MAGNETIC BEARINGS AND DRY GAS SEALS— A PROCESS COMPRESSOR APPLICATION

by

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ABSTRACT

The design considerations and features along with manufacturing and operating experiences encountered on a multistage centrifugal process compressor incorporating a dry gas seal and active magnetic bearings immersed in the process fluid are discussed.

Starting with clearly defined process requirements, a close user-manufacturer working relationship was essential to assure that all of the potential operating conditions associated with hydrogen recycle service were evaluated and that the machinery was designed to accommodate them.

Although the risks associated with the initial application of advanced concepts to a particular process cannot be totally eliminated, they can be minimized and successfully managed. Problems which occurred at several stages of the project are discussed, along with their solutions.

Operating results to date are reviewed to provide a basis for continuing improvement in future installations of advanced centrifugal compressors.

INTRODUCTION

In the early 1980s, it became apparent that the technologies for active magnetic bearings and dry gas seals were developing and being successfully applied.

In 1985, Shell Canada Limited conducted a detailed evaluation of these technologies, reviewing the hardware, and applications, and concluded that these technologies were viable and could be applied to refinery process compressors. Potential advantages were seen to be: improved reliability, reduced capital cost, and reduced maintenance cost.

An opportunity arose at the Montreal East Refinery to install a new compressor in Reformer Hydrogen Recycle Service. The project started as an opportunity to replace a compressor turbine drive with a motor drive to reduce energy costs. To minimize unit downtime, it was economic to build a new compressor in parallel with the existing one. It was possible, then, to build and install a compressor utilizing the new technologies, while keeping the existing compressor as a back-up to minimize risk.

In 1986, Shell Canada, Limited and Imo Industries, Incorporated's Delaval Turbine Division, entered into a contract for the design and manufacture of a completely "dry" centrifugal compressor for this application. To achieve the "dry" prerequisite it would be necessary to equip the compressor with active magnetic bearings (AMB) and dry gas seals. Additional discussions were held to define the final approach to the compressor configuration. One of the questions addressed the possibility of mounting the active magnetic bearings in the process gas. If this could be achieved, it would be possible to design the unit with only one shaft end seal. After due consideration it was decided that the process was clean enough and dry enough to elect this option.

Placing the active magnetic bearings in a process gas environment was a significant departure from previous "dry" compressor practice. Since shaft seals have been a traditional weak spot in compressors, and since dry gas seals appeared viable, but did not have a long history in refinery applications, the elimination of one shaft seal seemed desirable.

MACHINE DESIGN

Configuration

To meet the specified process conditions, a three stage compressor rotor was selected. Since an induction motor was specified as the driver, a speed increasing gear was needed to meet the required compressor operating speed. The basic compressor design is similar to the typical vertically split (barrel) compressor that would be used in hydrogen service. The similarity ends however, with the installation of the active magnetic bearings and the single dry gas seal. As can be seen in Figure 1, the entire bearing system is located inside the pressure housing. On the inboard (coupling) end the dry gas seal is located outermost on the shaft next to the coupling. The thrust end is completely enclosed. In operation, seal gas is injected between the single dry gas seal and the inboard end radial bearing. Some of the gas leaks through the seal and is vented to flare. The rest flows through the inboard radial active magnetic bearing, through a labyrinth that controls the flow and into the process. A nitrogen buffered double carbon ring is used to prevent the gas leaking to the outside from entering the atmosphere through the coupling guard. The configuration of this seal is shown in Figure 2. The double carbon ring also acts as a backup seal in the event of failure of the primary seal [1].

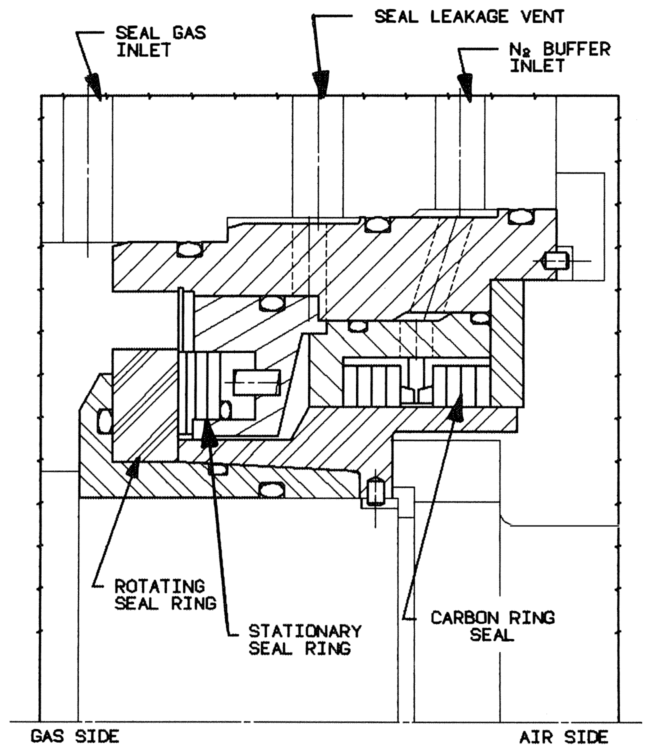

Figure 2. Dry Gas Seal Longitudinal Section.

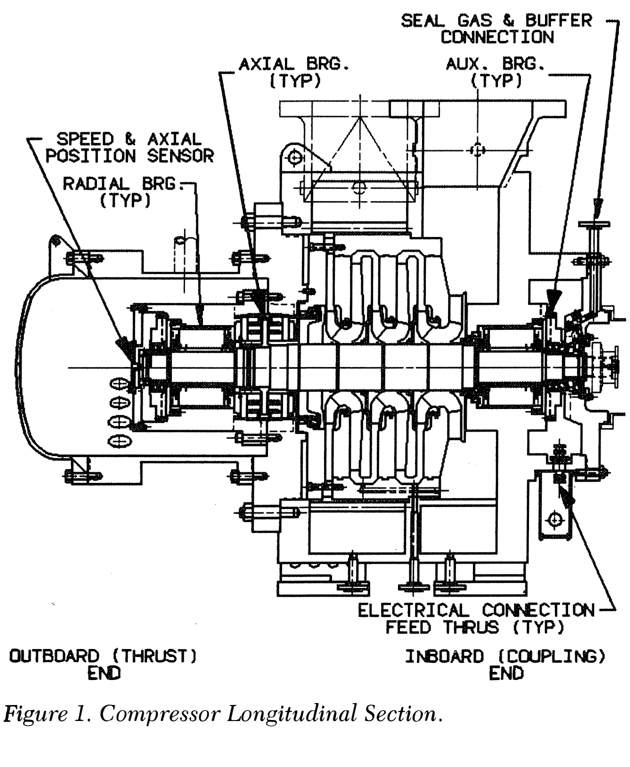

Figure 1. Compressor Longitudinal Section.

On the outboard end of the machine, leakage through the balance piston is directed downward toward the shaft and then out through the inboard axial bearing air gap. The flow is assisted by the windage effects of the rotating axial bearing thrust disk. The flow is channeled into the pressure housing through several openings provided in the bearing bracket. A pipe from the pressure housing to the compressor inlet chamber recirculates this flow back to the suction chamber. The windage effects in the outboard axial bearing air gap circulates gas from the pressure housing through the outboard radial bearing and the outboard axial bearing air gap back to the pressure housing through the bracket openings.

The seal gas flow and the balance piston flow are used to cool the active magnetic bearing stators.

MECHANICAL REQUIREMENTS

Bearing Alignment

In a typical compressor design (or any other rotating machinery using tilt-pad journal bearings) the rotor, at rest, sits in the bottom of the bearing. On start-up, the rotor rises on the oil film. When this occurs, the rotor moves "up," perhaps 1.5 to 4.0 mils, depending on the features of the particular rotor/bearing system. Typically, these bearings might have a diametral clearance of 5.0 to 12 mils. It is obvious that, in operation, the rotor does not run on the geometric center of the bearing clearance.

The active magnetic bearings used in the subject compressor have a radial bearing diametral clearance (air gap) of 40 mils. At rest, and de-energized, the rotor sits on the bottom, the same as if resting on a tilt pad bearing; however; after final tuning, and when the bearings are energized, the rotor levitates and is then held by the magnetic fields on the exact geometric center of the bearing diametral clearance.

With this in mind, the decision was made that it was necessary to control the relative locations of the centers of all the locating bores to a more stringent set of tolerances than normally used. To accomplish this, the compressor case, end cover and outboard bearing bracket were finish machined to the point necessary for assembly. The premachined parts were assembled and dowelled to insure repeatability of assembly. The important locating bores were then line bored to the specified tolerances.

A similar situation exists with the axial thrust bearing. When energized, the axial bearing rotor (thrust disk) is located on the center of the air gap. The air gap on the subject unit is 56 mils.

A typical oil lubricated axial bearing "rides" on a film of oil perhaps one or two mils thick. When in operation, the rotor moves to one side or the other of the built in end float, depending on axial thrust loads. Depending on the particular rotor bearing system, this end float can be from 10 to 20 mils. This means that during operation, the rotor can move axially nearly the amount of this axial clearance, depending on the aerodynamic loading. On the other hand, when the active magnetic bearing is energized the axial bearing rotor moves to the center of the built-in air gap and will stay at this location as long as the axial forces do not exceed the load capacity of the active magnetic bearing. In designing the mounting for the axial bearing stators, the housings were configured to allow for axial shimming at installation. This permitted accurate setting of the air gap so that in operation the rotor would be located in the correct aerodynamic position.

In addition to the hardware required to run a rotating machine on active magnetic bearings, some additional equipment was supplied. This is necessary to protect the machine in the event of certain system failures that can occur in a typical machinery installation.

Auxiliary Power

In an oil lubricated machine, if total power is lost, the unit may be able to coast down on the residual oil film on the bearing pads; or if this type of power loss is expected to be a common occurrence, an overhead gravity tank can be supplied to provide some oil during coast down. This is generally the option of choice.

The active magnetic bearing machine presents a different problem. The rotating and stationary bearing assemblies are constructed of "stacks" of laminations (very much like an electric motor stator and rotor) with a large air gap. If total power loss occurs, the rotor running at speed "crashes" into the stator resulting in severe damage to both. Provisions are therefore made to prevent this possibility.

An auxiliary power supply is available to keep the magnetic bearing system energized if the main power is interrupted. This auxiliary power may be in the form of batteries or any other reliable source. The capacity of this auxiliary power is determined by the needs of the particular installation. For this machine, backup batteries with ½ hour capacity were provided.

Auxiliary Bearings

Auxiliary bearings are also supplied to support the rotor. These bearings serve two functions: one is to support the rotor when the bearings are not energized; and, the other is to control the rotor and prevent rundown damage in the event of a loss of both normal and auxiliary power.

These bearings are preloaded, angular contact ball bearings with a specified preload. They are sized to suit the particular installation. The outer race is clamped in a holder to provide the preload and to permit installation in the compressor stator. The holder is fitted such that the bore of the auxiliary bearing is held concentric to the bore of the active magnetic bearing stator. The holders are mounted outboard of the active magnetic bearing stators and as close as feasible to them. The shaft under the auxiliary bearing is fitted with a replaceable sleeve. The outside diameter of the sleeve is sized to give a diametral clearance to the auxiliary bearing bore that is approximately one-half the active magnetic bearing clearance (i.e., in this case, 20 mils). Therefore, on complete loss of power to the active magnetic bearing coils, the rotor will drop and run down on the auxiliary bearings rather than the laminations. In addition to the radial clearance, the auxiliary bearing inner race on one end of the machine is located between two axial stops fixed to the shaft. On power loss, these stops limit axial rotor motion and prevent contact between the axial bearing rotor and stators. This clearance is also set at 20 mils. The various hardware items required for the active magnetic bearing/rotor system are shown in Figure 3.

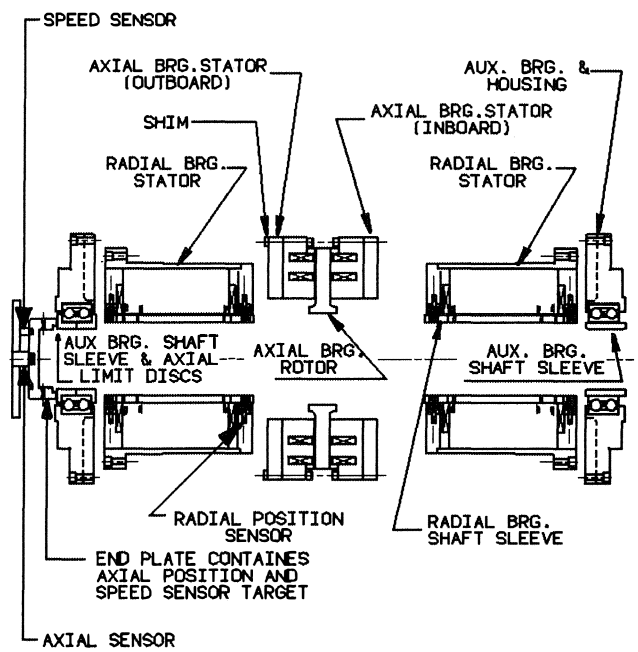

Figure 3. Active Magnetic Bearing (AMB) Components. (Control panel not shown).

Automatic Balancing

One of the features available in an active magnetic bearing system is referred to as automatic balancing. This terminology attempts to define a system's operation in familiar terms. We should understand that the system cannot actually "balance" a rotor. However, it can take advantage of the fact that a rotating body, properly controlled, can be made to rotate about its center of gravity rather than its geometric center. This effectively eliminates any rotational forces due to the rotor's inherent imbalance and, hence, reduces the bearing loads. To make use of this feature imposes certain restrictions on the rotor design. Since the automatic balancing works by reducing the bearing stiffness coefficients to very low levels, it is imperative that there are no rotor natural frequencies coincident with the running speed at any stiffness in the stiffness range of the bearing. A graphical representation of this requirement is shown in Figure 4.

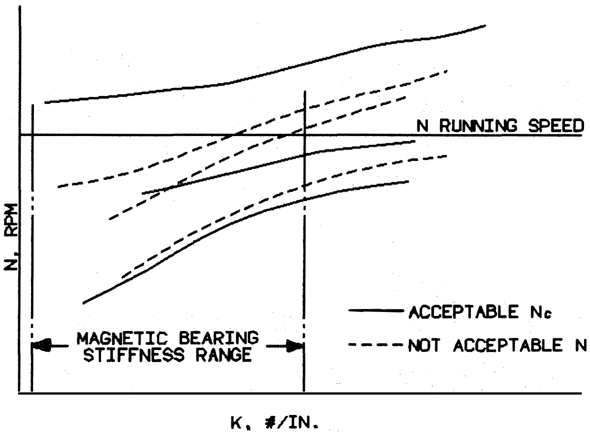

Figure 4. Typical Speed/Stiffness Curve for Rotors Supported on Active Magnetic Bearings.

Analysis indicated that by stiffening the shaft ends in the area of the bearing rotors we could meet the natural frequency criterion for this machine. To accomplish this, we departed from the normal radial active magnetic bearing rotor design. Rather than use a shaft sleeve with bearing lamination on the OD, we elected, after discussions with the bearing vendor, to mount the laminations directly on the compressor shaft. The lamination assembly was mounted on the compressor shaft with a light shrink fit and clamped axially with a check nut. The outside diameter of the laminations were finished to size according to the bearing design requirement. This approach permitted an increase in shaft diameter in the bearing area without increasing the bearing diameter.

Interchangeability

In addition to the mechanical details of the compressor design, one of the customer requirements was to use the same active magnetic bearing stator design as that being supplied with a "dry" compressor on order with another compressor manufacturer. The operating speed and rotor weight of both units were nearly the same. Use of the same active magnetic bearing stators would result in the ability to use many of the same spare parts in both units; reducing the number of spares required. There were configuration differences between the two compressors but by working with the active magnetic bearing vendor we were able to develop interfaces compatible with both compressors. The purchase specification for the active magnetic bearings was written to accept this compromise. It also specified that the active magnetic bearings were to be suitable for use in the expected process gas and at the expected operating temperature.

Other Mechanical Requirements

The remainder of the design is similar to any other centrifugal compressor. Fits, finishes, materials, etc., were selected to be compatible with the process and provide reliable service.

Maintainability

Every effort was made to simplify assembly and disassembly of the compressor. To this end, a number of special tools were designed. Throughout the various stages of assembly and testing at the factory, and later in the field, the tools proved their worth. To illustrate the logic behind the tools, one in particular is discussed here.

The radial bearing stator weighs approximately 300 lb. Assembly and disassembly into the compressor requires that it be pushed into its mounting location while passing over the bearing rotor already installed on the shaft. In order to align the bearing stator with the shaft and minimize the chance of damage to both the bearing stator and rotor, an assembly tool was required. This tool is simple in configuration, consisting of a tubular shape of predetermined length. The outside diameter of the tool has a thick, baked-on coating of hard nylon. During assembly, the tool is mounted on the shaft and held in place with a nut. The radial bearing stator is lifted with a crane and slid over the tool. Appropriate studs, nuts and washers are used to jack the stator into place. This arrangement is shown in Figure 5. The outside diameter of the tool is larger than the outside diameter of the bearing rotor and smaller than the inside diameter of the bearing stator. The stator outside diameter is stepped in such a manner that as the stator is pulled into its housing; the outside diameter fits mate before the inside diameter moves beyond the tool. This effectively prevents contact between the active magnetic bearing stationary and rotating parts during assembly.

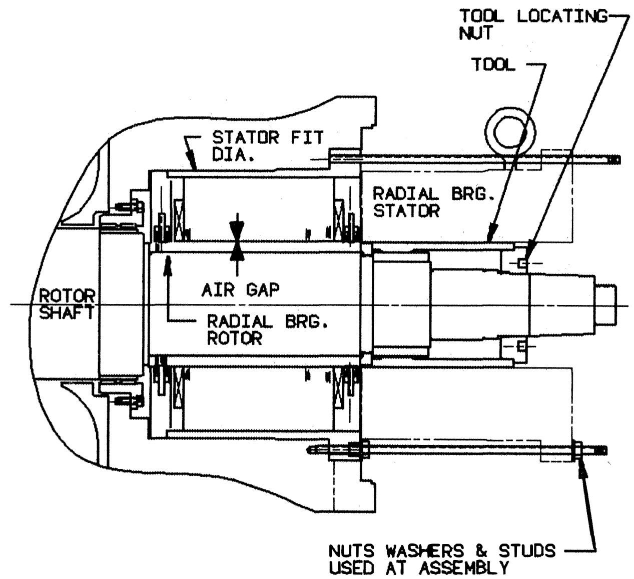

Figure 5. Typical Tool Used to Assemble Compressor. Installation of radial active magnetic, bearing stator shown.

Manufacturability

Earlier, the effort made to ensure the accuracy of the active magnetic bearing locating fits was mentioned. With standard manufacturing practice, each part of a compressor housing is machined individually and designed with a pilot fit to match the other parts. This approach results in the total tolerance from one end of the casing to the other being an accumulation of the tolerance at each joint. Since this compressor was being originally designed for the use of active magnetic bearings and dry gas seals, the engineers elected to develop an approach to design and manufacturing that would minimize the overall stack-up and accumulation of tolerances, eliminating potential problems in aligning the magnetic parts. Therefore, a procedure was developed that called for premachining some of the compressor components, assembling these components and then final machining of the remaining fits.

The compressor casing was finish machined except for the locating fits for the active magnetic bearing and dry gas seal stators. The locating fits were semifinished to allow for final machining later. The compressor case bore that locates the aerodynamic assembly was defined as the datum for all final machining operations. The end cover was finish machined. The outboard bearing bracket was semifinished except for the mounting surface and bolting provision. These three parts were assembled and dowelled to ensure repeatability during future maintenance. The mounting and locating surfaces for the active magnetic bearing stators were then finish machined to the specified tolerances. The arrangement is shown in Figure 6.

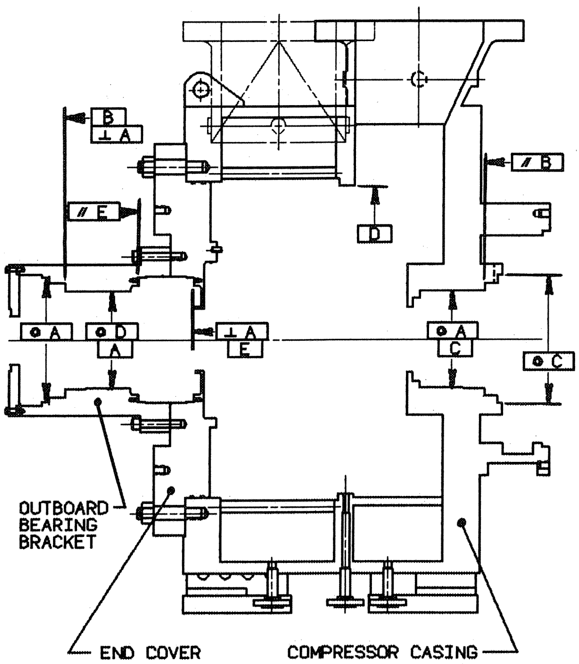

Figure 6. Geometric Tolerancing of Case Assembly Machining.

It is important to note that throughout the design stages of this project, periodic meetings were held with manufacturing operations to discuss the concepts being considered and to obtain their input and advice. This avoided the surprise of seeing them for the first time on a released manufacturing drawing, as well as reducing the impact of the changes on planning activities. In addition, everyone in the plant was aware of the advanced design of this unit and were interested in its progress. These meetings did much to smooth the flow of the job through the manufacturing process.

Similarly, several meetings were held with the end user to keep him informed of progress on the job and to discuss various design decisions such as the method of bringing the electrical connections out of the pressure casing.

Interface Connections

The active magnetic bearing stators are similar in design to an electric motor stator. The magnetic field is controlled by an electronic system that gathers data from sensors located at the bearings. Resistance temperature detectors (RTDs) in the bearing stators monitor the temperature. The thrust stators are also equipped with flux sensors. There are also a speed sensor, an axial position sensor and a temperature compensating circuit. These add up to 68 electrical connections (including grounds) that must be brought outside the pressure housing to be connected to the control panel. To make the penetration and to facilitate maintenance, a commercially available bulkhead connector was selected. This "feedthrough" consists of a sealed bulkhead fitting with an MS style connector on each end. The internal connectors are hard wired to the bearing stators; and, the external connectors are wired to the control panel. These connectors were selected with contact sizes suitable for the current and voltage being carried. Power leads and sensor leads were separated to minimize the possibility of electrical interference. Each radial active magnetic bearing stator required three connectors: two for power and one for sensors and RTDs. Each thrust stator required two connectors: one for power and one for flux coil and RTDs. The axial position sensor, the speed sensor and the temperature compensating circuit were wired to one connector. A total of 11 connectors were required. Each connector was specified with several extra contacts to allow for spares. A typical connector assembly is shown in Figure 7.

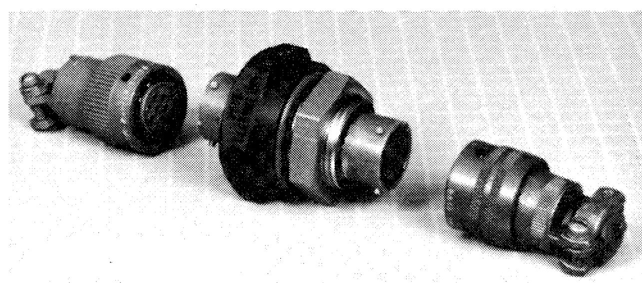

Figure 7. Typical Feedthrough Bulkhead Connector.

These connectors are located in specially machined areas on each end of the compressor. Eight connectors on the outboard (thrust) end are located in the outer pressure housing. Three connectors on the drive (coupling) end are located in the case (Figure 1). Fabricated junction boxes are provided to protect the connectors and to provide a mounting surface for conduit. The junction boxes and conduit can be seen in Figure 8.

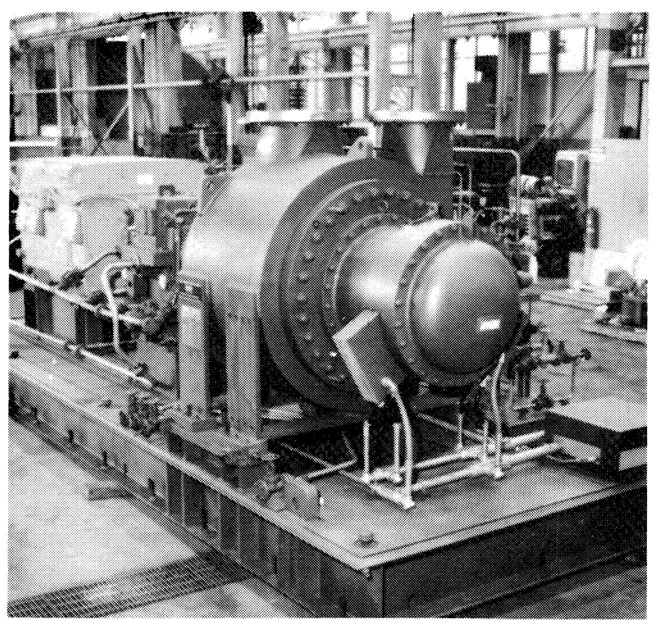

Figure 8. Motor, Gear, and Compressor on Bedplate Ready for Shipment.

The specification for ordering the connectors required that they be suitable for use in the expected process gas, that they be leak tight at a pressure differential equal to the compressor maximum working pressure and that they be suitable for service at a temperature at least equal to the compressor design discharge temperature.

Electronic Equipment Reliability

The control panel that operates an active magnetic bearing system consists of many electrical and electronic parts: power converters, power amplifiers, circuit boards, etc. Many of these components generate significant amounts of heat which must be dissipated. To do this, those components that generate the most heat are mounted on a copper heat sink that is water cooled. The water cooling is provided by a small, stand-alone chiller. The other components are cooled by air circulated in the panel by small fans. Back-up cooling water connections have been provided to allow cooling of the panels in the event of chiller problems.

Failure of any one of several components in the control panel can result in an unscheduled machine shut down. Depending on the failure, this could result in extensive downtime. Although a design review indicated a high mean time between failures (MTBF) for the electronics and user surveys had indicated few problems; to meet the high reliability required, and minimize the risk of extended downtime, Shell specified that two complete and identical control panels be supplied. These two panels would be connected to the compressor through a third "switching" panel. This switching panel allows operation from either of the two control panels and isolates them from each other. In the event of a failure in the active control panel, the compressor is shut down and brought to a standstill. The switching panel is manually switched from the failed panel to the standby panel and the compressor restarted. The out-of-service panel can be worked on and repaired as required, without affecting the operation.

SHOP ERECTION AND TEST

Mechanical Assembly

Generally, the first assembly of the compressor prior to testing proceeded as planned. The various assembly tools worked well and few revisions were required. Several cautions were added to the assembly procedure but these were of a common sense nature.

The most difficult part was convincing the mechanics that the specified tolerances for location and squareness were important and needed to be achieved. Setting the axial bearing rotor on the center of a 56 mil end float was, after all, something quite different from normal practice, especially since the location that had to be controlled from this setting was the end float of the outboard rundown bearing.

Obtaining the required clearance between the axial stops for the outboard rundown bearing was simple. It is controlled by the length of the shaft sleeve. When the axial bearing (thrust) rotor is positioned on the center of its 56 mil air gap, the sleeve is positioned so it is centered on the inner race of the outboard rundown bearing. The sleeve is located by the inner axial stop. The axial stop is located by a shaft shoulder. This is shown in Figure 9. Only after several explanations using the hardware and sketches as a guide, did the mechanics fully understand the requirements.

Subsequent disassembly and reassembly operations proceeded quite smoothly. By the time the unit was ready for shipping the erecting crew had become quite proficient. At that stage the turnaround time was not significantly different than a "standard" machine.

Electrical Installation

Active magnetic bearings are electrical components driven by a sophisticated control panel. The location of the machine with the bearings installed may be quite far from the location of the

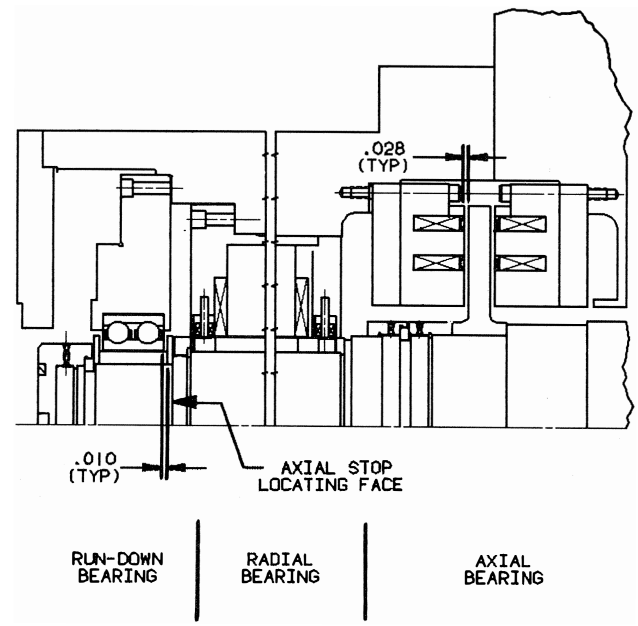

Figure 9. Axial Limit Stops—Outboard Rundown Bearing.

control panel. It is important that when the active magnetic bearings are tuned, the set up closely represents the actual installation (i.e. wire length). To this end, the customer-supplied interconnecting wire was used to set up for the shop test. Temporary wire ways constructed of wood were used to hold and protect the approximately 300 ft of wire required. Each wire was labelled to ensure proper reconnection in the field. The job feed-through connectors were used and the mating plugs were hard wired to the active magnetic bearing stators.

Tuning

Requirements for tuning were discussed with the active magnetic bearing vendor before the hardware was completed.

Two stages of tuning of the electronic systems are required: an initial static tuning to center the rotor in the bearings, and a dynamic tuning to adjust the stiffness, damping and response. One of the more time consuming tasks was finding the initial center position. To minimize this time span we manufactured a "tuning tool" kit. This kit consisted of two aluminum "blocks" sized to replace the rundown bearings but with an important difference. The block diameters were sized to be a snug fit in the bearing holder and on the shaft sleeve. The length of the outboard (thrust) end block was sized so that it was also clamped between the rundown bearing axial stops. With these tools installed, the rotor was blocked on the geometric center of the radial active magnetic bearings and with the axial rotor on the center of the air gap. This approach was very successful. However, any time made up at this stage was lost during the test runs.

Shop Testing

During Tuning

After initial static tuning, the unit was started and run for the final dynamic tuning. During the first full speed run an apparent instability occurred. It appeared as a gradual increase in vibration amplitude from about 0.8 mils to 5 mils at which time the control panel tripped the unit on both high amplitude and coil overcurrent. The frequency of the signal was predominantly rotational. The active magnetic bearing technicians made every attempt to electronically isolate the problem, with little success.

At that point, the engineers decided that perhaps the rotor was not adequately balanced.

Since there were two rotors, they were switched, but the second rotor ran, disappointingly, with similar results. While the second rotor was being run, the first rotor was check balanced at running speed. This indicated that it was unlikely that imbalance was causing the high vibration.

It was known during the design process that a potential control problem existed due to a node located between the bearing and the sensor. This node was associated with the shaft third node (first free-free node) which was approximately 20 percent above maximum operating speed, but well within the band width of the magnetic bearing control loop. The magnetic bearing vendor had provided for this potential problem by installing two sets of radial position sensors, one inboard and one outboard of each radial bearing. Temporary connection of the inboard sensor on the coupling end did reveal a node at running speed. This in effect meant the control loop was responding "in the wrong direction" and slowly feeding energy into the shaft at 1× running speed. The decision was made to switch to inboard sensors. Both rotors were then run successfully.

During Mechanical/Performance Testing

The compressor was finally ready for acceptance testing. After some preliminary runs to shake down the test loop, full scale testing was undertaken on a closed loop with an equivalent helium-nitrogen gas mixture.

Given the nature of this system and given the tuning problems, the first full scale test was closely monitored. Data was recorded at close intervals. All readings were within normal limits, when approximately one hour into the run, the test engineer noticed a sudden change. He immediately reached for the emergency shutdown switch. Before he could reach it, the control panel shut the unit down on high current, and the compressor came to an immediate stop.

On disassembly, the outboard active magnetic bearing stator was found to be welded solidly to the rotor. A review of the recorded temperature data indicated that the active magnetic bearing temperatures had not exceeded allowable limits until several minutes after the shutdown.

Fortunately, an engineer from the active magnetic bearing vendor was onsite to witness the test when the failure occurred. Subsequent study of the active magnetic bearing stator design and the failed hardware, indicated that the stator windings had pushed out of the laminations, contacted the rotor and shorted out. The failure had most likely been initiated by the expansion of the encapsulating epoxy due to the temperatures reached in the stator.

The active magnetic bearing stators were redesigned and rebuilt which required approximately twelve weeks, and involved all of the parties. The encapsulating epoxy on the axial bearing was changed. The radial bearing stators were rebuilt using a conventional motor winding varnish system.

With the new active magnetic bearing stators installed the unit was retested with good results. On completion of the testing (both rotors) and after acceptance by the customer, the unit was shipped to the job site. The compressor, speed increasing gear and motor installed on the bedplate are shown in Figure 8, prewired and ready to be shipped.

FIELD INSTALLATION

The compressor was installed in a conventional two-sided roofed enclosure. Due to the lack of a lubrication and seal oil system, it was possible to install the compressor at grade, saving about $100,000, on the installed cost. The electronic panels were installed in a nearby electrical building.

Installation and check out proceeded without significant incidents. As would be expected, getting all the wiring sorted out was a challenge. It was a pleasure, however, not to struggle through the flushing and cleaning of a lube and seal oil system.

Field tuning of the electronic systems was completed in two days.

The machine was successfully run for several hours on air to ensure that everything worked well; it was now ready to put into service.

OPERATING AND MAINTENANCE EXPERIENCE

Things That Went Well

Several aspects of this project are considered as successes.

Dry Gas Seal

The dry gas seal has worked effectively, with no problems. It has proven to be easy to operate.

Active Magnetic Bearings

The basic principles of active magnetic bearings works. The compressor has accumulated over 7500 hr of operation.

Auto-Balancing

This function works well. The machine operates at less than 0.5 mil vibration, and low forces are transmitted to the casing.

Operational Problems

Liquids. The problems on this compressor have come from an area least expected—liquids. Liquids in the machine have lead to a deterioration of the active magnetic bearing windings and the internal wiring. The resulting low ground impedance and ground faults have caused a series of problems with the electronic circuitry and circuit boards.

The liquids are water and hydrocarbons condensed from the process gas. The condensate has an aromatics content of over 20 percent which has softened and weakened the insulating varnish, and a water component which has caused ground paths at the connections between the lead in wires and the magnets.

Accumulations of liquids in low points within the machine could not initially be drained off. Many of the connections between the lead in wires and the magnets and sensors are in the lower part of the bearings. These connections were immersed in the accumulated liquids, accelerating the deterioration.

The sealed bulkhead "feedthrough" connections (Figure 7) have an internal plastic part which supports the pins and sockets. These connections were also at the bottom of the machine and became immersed in liquid. Some of the plastic parts in the connectors have been attacked by the aromatics, resulting in a loosening of the pins and sockets, and a deterioration in the quality of the connection. Subsequent evaluation has found that the damaged components were made of an off-spec plastic.

The ground paths provided by the weakened insulation and liquids caused high currents in the control circuitry resulting in several card failures. The electronic panel has a number of circuits which assist in the diagnosis of problems; however, this type of problem was not foreseen and was difficult for the repair teams to diagnose when the machine tripped.

Fuse

There has been one trip from a mechanical failure of a fuse in the control panel.

Overcooling

One trip has occurred due to over cooling of the control cabinet. The thermostat on the glycol chiller was set too low, resulting in moisture condensation on the electronic parts, and consequential failure.

Detailed Operating History

June 1988

The compressor was commissioned with the reformer operating on the old turbine-driven recycle compressor. It tripped immediately on excessive rotor shift. The machine had become flooded with liquid which had condensed and accumulated in the antisurge recirculation line. All the active magnetic bearings were grounded. The compressor ends had to be opened to drain the bearing cavities, as there were no drain valves. Drains were added. Welding machines connected to the magnet windings were used to provide heat to dry out the windings.

The compressor was restarted in full circulation while the turbine-driven compressor was reduced in speed and tripped. The new motor driven unit came on line smoothly as the antisurge controller closed the recirculation valve.

July 1988

The compressor tripped on excessive rotor shift. Overcooling of the control panel and condensation on the electronics caused a control failure.

Later in July 1988

The compressor tripped, indicating excessive rotor shift. A power amplifier fuse failed. Since the amplifier was okay, the fuse failure has been attributed to fatigue.

September 1988

While circulating nitrogen during a catalyst regeneration, the machine tripped on high thrust bearing temperature.

The cooling gas flow through the outboard bearing was hotter than normal (220°F *vs* 175°F); and the thrust bearing was drawing more current due to the higher thrust load. The actual operating conditions were different than the design nitrogen case. In future regenerations, a cold nitrogen purge will be fed into the bearing housing.

November 1988

During a process upset, the machine tripped on high level in the suction drum. The compressor was again flooded and the inboard bearing was grounded. The fault was traced to liquid trapped in the sealed bulkhead "feedthrough" connectors.

December 1988

While the reformer was down for major repairs, the compressor magnets were kept energized. The rotor became unstable, and control was transferred to the standby control panel. A resistor in an oscillator circuit had burned out.

January 1989

As the reformer was being started, the compressor rotor again became unstable. Two of the bearings were replaced due to grounds: one thrust bearing; and the outboard radial bearing. The compressor stayed down until April.

April 1989

After a reformer turnaround, the compressor could not be started, due to instability on both control panels. One circuit card in each panel had failed.

June 1989

The machine was placed in service and ran until October 1989, when it tripped on excessive rotor shift. Again two bearings were grounded. The machine has been out of service since that time.

Operators/Electricians' Comments

In addition to the previous comments, the operators and maintainers of this compressor had these comments:

- The electronics are very sensitive; they even have to be tuned to the wire length. It is uncomfortable to deal with this amount of sensitivity.
- Panel testing after a repair is not really possible without shutting down the compressor. The chokes provided for testing are not a good simulation of the real machine.
- The machine often trips without warning when there are active magnetic bearing system problems, even before vibration or current indications are high. It behaves like a light bulb, it's either on or off.

CURRENT STATUS

At the time of writing, the owner, compressor vendor, and active magnetic bearing vendor are working on changes to eliminate the liquids related problems. The following changes are being made:

- Connections on the thrust bearing between the lead-in wires and the magnet coils are being improved, and will be encapsulated.
- The thrust bearing will be rotated so that all the connections are above the centerline.
- Thrust bearing encapsulation will not be changed.
- Radial bearing windings will be changed to a fiberglass/epoxy insulating system, provided that chemical resistance tests, currently in progress, are passed.
- Some coil-to-coil connections in the radial bearing windings will be eliminated.
- Connections between the radial bearing lead-in wires and the windings will be improved and encapsulated.
- Radial bearing connections will be shifted above the centerline.
- The sealed bulkhead "feedthrough" connectors will be replaced with ones of a proper material.
- The outboard and inboard bearing housing will be rotated such that the "feedthrough" connectors are above the centerline.
- A dry hydrogen purge will be added to the outboard bearings to prevent accumulation of condensable vapors.

SUMMARY

This project has been successful in many ways. The active magnetic bearing and dry gas seal technologies have been demonstrated and found workable. The anticipated capital cost savings in installation have been proven. The effectiveness of close user-manufacturer relationships have been demonstrated to the parties involved, and these relationships are continuing in the development of solutions to the current problems.

The anticipated reliability and operating cost benefits have not yet been proven on this machine, due to the unforeseen severity of the liquids problems. The participants are confident that the changes currently in progress will resolve these problems.

REFERENCE

1. Junge, M., and Conquergood, C. P., "Secondary Sealing Methods for Dry Gas Seals," Revolve 89 Conference, Calgary, Alberta, Canada (1989).