MAGNETIC BEARING OPERATING EXPERIENCE

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John Sears has over 20 years experience and training in the turbomachinery industry. He started with Nova Corporation of Alberta in 1971 and began a career with the operational and technical maintenance group in the maintenance and repair of their compressor station equipment. His focus with the new technologies, dry seals, and magnetic bearings, began in 1982. In 1987, he joined the Engineering group, Hardware Applications, which was responsible for the design and development of these products. He became the supervisor of the Project and Design group in 1989 and was responsible for the application and support of the dry gas seals and magnetic bearings throughout the Nova group of companies. He is currently the Director of Applications Engineering in Revolve Technologies Inc.

Stan Uptigrove graduated from the University of Calgary with a B.S. degree in Mechanical Engineering. He worked as a Project Engineer in the Gas Transmission Division of Nova Corporation of Alberta for eight years. For the last nine years, Mr. Uptigrove has been extensively involved in the design and installation of dry seal and magnetic bearing systems. Since the formation of Revolve Technologies Inc. in January 1993, he has been actively involved in the company's success with dry seal and magnetic bearing systems. As Director of Marketing for Revolve, he plays a key role in the company's management team and more specifically in the commercialization of these technologies. Mr. Uptigrove has also been very active as Chairman of the Revolve Conference (which highlights these technologies) in conjunction with the National Petroleum Show and the ASME, OMAE Division.

ABSTRACT

Over the past 10 years, magnetic bearings have been making significant changes to the rotating equipment industry. As magnetic bearing systems become more accepted and the numbers of applications increase, so do the operating experience and advancements. Revolve Technologies Inc. (Revolve), formerly the Hardware Applications Group within Nova Corporation of Alberta, has been involved in over 35 magnetic bearing applications from design and installation through operation and trouble shooting. The applications included the world's first compressor operating on magnetic bearings, along with the first power turbine and the first overhung compressor similarly equipped. The experience includes other applications of magnetic bearings including turbo expanders and pumps. Utilizing this background in the technology, three key areas of experience will be presented in order to assist future users and designers in understand the technology. The three areas are:

- Design and application concerns
- Operating experience
- Troubleshooting

Design and Applications Concerns

Starting with an overview of the different types of applications, some of the unique issues encountered are elaborated on. A few case studies are presented on design and application issues in order for others to learn from these experiences and avoid duplications of potential problems.

Operating Experience

Reliability and availability data from many of these applications is presented in detail. The experience gained in technology implementation is discussed, along with some points to ensure field acceptance. Understanding and feeling comfortable with any new technology by all parties involved is key to its acceptance. Recommendations on how this can be achieved are presented.

Troubleshooting

The different problems encountered in operation, and their resolutions, are reviewed. Many of these problems have been taken into account and eliminated in newer product designs, but as more manufacturers get into the business, these same problems should be understood so they may be avoided. Therefore, understanding these problems and their solutions is key to both the end users and the manufacturers, in order to obtain better reliabilities and for the technology to gain widespread acceptance.

DESIGN AND APPLICATION CONCERNS

Overview

Magnetic bearings have been applied to a number of different types of machinery ranging from turbomolecular pumps and high speed spindles to large centrifugal compressors. Small canned pumps, large centrifugal pumps, electric motors, large industrial fans, turboexpanders, turbines, and turbine engines have all been installed or tested with magnetic bearings. In turbomachinery, the two most common applications to date have been turboexpanders and centrifugal compressors. Applications are also rapidly growing in pumps due to changing environmental regulations, the advantages associated with eliminating bearing lubrication systems and in some cases, seal systems.

There have been many papers presented which explain the basic concepts of how magnetic bearings work [1, 2, 3, 4, 5], so this will not be covered herein. The basic components of a magnetic bearing system used in industrial turbomachinery applications will generally consist of the following; two radial bearings, one axial or thrust bearing, position sensors, a control system, and auxiliary systems that include a landing system, and purge system. Complex cooling systems for amplifiers should no longer be a concern, due to advancements in switching amplifier technology used in the magnetic bearing power system.

Magnetic Bearing Application and Design Advantages

A few different configurations of magnetic bearings are shown in Figures 1, 2, 3, 4, and 5 in different types of turbomachinery applications. These include an overhung compressor, a beam or between bearing compressor, a turboexpander, and a multistage pump. As with most applications to date, each of these applications have some unique design features which have added to the value of the magnetic bearing in the specific application.

A typical beam-type centrifugal gas compressor with the impeller between the two radial bearings is shown in Figure 1. The thrust bearing is located outboard of the two radial bearings. Position sensors for each radial magnetic bearing are located adjacent to the magnets. The auxilary landing system can also be seen and is used to support the rotor when the system is delevitated. The auxiliary bearing closest to the thrust bearing is also designed to support thrust loading.

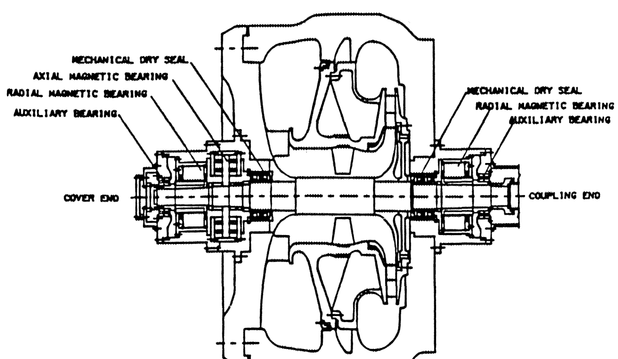

Figure 1. Beam Type Centrifugal Compressor.

An overhung centrifugal compressor is depicted in Figure 2. Overhung compressors in high pressure applications experience large thrust loads during pressurization and subsequent startup of the unit. This is caused by process gas pressure acting on the overhung end of the shaft and can result in extremely high forces for any type of thrust bearing. The configuration shown utililzes a single stage dry gas seal on the overhung end of the shaft as a thrust balancing device. A closed loop control system regulates gas pressure at the end of the shaft to minimize loads on the thrust bearing. It uses the magnetic bearing as the measuring device for the required pressure to maintain a given thrust load. For example, during startup, the control system would depressurize the overhung end of the shaft with the pressure increasing as the unit increases head. This design significantly reduced the maximum thrust force from over 40,000 lb to less than 8,000 lb.

The same idea of using dry gas seals in combination with the magnetic bearings to control thrust forces has also been applied

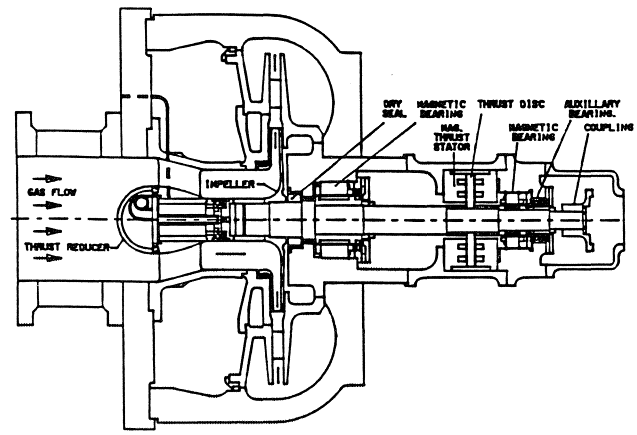

Figure 2. Overhung Centrifugal Compressor.

to beam-type compressors. This eliminates the need for a balance piston line and, thereby, increases the unit efficiency by eliminating the recirculation of gas through the balance piston. The layout of a unit is shown in Figure 3 with magnetic bearings and a balance seal. The control loop takes measurements from the thrust magnetic bearing and controls the inter stage pressure between the two dry seal stages which have different diameters. By eliminating the need for a balance piston line the overall length of the shaft can also be reduced.

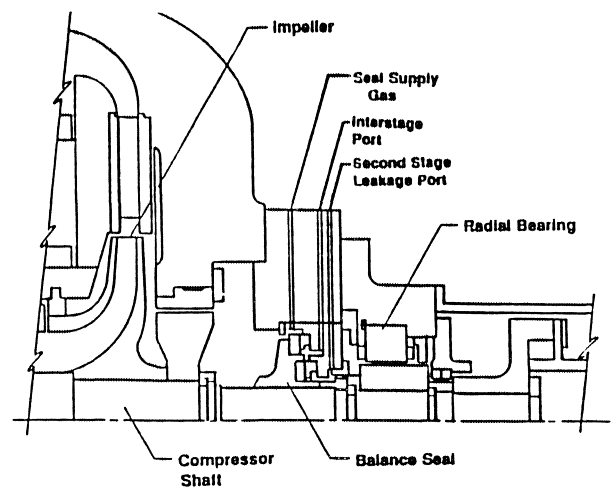

Figure 3. Thrust Reducing Balance Seal.

In turboexpander applications a substantial portion of the footprint is taken up by the oil systems. Removal of the oil system frees a large amount of space and eliminates a lot of complex and costly equipment. This makes the conversion to magnetic bearings very economically attractive and is one of the reasons why turboexpanders lead the way in terms of the largest number of industrial applications. A turboexpander application is shown in Figure 4 with magnetic bearings operating within the gas and, thus, no need for seals. Similar designs have also been used in electric motor driven centrifugal compressors where the motor is placed between the magnetic bearings and also operates in the gas stream [6].

Pump applications are also utilizing magnetic bearings to eliminate seals and, thereby, eliminate leakage. A multistage pump is shown in Figure 5 with magnetic bearings. In this case,

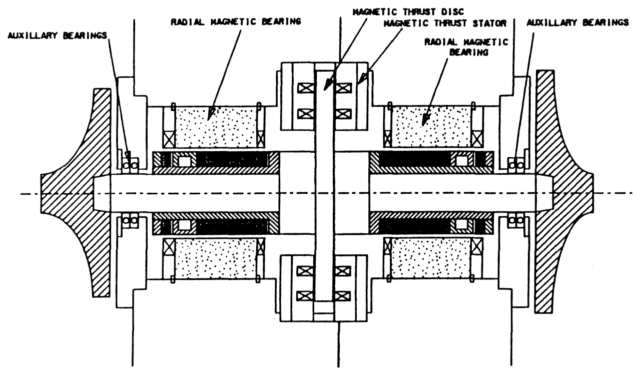

Figure 4. Turboexpander Schematic.

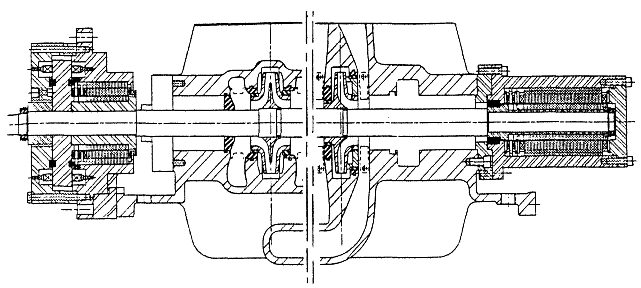

Figure 5. Multistage Pump with Magnetic Bearings.

only one end uses a canned magnetic bearing. The magnetic bearing operates through a stainless steel sleeve and is not in contact with the liquid. In this case, the other end of the shaft still has a seal. This seal has been eliminated on some designs by having the motor drive an intergal part of the canned system. This can be accomplished by designing the motor shaft with the same type of stainless steel sleeve.

These designs show only some of the unique concepts that are now being utilized through the application of magnetic bearings. There is no question that magnetic bearings are changing the way rotating equipment is designed, and also making significant improvements to efficiencies, wear, and environmental impact. With each new design or application, one must also be aware of the potential problems and take precautions to reduce or eliminate them.

Concerns and Potential Problems

The following areas describe some of the potential design issues and application concerns associated with magnetic bearings which, if properly designed, can be avoided. There are no clear cut answers which cover all cases but a good understanding of the different potential problems, and learning from past experiences will help avoid duplicated mistakes.

Bearing Locations

A number of factors are involved in evaluating a given layout. Considerations include space constraints, the bearing span and thrust bearing location inboard or outboard of the radial bearing. Magnetic bearings have a lower specific design load capacity than do hydrodynamic bearings (60 to 70 psi, *vs* 350 to 500 psi). Therefore, magnetic bearings occupy a larger amount of space to achieve the same load capabilities as do oil bearings. Care must be taken for proper integration of this increased size into turbomachinery designs. There is a trade off of rotordynamic considerations with the radial bearing span and the difficulty in removing and installing the thrust bearing (i.e., if the thrust is outboard of the radial bearing, it must be removed to get at the radial bearing. Yet, if it is placed inboard, it increases the radial bearing span and weight between the span). This is further complicated by the location of the backup bearings, which is described later.

To summarize, bearing load capacity, size, location, and ease of installation all have to be taken into account with careful and thorough evaluation of the rotordynamic impacts.

Radial Sensors

In a magnetic bearing, the location of the sensor is very important, since it is at this location where the motion of the shaft is measured and determines the corresponding shaft control or movement. If the difference in the signal from the sensor and the bearing is simply one of magnitude, as shown on the right of Figure 6, the effect is to alter the gain of the loop. This effects the stiffness and damping values of the system.

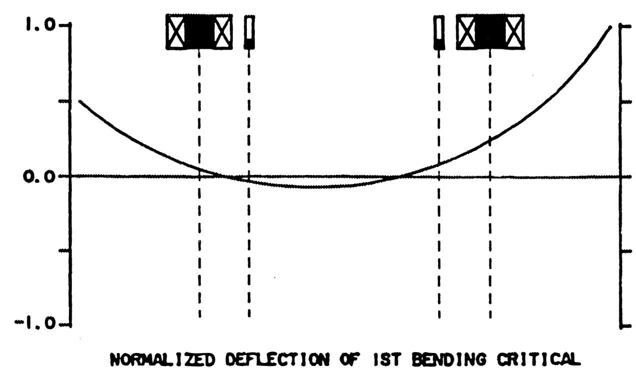

Figure 6. Bearing and Sensor Location.

A more severe problem can occur if there is a phase change between the bearing and sensor, as shown on the left of Figure 6. This can actually lead to negative stiffness and ultimately instability. This will occur if the nodal point is located between the bearing and the sensor. Instability will also occur if the nodal point is very near the sensor, since insufficient control will most likely occur.

Some flexibility is available in dealing with these scenarios. The sensor can be placed on either side of the bearing, provisions can be made to move it to either side in the design, or some designs use a sensor on each side of the bearing and average the two signals to establish what the motion is at the bearing. Other development work is also being pursued to develop a sensor that can operate in the magnetic field or to eliminate the need for the sensor by utilizing the bearing characteristics to determine where the shaft is. There is no substitute for good engineering design to ensure the sensor and bearings are in the best possible location for any given shaft layout.

Axial Sensor

With axial sensors, it is also good practice to have the sensor as close to the bearing as possible. The reasons in this case are more to do with thermal growth than the flexibility of the shaft. The problem is that the best place to place an axial sensor is generally at the center of the end of the shaft. This reduces any error from any end-face runout. If the axial sensor is located a substantial axial distance from the thrust bearing, concern should be raised over whether there will be any thermal differential between the shaft and the housing during operation (Figure 7).

There are a number of considerations that can affect thermal differential between the sensor and bearing. These include the process, axial, or radial bearing heating from windage, resis-

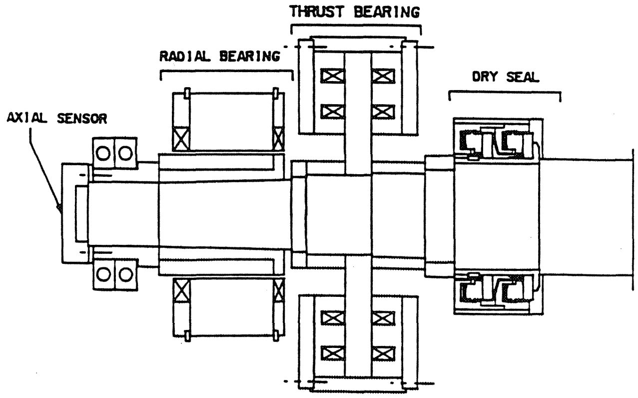

Figure 7. Axial Sensor Location Substancial Distance from Magnetic Thrust.

tance or eddy current losses, or even ambient conditions. This can result in a loss of clearance, risking rotor/stator contact and possibly causing failure while in operation.

As mentioned, the best solution to this is to locate the sensor as close to the bearing as possible. However, as this involves tradeoffs, compensation for the thermal growth can be provided in the machine buildup, so the machine runs in its proper position during operation. This will require offsetting the zero position an amount equal to the anticipated differential growth.

Auxiliary Landing System

Of all the mechanical components, it is the auxiliary landing system that requires the most frequent inspection, especially if there has been an emergency shutdown related to excessive rotor excursions. Should this occur, the auxiliary landing system may have been damaged, and it is advisable to check the clearance between this system and the shaft. This can de done electronically, and if it yields suspect results, then a mechanical inspection may be required. Therefore, it is best to have the auxiliary bearings as accessible as possible, which is best achieved if they are located outboard of the bearings. Rotordynamic constraints may prevent this from being the most attractive option.

In addition, similar concerns exist around the thermal growth issue described under the axial sensor section above. Again, the location of the axial sensor, the thrust bearing, and the auxiliary bearing must all be taken into account with the potential of thermal growth between them. If, for example, the position of the axial sensor and the thrust bearing are both inboard of the radial bearing and the auxiliary bearing is placed outboard, the thermal growth of the shaft must be taken into account. The auxiliary bearing can then be offset in the axial direction during installation to compensate for the thermal growth and thereby be centered during operations.

Angular contact rolling element bearings are the most common type of backup bearing. This is not to say that they are without problems. One advantage of the angular contact bearing over the radial contact bearing is the ability to apply a preload. Preload is the application of an axial load to the matched set, causing the rolling elements to ride up in the grooves slightly higher than nominally designed. This provides increased stiffness resulting in a greater ability to withstand shock loading in the axial direction and also increases the rolling resistance. This is important to reduce the chance of turning due to windage which will significantly increase the life of the rolling elements.

The auxiliary landing system is one area which still sees some operating problems due to high impact loading or contact during operation. Sleeve type backup bearings have also been tried and an area that could use some additional development work to optimize. It must also be emphasized that these bearings are only utilized when a system upset occurs, which causes the shaft loads to exceed the magnetic bearing capability, or loss of power to the magnetic bearings occur for any reason. Therefore, in steady state operating conditions, there should not be a problem with the auxiliary bearings. The two types of rundown configurations presently used are shown in Figure 8.

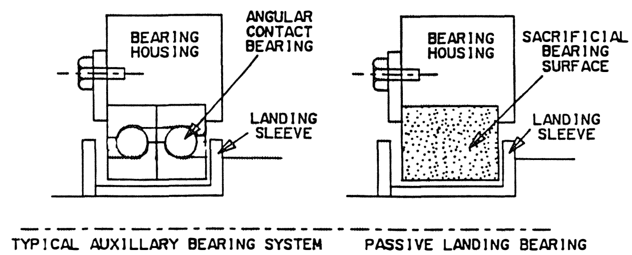

Figure 8. Auxilary Landing Bearings.

Radial Rotors

If possible, the outside diameter of the bearing rotor should be made smaller than all components located inboard, including sleeves, thrust discs, seals, impellers, etc. Then the bearing rotor would only need to be removed from the shaft if damaged. In reality this is not always possible. Therefore, the rotor should be designed in a manner such that removal is straight forward.

There is more to bearing rotor design than it would first appear, due to the unusual difficulties associated with a compressed stack of metal laminations. This stack must be kept sufficiently compressed to withstand the cyclic loads imposed during operation, which include centrifugal force from rotation, mechanical stress from the fluctuating magnetic field, plus thermal stress from eddy currents. In addition to this is the intermittent mechanical strain imposed during installation and removal. Over this range of loads, rotor surface runouts must be maintained within precision tolerances. Rotor design must adequately reflect all of these considerations. Some examples are shown in Figure 9 of radial rotor designs made to be either permanent or removable.

Load Capacity

One common issue encountered with new applications of magnetic bearings is a lack of detailed knowledge of all dynamic loads experienced by the machine. In some cases, the bearings were sized to handle 1.5 times the static load, which was more than sufficient to cover the loads imposed over the entire operating range. In other cases, this was insufficient.

One solution is to select a larger bearing with greater static load capacity. This has additional tradeoffs in the physical size of the bearing, required length of shaft and the dynamic response time. Other solutions are to establish and understand fully the reasons for the unknown static and/or dynamic loads and to modify the machine design to eliminate or reduce these loads. Utilizing the magnetic bearing as a design tool, the OEMs have an opportunity to design a much more stable and aero dynamically balanced machine.

Correct sizing of the bearings is obviously dependent upon knowledge of the load characteristics of the particular turbomachine in the intended application. Users are advised to be thorough in questioning this aspect of application design. Analytical methods of load prediction from process fluid turbulence are not always accurate; therefore, review of past experience,

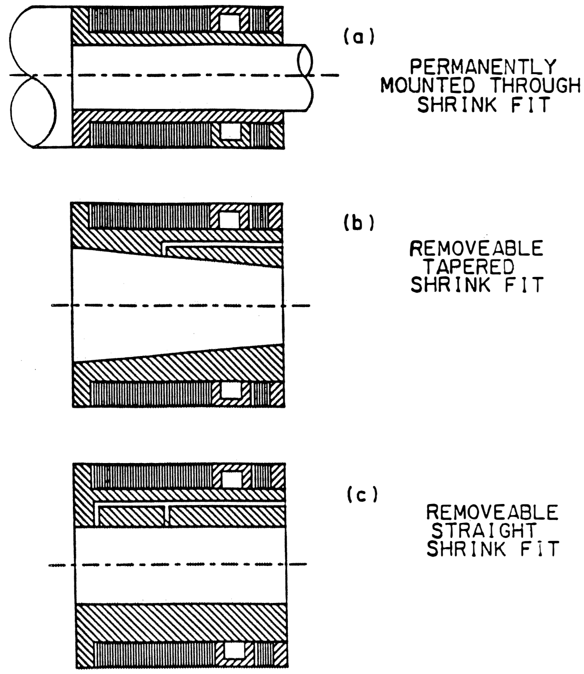

Figure 9. Radial Bearing Rotor Mounting.

possibly augmented by instrumentation of machines, may still be the best way to proceed at this point.

OPERATING EXPERIENCE

It is safe to say that many first and second time users of magnetic bearings have had very good experience with very few problems and high reliabilities. To name a few, they include companies like Marathon Oil [7], Alberta Natural Gas, who recently commissioned two natural gas transmission compressors, and Elf Aquitane, with reliability in the 99 to 100 percent range [8, 9].

Just as many have experienced operating problems, including Shell Canada, Tenneco and Nova. Most of Shell and Tenneco's problems were associated with putting magnetic bearings directly into the gas stream. In one case, the compressors were designed with only one dry gas seal on the coupling end, with the radial and thrust bearing operating outside. On the other end, the magnetic bearing was operating in the gas in order to eliminate the additional seal. The other case is that of a compressor where the magnetic bearings and the electric motor are all running in the high pressure gas with no seals [6]. In both cases, some problems occurred largely due to liquids migrating into the magnetic bearing and causing electrical shorts over time. These problems were overcome by changing the insulation and epoxy coating on the windings. Another possible solution would be to "can" the bearings, as shown in Figure 5, where the bearings are applied to a multistage pump.

Revolve recently completed a detailed study on the operating history of magnetic bearing systems applied to natural gas pipeline compressors operated by the Alberta Gas Transmission Division of Nova Corporation of Alberta (Nova). This included as many as 31 turbocompressor units equipped with magnetic bearings. This study formed a part of the guidelines published by the Electric Power Research Institute (EPRI) [10], for the application of magnetic bearing systems to equipment used in the electric power industry. The following are excerpts from these studies.

The study was complicated by the fact that many incidents initially tagged as magnetic bearing related, were actually the result of a magnetic bearing alarm or trip initiated by an event such as machine surge, ingestion of a foreign object, out-of-balance condition or other incident related solely to the rotating equipment itself and not to the magnetic bearing.

The study concentrated on the last three years from 1991 to 1993, and in summary showed a rate of improvement in the availability of just the magnetic bearing systems that should show a stabilized performance in another year or two in the high 99.5 percent plus range (Figure 10).

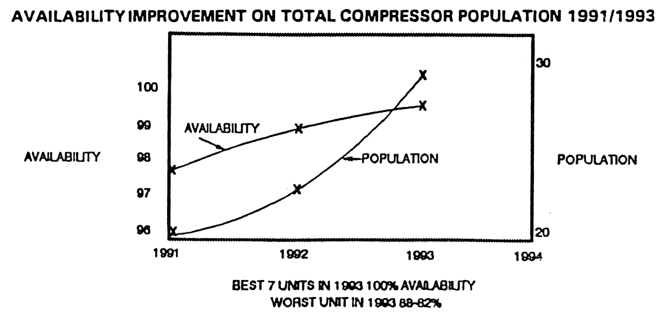

Figure 10. Taken from EPRI [10].

During 1993, seven units out of 29, or 24 percent, showed 100 percent availability for the whole year, indicating that with some combination of design, fabrication, operation and maintenance skills the goal of near 100 percent reliability is possible.

It should be stated that during this same time the average usage of all the magnetic bearing units also increased from 46.15 percent to 71.56 percent. So the overall system operation was also undergoing some major changes.

The overall unreliability of the magnetic bearing was first subdivided into four main areas, mechanical, controls, chillers, plus power supplies and backup batteries. Of these, the power supplies and backup batteries provided virtually no unreliability. The chillers on the other hand provided a large portion, but are no longer required on the current bearing control systems offered by any bearing manufacturer. In 1993, the chillers caused 26 percent of the events perhaps explaining how so many units could achieve 100 percent availability while the average was somewhat lower, as some of the later units did not have this feature.

The control system resulted in 67 percent of the events in 1993. The majority of these failures (78 percent) are of components that could be substituted by software in the digital systems that are now being offered, and hence this source of failure could be significantly reduced. The other 22 percent were failures of components that will still be required, such as DC/DC converters and amplifiers, and for which the vendor has been offering improved components with a higher reliability.

This leaves the last area, the mechanical system, with the latter containing the magnetic bearing, the position sensor, the magnetic bearing rotor, and the auxiliary bearing which contributed to 7 percent of the events. This system has often been compared to an electric motor in terms of its potential reliability. This comparison is valid so far as the bearing components are concerned, with the laminations and windings resembling those of an electric motor.

Apart from one unit where problems arose due to the QC of the metallurgy of the bearing laminations and some problems with connectors on the axial position sensors, the reliability of the bearings and the position sensors has been very good.

There were several other areas however, where the mechanical system reliability was not as good as desired. This was due to the following failure modes:

- Auxiliary bearing failure due to turning, as a result of excessive contact during normal magnetic bearing operational.

- Auxiliary bearing failure due to whirling following a full speed drop.

- Need for auxiliary bearing replacement due to in-service wear but within design specification.

- Excessive shaft deflection causing rotor/stator or compressor component rub during operational transient or during full speed drop.

- Shaft out-of-balance following the installation or replacement of the magnetic bearing rotor causing either shaft motion or bearing current trips and alarms [10].

All of the above failure modes should be able to be avoided in a sound design. The study goes on to describe methods of eliminating such problems. These include issues which were already discussed in section one including preload on auxiliary bearings and detailed rotordynamics studies including the ability to quantify the whirl process on the auxiliary bearings and shaft deflection during a full speed drop.

An additional method of potentially eliminating damage during plant transients or full speed drop with a flexible rotor would be to have additional landing sleeves for each radial bearing at the inboard end, a midspan damper, or sleeve backup bearing in the center of the machine. These concepts would prevent large deflections and potentially help with the rotordynamic stability.

TROUBLESHOOTING

To troubleshoot any system, use of an orderly approach to the analysis of technical problems is paramount. The resources required to resolve problems are dependent on the complexity of the problem and the expertise of the first line of defense personnel.

The magnetic bearing system serves two purposes in turbomachinery applications; it supports the rotating components and also provides diagnostic monitoring. A magnetic bearing system has the capability of monitoring a parameter not normally provided on the wet systems, that being the exact static and dynamic loads present, and in what direction they are acting upon the aero or rotating assemblies, at any given moment. This can, over the life of the equipment, greatly increase installation efficiency and reliability, plus aid in understanding and correcting the design deficiencies of the equipment and systems of the past.

Troubleshooting Methodology

Before any abnormal condition can be recognized, we must first understand and be able to describe normal operation. One must know how the system functions and be familiar with the data expected from the monitoring system.

Commissioning Data

To determine if a system problem is related to installation, data must be compared against that recorded during installation and commissioning. These data must include verification of proper assembly procedures, and include important mechanical data such as fastener torques, clearances, load capacities, etc. By using a commissioning manual especially designed to tabulate all such data, the troubleshooting of a system is simplified.

Operating Data

The operating data that identify normal operating conditions, sequences of events prior to malfunctions or trips, and the subsequent train of system events, is the greatest aid to expediting troubleshooting. All available information should be reviewed; however, to be effective, only factual information should be considered. Any data based on speculation should be eliminated from the history report.

Analysis

Once the factual information is tabulated, the system problems can be identified and the failure symptoms defined. With the failure symptoms defined all possible causes must be reviewed. Systematically eliminate possible causes and perform the easiest and quickest checks first. Ensure the action taken is not detrimental to the rest of the system. Following analysis of a failure, it may be evident that there is more than one action to be taken to completely rectify the system problem. Documentation of corrective action taken and results thereof will assist future efforts by providing a database for referral.

MAGNETIC BEARING SYSTEM TROUBLESHOOTING

This will focus on the types of trouble that can either occur in, or be detected by, a magnetic bearing system.

Possible Origins of Malfunction

The magnetic bearing system may trip due to an internal malfunction, or an external malfunction resulting in excessive loads being applied to the magnetic bearing system. Further breakdown of these two classes of malfunctions yield the following short list of suspects to be considered during troubleshooting:

Internal

- Power source for the control system
- Control system (panel)
- Interface connections
- Magnetic bearing hardware

External

- Aero assembly components
- Equipment mechanically connected to the aeroassembly that may transmit loads/vibration back to the aeroassembly, which is supported by the magnetic bearing system
- Stationary components surrounding the magnet suspended shaft
- Associated systems

Axis of Trip Initiation

The cause of a shutdown may be narrowed down by identification of the bearing axis which initiated the trip. On a five axis system, the ability to identify which axis initiated the trip can reduce troubleshooting time significantly. End users should pay particular attention to the diagnostic capabilities of a magnetic bearing system to aid them in troubleshooting. The value of this feature of magnetic bearing technology is now being recognized by end users and manufacturers alike.

Sequence of Events and Signatures

The sequence of events prior to the shutdown, during the shutdown, and after the shutdown must be carefully analyzed to establish what was the cause and what events resulted. Such analysis can also identify not only what was the original problem, but what other damage may have occurred during the coast down of the equipment.

The actual signatures of the shutdown as recorded by real time analytical equipment may be necessary to further eliminate possible causes to the problem. In analyzing the problem, it is best to have the origins of possible malfunctions discussed if monitored by real time data collection equipment, it makes it easy to identify clearly the sequence of events. However, by monitoring magnetic bearing currents and their vibration frequencies and amplitudes, one can determine with a good degree of confidence the origin of the problem and the probability of damage to the equipment.

The frequency of the vibration helps in determining whether the problem is in the control system, the rotating components, the equipment stationary components, or the associated system such as surge related. The level of currents and whether this current is DC or AC on the magnetic bearings also provides information as to the possible cause. By comparing the sequence of events, frequencies of vibration, amplitude of vibrations, and the current levels at each magnetic bearing axis, the cause of the trip can be narrowed down if not exactly located, and corrective action can then be taken.

If real time data is not available to provide the information necessary to determine the cause or subsequent damage as a result of the malfunction, then prestart checks must be conducted to provide some level of confidence that the machinery is safe to restart. The control system can be activated to check shaft movement and control within the auxiliary bearing clearances on all axes. Provided that the magnetic bearing reacts normally in this static condition, and no other indication of problems are evident, then a restart may be attempted. If possible, real time analytical equipment should be connected in the event that the problem was not a transient condition that has passed through. It should be mentioned that some of the magnetic bearing systems now provided data capture features to help with diagnostics.

CASE HISTORIES

Following is a brief discussion of some of the problems that have been detected and solved with the aid of magnetic bearing technology. The examples are all from centrifugal gas compressors in natural gas pipeline service. They include:

Surge Related Deficiencies

Field testing of selected compressors was conducted in 1991 to determine their response to surge, after four incidents on other similar compressors where mechanical damages occurred. Real time data was used to monitor the magnetic bearing signatures and the sequence of events from the associated recycle system. Testing revealed that upon shutdown, the units entered surge producing excess shaft vibration. Further investigation revealed the recycle valve and associated piping was undersized and/or was too slow to open.

One shutdown produced a full flow reversal with pressure pulsations occurring at a frequency of 5.0 Hz for well over a 10 second duration. Corresponding shaft response resulted in excess vibration on all axes producing auxiliary bearing deflection in both axial and radial directions. Rapid opening of the unit recycle valve (RV) also produced pressure waves at both suction and discharge ends of the compressor. However, except for one trip where the RV opened in less than 200 milliseconds (msec), this did not appear to result in uncontrolled shaft motion.

Subsequent testing has provided data that suggests that, provided recycle system sizing is correct, RV actuation to fully open within 500 to 750 msec is sufficient to keep the units out of surge and the resultant rotor excursions from occurring. Modifications were carried out to effect such a standard throughout the system.

Mechanical Rubbing of Impeller and Balance Piston Seals

Mechanical rubbing of labyrinth seals can be the result of a number of circumstances, including:

- Design deficiencies such as incorrect rotordynamic analysis, loose or insufficiently rigid fits, etc.
- Manufacturing deficiencies such as poor alignment of compressor bore, or eccentrically machined components.
- Operating conditions such as surge or liquid slugs causing extreme vibration.

Real time data tapes from numerous compressor shutdowns indicated that significant vibration was occurring during the rundown through the first critical speed of the compressor rotor. Disassembly of the compressor showed signs of rubbing of the impeller and balance piston labyrinth seals, ranging from light to sometimes severe.

The pattern of the rubbing identified that the shaft was bowing in the center, between the bearings. From evidence witnessed during disassembly and from analysis, it has been concluded that internal rubbing is occurring when the magnetic bearing vibration signature indicates the first critical is being excited.

Balance Piston Sizing

The current supplied to the active thrust bearing provides a direct relationship to the load in the axial direction during all modes of operation. As pressure differential across the unit is increased, the sizing of the balance piston is critical to limit the load being transmitted to the thrust bearings.

In the past, as with radial oil bearings, excessive loading may manifest itself as a higher bearing or oil temperature. In many instances, since the overload was present right from initial startup, this potential warning would be absorbed by the oil cooling system. There was no signal that the bearing was being overloaded and operation continued until the bearing mysteriously failed after a much shorter life than expected. Simple replacement offered no improvement.

With a magnetic bearing, problems with balance piston sizing have manifested themselves instantaneously; end users have seen the load grow proportionally with head, until warning and then trip occurs. Both undersized and oversized cases have been encountered and rectified accordingly.

Damaged Exit Guide Vanes

A similar case exists in the radial direction. As compressor head grows with speed, instances occurred where radial loads in a particular direction were observed. In one case, this resulted in upward forces greater than the weight of the rotor assembly. The damaged exit guide vanes also caused dynamic instabilities resulting in oscillating forces being applied to the shaft with subsequent vibration.

Hydrotest Plug Detection

Startup of a large beam-type compressor was accompanied by active thrust bearing current levels which allowed the package to attain idle speed for the warmup cycle. Upon completion of warmup and rampup into the operating band, the thrust bearing current levels rose rapidly, until warning and shutdown limits were exceeded. The unit could not attain minimum operating speed.

Analysis of the direction of the load and possible process pressure levels within the compressor internals isolated the possibility that the external balance line was blocked, preventing balance of the thrust loads with the balance piston.

Subsequent disassembly revealed that entry to the balance line was blocked by a plug used for hydrotest.

Impeller/Shaft Interference Fits

Startup of a natural gas pipeline compressor was attained successfully after thoroughly following all recommended commissioning procedures. Machine performance was initially acceptable.

This was followed by a sudden and noticeable increase in radial vibration exceeding shutdown levels. Attempts to restart were successful in bringing the compressor back up only to minimum operating speed, insufficient for the application.

Troubleshooting identified no problems with the control circuitry. Evidence gathered led to the belief that the problem lay with mechanical integrity of the rotor assembly.

The rotor assembly was removed and measured as-is prior to disassembly and rebalancing. Runouts recorded indicated a bowing of the rotor of 0.05 mm over a span of 2.26 m. At the operating speed, this was enough to cause an imbalance of 580 g-in., more than 14 times the maximum allowable under API 617.

Further examination revealed a unique impeller to rotor fit involving a double-step interference with relief in between. It was thought that this fit had 'pinched' the rotor, causing it to bow. This must have occurred after commissioning as all runout measurements and balancing was within spec. The potential for the fit to 'bite' on the rotor was there, but must have been catalyzed by a process upset, such as surge.

As a result, the fit was modified by relieving one of the interference fits to size-on-size. Subsequent installation, commissioning and startup yielded positive results and the problem has not yet recurred after more than two years of operation.

CONCLUSION

In summary, many new applications have taken place in the past three years, and the reliability and availability continue to increase to very high levels (99 percent plus). The key is to learn from past experiences, design around potential problems and feed back any problems encountered to the equipment and bearing manufacturers along with other potential end users. Through these efforts we will continue to see further improvements in reliability and availability.

The rapid change, improvement, and advancements being seen in the electronics and computer industries are also a reality in the rotating equipment industry. Magnetic bearings will continue to improve and become less expensive with advancements in computers. With broader application, reduced costs and new, advanced features they will become even easier to operate and more user friendly. New concepts and new designs which utilize the present and future potential advantages of magnetic bearings are continually being introduced and applied to turbomachinery. It is this process of improvement in and advancement of technology that has made the use of magnetic bearings in turbomachinery a reality.

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