of magnetic bearing technology.

13.4.1 Thrust-Reducing Seals

To take advantage of current dry seal and magnetic bearing technology, Revolve is focusing considerable attention on thrust-reducing seals for both overhung- and beam-type compressors. The first thrust-reducing seal for overhung compressors was installed in 1988, making possible the application of magnetic bearings in this type of compressor. Overhung compressors typically undergo much higher thrust loading upon startup than do beam-type compressors because their pressure forces are not balanced. Without the thrust-reducing seal, the size of the axial magnetic bearing required would have been prohibitive. The seal enabled reduction of the bearing to a size easily incorporated into the housing.

The compressor selected for this application, originally of radial inlet design, was retrofitted to an axial inlet configuration (Fig. 13.12). Basically, a tube encloses a volume projecting from the eye of the impeller, roughly equivalent to the cross section of the shaft. A dry seal located at the eye of the impeller is used to isolate this volume from the compressor process cavity (Fig. 13.13).

Thrust control begins on startup with the tube at atmospheric pressure. As head builds across the compressor and impeller, a net thrust results in the direction of the flow into the compressor, thereby loading the Z2 axial magnetic stator (Fig. 13.14). The increasing current signal is processed through an electropneumatic transducer that allows discharge pressure

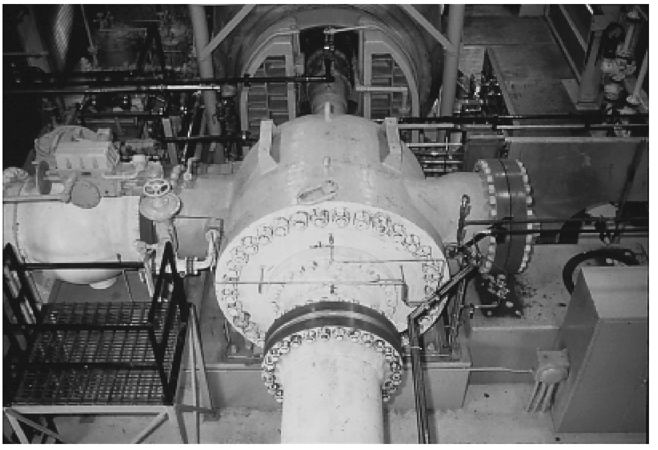


FIGURE 13.12 Axial inlet compressor being retrofitted with magnetic bearings. *(Revolve Technologies, Calgary, Alberta, Canada)*

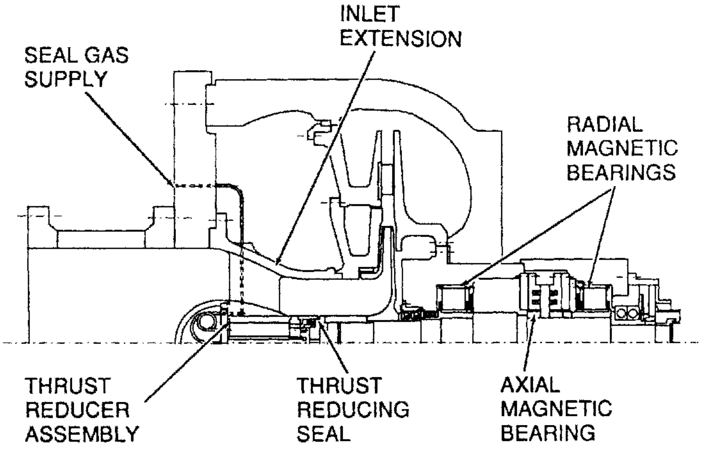


FIGURE 13.13 Axial inlet thrust reducer. *(Revolve Technologies, Calgary, Alberta, Canada)*

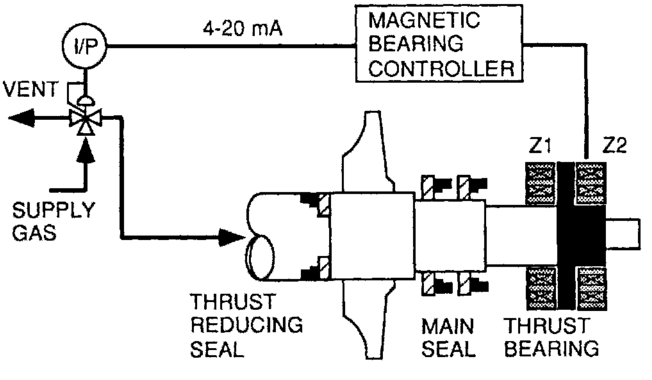


FIGURE 13.14 Control for an overhung thrust reducer. *(Revolve Technologies, Calgary, Alberta, Canada)*

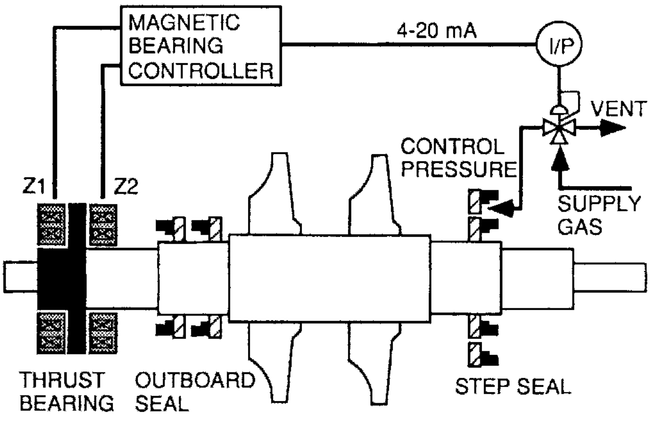


FIGURE 13.15 Control for a beam-type thrust reducer. *(Revolve Technologies, Calgary, Alberta, Canada)*

seal supply gas into the tube. This pressure increases, counteracting the thrust load on the shaft, and reduces axial bearing current until equilibrium is attained.

For beam-type compressors, a thrust-reducing seal was installed in 1991. This seal acts on the current signal from the active stator of the axial magnetic bearing (Fig. 13.15) in much the same way as does the system for overhung compressors. Seal stages of different diameter, combined with an active pressure regulator and discharge pressure seal supply gas, provide an envelope of variable axial load. This is used to counteract the net thrust on the shaft imposed by pressure differential across the impeller and gas momentum forces.

These loads are normally counteracted by a balance piston. As discussed in Section 12.5, a balance piston, or balance drum, is a cylinder of predetermined size fitted onto the shaft adjacent to the discharge side of the impeller. The pressure differential across the impeller and resulting net thrust is balanced by a reversal of the same pressures across this cylinder. This is done by fitting a labyrinth seal to the outside diameter of the cylinder and allowing a stream of gas from the discharge side of the impeller to flow across the seal and then through a *balance line* back to the suction side for recompression. Use of a thrust-unloading seal can minimize or completely eliminate the need for a balance piston leakage and thereby increase the overall efficiency of the compressor. Balance piston leakage can be as high as several percent of the flow through the compressor.

13.5 INTEGRATED DESIGNS

Magnetic bearing technology yields information previously unknown, such as the effect of internal aerodynamic design on bearing loads. The uncertainty associated with this design factor is one reason that magnetic bearing technology has not gained greater acceptance. By combining the operational knowledge gained to date with recent advances in numerical analysis hardware and software, a predictive model for bearing loads could be devised that would reduce or eliminate this uncertainty.

To date, the retrofit of these technologies has served to illustrate the practicality, robustness, and limitations of various configurations. This does not mean that the optimum has been reached for all factors affecting operational efficiency and serviceability. Thrust-reducing

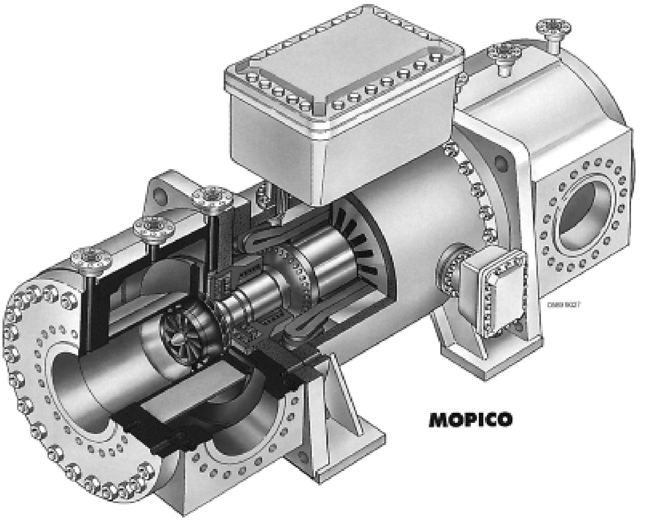


FIGURE 13.16 Sulzer Mopico motor pipeline compressor, incorporating magnetic bearings. *(Sulzer, Ltd., Winterthur, Switzerland)*

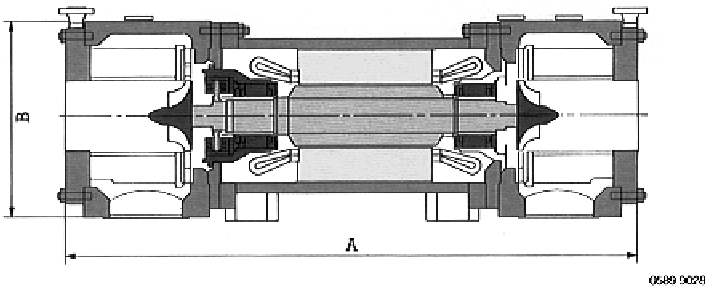
seals provide an excellent example. The value of magnetic bearing signals in active control loops governing other aspects of turbomachinery performance has been demonstrated with thrust-reducing seal applications. Of the many factors that influence turbomachinery design, Revolve and such large-scale users as Nova Corporation believe that these technologies can assist in attaining greater compressor efficiencies when incorporated at the conceptual stage.

Dry seal and magnetic bearing technologies offer significant advantages over the systems they replace. The vast majority of problems encountered have been solved and are not related to the technologies themselves. Recent advancements in magnetic bearing control technology and thrust-reducing seal applications take greater advantage of the benefits offered by these technologies. Where they have been incorporated into the design concept stage of turbomachinery development, they have proven to lead to interesting and advantageous designs.

Two of these advantageous designs are embodied in the Sulzer motor pipeline compressor (Mopico) and Sulzer-Acec high-speed oil-free intelligent motor (Hofim) compressors. The Mopico gas pipeline compressor features a high-speed two-pole squirrel-cage induction motor. Motor and compressor are housed in a hermetically sealed vertically split forged steel casing (Figs. 13.16 and 13.17). The center section contains the motor and bearings, and each of the end casing sections houses a compressor wheel, a fixed-vane diffusor, and inlet and discharge flanges.

Mopico compressors can be operated in series or parallel (Fig. 13.18). Magnetic radial bearings and a double-acting magnetic thrust bearing maintain the runner in position. The motor is cooled by gas metered from the high-pressure plenum of one of the compressor housings. Hence, the Mopico runs completely oil-free.

The speed and thus the discharge rate of the Mopico unit is controlled by a thyristorized variable-frequency drive. This drive was developed by Ross Hill Controls Corp. of Houston, Texas. It uses thyristors that can be switched out. These enable pulse-free run-up without current peaks and an operating speed range of 70 to about 105%.



Dimensions and weights

Frame size	Power kW	A mm	B mm	Weight kg
RM 28	2000	2500	900	9 000
RM 35	3500	2900	1000	11 000
RM 40	6000	3300	1200	13 000

FIGURE 13.17 Dimensions, weight, and simplified cross section of a Mopico compressor. *(Sulzer, Ltd., Winterthur, Switzerland)*

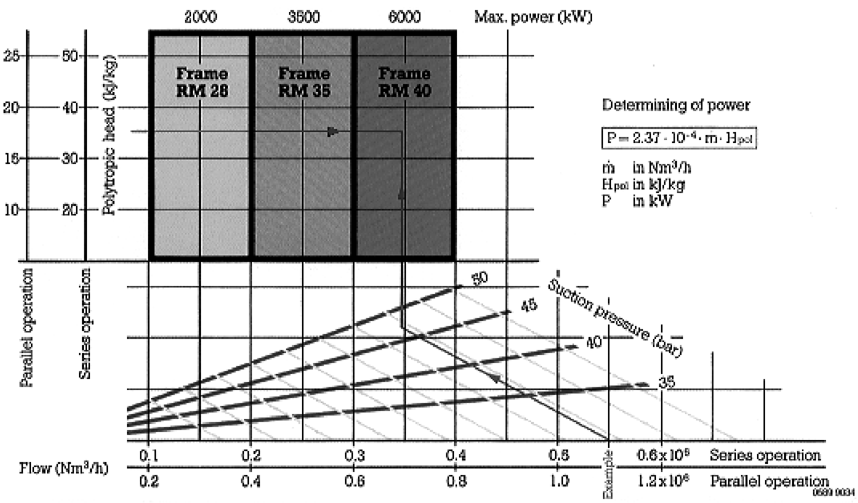


FIGURE 13.18 Application ranges for Mopico compressors. *(Sulzer, Ltd., Winterthur, Switzerland)*

Refer to Fig. 13.19 for an overall installation schematic and note that the following conditions can be complied with through the new combination of elements:

- Low installation, maintenance, and energy consumption costs
- Broad operating range at high economic performance
- Compatibility with existing compressors
- Unattended remote control

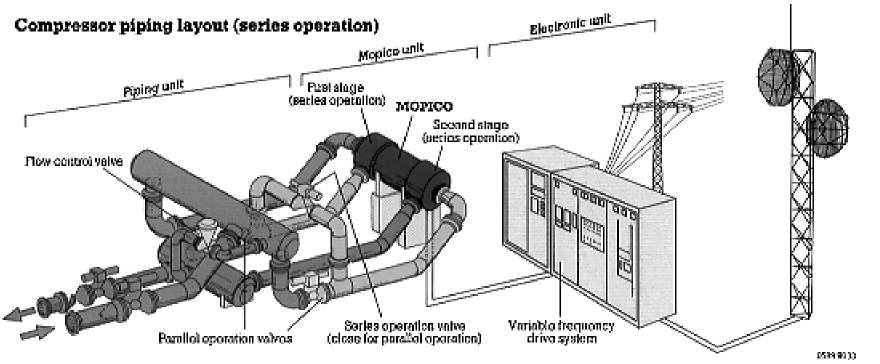


FIGURE 13.19 Installation schematic for a Mopico compressor. *(Sulzer, Ltd., Winterthur, Switzerland)*

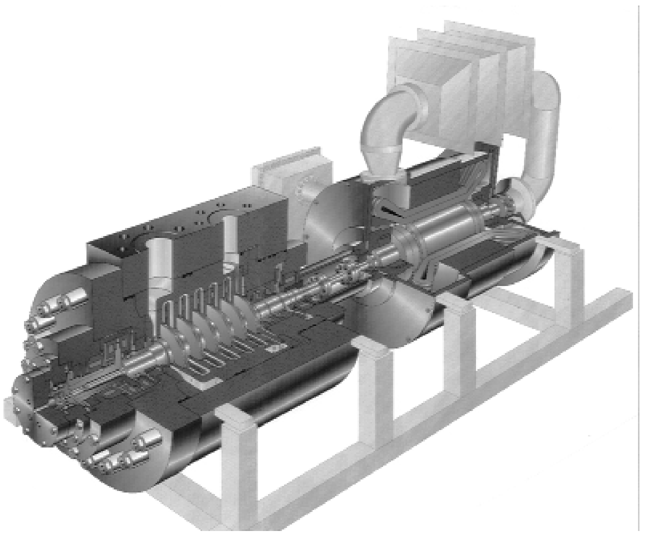


FIGURE 13.20 Hofim high-speed oil-free intelligent motor compressor. *(Sulzer, Ltd., Winterthur, Switzerland)*

- Emissionless and oil-free
- Possibility of outdoor installation

Based on cost per installed horsepower, the cost of the Mopico compressor is only about two-thirds that of a gas turbine unit and less than half that of a low-speed reciprocating compressor. It is some 10% less than that of a conventional centrifugal compressor with dry seals, magnetic bearings, and a direct-drive high-speed induction motor with variable-frequency drive.

On the other hand, a conventional centrifugal compressor with motor gear drive and variable inlet guide vanes is less expensive. This system is, however, unacceptable for pipeline application because of poor efficiency under low-pressure-ratio conditions.

Sulzer-Acec's Hofim is an equally promising new design concept. This high-speed oil-free intelligent motor compressor is shown in Fig. 13.20. It features separate motor and compressor directly coupled, with both machines supported on magnetic bearings. The prototype application is in a natural gas storage facility, for which the key parameters are:

Motor nominal speed	20,000 rpm
Motor nominal power	2000 kW
Compressor inlet pressure	5000 kPa
Compressor discharge pressure	15,500 kPa
Compressor speed range	70–102%
Compressor flow	38,000 $N \cdot m^3/h$

The motor is an asynchronous squirrel-cage induction machine driven by a solid-state variable-frequency drive. The compressor is a six-stage barrel machine of fully modular design. Dry gas seals are used to minimize internal and external leakage. An active balance system controls residual thrust of the entire unit to a level compatible with the capacity of the axial magnetic bearing. The unit was developed jointly by four companies in conjunction with the European Brite program.

13.6 FLUID-INDUCED INSTABILITY AND EXTERNALLY PRESSURIZED BEARINGS*

13.6.1 Instability Considerations

In the 1950–1990 period the Bently-Nevada Company played a role of unparalleled importance in understanding and monitoring machinery vibration. In the late 1990s and after the company became part of the General Electric Company, Donald Bently devoted more time to the development of externally pressurized bearings. Externally pressurized bearing technology solves an environmental problem with a simple, reliable, and highly efficient bearing that can actually improve the rotordynamic performance of many types of high- and low-speed machinery. Although used previously only on hydro turbines, this technology holds real potential for use in water-injected twin-screw compressors and possibly, certain types of centrifugal process gas compressors. This is why the reader needs to become acquainted with the concept.

13.6.2 Fluid-Induced Instability

Fluid-induced instability can occur whenever a fluid, either liquid or gas, is trapped in a gap between two concentric cylinders, one of which is rotating relative to the other. This situation exists when any part of a rotor is completely surrounded by fluid trapped between the rotor and the stator: for example, in fully lubricated (360° lubricated) fluid-film bearings, around impellers in pumps, or in seals. Fluid-induced instability typically manifests itself as large-amplitude, usually subsynchronous vibration of a rotor. This vibratory excursion can cause rotor-to-stator rubs on seals, bearings, impellers, or other rotor and stator parts. The vibration can also produce large-amplitude alternating stresses in the rotor, creating a fatigue environment that can result in a shaft crack. Fluid-induced instability is a potentially damaging operating condition that must be avoided.

During the 1980s, Don Bently and Agnes Muszynska showed that whirl and whip fluid-induced instabilities were actually manifestations of the same phenomenon, not separate

* Contributed by Don Bently and Carlo Luri, Bently Pressurized Bearing Company, www.bpb-co.com.

malfunctions, as previously believed. This groundbreaking work, modeling the two malfunctions in a single harmonized modern algorithm, was summarized in the April 1989 issue of the Bently-Nevada publication *Orbit* [1].

Fluid-induced instability can originate in bearings or seals, but occurs most often in fluid-film bearings. It appears suddenly and without warning as the rotor speed reaches a particular threshold speed, which the Bently Pressurized Bearing Company appropriately calls the *Bently–Muszynska threshold of instability.* Through rotor stability analysis, we can obtain a very useful expression for the threshold of instability, Ω_{th}:

$$\Omega_{\text{th}} = \frac{1}{\lambda}\sqrt{\frac{K}{M}} \tag{13.1}$$

where λ is the fluid circumferential *average* velocity ratio, a measure of fluid circulation around the rotor; *K* is the rotor system spring stiffness; and *M* is the rotor system mass.

There is an important point regarding this equation: If the rotor speed is less than Ω_{th}, the rotor system will be stable. Or to look at it another way, if Ω_{th} is above the operating speed, the rotor system will be stable. Thus, to ensure rotor stability, all we have to do is keep the threshold of instability above the highest anticipated machine operating speed.

Rotor dynamic stability analysis itself is a separate topic, best performed using a technique called *root locus analysis*. Pioneered by Walter Evans, a brilliant control systems engineer, root locus represents one of the most important contributions ever introduced to the field of control. It is covered in Evans's own text [2] as well as in numerous other modern texts [3–5].

Externally pressurized bearings counteract fluid-induced bearing instability. Also, pressurized water bearings are ideal for certain hydraulic turbines and hold great promise for water-injected screw compressors. Whereas oil whirl can often be handled by bearing stiffness modifications, the same fix may not work for its cousin, oil whip. Oil whip–related instabilities cannot generally be addressed by bearing stiffness modifications, but must be addressed by rotor stiffness changes instead. In this segment of the book, we relate Bently's approach to using externally pressurized bearings as midspan seals to effectively eliminate whip. With externally pressurized bearings, there is no longer any reason for a machine to suffer from either whirl or whip.

Controlling Lambda One way that the threshold of instability is commonly raised is to reduce λ. It can be seen from Eq. (13.1) that if we reduce λ, we will increase Ω_{th}. The fluid circumferential average velocity ratio λ is a measure of the amount of fluid circulation in the bearing or seal. Hence, it can be influenced by the geometry of the bearing or seal, the rate of end leakage out of the bearing or seal, the eccentricity ratio of the rotor in the bearing or seal, and the presence of any pre- or antiswirling that may exist in the fluid. Fluid-induced instability originating in fluid-film bearings is commonly controlled by bearing designs that break up circumferential flow. Examples of such bearings include tilting pad, lemon bore, elliptical, and pressure dam bearings. The value λ can also be reduced by antiswirl injection of fluid into the offending bearing or seal.

Controlling Stiffness Fluid-induced instability can also be eliminated by increasing the rotor system spring stiffness, *K*. Before we show how an externally pressurized bearing or seal can be used to control fluid-induced instability by increasing *K,* we need to discuss how the various sources of spring stiffness combine to produce the symptoms of whirl and whip.

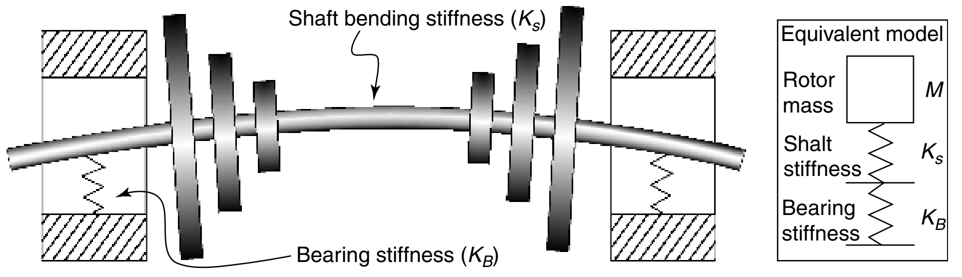


FIGURE 13.21 Series spring combination with mass.

Spring Model A flexible rotor can be thought of as a mass that is supported by a shaft spring, which in turn is supported by a bearing spring (Fig. 13.21). Thus, K actually consists of two springs in series, the shaft spring, K_S, and the bearing spring, K_B. For these two springs connected in series, the stiffness of the combination is given by these equivalent expressions:

$$K = \frac{1}{1/K_S + 1/K_B} = \frac{K_B}{1 + K_B/K_S} = \frac{K_S}{1 + K_S/K_B} \tag{13.2}$$

For any series combination of springs, the stiffness of the combination is always less than the stiffness of the weakest spring: The weak spring controls the combination stiffness. For example, assume that K_B is significantly smaller than K_S. Thus, K_S is much larger than K_B, and the term following the second equals sign can be used. As K_S becomes relatively large, K becomes approximately equal to K_B. For this case, the system stiffness, K, can never be higher than K_B; in practice, it will always be less. A similar argument can be used with the rightmost equation when K_B is relatively large compared to K_S; the system stiffness will always be lower than K_B.

13.6.3 Eccentricity and Stiffness

Let's assume that the source of the fluid-induced instability is a plain cylindrical, hydrodynamic bearing, an example of an *internally pressurized bearing*. Typically, when the journal is close to the center of the bearing (the eccentricity ratio is small), the bearing stiffness is much lower than the shaft stiffness. In that case, the ratio K_B/K_S is small, and the term following the second equals sign of Eq. (13.2) tells us that the combination stiffness is a little less than K_B. In other words, at low eccentricity ratios, the bearing stiffness is the weak stiffness and controls the combination stiffness.

On the other hand, when the journal is located relatively close to the bearing wall (the eccentricity ratio is near 1), the bearing stiffness is typically much higher than the rotor shaft stiffness. Because of this, the ratio K_S/K_B is small. Then the term following the third equals sign of Eq.(13.2) tells us that the combination stiffness is a little less than K_S. Thus, at high eccentricity ratios, the shaft stiffness is the weak stiffness and controls the combination stiffness.

Fluid-induced instability *whirl* begins with the rotor operating relatively close to the center of the bearing. The whirl vibration is usually associated with a rigid body mode of the rotor system (Fig. 13.22*a*). During whirl, the rotor system precesses at a natural frequency that is controlled by the softer bearing spring stiffness.

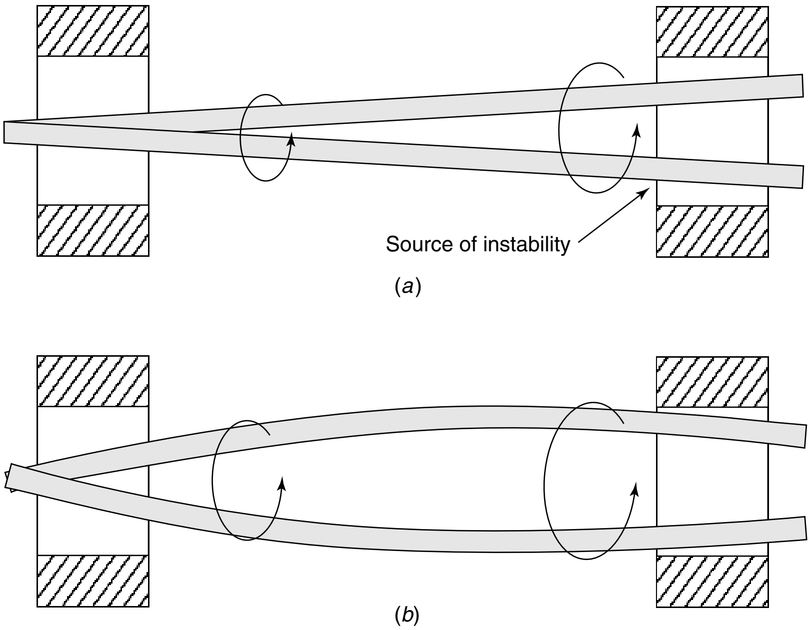


FIGURE 13.22 (*a*) Conical whirl mode shape; (*b*) bending whip mode shape.

Whip is an instability vibration that locks to a more or less constant frequency. The whip vibration is usually associated with a bending mode of the rotor system (Fig. 13.22*b*). In this situation, the journal operates at a high eccentricity ratio, and K_B is much higher than K_S. K_S is the weakest spring in the system and controls the natural frequency of the instability vibration.

To summarize, at low eccentricity ratios, the bearing stiffness controls the rotor system stiffness. Therefore, any changes in bearing stiffness will show up immediately as changes in the overall rotor system spring stiffness, K. On the other hand, at very high eccentricity ratios, the constant shaft stiffness is in control, and the overall rotor system spring stiffness will be approximately independent of changes in bearing stiffness.

13.6.4 Externally Pressurized Bearings and Seals

Conventional hydrodynamic bearings generate the rotor support force through the dynamic action of fluid drawn around by the rotation of the shaft journal. For that reason they are called internally pressurized bearings (Fig. 13.23). These bearings are normally designed to operate in a partially lubricated condition. They are vulnerable to fluid-induced instability problems because they can become fully lubricated if the shaft journal operates at a low eccentricity ratio, as can happen due to misalignment or an unanticipated high radial load.

Externally pressurized bearings operate in a fully lubricated condition by design. The spring stiffness of these bearings depends strongly on the pressure of the lubricating fluid supplied to the bearing. This pressure is generated by an external pump: hence, the name. By varying the pressure supplied to the bearing, it is possible to control the spring stiffness of the bearing, providing the possibility of variable stiffness control of a machine.

Seals can also act like bearings and have been responsible for triggering fluid-induced instabilities, even in machines supported by instability-resistant bearing designs, such as tilting

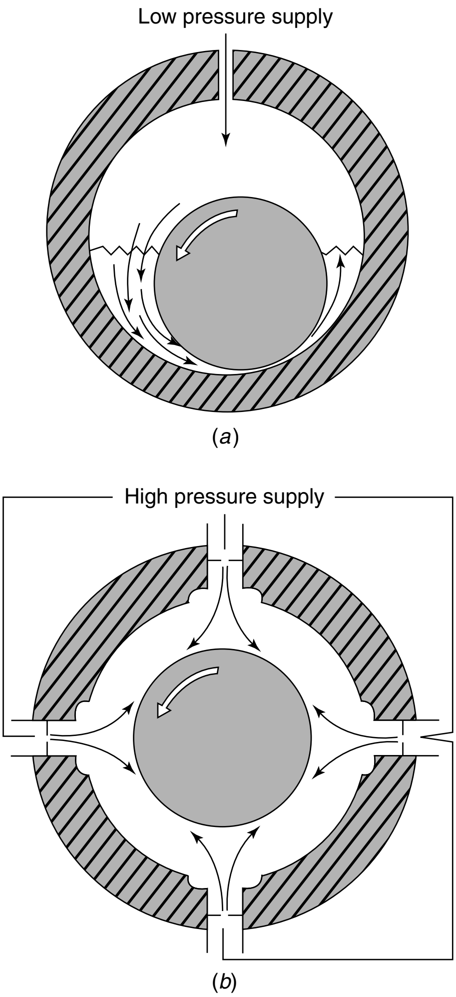


FIGURE 13.23 Cross sections of (*a*) hydrodynamic (internally pressurized) and (*b*) externally pressurized bearings.

pad bearings. Seals can also be externally pressurized with either gas or liquid, and they present the same possibilities for variable-stiffness operation.

Ironically, the use of external pressurization in bearings occurs worldwide, especially in Europe, as many large machines use jacking oil to lift the rotor during startup, before rotative speed can develop a self-sustaining oil wedge. Unfortunately, it is widely believed that external pressurization at operating speeds degrades, rather than enhances, rotor dynamic stability. For this reason, the *jacking oil* pressure is removed once the rotor reaches an appropriate rotative speed. As shown here, external pressurization of a properly designed bearing *enhances* stability and can *eliminate* whirl and whip. Thus, externally pressurized bearings and seals offer a new approach to the control of fluid-induced instability.

In *whirl*, the bearing stiffness is the weak stiffness of the system. We can increase the pressure of an externally pressurized bearing, increasing the bearing spring stiffness, K_B, and the system spring stiffness, K. The result is an increase in the threshold of instability, Ω_{th}. Thus, it is possible to design a rotor system using externally pressurized bearings that prevent fluid-induced instability whirl.

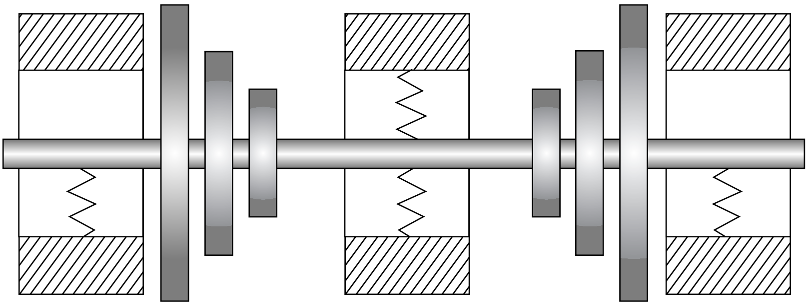


FIGURE 13.24 Side view of a rotor system with an additional spring at the midspan seal location.

In *whip*, the bearing stiffness is very high, and the shaft stiffness, K_S, is the weak spring in the system. In this situation, increasing the bearing pressure and stiffness *will have no effect* on the overall system spring stiffness, K. It is controlled by the shaft stiffness, which cannot be changed. However, we *can* add an additional spring to the system that acts *in parallel with* the shaft spring. This can be done by pressurizing a seal at or near the midspan of the rotor shaft (Fig. 13.24).

The total stiffness of two springs in parallel is simply the sum of the two stiffnesses. With an externally pressurized center seal with stiffness K_{seal}, Eq. (13.2) becomes

$$K = K_{\text{seal}} + \frac{1}{1/K_S + 1/K_B} = K_{\text{seal}} + \frac{K_S}{1 + K_S/K_B} \tag{13.3}$$

Thus, increasing K_{Seal} has the effect of increasing directly K and increasing the threshold of instability, Ω_{th}. Pressurizing the seal is equivalent to increasing the stiffness of the rotor shaft.

These two approaches to instability control follow the same basic principle: The stiffness of the weakest component is increased using externally pressurized bearing or seal technology. In whirl, the relatively weak bearing is stiffened; in whip, the relatively weak shaft is stiffened. Externally pressurized bearings and seals offer one additional technical advantage. The working fluid can be injected radially or tangentially. If fluid is injected tangentially against rotation (antiswirl injection), λ is reduced. This has a direct and positive effect on rotor system stability. Antiswirl injection is easy to design into an externally pressurized application.

By applying all of these approaches in the design of a machine, it is relatively easy to produce a machine that is immune to fluid-induced instability problems. This has been demonstrated both in the lab and in public venues by Bently Rotor Dynamics Research Corporation. Through the use of externally pressurized bearings and seals, both whirl and whip can be eliminated *without* introducing the disadvantages inherent in other commonly applied technologies or approaches, such as tilting pad bearings, intentional misalignment (including gravity), and sleeve bearing variations, such as tapered land, lemon bore, pressure dam, and others. These approaches introduce other problems (moving parts to wear out, greater mechanical losses, increased machine stress), and none are able to address whip instabilities. In addition, externally pressurized bearing and/seal technology provides advantages far beyond the elimination of whirl and whip and can replace other bearing types, such as magnetic and rolling element, overcoming their numerous disadvantages. It can be used to provide a variable-stiffness machine, easily adjusted in the field manually or using automatic controls it can be used with a variety of working fluids (gas or liquid), allowing "oil-free" operation, sometimes with the process gas itself; it allows greater use of vertical machine designs rather than primary reliance on horizontal designs, which use gravity as a stabilizing preload; and it can be used at very slow rotational speeds since it does not depend on rotation to develop a supporting fluid wedge. These observations allow us to consider practical applications next.

13.6.5 Practical Applications

Low-pressure oil-lubricated hydrodynamic bearings were specified 100 years ago as original equipment on Southern California Edison's hydro turbines. Although low-pressure oil bearings have been adequate for this and many other installations, the advent of stricter environmental regulations and the cost of safety, health, and environmental compliance make continued use of these bearings in any hydro turbine application a risky proposition. Recent improvements have allowed water-lubricated externally pressurized bearings to replace oil-lubricated hydrodynamic (internally pressurized) bearings in many applications. For the water power industry, the primary advantage of externally pressurized bearings is the option to use an environmentally benign fluid such as water for lubrication without loss of load-carrying capacity, efficiency, or reliability. A properly engineered externally pressurized bearing also results in better stability and control of the vibration response to dynamic forces acting on the rotor.

Hydrostatic lubrication applied to journal bearings can support load even with little or no relative motion between the rotating and stationary parts of the bearing because it uses external pressure to form the supporting layer of fluid. A modern example is high-pressure jacking oil systems (usually, Vickers pumps) used to reduce wear during the slow-speed startup phase of large turbines. Hydrostatic journal bearings are fully lubricated around 360° of the bearing surface and tend to operate at relatively low eccentricity (near the center of the bearing clearance).

As technology advanced and rotating speeds increased in the early twentieth century, many machines exhibited unstable behavior and high vibration. Experiments showed that flooding simple sleeve bearings with lubricant caused the bearings to exhibit instabilities. These fluid instabilities became known as whirl and whip phenomena. Whirl and whip instabilities are characterized by high subsynchronous vibrations commonly occurring at a frequency just below 50% of running speed.

Since fluid instability was associated with flooded bearings, it was erroneously assumed that a fully lubricated hydrostatic bearing would be inherently unstable. To improve stability, bearings were operated partially lubricated at low pressure. High-pressure hydrostatic lubrication was avoided for all but the slowest-speed applications. Over time, several features, such as the axial groove, elliptical, offset half, pressure dam, and tilt pad, were added to the plain hydrodynamic journal bearing to raise the instability threshold. These features helped but were never able to fully solve the problem of fluid instability.

13.6.6 Rotor Model, Dynamic Stiffness, and Fluid Instability

Rotor systems can be modeled as spring systems using *Hooke's law,* which states that the static displacement of a spring is directly proportional to the force applied ($F = Kd$). Dynamic rotating systems are a little more complex than simple static spring systems, where the stiffness is represented by a single spring constant (K). On a rotating shaft where the rotor mass is supported by fluid bearings, the radial motion can be represented by

$$r = \frac{\Sigma \mathrm{F}}{\mathrm{DS}} \tag{13.4}$$

where r = radial displacement or vibration vector
ΣF = sum of the force vectors
DS = dynamic stiffness vector

Because we are operating in a rotating system, we define the quantities in polar coordinates. Each term can be described by a vector with magnitude r and phase angle δ. If we trace the value of r over time, we obtain an orbit plot that represents the path that the shaft centerline takes through space. In practice, the orbit can be monitored using two orthogonally mounted proximity probes near the shaft surface. A Keyphasor signal gives us a once-per-turn reference point that can be superimposed on the orbit to give us important information about the phase angle of the vibration vector.

The sum of the force vectors includes all of the forces acting on the rotor system. These forces can be static (unchanging in direction and time) or dynamic (exhibiting changes in magnitude or direction with time). Common examples of forces on rotor systems are gravity (static force) and unbalance (dynamic force).

The dynamic stiffness is derived from *Newton's second law*: The sum of forces acting on a body equals the mass times the acceleration. By making certain assumptions (the rotor system parameters are isotropic, gyroscopic and fluidic inertial effects can be ignored, and linearity), we can create a simplified equation of motion based on a free-body diagram (Fig. 13.25). The free-body diagram shows a rotor with mass M rotating at a speed Ω in a direction X to Y. The displacement from equilibrium is represented by the term r, velocity by the term $\dot{r}$ and acceleration by the term $\ddot{r}$. The spring term F_S points back toward the equilibrium position, the damping term F_D acts opposite to the velocity vector and the tangential stiffness force F_T acts at 90° from the displacement vector, $\mathbf{r}$, in the direction of rotation. The perturbation

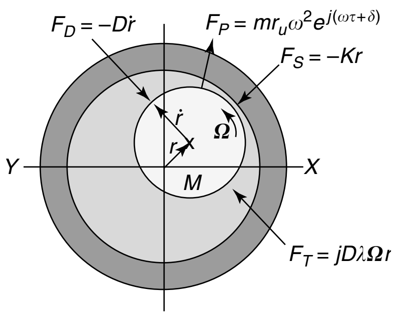


FIGURE 13.25 Free-body diagram for a simple rotor system.

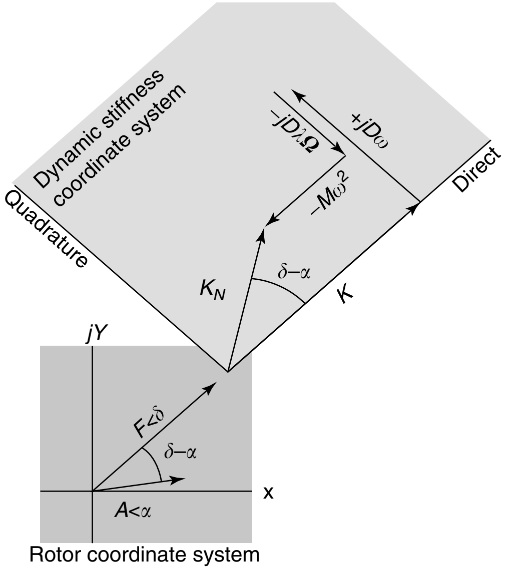


FIGURE 13.26 Representation of dynamic stiffness using vectors.

force F_P is shown acting at angular position δ at frequency ω. The term mr_u represents the unbalanced mass (m) operating at distance (r_u) from the center of the rotor. Newton's second law sets the sum of these forces equal to the mass M times the acceleration. Rearranging this equation to solve for dynamic stiffness results in the following:

$$\mathrm{DS} = K - M\omega^2 + jD\omega - jD\lambda\Omega \tag{13.5}$$

As shown in Fig. 13.26, dynamic stiffness can be viewed as the sum of four vectors. The four components of dynamic stiffness can be described in physical terms. The first term, K, is the simple spring constant. The positive value of K indicates that it acts opposite to the direction of applied force. The second term, $-M\omega^2$, is the mass stiffness that occurs because of the inertia of the rotor. The fact that this term is negative indicates that it has a destabilizing effect on the rotor system. Both of these terms are known as *direct terms* because they act parallel to the direction of applied force. The third and fourth terms used in this equation are known as *quadrature terms*. These terms are preceded by the symbol j to indicate that they act at 90° to the direction of applied force. The term $jD\omega$ is the fluid damping term. The fluid damping term is positive, indicating that it has a stabilizing effect on the rotor. The tangential stiffness term, $-jD\lambda\Omega$, acts opposite to the damping term and is potentially destabilizing. The tangential stiffness term is proportional to the fluid circumferential average angular velocity $\lambda\Omega$.

The tangential stiffness term introduces to our rotor model the term λ (lambda), which describes the fluid circulation around the circumference of the bearing journal. Whenever a viscous fluid is contained between two surfaces moving at different velocities, the fluid will be dragged into relative motion. Because of friction, the relative velocity of the fluid at the surfaces will be zero. Because the surfaces are moving at different rates, the fluid will develop a velocity profile similar to the one represented in Fig. 13.27.

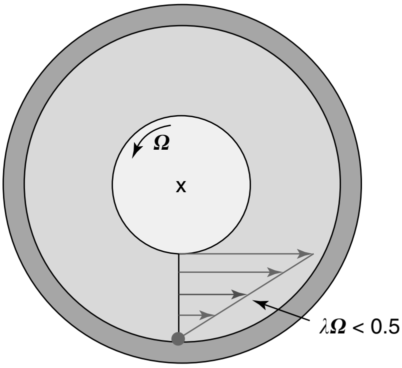


FIGURE 13.27 Average fluid angular velocity represented by $\lambda\Omega$.

Returning to the equation for radial position of our spring–mass–damper system,

$$r = \frac{\Sigma F}{\mathrm{DS}} \tag{13.4}$$

the maximum displacement will occur when the value of dynamic stiffness is small for any nonzero value of applied force. *Instability*, defined as the threshold where r becomes bounded only by the mechanical constraints of the system and nonlinear effects, occurs when the value of the dynamic stiffness in our model equals zero. Because the dynamic stiffness consists of both direct and quadrature terms, this condition is satisfied when $K = \mathrm{M}\omega^2$ and $\omega = \lambda\Omega$. Combining these two conditions results in the Bently–Muszynska formula for the threshold of instability:

$$\Omega_{\mathrm{th}} = \frac{1}{\lambda}\sqrt{\frac{K}{M}} \tag{13.6}$$

This simple equation provides an excellent starting point for comparing the stability of a variety of fluid bearing designs. As mentioned earlier, the Bently–Muszynska model predicts that the stiffer the bearing support and the lower the value of λ, the higher the threshold of instability or stability margin.

13.6.7 Root Locus Stability Analysis

The best way to evaluate the stability of rotor systems and bearing designs is to use the graphical technique known as the *root locus method*, which graphs the roots of the following characteristic equation of the rotor system:

$$M\ddot{r} + D\dot{r} + (K - jD\lambda\Omega)r = 0 \tag{13.7}$$

The solution of the characteristic equation takes the following exponential form:

$$r = Re^{st} \tag{13.8}$$

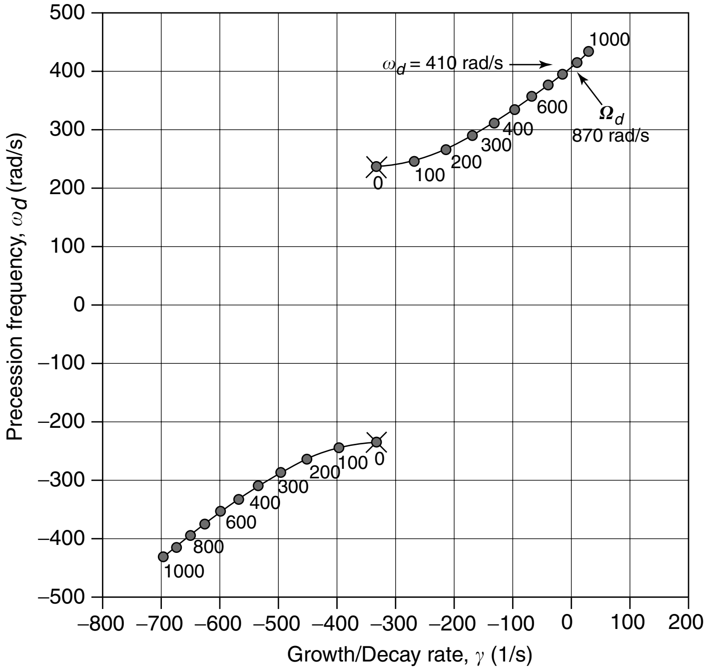


FIGURE 13.28 Root locus plot showing the forward and reverse precession roots.

Solving for s when R, λ, and Ω are nonzero, we obtain two complex roots:

$$S_1 = r_1 + j\omega_d \quad \text{and} \quad S_2 = r_2 - j\omega_d \tag{13.9}$$

Because the solution for r (displacement) has an exponential form, values will grow unbounded with time when γ is positive and decay when γ is negative. Negative γ roots indicate that the rotor system is stable and will return to an equilibrium position when disturbed. Positive γ indicates instability; no real machine can operate in the region of instability. When γ is equal to zero, r will remain constant. This represents behavior at the threshold of instability. Figure 13.28 shows a typical root locus plot over a range of operating speeds from 0 to 1000 rad/s. Based on this plot, the threshold of instability is predicted to be 870 rad/s. Root locus plots can be used to evaluate the effect on stability for a wide range of machine parameters, making it an extremely useful tool for rotordynamic analysis.

13.6.8 More About Externally Pressurized Bearings

In contrast with hydrodynamic bearings, which rely on the relative motion between the rotating journal and stationary bearing to create a pressure wedge to support the load of the shaft, externally pressurized bearings (EPBs) rely on an externally generated source of pressure. Therefore, they have several unique features and advantages that make them superior to low-pressure hydrodynamic journal bearings and even the latest modified geometry bearings, such as the lobed and tilt-pad bearings.

The distance to the bearing wall is measured by the eccentricity ratio (0 represents a centered journal, 1 a journal touching the bearing wall). As shown in Fig. 13.29, a hydrodynamic

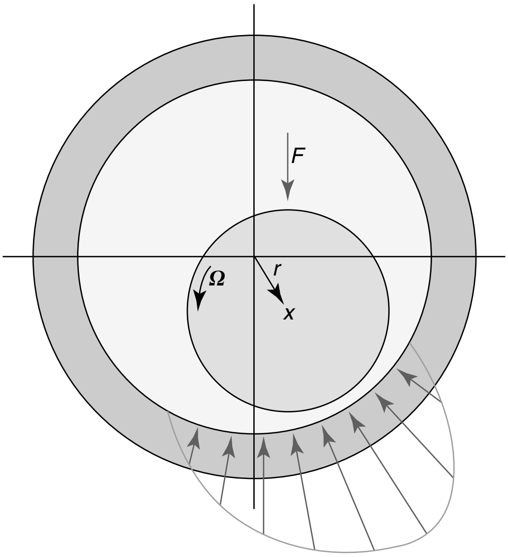


FIGURE 13.29 Pressure wedge supports the load of a hydrodynamic bearing operating at high eccentricity.

sleeve bearing must operate off center to create the pressure wedge that supports the bearing load. The stiffness of the hydrodynamic bearing is lowest when the eccentricity ratio is near zero. Misalignment during installation or changing radial loads can result in a hydrodynamic bearing that is forced to operate at low eccentricity ratio and low stiffness. Since we showed earlier that the threshold of instability and the vibration response are related to stiffness, a normally stable hydrodynamic bearing can have problems when operating at a low eccentricity ratio. A partially lubricated hydrodynamic bearing (which would have a low λ value) can become fully flooded at low eccentricity ratio, resulting in an increase in λ.

Because its stiffness derives from external pressure, an EPB can retain good stiffness and damping properties even at a low eccentricity ratio. The relationship between eccentricity ratio and bearing stiffness for several bearing types is shown in Fig. 13.30. Although both the EPB and the hydrodynamic bearings have good relative stiffness at high eccentricity, only the EPB retains its stiffness at zero eccentricity. A pure hydrostatic bearing operating at very slow speed has good stiffness at zero eccentricity but does not have the advantage of hydrodynamic lubrication at high eccentricity.

The superior stiffness value of an EPB makes it less prone to instability even when subject to misalignment, changing radial loads, or operation at low eccentricity. This feature makes the EPB ideal for installation on the vertical machines used in many large hydro installations. In vertical hydro turbines, the lack of gravity side load makes it possible for the hydrodynamic guide bearing journals to be forced to operate at low eccentricity. This is often the case during transient loading conditions that occur during startup or shutdown.

Figure 13.31 shows the typical pressure profile of a four-pocket EPB design. The pressure profile around the EPB differs from that of the low-pressure bearing because it acts around 360° of the circumference. The unique feature of the EPB is the pressure profile that acts along the longitudinal axis of the bearing. This axial pressure gradient drives flow out

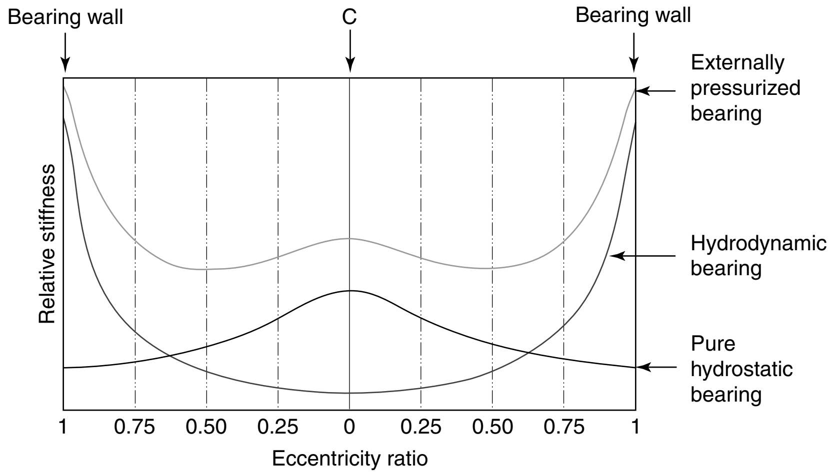


FIGURE 13.30 Relative stiffness as a function of the eccentricity ratio.

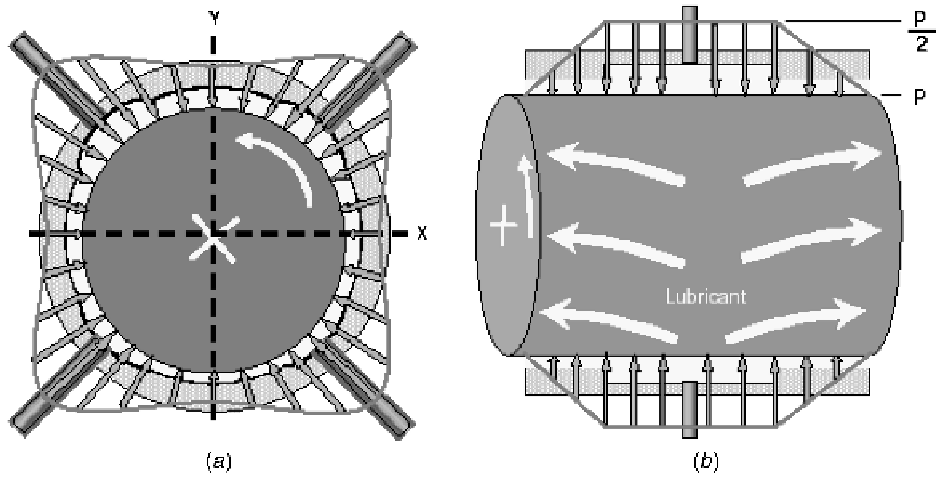


FIGURE 13.31 Pressure profile of a four-pocket externally pressurized bearing around the circumference (end view, *a*) and along the longitudinal axis *(b). p/2* is the supply pressure and *p is the ambient pressure at the exit or 0 psig*.

of the end boundaries of the bearing geometry, effectively reducing λ (the average circumferential velocity ratio).

External pressurization controls stiffness and λ, the two parameters that contribute directly to the stability of the bearing. Not only are these two parameters controllable during the bearing design process but they are also controllable by the operator online. The properties of the bearing can be changed after installation by changing the external supply pressure (stiffness and λ) or the supply temperature (damping).

Almost all types of hydrodynamic bearings in use today rely on petroleum-based lubricating oils. Physical properties of lubricating oils facilitate formation of the pressure wedge

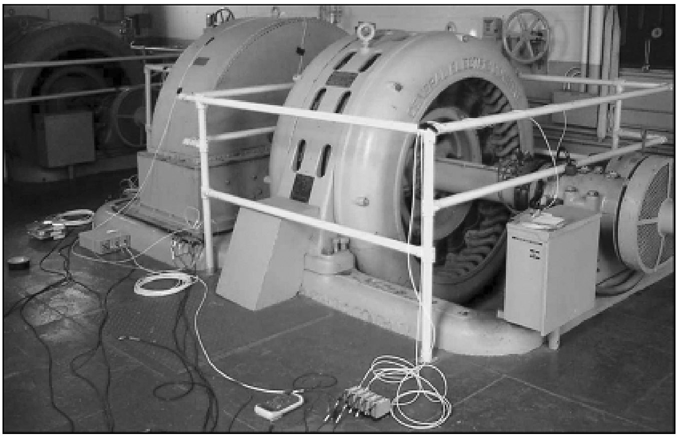


FIGURE 13.32 Pelton wheel (left) and generator (right) instrumented with temporary proximity probes for diagnostic analysis.

that supports the load in an internally pressurized bearing. Water has poorer lubricant properties and does not work well in hydrodynamic bearings.

An EPB can operate with conventional petroleum lubricants but can also operate with other incompressible fluids, such as water. Water can be used to support load in the EPB because the internal pressure profile is dependent on an external pressure source and not primarily on the relative motion between bearing and journal. The EPB makes it possible to choose a bearing fluid not solely on its lubricant properties but also for its compatibility with the process and the environment.

In hydropower applications, two convenient sources of pressurized water can be used to supply the EPB. If a clean water source with sufficient head is available from the penstock, it may be used to supply the EPB. Typically, filtration in the range 3 to 5 μm is necessary to protect the close tolerances of the EPB. The pressure necessary for EPB operation must be calculated by the bearing designer on a case-by-case basis. Typically, the pressure needed is in the range 150 to 1000 psi, or between 350 and 2300 ft of static head.

If a suitable source of water is not available from the environment, pressurized water can be produced by a closed-loop fluid delivery system. This system is composed of several components, typically a high-pressure pump, a high-pressure filtration unit, and a low-pressure pump and heat exchanger on a kidney loop to control the fluid temperature. Sensors are employed to monitor the fluid temperature, pressure, and flow. Controls and interlocks can be employed to ensure that the bearing is operating at the desired design conditions. A pressurized accumulator is generally employed to allow for controlled shutdown of the machine should the system for any reason experience a sudden loss of pressure.

13.6.9 Field Data Collection

To obtain baseline data for the hydrodynamic oil bearings, field data were collected from a hydro turbine at Southern California Edison's Lytle Creek generating facility. A typical machine train is composed of two major components: a Pelton waterwheel and a General Electric three-phase generator (Fig. 13.32). The generator and Pelton wheel are mounted on a common horizontal shaft supported by three bearings. Bearing 1 is located outboard of the Pelton wheel, bearing 2 is located between the Pelton wheel and the generator, and bearing 3 is located outboard of the generator.

The bearings appear to be those supplied with the original equipment. They utilized soft metal babbitt faces with diagonal grooves to help distribute the oil. Soft metal is used to avoid damaging the shaft on startup and shutdown. A leather slinger was employed on the shaft to contain the oil in the bearing pedestal. Based on nameplate information, it appears that the Pelton wheel and generator were manufactured in 1909.

A common occurrence on these types of machines is oil leakage from the bearing pedestals. This problem is prevalent at bearing 2, which is in close proximity to the Pelton wheel (Fig. 13.33). The housing of the Pelton wheel operates under slight vacuum, causing oil to be sucked into the water discharge. Bearing 2 also carries most of the load of the machine. For these reasons, it was decided to replace the hydrodynamic oil bearing in the middle of the machine with an externally pressurized water bearing.

Because of the historical nature of these machines, Southern California Edison requested that no external modifications be made to the bearing pedestals. The EPB retrofit, which consisted of new bearings and "backers", was designed to fit inside the original pedestal, with small penetrations for pressurized water delivery and return.

The properties of the original oil bearing are as follows:

- Length 18 in.
- Diameter 6 in.
- Bearing construction Babbitt-lined steel with diagonal oil grooves

The field data collected from this machine revealed no surprises. The machine appeared to be well behaved and exhibited rotor dynamic data consistent with a heavy, slow-speed horizontal machine. Proximity probes mounted near the bearing housings and a once-per-turn

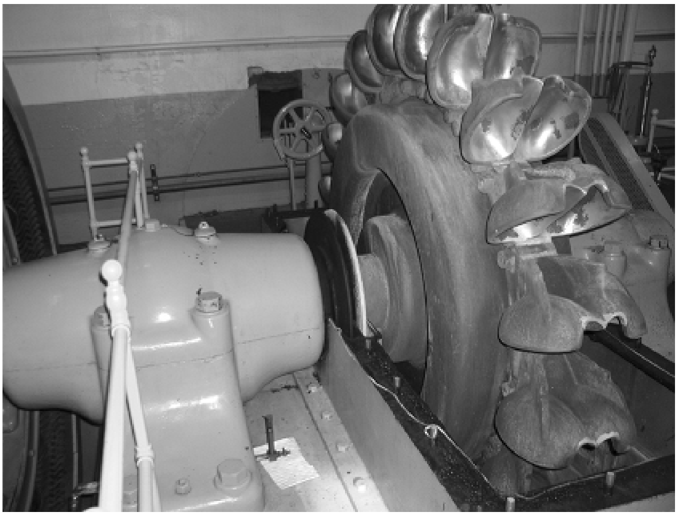


FIGURE 13.33 Center pedestal (bearing 2) in close proximity to the Pelton wheel (right).

Keyphasor signal were used to generate the orbit plot shown in Fig. 13.34. This plot shows the filtered 1X (synchronous) vibration, a nearly circular orbit with 0.002 in. (2-mil) peak-to-peak vibration. Field data from bearing 2 was difficult to interpret, due to axial movement over a taper in the shaft at the probe locations. Field data from bearing 1 were reported as representative of the shaft vibration.

The data plot shown in Fig. 13.35 is a transient data plot showing the average shaft centerline position from startup to operating speed. This reveals a moderate rise in the shaft centerline from -1.5 mil (-0.0015 in.) to $+\ 0.5$ mil (0.0005 in.) over a speed range consistent with hydrodynamic bearing performance.

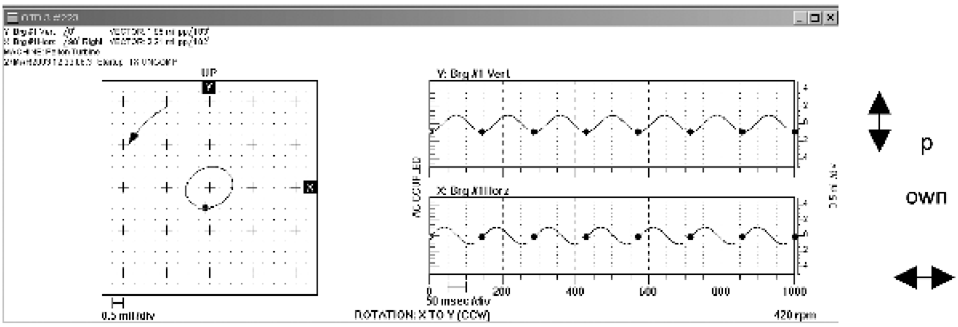


FIGURE 13.34 1X orbit plot from bearing 1 *(outboard of Pelton wheel).*

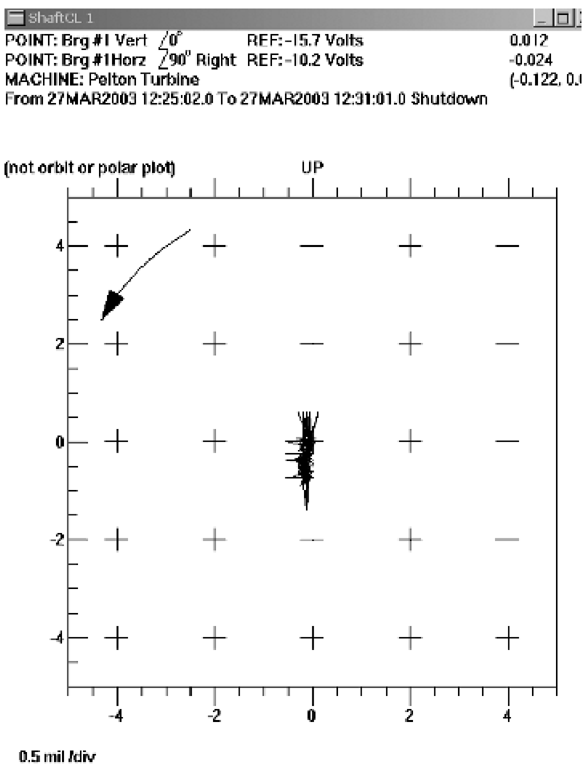


FIGURE 13.35 Average shaft centerline position of the original oil bearing.

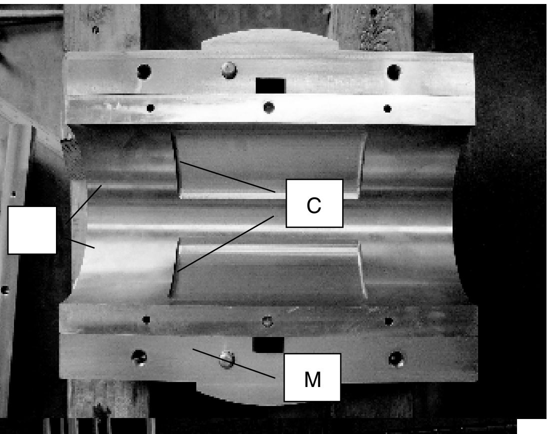
13.6.10 Test Stand Data

To validate the pressurized water bearing design, a nearly full-scale model of the turbine generator machine train was constructed in Minden, Nevada (Fig. 13.36). The original number 2 bearing pedestal from Southern California Edison's Mill Creek site was used to support a pressurized water bearing and replacement backer. The simulated machine train uses two rotating wheels, with a weight of 3300 lb each, to model the turbine and generator. The total rotating weight with shaft is close to 8000 lb. The shaft is driven by a variable-speed electric motor at 450 rpm. Two roller bearings are used on either side of the wheels to simulate bearings 1 and 3.

The test stand pressurized water bearing was designed to carry slightly more than half the total load. Pressurized water is generated by a closed-loop fluid delivery system that supplies 15-gpm flow at 700 psi. The properties of the pressurized water bearing are as follows:

- Length 14 in.
- Diameter 6 in.
- Bearing construction Bronze

(a)



(b)

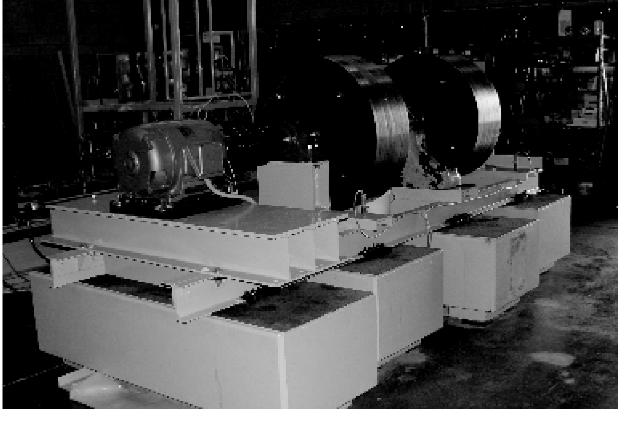


FIGURE 13.36 (*a*) Upper half of the pressurized water bearing (the lower half is similar). Note the hydrostatic pockets, orifice locations, and pressurized water manifold. (*b*) Bearing test stand in Minden, Nev.

Data collected from the test stand are shown in Figs. 13.37 and 13.38. The 1X filtered orbit shows a very small peak-to-peak vibration of about 1 mil (0.001 in. or 0.025 mm). About 4 lb (1.8 kg) of unbalance mass was added to the drive-end wheel diameter to obtain this vibration amplitude. Vibration without the unbalance mass (not shown) was almost undetectable. These data were used to calculate the stiffness value for the pressurized water bearing. The stiffness was estimated at 900,000 lb_f/in., significantly higher than the value of the original hydrodynamic oil bearing.

Similarly, the shaft centerline plots show very little change in position of the shaft centerline (shaft eccentricity) from startup through running speed. Analysis of the full-spectrum vibration plots (not shown) verifies that no abnormal subsynchronous vibrations are present in the operating data. The data collected confirm that an EPB can operate near zero eccentricity with excellent stiffness and no sign of fluid instability.

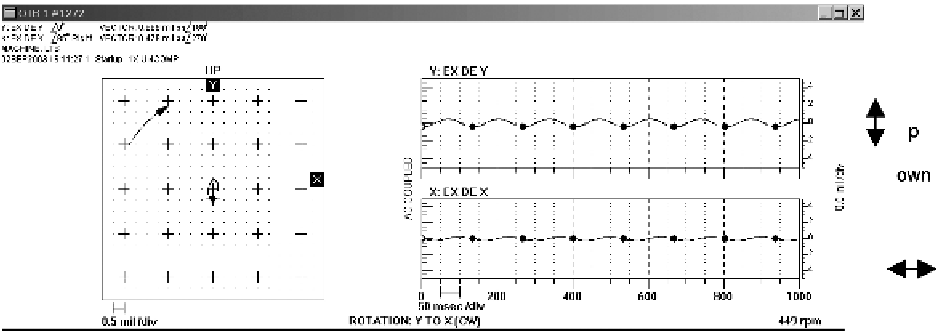


FIGURE 13.37 Shaft orbit at the drive-end (DE) side of the bearing 2 pedestal.

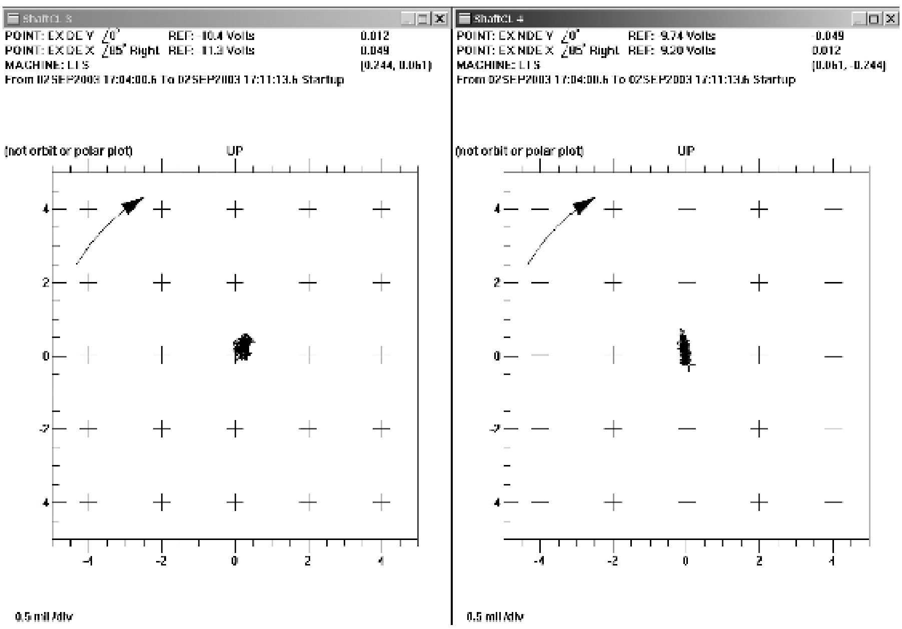


FIGURE 13.38 Shaft average centerline plot of test stand bearing 2.

13.6.11 Conclusions

Externally pressurized bearings solve the age-old problem of fluid bearing instability. Century-old principles of hydrostatic bearing lubrication are applied in a manner that enhances bearing stiffness and reduces λ. In doing so, the externally pressurized bearing can positively influence the two bearing factors that have the greatest impact on the stability of rotating machinery.

By solving the problem of fluid instability, machine designers and operators can apply the technology in numerous new and existing rotating machinery applications. It is thus evident that reliability-focused and environment-conscious users are in a position to take advantage of the numerous benefits the product has to offer. Among these benefits are the following possibilities:

- Using traditional hydrocarbon oils or alternative fluids such as water for bearing lubrication
- Excellent efficiency due to low frictional losses and stable operation, with very little average shaft centerline movement over the operating speed of the machine
- Low maintenance and excellent reliability by eliminating bearing wear on startup and shutdown

Although this overview highlights the application of the externally pressurized water bearing in a radial load application, the technology works equally well on vertical machines. It can be applied successfully to thrust bearings, resulting in similar benefits and advantages. This application of the technology would be ideal in the many vertical-configuration Francis turbines used worldwide. Many potential applications of pressurized bearing technology can be found in the power, petrochemical, and industrial machinery markets. The simple principles described in this section can be applied universally to high-speed gas turbines using compressed gas bearings or low-speed hydro turbines utilizing pressurized water or more traditional oil-lubricated bearing systems. We see a real possibility that water-injected twin-screw compressors will find increasing use in industry.

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