SPECIFYING, MANUFACTURING, AND TESTING A CRYOGENIC TURBOEXPANDER MAGNETIC BEARING SYSTEM

by

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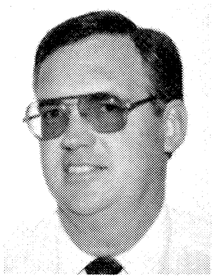

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Prior to this position, Mr. Andrews was the Chief Engineer of ACD, Inc. (formerly Airco Cryogenics), where he was involved in all aspects of high speed turbomachinery design and testing.

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ABSTRACT

The state-of-the-art magnetic bearing system, as it exists today, has several unique characteristics which tend to make certain applications of this technology more favorable than others. One application which seems quite favorable is the use of magnetic bearing systems in turboexpanders.

One such project is discussed, beginning with the initial concept and ending with the rigorous testing of the finished machines. Several problem areas and the associated corrective actions are discussed.

INTRODUCTION

When new and relatively unproven technology is applied to any existing piece of turbomachinery, the first question which must be raised is, "What is the risk-to-benefit ratio?" This project was no exception. When a large chemical company decided to construct a new plant, they elected to investigate numerous new technologies which would potentially make the plant safer, more reliable, and more efficient. One of these new improvements was the use of magnetic bearings in their cryogenic turboexpanders, which produce the lowest temperatures available within the plant.

The present cost of a magnetic bearing system is relatively high, and as such it tends to be applied to "engineered" products as opposed to "commodity" products (such as ANSI pumps), which sell for far less than the cost of the magnetic bearing system alone. The turboexpanders involved in this project are engineered products.

Also, the use of a magnetic bearing system for upgrading an existing machine is rarely cost-effective, since the capital has already been spent on a lube oil system. However, on a new system, such as the one presented here, the cost of a first-rate, pressurized lube oil skid was very similar to the cost involved in providing a magnetic bearing system. This meant that the capital cost of the system would be competitive.

The benefits available through the use of the magnetic bearing system in the machine include lower power losses in the bearings and support systems, no possibility for oil contamination, and a smaller installed machine "footprint." Of these, eliminating the possibility for oil contamination of the cryogenic process proved to be the most important benefit for this project.

After looking at several manufacturers throughout the world, the Mafi-Trench Corporation was awarded the contract to build two large turboexpanders with magnetic bearings. The high pressure unit is 1337 hp (997 kW) and the low pressure unit is 2492 hp (1,858 kW). Both machines have a design speed of 17000 rpm, with overspeed ratings of 21,900 rpm and 20,900 rpm, respectively.

As a precautionary measure, bearing housing castings to accommodate a conventional oil bearing assembly were ordered so that, in the event of an insurmountable problem occurring during the project, the job schedule would not be severely penalized. Also, financial agreements were worked out in advance, so that all parties would share in the success or failure of the project. This is the type of cooperative arrangement which the purchaser and the manufacturer are beginning to use in an effort to improve buyer-seller relations. The method fosters teamwork, since all parties work towards a common goal.

A "go/no go" date was set, such that all parties would evaluate the progress of the project at that point and decide whether or not the project should continue as a magnetic bearing system, or switch to a conventional oil lubricated system.

BEARING SPECIFICATION

When incorporating any new technology, it is a natural tendency to "take what is offered" by the supplier. With this in mind, a conscious effort was made to write a specification for the purchase of the magnetic bearings which did not encroach upon their freedom to design the best possible bearing for the system, but at the same time set up a relatively rigorous list of tests which the new bearings would have to pass. These tests included unbalance response tests, bearing load tests, and a minimum of two complete delevitations of the rotor such that the backup bearings were forced to withstand the full impact of the rotor at the maximum continuous speed of 19000 rpm.

MACHINE DESIGN

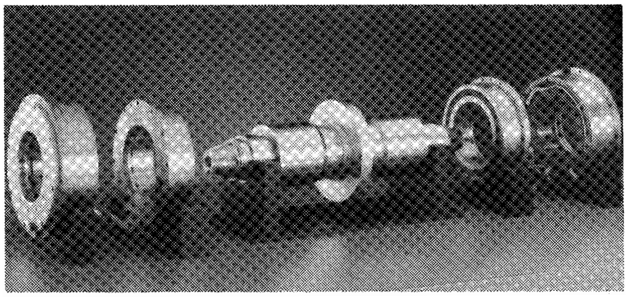

Figure 1. Magnetic Bearing System with Shaft.

The design of a magnetic bearing system (Figure 1), like any other design, consists of putting together the right series of compromises so that the overall design is optimized for its intended use. Some of the more important design considerations are discussed herein.

Magnetic Journal Bearings

One of the major limitations of oil lubricated bearings is their propensity to become unstable at high speeds. A standard cylindrical bore bearing with three axial grooves will typically run well up to a journal tangential velocity of about 260 ft/sec (79 m/sec). A tilting pad journal bearing, being inherently more stable, will typically run well up to about 330 ft/sec (100 m/sec), before the heat generated due to high horsepower losses becomes intolerable and/or other problems occur.

The typical magnetic bearing (Figure 2) is limited, generally speaking, by the strength of the materials used for the shaft laminations. For "standard" materials, this is about 660 ft/sec (200 m/sec), and can be pushed even higher with "special" materials and designs. Clearly, this seems almost too good to be true, and, alas, there is indeed a catch. Because the magnetic bearing cannot support as much load for a given projected area as a standard oil film bearing, the journal size of the magnetic bearing must be considerably larger, thus giving up some of its speed advantage. Then, due to the decrease in shaft stiffness resulting from the installation of the laminations, the journal diameter must be made even larger to achieve similar rotordynamic properties. In general,

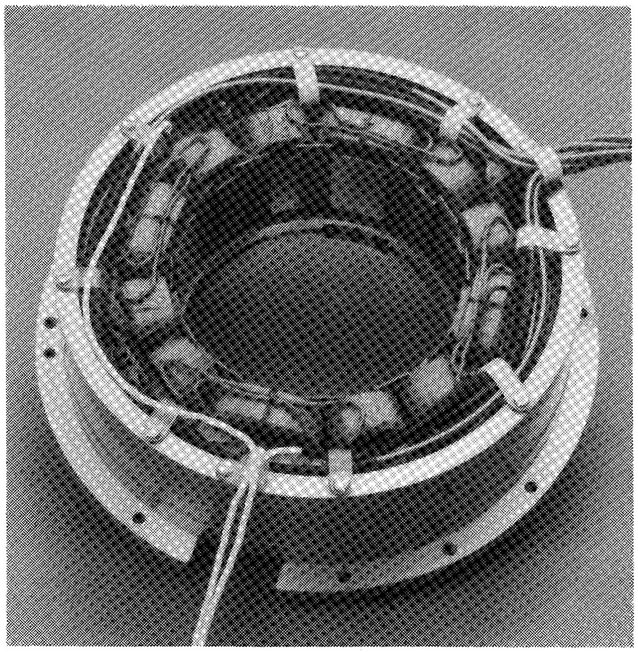

Figure 2. Radial Magnetic Bearing with Integral Radial and Axial Position Sensors.

magnetic bearings can extend the speed range of a given rotor, but not nearly as much as might be expected at first glance.

Magnetic Thrust Bearings

The major concern with magnetic thrust bearings (Figure 3) is the low load carrying capacity they possess compared to the equivalent size oil film thrust bearing. Unlike the magnetic journal bearing, it is a great deal more difficult to compensate for this by increasing the bearing size, since the stress and bore growth in the thrust disk soon become the limiting factors.

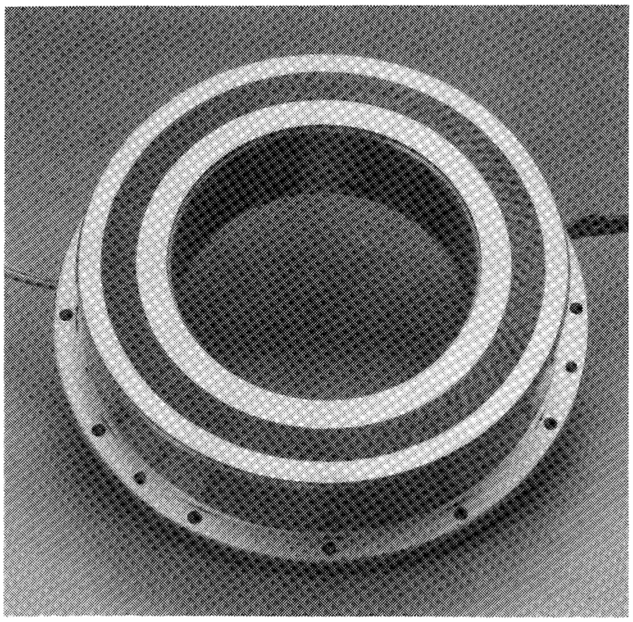

Figure 3. Magnetic Thrust Bearing.

The best way to minimize this problem is to pay careful attention to the external pneumatic thrust balancing controls. After engineering the maximum thrust bearing size into each machine, a pneumatic thrust balancing control system was designed to carry the majority of the thrust load under all operating conditions.

Backup Bearings

Each set of backup bearings consists of a pair of preloaded angular contact ball bearings mounted in a face-to-face configuration. This arrangement allows the bearings to take both a radial and an axial load, while the preload is sized to generate sufficient drag on the bearings to prevent rotation due to windage. The races are steel with a special dry lubricant coating, and the balls are ceramic. The radial clearance during operation is about 0.006 in (0.15 mm), and the axial clearance is about 0.010 in (0.25 mm).

The landing surface on the shaft under the inner race of the ball bearings is chrome plated to resist damage during a delevitation. A replaceable sleeve was considered, but concerns about the loss of shaft stiffness and the possibility of the sleeve dilating during a delevitation due to heat generation prompted the selection of the chrome plating. It was felt that a sleeve could always be added later if the chrome plating was not successful, whereas if the sleeve design was not successful, a new shaft would be required. Keep in mind that these machines, like most turboexpanders, are designed to operate below the first bending critical. This means that any reduction in stiffness due to a sleeve forces the operating speed closer to the first bending critical speed.

The landing surface for the backup bearings in the axial direction is a replaceable, hardened steel ring. This ring is replaceable due to the potential for fretting during a delevitation. Unlike the radial bearing which experiences an initial "skidding" and then a relatively pure rolling motion, the thrust surface experiences a continuous "in and out" radial movement relative to the backup bearing inner race due to the 0.012 in (0.3 mm) diametral clearance between the backup bearing and the shaft.

Labyrinth Seals

All seals in the machine were designed to be self-centering to allow the seals to survive a delevitation of the rotor. Extensive pressure balancing was required to prevent seal lock up due to the pressure differential acting across each seal.

Test seals were required for the expander wheel seals, since the test was not done at the actual cryogenic temperatures. The test seals were sized to produce the same clearance under test conditions as the actual clearance will be under cryogenic conditions with the proper wheel seal.

Tapered Impeller Attachment

The high tip speeds normally found in turboexpanders causes each wheel to experience high stresses in the bore. This in turn forces the bore to expand during operation and a large unbalance can occur, which can neither be detected nor corrected while the rotor is in the balance machine. For small bore growths due to low speeds, this can sometimes be handled with an interference fit between the shaft and the wheel. This solution, however, makes working on the machine much more difficult, and there is always the risk of galling the bore. Also, since most turboexpander wheels are heat treated aluminum, the use of a torch can be quite damaging if improperly used.

The two machines for this project are designed with the turboexpander manufacturer's standard tapered shaft attachment (Figure 4), except that the taper angle is larger to accommodate the large bore growths due to the high tip speeds. This shaft attachment maintains the wheel alignment as the bore grows by allowing controlled axial movement up the taper. The manufacturers stan-

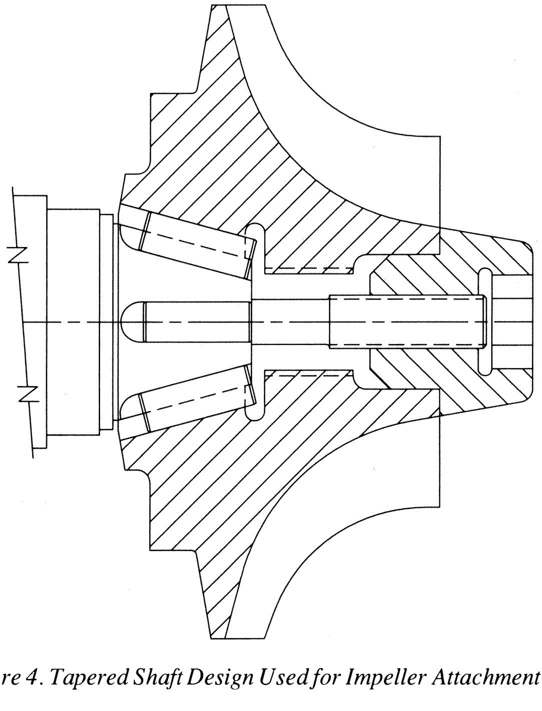

Figure 4. Tapered Shaft Design Used for Impeller Attachment.

dard round keys and keyways are used to minimize the stress concentration factor at the bore.

Bearing Shims

The design allows for axial shims to be used at all bearing locations. Axial shims are required at the radial bearing locations because the axial sensors are attached to the radial bearings, thus requiring precise axial alignment. Only one of the two thrust bearings must be shimmed to obtain the proper clearance.

Assembly Tooling

Each machine is assembled primarily in a vertical direction in order to minimize any chance of damaging the magnetic bearings and sensors. Special "dummy" plates hold the rotor in the proper position while the bearings are installed. Special sleeves were made to protect both the bearings and the shaft laminations, but during the initial assembly they were found to be unnecessary.

Thermal Growth Calculations

A detailed thermal analysis was conducted to investigate both the appropriate amount of seal gas required for cooling the magnetic bearings and to determine the differential axial expansion between the rotor and the housing, so that an interference would not occur due to the radical temperature change from ambient to operating conditions. The final design allowed an operating axial clearance of about 0.010 in (0.25 mm). The seal gas flow is routed through the magnetic bearings to allow the heat generated due to the magnetic bearing I^2R losses and the windage losses due to disk friction to be removed. The main source of heat generation was not calculated to be the losses from the magnetic bearings, but rather from the heat generated due to windage. This was later confirmed on the test stand by allowing the unit to remain levitated without rotation for an extended period of time and comparing the results with similar time periods at both full and half speed operation.

Aerodynamic Design

The expander power produced by each radial inflow turbine used in this project is absorbed by a corresponding centrifugal compressor stage. Of the two machines, the low pressure unit was the more "aggressive" design aerodynamically, dropping an isentropic head across the expander of 66.13 Btu/lbm or 51460 ft-lbf/lbm (153.8 kJ/kg or 153,814 Nm/kg) and generating a polytropic head across the compressor of 47.39 Btu/lbm or 36877 ft-lbf/lbm (110.2 kJ/kg or 110,225 Nm/kg). Like all new high tip speed wheels produced by the manufacturer, finite element analysis was used to ensure that the stress levels did not become unmanageable.

Piping Strain

The low pressure unit had the unusual requirement that the entire machine be free to float with the piping once it was installed in the field. This was due to the piping layout and the large temperature swing from ambient to operating conditions. The requirement was met by allowing the expander support to act as a "wobble leg" and installing reinforced fluorocarbon pads under the compressor housing.

TUNING

The tuning methods used by the magnetic bearing supplier is proprietary and will not be discussed except in very general terms. Prior to building the first machine (the high pressure unit), many stories were heard describing the possible horrors of tuning. However, when building the first machine the engineers were pleasantly surprised at their success, even though the tuning took a little longer than expected. The initial tuning values which were calculated by the bearing supplier were installed in the control cabinet and proved to be acceptable through all tuning checks with only one minor change. These same values ran well throughout all subsequent mechanical and performance tests of the machine.

The second machine (the low pressure unit), however, was not quite as well behaved. Several iterations were required to allow stable operation throughout the entire operating range. The problems mainly centered around the excitation of the third, fourth, and fifth lateral critical speeds and certain blade and disk natural frequencies.

In a way, the problems encountered on the second machine had a very favorable impact, since a great deal of practical information was gained which will be employed on future machines. This information is primarily related to the manner in which the choices made during the aerodynamic and machine design phases effect the interaction and excitation of these components by the magnetic bearing system. This information is considered to be proprietary to the manufacturer and will not be discussed further herein.

TESTING

Numerous tests were performed on the machines to prove the design both to the customer and to the manufacturer. The highlights of some of these tests (mainly on the high pressure unit) and their results will be discussed later.

Thrust Bearing Capacity Check

As discussed previously, for a high speed system, the thrust capacity of a magnetic bearing is limited mechanically by the stress in the thrust disk. In the manufacturer's standard thrust bearing design, the load is constantly monitored and pneumatically adjusted to prevent damage. A similar arrangement was designed for the magnetic bearing system, the major difference being the increased sensitivity required due to the lower load capacity of the magnetic bearing.

To verify the load capability of the magnetic thrust bearing, a variable load of known magnitude was applied to the shaft and the current required to support this load was recorded. The results were quite favorable, yielding a smooth, linear curve (Figure 5) up to saturation of the iron at about 3100 lbf (1400 kgf). The normal design load is approximately half of this value.

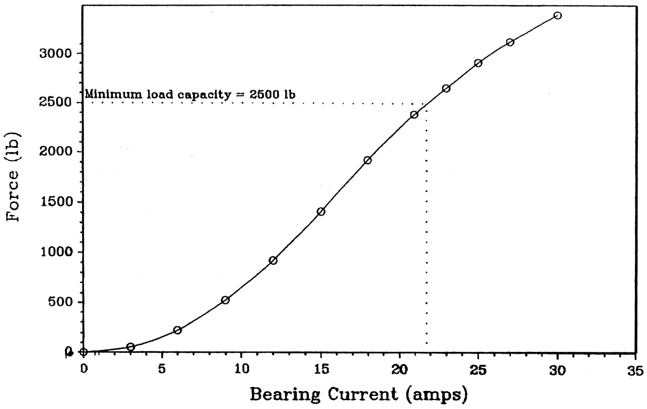

Figure 5. Magnetic Thrust Bearing Load Versus Current Curve.

Battery Backup Test

Prior to running each machine, tests were conducted to ensure that the battery backup system would indeed continue to support the rotor when the power to the control cabinet was turned off. This proved to be useful, since a problem was discovered and corrected in the control cabinet for the low pressure machine.

Compressor Performance Test

The compressor test on the high pressure machine was chosen as the first performance test, because the ASME PTC-10 test speed was 10700 rpm (about half speed), and would allow confidence and experience to be built up while gathering meaningful data in a relatively low risk situation. Also, the manufacturer of the ceramic ball bearings had slipped the delivery date several times, so the bearings weren't available in time for the initial testing. Standard steel ball bearings were installed as a temporary replacement during the initial low speed testing.

The compressor test went very well in virtually every aspect. The machine ran very smoothly, the compressor head and efficiency matched predictions with a good surge margin, and the test crew became more comfortable working with the magnetic bearing control system.

The only disappointment was a misunderstanding between the bearing supplier and the turboexpander manufacturer concerning the vibration signal outputs from the magnetic bearing control cabinet. The signals available to monitor the vibration are "real time" signals. This is fine for analysis, but the data acquisition system used during the tests did not have an input option for reading and recording low level peak-to-peak AC voltages. A simple circuit was constructed to perform the task of producing a DC output proportional to the peak-to-peak AC waveform.

Compressor "Smart" Dummy Wheel

A dummy wheel, which approximates the weight and center of gravity of the actual compressor wheel, is required during the mechanical run tests to prevent an excessively high compressor discharge temperature due to the high speeds involved. Blades were machined into each dummy wheel and a nozzle block was built to allow air to be used as a means of braking the turbine to simulate the drag of the compressor wheel during rundown tests.

The device was tested at 10700 rpm, and the results compared very favorably with the rundown tests performed using the actual compressor wheel. The "smart" dummy wheel is also used as a means of loading the turbine during the expander performance test. All tests of this design proved successful, although the noise generated by the dummy wheel at off-design speeds was louder than expected.

One final feature of the "smart" dummy wheel is the addition of a series of tapped holes of precise size and location such that convenient increments of unbalance can be added to the wheel to determine the ability of the magnetic bearing system to handle these additional loads. The results of these tests will be discussed later.

Mechanical Run Test

When the ceramic bearings finally arrived, they were installed in the high pressure machine and the first full speed mechanical run tests were performed. The machine ran extremely well throughout its entire speed range. The magnetic bearings ran warmer than expected, but this was subsequently traced to excessive sound insulation which was installed to reduce the noise due to the braking action of the dummy wheel. It was discovered that the braking action was much quieter at the higher speeds, nearer to its design point, and that the sound insulation could be removed under these conditions.

The oscilloscope orbits of the vibration signals were not always circular. In fact, they seemed to undergo distinct changes throughout the speed range. The magnetic bearing manufacturer assured the turboexpander manufacturer that this was normal. Plots of amplitude versus frequency indicated a nice, clean spectrum (Figure 6). Overall, those involved were very pleased with the results.

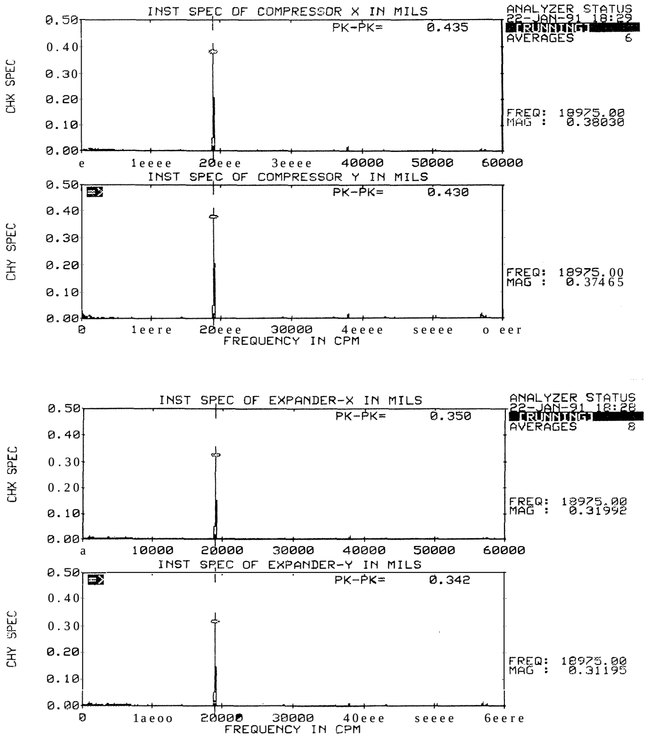

Figure 6. Amplitude Versus Frequency Plot for Magnetic Bearing System.

The low pressure machine was not as successful in its early attempts at full speed operation. At the time of this writing, speeds in excess of the design speed have been attained, but additional tuning will be required before the mechanical run test can be successfully completed.

Unbalance Response Test

One of the concerns that the turboexpander manufacturer had was the ability of the magnetic bearing system to withstand a relatively large rotating unbalance. Systematic runs were made starting from a balanced condition, increasing up to over eight times the ISO 1940 G-2.5 level. These tests indicated that the system was indeed capable of successful operation at relatively high unbalance levels.

Delevitation 1

The test which everyone was eager, yet apprehensive, to see was the delevitation test. This test involved running the machine at the maximum continuous speed of 19000 rpm, then simultaneously tripping the machine and disarming the magnetic bearing system. The purpose of this test was to prove that the backup bearings could sustain, in a stable manner, the dynamic forces associated with a full speed "crash down."

All vibration data was recorded on a 14 channel tape recorder, while the currents, temperatures, etcetera, were recorded by a computerized data acquisition system. Cascade plots were captured during the coastdown on a computerized vibration analysis package. X versus Y oscilloscope traces (orbits) of the shaft centerline were observed throughout the test.

The orbits were perhaps the most revealing means of viewing the delevitation. Three distinct patterns were observed during the coastdown on the backup bearings. First, a whirl was observed, as one might imagine when the shaft speed is greater than the speed of the inner race of the backup bearing.

The other two orbit patterns were what one would expect to see as the inner race of the backup bearing catches up with the speed of the shaft. At times, a rocking motion was observed as the shaft attempted to whirl but the backup bearing would not provide sufficient resistance to allow this to happen. The result was an oscillating motion produced as the shaft would "climb up" the inner race of the backup bearing, only to "slide" back down due to the nearly equal speeds of the two components. As this oscillatory motion ceased, the orbit would essentially be reduced to a "point," as one would expect from a system in which both the speed of the shaft and the speed of the backup bearings are equal. The result is that the shaft will "sit" in the bottom of the backup bearing, thus yielding a "point" for an orbit.

After disassembly of the machine following the first delevitation, the machine was re-assembled using all of the original components. Pictures were taken to document the condition of the parts for comparison with the second delevitation. The damage consisted of a series of scuff marks on the backup bearing inner bore and the shaft (Figure 7), where these two parts contacted during the delevitation, and some light fretting on one of the replaceable hardened steel thrust rings where it contacted the backup bearing during delevitation.

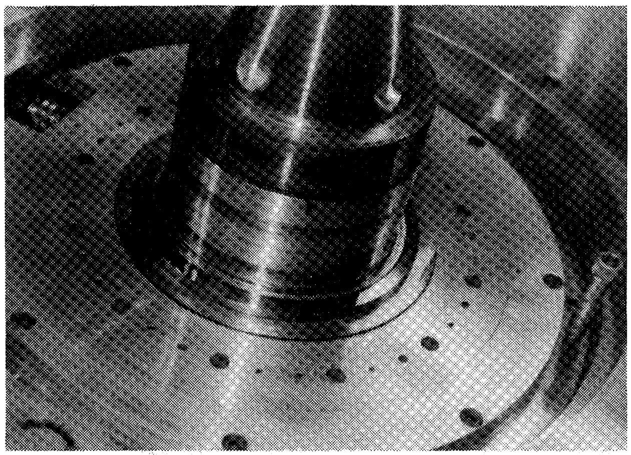

Figure 7. Scuffing Damage on Shaft Due to Contact with Backup Bearings during First Delevitation.

Delevitation 2

The results of the second delevitation were as good as the first. In fact, in some ways it went better. The orbits were somewhat smoother and the cumulative scuffing of the contacting parts due to both delevitations (Figure 8) was only slightly worse than the damage due to the first delevitation alone. It was estimated that 90 percent of the damage observed after the second delevitation was due to the first one, and only 10 percent was due to the second one.

It should be stressed that the machine could have been immediately restarted following either delevitation. The disassembly and inspection following each one was done not because of any problems, but because all parties had agreed to do it as part of the test procedure.

Overspeed Test

Due to the relatively high stresses in the thrust disk, the machines were subjected to an overspeed test which matched the overspeed ratings for the wheels in each machine. The high pressure unit was run at 21900 rpm for one minute, while the low pressure unit was run at 20900 rpm for the same length of time.

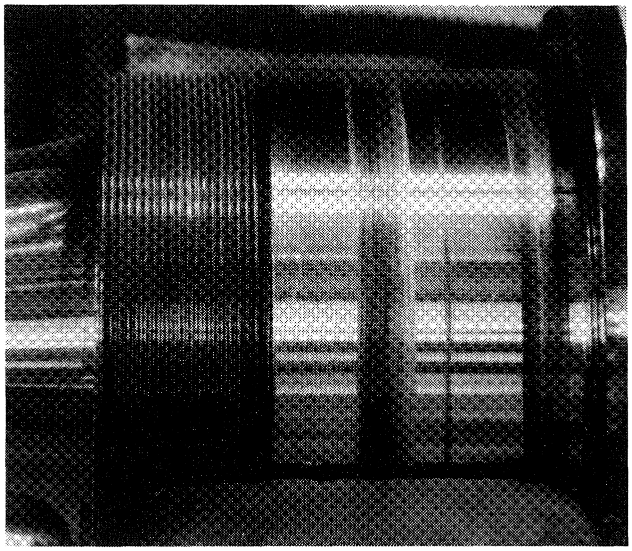

Figure 8. Scuffing Damage on Shaft Due to Contact with Backup Bearings during Second Delevitation.

Automatic Balancing System Test

One of the features of the magnetic bearing system is the ability to use a tracking filter to reduce the control system's response to synchronous vibration. This allows the rotor to turn about its mass center instead of the geometric center measured by the position sensors with a resulting reduction in forces due to unbalance. On the high pressure machine, this feature had virtually no effect (note that the feature can be disarmed by removing the cards containing the ABS circuits), while on the low pressure machine it appeared to increase the stability of the system, even though it did not significantly change the synchronous vibration as might be expected.

MISCELLANEOUS PROBLEMS AND COMMENTS

- The sensor laminations are subject to pitting corrosion (rusting) and damage due to handling. To minimize this the area was undercut and a protective, nonmetallic coating was applied.

- The alignment of the backup bearings with the magnetic bearings and the position sensors is very critical. This alignment should be checked during assembly of the machines, which requires a control cabinet. Since this would not be practical once the units are in the field, the turboexpander manufacturer developed a circuit to check the alignment without the cabinet. This would allow future overhauls to take place without having the control cabinet available.

- The control cabinet must be mounted within a cable length of about 325 ft (100 m) from the actual machine, so that the cable length does not adversely influence the magnetic bearing system. Also, the nature of the components in the cabinet is such that a reasonably clean and dry environment is required. These are not big problems, but they must be considered.

- The magnetic bearings are built in a very similar manner to electric motors. Two problems arose from using this technology. First, significant rework of the bearings and shaft laminations were required, due to poor quality machine work. Electric motor shops just do not seem to work well with the tight tolerance levels found in magnetic bearings. Second, the imbedded resistance temperature detectors (RTDs) used to monitor the winding temperatures were not replaceable, a problem frequently encountered in electric motors.

- The axial sensors and laminations were built such that a strong false vibration signal was produced at four times running speed. This false signal was then acted upon by the control electronics as if it were real. It was not clear that this caused any real problems with the hardware. At the time of this writing, additional tests are being planned to determine if this is acceptable or not.

A similar concern is the orbit shape itself. What is considered a good "round" orbit with a magnetic bearing requires some redefinition of the word "round." While nice, round orbits were obtained at certain speeds, at other speeds they were laced with small, controlled loops of higher frequency activity. The machine did not seem adversely effected by this, but those concerned are still looking into the situation.

- The control cabinet provides an output of the "raw" vibration signal for analysis. It does not, however, provide outputs which are proportional to the overall peak-to-peak vibration. Because the sensor sensitivity is about 3.125 times a standard eddy current probe output, standard monitors cannot be used. However, with the newer monitor systems that have adjustable scale factors, this is not a problem.

- While the chrome plated surface of the shaft under the backup bearings worked well, it would seem that a replaceable sleeve would be better if the rotordynamics allow it. Worries about the heat generated during a delevitation causing the sleeve to become loose on the shaft seem to be unfounded based on this experience.

PRESENT STATUS

At the time of this writing, the high pressure unit is complete concerning all contractual testing, although additional delevitations are planned on an experimental basis. The performance of this entire machine, including both aerodynamic and mechanical testing, has been excellent.

The low pressure unit has been aerodynamically tested with outstanding results (Figure 9). The tuning of this system has not gone as well as the high pressure system, but is expected to be completed in the very near future. Until this tuning is complete, the mechanical high speed testing of this machine cannot be performed.

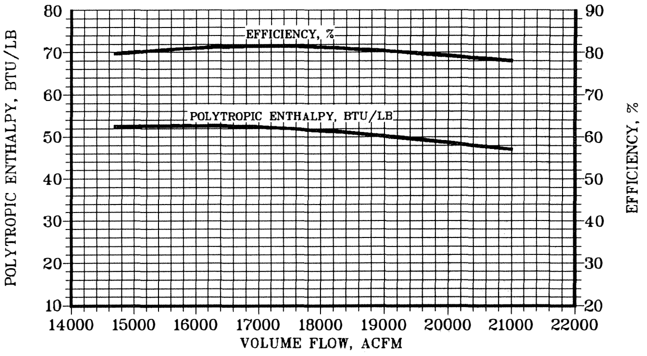

Figure 9. Aerodynamic Performance Test Results for the "Aggressive" Compressor Wheel on the Low Pressure Unit.

An attempt will be made to provide one set of tuning parameters which will work with either machine configuration. This will make maintenance and repair of the control cabinets easier for all of the parties involved.

A separate axial eddy current probe will be temporarily installed during the mechanical test of the low pressure unit to determine how seriously (if at all) the false vibration signals mentioned previously are effecting the shaft.

CONCLUSIONS

Only time will tell what part magnetic bearing systems will play in the future of turbomachinery. It is the authors' opinion that this technology will continually improve, its cost will gradually decline, and many more applications will become economically viable.

One thing, however, is certain. The benefits of magnetic bearing systems cannot be denied, and certain applications which stand to benefit from these systems will continue to drive the technology to higher and higher plateaus. The problems which exist today will surely be solved tomorrow. After all, isn't that what engineering is all about?