FULL-LOAD TESTING OF AN ALL-ELECTRIC CENTRIFUGAL COMPRESSOR FOR MISCIBLE GAS INJECTION

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In 1990, he transferred to the turbocompressor division of Sulzer to lead the aerodynamic and mechanical development group and later the combined engineering and development departments. During this period, he was closely involved with the development and market introduction of the MOPICO pipeline compressor and the related HOFIM systems—both oil-free, high-speed turbomachinery concepts.

Mr. Marriott served an apprenticeship in Aeronautical Engineering with Bristol-Siddeley and graduated in this discipline at the Bristol College of Advanced Technology in England.

ABSTRACT

A two casing high-pressure injection compressor train driven by a directly coupled high-speed induction motor was tested to demonstrate satisfactory electrical, aerothermal, and rotordynamic performance. Some key test aspects are discussed and illustrated with typical results. These include the influence of misalignment

and thrust reaction forces on magnetic bearing loading, compressor rotor stability in magnetic bearings, rotor behavior in the landing bearings during simulated magnetic bearing failure, performance of the active thrust control system under transient and simulated failure conditions, and controls tuning of the variable frequency drive. Observations and difficulties encountered during commissioning and plant startup in the field are also discussed.

THE POINT MACINTYRE EOR PROJECT

The Point MacIntyre Enhanced Oil Recovery (EOR) objective is to introduce a miscible injectant (MI) into the Point MacIntyre reservoir at a nominal rate of 50 mm scfd to ultimately recover an additional 32 million barrels of oil. Point MacIntyre is one of the largest fields in the United States in terms of production, and reached a facility-constrained plateau of 165,000 barrels of oil per day in 1996. The original oil in place is estimated to have been approximately 800 million barrels. The field is produced through the Lisburne Production Facility at Prudhoe Bay on the North Slope of Alaska. The Lisburne facility is operated by BP Exploration Alaska, Inc., and is owned by BP, ExxonMobil, and Philips Petroleum.

The MI gas is produced at the Lisburne natural gas liquids (NGL) plant. The process streams used to create the MI gas are the vapors from the economizer drum and the depropanizer tower. These "rich" solvent components are compressed with the HOFIM compressor train from 16.8 bara (244 psia) to approximately 283 bara (4100 psia). Residue gas from Lisburne (20.5 molecular weight natural gas) is blended with the "rich" solvent stream to conform to minimum miscibility pressure criteria for Point MacIntyre, i.e., 27 molecular weight. Blending of the rich solvent and residue gases takes place either upstream or downstream of the new MI compressor.

The MI gas is shipped from Lisburne via a 12-mile long pipeline to the two Point MacIntyre drill site facilities. The MI is then injected into the reservoir using existing water injection wells under a water-alternating-gas scheme.

MOTIVATION OF COMPRESSOR CHOICE

Project motivation for the use of this innovative approach was driven largely by the compactness of the design. The high-pressure injection compressor train weighs only 27 tons versus 90 tons for the conventional format. Additionally, the size of the machine is proportionally less. This allowed packaging into two modules totaling 180 tons, versus 600 tons for the conventional case, which would have required the module to be barged to site. The compressor modules were constructed in Anchorage, Alaska, and then trucked 800 miles over paved and gravel roads to the North Slope. This provided substantial schedule flexibility over barging the equipment, since barges are constrained to a 6-week window during the Arctic summer when the Beaufort Sea is ice-free.

The smaller footprint of this compressor train also allowed for the design of a much smaller module that made a substantial contribution to reducing project costs. Equipment modules on the North Slope provide weather protection for equipment and personnel. Costs for the heating/ventilation/air-conditioning (HVAC) system and the fire and gas systems for such modules are directly related to the size of the module. Therefore reducing the size of the equipment has a beneficial impact on the cost of these ancillary items in addition to reducing the cost of the steel required to build them. Figures 1 and 2 give an idea of the compactness of the two modules.

CONFIGURATION OF THE COMPRESSION SYSTEM

The overall configuration of the compression system is based largely on the design of the original high-pressure injection demonstrator, which commenced operation in 1993. In comparison to the demonstrator, the Point MacIntyre system features a number of significant differences in operating conditions and parameters,
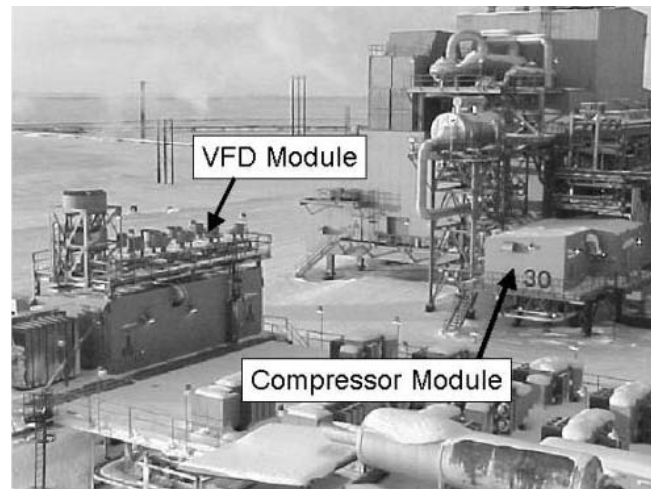
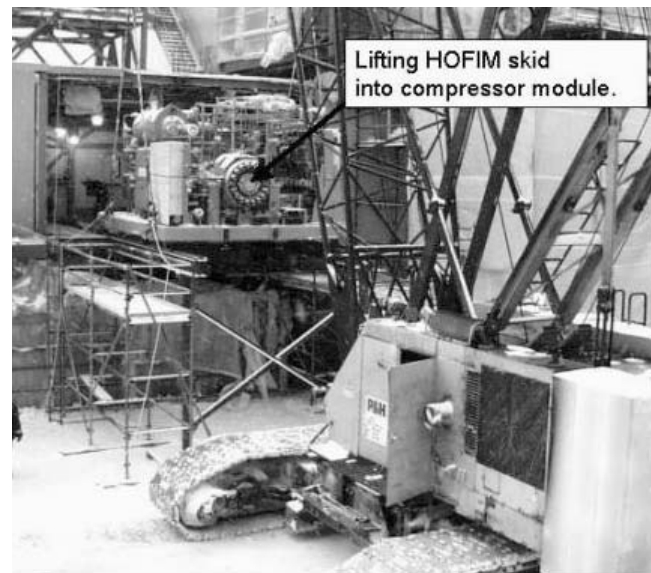

Figure 1. Module Site.

Figure 2. Lifting the Skid into the Module.

which entailed further development of the original concept. Details of the motor-compressor and drive systems of the demonstrator can be obtained from Gilon and Marriott (1993) and Graham (1993).

The Motor-Compressor Train

The high pressure ratio of the Point MacIntyre application required two compressor casings with an intermediate cooler. During initial system design, a number of possible arrangements were investigated. The final choice was a double-ended motor with casings at each end, in "in-line" arrangement. All three machines are axially rigidly coupled, the residual thrust of the train being compensated by an active thrust control system. The shaft system is thus supported on six radial magnetic bearings, with a single axial magnetic bearing in the motor. The axial magnetic bearing absorbs rapid thrust transients and provides the input signal to the programmable logic controller (PLC) of the active thrust controller. In view of the high dew point of the MI gas, it was decided that all critical sealing and cooling functions (dry gas seals and magnetic bearings) would be carried out using lean residue gas (20.5 molecular weight).

The high-speed motor design is essentially the same as the demonstrator of 1993, i.e., fully laminated, shaftless rotor and special low loss stator winding. However, due to the double-ended

drive design it was not feasible to use an integrated cooling fan. Instead an electrically driven external cooling fan is used.

Figures 3 and 4 show the cross section of the low-pressure (LP) and the high-pressure (HP) compressors, respectively. Both the LP and the HP compressors have six impellers. Figure 5 shows the skid with the compressors mounted. The HP body is nearer the camera.
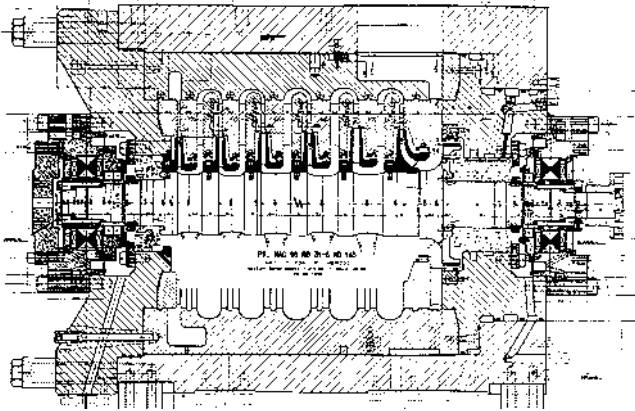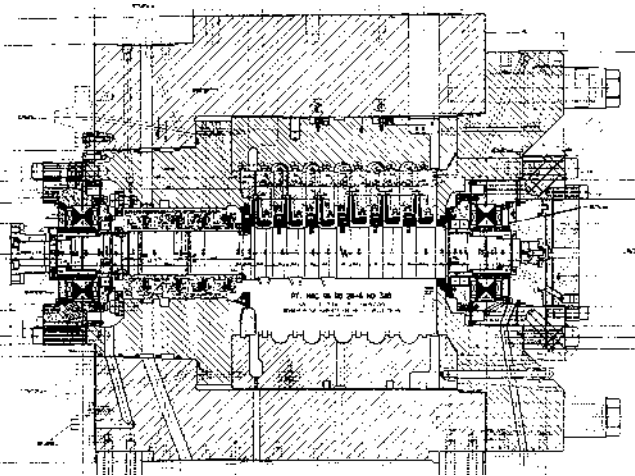

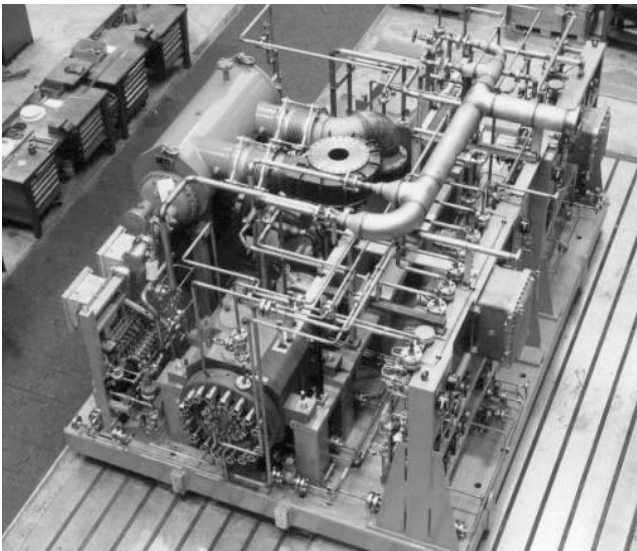*Figure 3. LP Cross Section.*

Figure 4. HP Cross Section.

Figure 5. Assembled Motor-Compressor Train.

The inboard seals of the LP and HP compressors are buffered with nitrogen. The same medium is used to cool the motor.

Figure 6 illustrates diagrammatically the arrangement of the shafts, bearings, seals, and thrust control devices. Figure 7 shows the thrust control scheme.
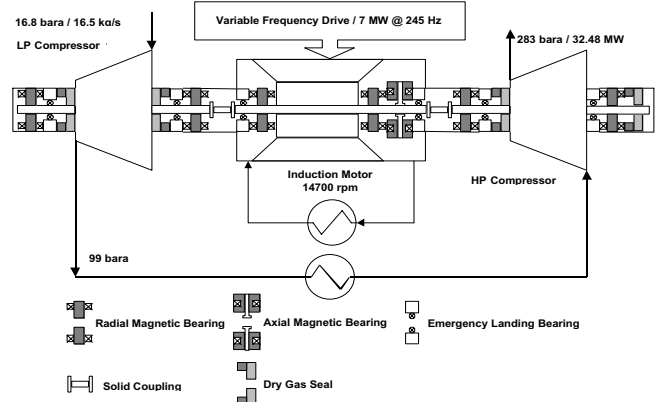

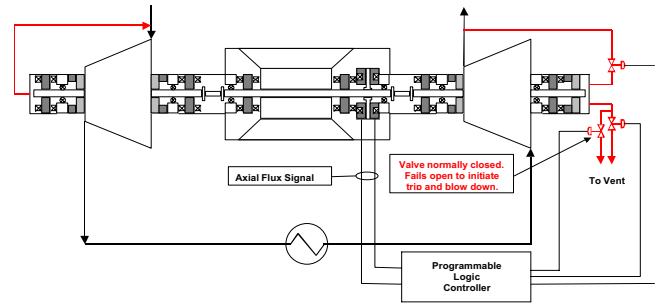*Figure 6. Arrangement of Shafts, Bearings, and Seals.*

Figure 7. Closed Loop Active Thrust Control.

The Drive System

Although compressors regularly operate at the speeds and powers required for the Point MacIntyre application, the motor and the variable speed drive pushed the frontiers of prior technology. Figure 8 shows the current worldwide experience in high-speed drives. Each dot represents a variable frequency drive (VFD) greater than 1 MW and above 3600 rpm speed. The smooth curve joins the hitherto limiting points from 5000 rpm at 32 MW to 20,000 rpm at 2 MW. It can be seen that the Point MacIntyre application lies well above this line, 74 percent more power than the previous limit at 14,000 rpm and 32 percent faster than any other 7.1 MW drive.
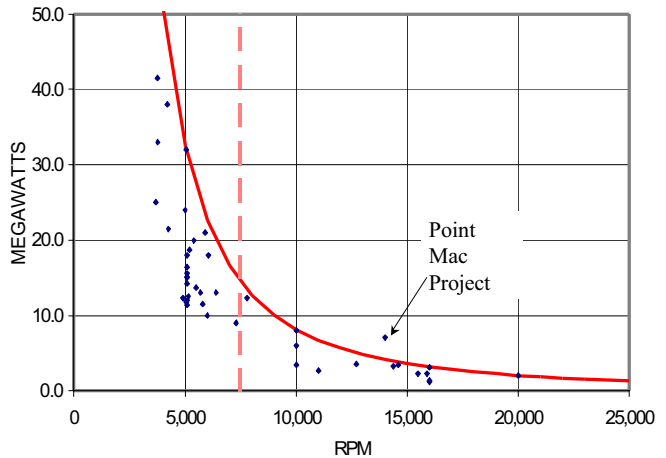

Figure 8. Worldwide Experience of High-Speed Electric Drives.

The technology used for this high-speed, high power drive is a capacitor compensated induction motor drive as shown in Figure 9. The power is taken from the 13.8 kV site bus and stepped down to 2600 VAC by the 9750 kVA transformer. The transformer has two secondary windings with 30 degree phase displacement to give a 12-pulse current wave shape in the supply, as required to meet the harmonic limits of IEEE 519 (1992). Further details can be found in McBride and Franks (2000).
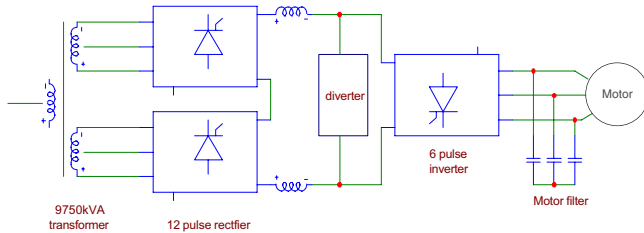

Figure 9. Simplified Diagram of Electric Power System.

The inverter uses fast turn-off thyristors to be able to operate at 245 Hz. Although the inverter is only a 6-pulse bridge, the motor filter capacitors trap all the high power current harmonics, thus the motor sees a reasonably smooth sine-wave current.

When incrementing the power and speed up to the levels needed for the Point MacIntyre application, a number of issues had to be addressed. As the frequency rises, the switching losses in the thyristors increase, so more cooling is required. In addition, the air-cooled reactors see eddy current losses proportional to the square of the frequency. Both aspects were considered during the design phase, but could only be proven during shop test heat runs. Neither was adequate at the first test, thus cooling water circulation was altered to provide cooler water to the thyristors, and the reactors were rewound with larger air circulation paths.

THE TEST PROGRAM

Testing of the complete transformer/variable frequency drive/motor/compressor system was carried out in a turbocompressor test facility in Zurich under essentially class III conditions, i.e., using inert gas mixtures formulated to suit the purposes of the individual tests. The program of tests carried out can be summarized as follows:

- LP compressor performance test, ASME PTC 10, class III (1997)
- HP compressor performance test, ASME PTC 10, class III (1997)
- Train mechanical running test, API 617 (1995)
- HP compressor full load test
- Active thrust control system failure test
- Rotor drop test (magnetic bearing failure)
- Gas leakage tests
- Motor pressurization, thermal, and electrical tests
- Variable frequency drive stability and undervoltage tests

The primary objectives of the test program were to simulate the major transient and stationary operating conditions and to demonstrate that system operability was satisfactory or that safe shutdown was possible. Test electrical supply conditions prevented operation of the train at full load; however approximately 80 percent of the full power level could be achieved during the HP compressor full load test.

Stability of system components: aerodynamic, rotordynamic, electrical, and thrust control, was considered to be a major acceptance criterion and tests were devised to investigate these aspects, e.g., injection of disturbance signals via the magnetic bearing system to demonstrate compressor rotor stability (log decrement).

The purposes of the HP compressor full load test were:

- Verify HP compressor stability across the operating range at power levels and gas densities comparable to the actual field condition.
- Demonstrate the compressor was free from rotating stall.
- Demonstrate acceptable mechanical performance of the compressor, magnetic bearings, seals, motor, and VFD at a load of approximately 6200 kW.

This full load test utilized an 80/20 mixture of nitrogen/CO_2 at a suction pressure of 115 bara (1668 psia) to load the HP compressor to approximately 2800 kW. Simultaneously the LP compressor was loaded to about 3400 kW with a 60/40 mixture of nitrogen/helium at a suction pressure of 40 bara (580 psia).

During this full load test, a sine sweep was introduced via the magnetic bearings and the log decrement was measured to verify acceptable stability. Figure 10 is a schematic of the stability test setup. The sine sweep was across a frequency range of zero to 200 Hz. The criterion for acceptability of the test was that the log decrement had to be at least 10 percent. Stability tests were conducted on the HP compressor at three operating points at 100 percent speed with one point being near the surge line. An additional stability test was conducted on the HP compressor at 105 percent speed.
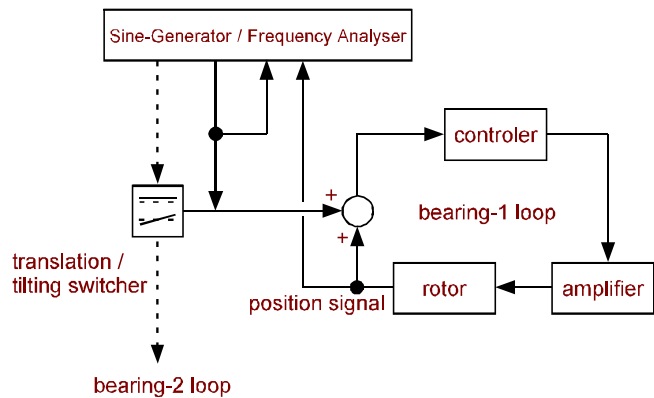

Figure 10. Injection of a Sine-Sweep into the Bearing Controller.

The setup of the compressor test loops is shown in simplified form in Figure 11. Figure 12 is a photo showing the overall test arrangement.

Stability tests were also conducted on the LP compressor at two different speeds, although these had to be done at loads less than design due to test stand limitations.

The rotor drop test was developed to demonstrate that, in the event of a failure of the magnetic bearing or controller, any of the three rotors would land on the landing ball bearings and the rotors would coast down without damage to the bearings or the machine. Almost immediately upon failure, the magnetic bearing will have zero damping and negative stiffness, so damping must be provided in the landing bearings. In this design, the landing ball bearings are full complement face-to-face duplex oriented angular contact bearings with damping provided by Borelli ribbons between the outer race and the bearing housing. Further details of this damping arrangement can be obtained from Schmied and Pradetto (1992).

During a third party rotordynamic audit, a transient analysis of the rotor drop was modeled that included various unbalance and aerodynamic loading cases. The modeling had indicated that we might expect impact on the backup bearings as high as 20 G's. Also the modeled cases had shown that with aerodynamic forces from the labyrinth seals, the HP compressor could go into a high amplitude whirl during the shutdown. Based on this analysis it was decided to drop each rotor individually with the rotor unbalanced to at least five times the specified balance of ISO 2.5 G by applying weights at the two couplings and at motor balance planes.
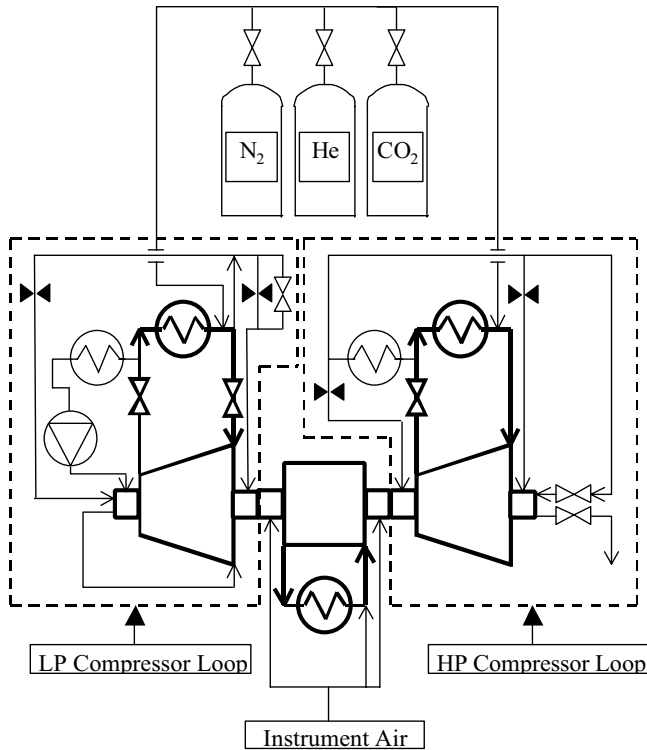
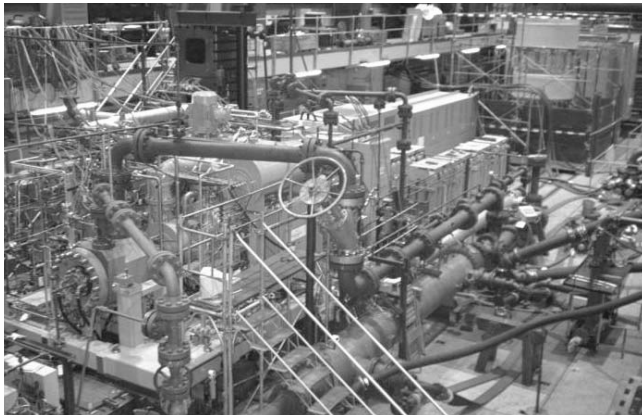*Figure 11. Compressor Test Loop Arrangement, Simplified.*

Figure 12. Shop Test—View from Compressor End.

Failure was considered unlikely for more than one magnetic bearing system at a time because its own controller cabinet controlled each of the three rotors, and each cabinet was provided with individual uninterruptible power supplies. Also the magnitude of impact forces was comparable regardless of dropping the three bodies individually or simultaneously according to the transient model.

The tests were conducted with the machine at 14,000 rpm and 5400 kW so that the compressors provided sufficient aero braking to slow the rotor down to less than 6000 rpm within a minimum time (around 10 seconds). Longer periods of operation on these nonlubricated ball bearings would have been damaging. A power failure of a single bearing cabinet was simulated by switching off the power to the cabinet while simultaneously tripping power to the drive. The magnetic bearings were reactivated after about 5 to 7 seconds because of concerns that aerodynamic braking might not be sufficient to slow the system down adequately to prevent thermal damage to the bearings. Each rotor was tested in this way.

During the drops, rotor vibration position and phase were recorded at each of the six bearings and processed through oscilloscopes and fast Fourier analyzers. After each drop test the

compressors were shutdown, cooled, and the radial and axial ball bearing clearances were reconfirmed. The magnetic bearing controller has a diagnostic feature that allows the landing bearing clearance to be checked after shutdown by levitating the rotor and moving it across the clearance. After all drop tests were completed, the two compressors were disassembled and all bearings and compressor labyrinths were visually inspected.

HIGHLIGHTS FROM TEST RESULTS

Overload of the Radial Magnetic Bearings

When substantial gas pressure was first introduced into the compressor casings, the magnetic bearing on the drive end of the LP compressor tripped off due to excessive current. This occurred while the rotor was levitated by the magnetic bearings but not rotating. The radial load capacity of the compressor bearings, with silicon iron armatures, was 2000 N (450 lb), which was more than double the load to be supported.

Each magnetic bearing has a proportional-integral controller that attempts to force the rotor to the center of the bearing. Because solid couplings connect the compressor and motor shafts, any deviation of the centers of the six radial bearings from a straight line causes reaction forces at the radial bearings, with the rotor acting like a statically indeterminate beam on six supports. Thus the increase in bearing current indicated that introducing gas pressure was causing the bearing centers to move off line.

The load supported by each bearing can be calculated from the current applied to the armature coils. Before gas pressure was introduced, these loads were as expected for the rotor weights plus the cold alignment offsets for thermal growth. This confirmed the initial alignment of the LP and HP compressor bodies, and the motor, was correct prior to introducing pressure.

Referring back to Figure 7, note that gas pressure from the LP suction acts on the gas seal at the outboard end of the LP shaft. The resulting axial thrust on the shaft is balanced by pressure on the gas seal on the opposite end of the shaft. The thrust controller adjusts this pressure to produce zero net thrust. However, the resulting reaction force on the LP and the HP compressor bodies is about 15 metric tons (17 tons), at rated pressures, pushing each compressor body outboard. This thrust is larger than a conventional train with balance pistons. Massive axial members at the bearing centerline height prevent bending of the baseframe due to the large axial force.

The axial reaction force from each body is transmitted to the inboard pedestal by a transverse key, shown in Figure 13, designed to allow for radial thermal expansion. This arrangement has been used successfully for years with solid coupled shafts. The key has a shim on its top side to support the body at the correct height for shaft alignment.

Figure 13. LP Compressor Support Feet.

Figure 14 shows that axial thrust and its reaction act on this key in opposite directions, but not at the same elevation. This produces a couple, which tends to rotate the key. Because the shim was on top of the key, any rotation lifts the compressor body causing the problem. To fix the problem, the shim was removed from the top of the key, so that key rotation would not lift the compressor. As shown in Figure 15, a shim was placed beside the key, directly between the foot and the pedestal, to set the vertical alignment. Alignment and magnetic bearing currents have been satisfactory since this change. The silicon iron armatures of the magnetic bearing were replaced with cobalt iron armatures, thus providing additional capacity against any other misalignment problems.
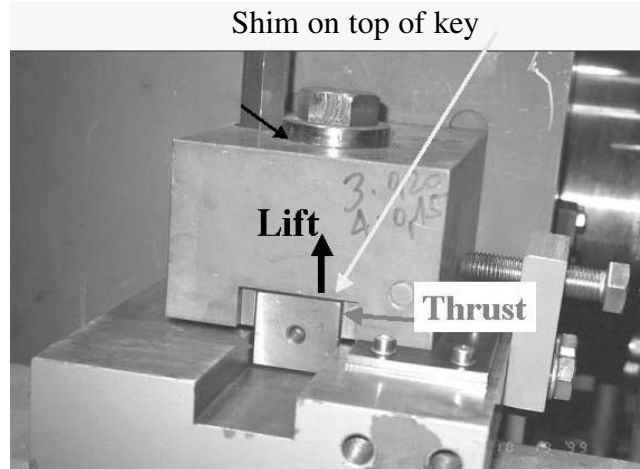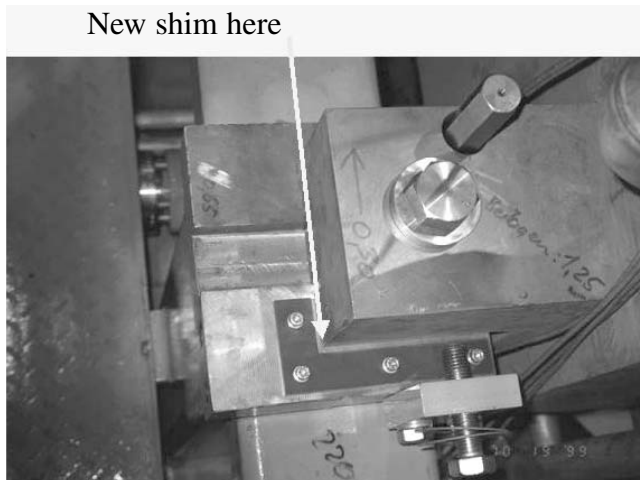

Figure 14. Problem Caused by Shim on Top of Key.

Figure 15. Fix—Move Shim.

Thrust Control Fail-Safe Test

During initial testing at high pressure, the thrust control failed and the drive shut down. While coasting to a stop, the resulting load on the landing bearing in the motor exceeded the bearing capacity with minor consequential damage. A review of the protective system recommended adding a fail-safe design, which would prevent excessive thrust in case of loss of thrust control.

This was accomplished using existing hardware. One port of the digital control valve now fails open, and the drive trips off on loss of thrust control. The port size is matched to the system pressure and volume characteristics to approximate the action of the controller while the compressor is coasting to a stop. The fail-open valve is shown in Figure 7. The digital valve is shown in Figure 16. (The valve is called digital because, starting with the smallest port,

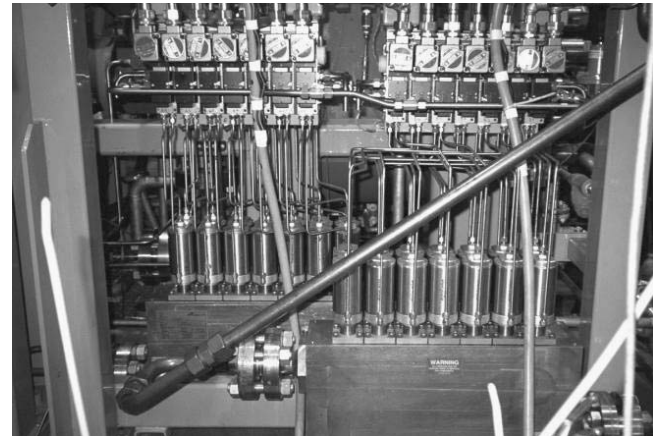each subsequent port has twice the area of the last. Also the ports must be either open or closed.)

Figure 16. Digital Thrust Control Valve.

The port size was calculated using the test loop piping volumes and the test gas composition and states. The calculation was confirmed by tripping the drive with the controller active and observing the digital valve, to see if the port area actuated by the controller was the same as the port area chosen for the blow-down. Figure 17 plots the test results of compressor speed and of port area as a function of the coastdown time. The outlet valve cycled between 1.5 and 3 mm^2 (0.0023 to 0.0047 in^2), confirming the calculated port area. Blow-down calculations for the field installation determined a different port size is required with the compressor installed at Point MacIntyre.
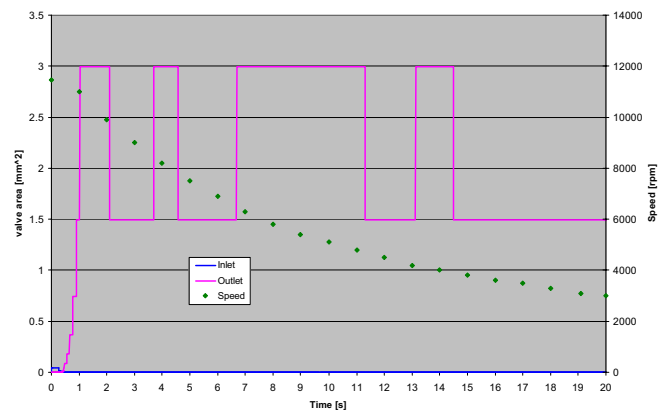

Figure 17. Valve Openings upon Trip.

To confirm that the thrust control system is fail-safe, the thrust controller power supply was shut off while the compressor was operating at high pressure and at 11,500 rpm. The compressor automatically tripped, as designed, and the fail-open port actuated. Figure 18 shows the resulting thrust, as measured from the magnetic thrust bearing flux, versus time during coastdown. RPM is also shown. The maximum measured thrust was less than 10,000 N (2200 lb) compared to the magnetic bearing capacity of 25,000 N (5600 lb), a comfortable margin.

If the compressor surges, the thrust will vary rapidly, and the controller will try to follow. Figure 19 shows the action of the thrust controller when surge was deliberately introduced during the high-pressure test. During the transient, the thrust was measured by the magnetic bearing flux. The thrust force exceeded trip limits, and the drive was tripped. A touch was recorded on the landing bearing, but no damage occurred. This test demonstrated that no damage would occur even if the surge control system failed to prevent surge.
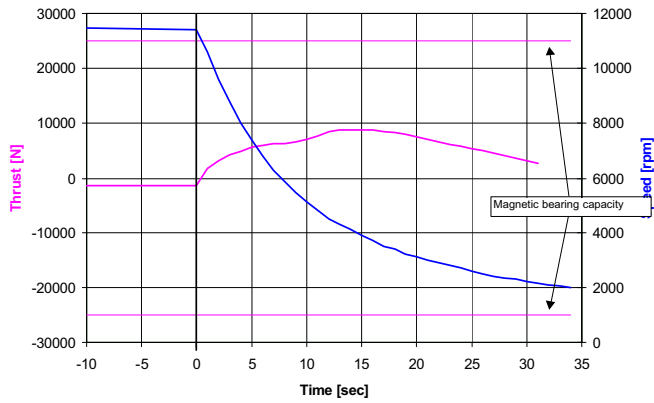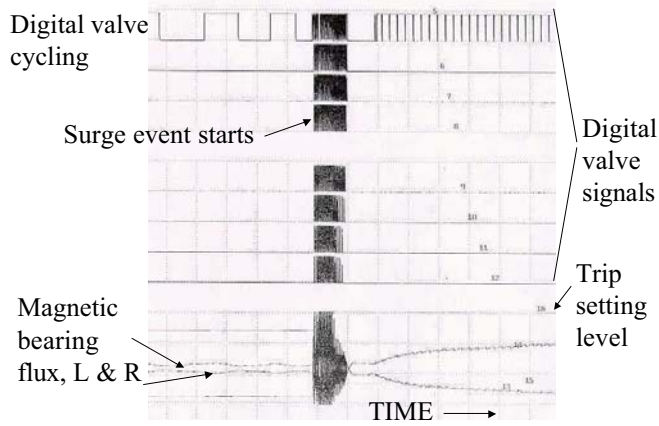
Figure 18. Thrust Panel Failure Test.

Figure 19. Surge Test of Thrust Controller.

Figure 20. Log Decrement Measurement During High Pressure Test.

Figure 21. HP Rotor Drop Test—Speed Versus Time.

Measurement of Rotor Stability (Log Decrement)

Centrifugal compressors operating at the pressure and gas density required for Point MacIntyre raise concerns about rotordynamic stability against self-excited subsynchronous vibrations. With magnetic bearings, the log decrement can be measured at full pressure, full speed operation, to confirm the rotordynamic design. A frequency-swept sine-wave signal was introduced into the magnetic bearing controller to perturb the rotor lateral vibrations. Considering the rotor as a linear dynamic system, the log decrement can be determined by the width of the dynamic response at the fundamental bending frequency. Figure 20 shows, for the 72 Hz mode, a response width of 14 Hz at the half-power point. The corresponding log decrement is 0.61, indicating satisfactory rotor stability.

No rotating stall could be found in the operating map of either compressor. Both the LP and the HP compressors have vaned diffusers on every stage.

Rotor Drop Test

As discussed under section "THE TEST PROGRAM" above, magnetic bearing control failure was simulated by turning off power to each of the three control cabinets, one test at a time. Thus one body drops onto the landing bearings, while the other two are supported by the magnetic bearings. Bearing control failure initiates a trip of the compressor train, which then coasts to a stop. The bearing control power was reenergized after the rotor response was demonstrated, as a practical step to avoid excessive landing bearing wear on test. Figure 21 shows the HP compressor drop, starting at 14,000 rpm at time zero, with the bearing to be reenergized at 7000 rpm. As the drive had tripped, the train coasts toward 0 rpm, but decelerates ever more slowly because the gas in the test loop is being released simultaneously.
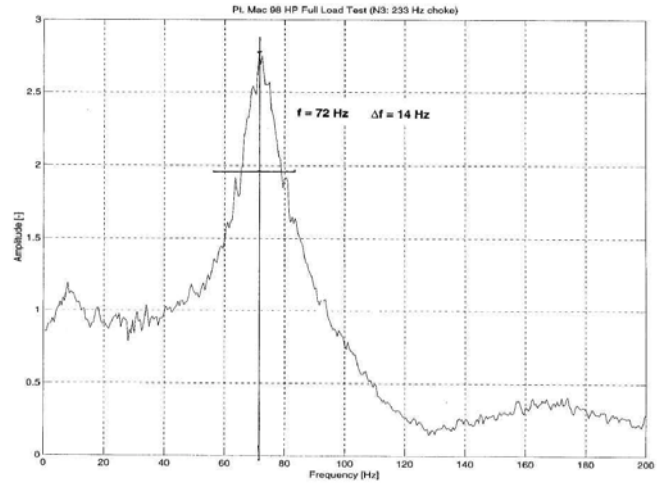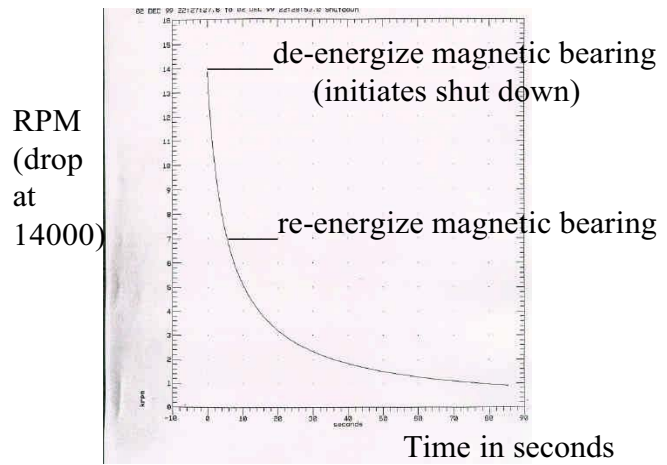
Figure 22 shows a waterfall plot of the vibration spectra during the HP drop test. Initially, at 14,000 rpm, two subsynchronous modes appeared, one at 70 Hz, and a second at 150 Hz. It is interesting to see two instability frequencies, instead of only one as commonly observed. When the magnetic bearing is reenergized, a transient is apparent due to the step function in force driving the rotor back to the center of the bearing. The subsynchronous modes drop out after the spectrum taken at 10,500 rpm. Subsynchronous noise is suppressed after the bearing is reenergized.
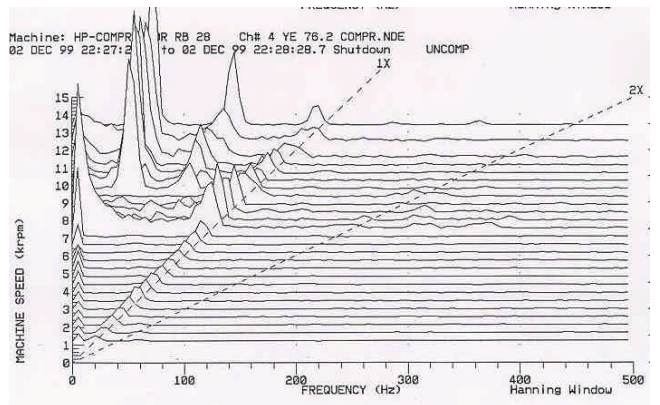

Figure 22. HP Rotor Drop Test—Vibration Spectra Waterfall Plot.

Figure 23 shows the orbit at the HP nondrive-end bearing during the drop test. The circular orbit at the center, of 20 µm (0.8 mil) diameter, is the synchronous response prior to the drop. The 150 µm (6 mil) radius represents the touch circle of the landing bearing. The 220 µm (9 mil) radius represents the maximum orbit of the landing bearing when the damper ribbon behind the bearing is fully deflected.
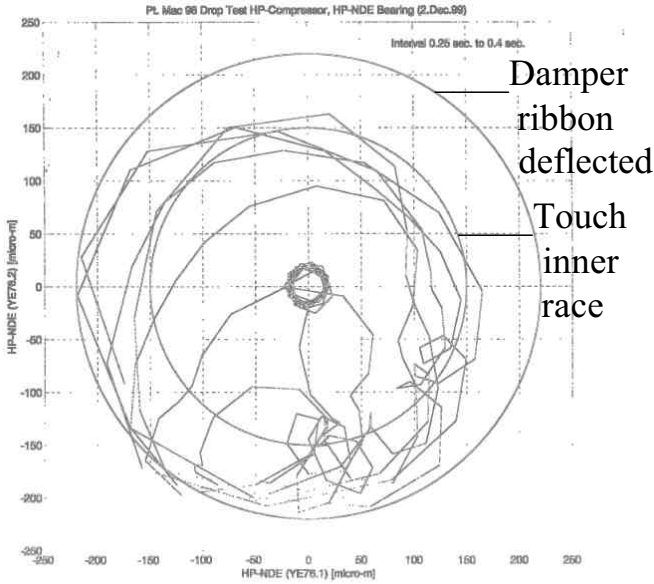

Figure 23. HP Rotor Drop Test—Shaft Orbit.

Figure 24 is a photograph of the rotor and its labyrinth seals after the drop test. Figure 25 is a photograph of the stationary portion of the labyrinth seals. Note the swirl brakes at the entry side of the seals. The labyrinths were not damaged by the drop test, showing the rotor vibrations were controlled by the landing bearings.
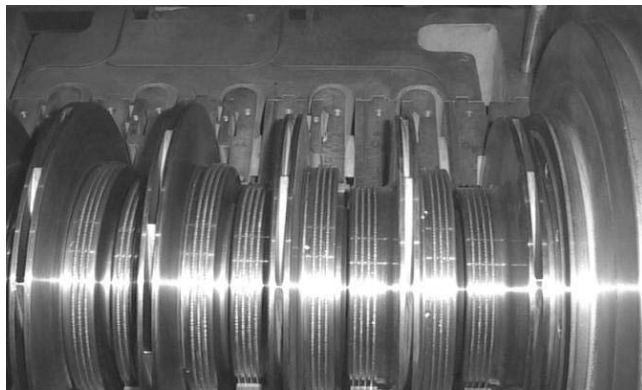

Figure 24. HP Rotor and Labyrinth Seals after Drop Test.

Drive Tuning

The entire drive and its supply transformer were shipped to the manufacturer in Zurich for combined testing with the motor and compressor. To achieve the tests required a special tap being added to the transformer at the design stage to get full secondary volts with the test station supply of only 11 kV at 50 Hz compared with the site condition of 13.8 kV at 60 Hz. In addition, since the source impedance was significantly higher than the normal site conditions, changes were required to the current loop gain to achieve the desired response times.

Starting up a VFD system on a motor with magnetic bearings requires a very cautious approach as the drive requires the motor to

Figure 25. HP Stator and Labyrinth Seals.

run to set the control responses, and yet the bearing controls need to be tuned before any significant speeds can be achieved. Both the VFD engineers and the bearing experts were present while the speed was progressively increased, and the compressor load had to be adjusted at each stage to provide only enough torque to ensure the stability of the whole system.

To maximize the benefit of prior experience, the software was adapted from standard versions in use on other high power drives but at lower frequencies. It was found on shop test that the high-speed software was limiting and some of the unnecessary features had to be removed to achieve the full 245 Hz frequency. Inevitably the changes had to be debugged progressively as each one appeared during the test program, but eventually the motor was run stably at full speed at about 6100 kW, which was the maximum load achievable on the test bed. Figure 26 shows the quality of the motor volts and amps at 245 Hz.
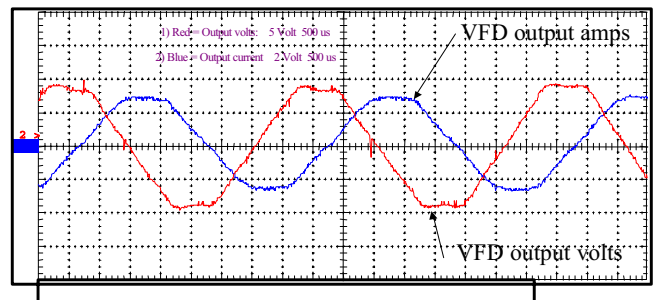

Figure 26. Motor Volts and Amps at Full Speed.

TEST SUCCESSES

The successes achieved during testing can be directly related to the innovative features of the compression system. Starting with the variable frequency drive: despite the problems encountered in tuning the controls software, it could be demonstrated on test that the design concept of the drive is fully adequate for a power-speed duty that is well outside previous experience (refer to Figure 8). The same can be said for both thermal and mechanical design of the induction motor. Although the demonstrator motor was designed for higher speed (20,000 rpm as opposed to 14,700 rpm), the power level for the Point MacIntyre application is more than three times higher. This is of particular significance for the cooling system of the motor.

A train with six radial magnetic bearings and axially rigid couplings was also a novelty. In such a system there is considerable potential for interaction between the individual bearing controllers.

However, the actual time required to tune the bearing system on test corresponded to the predictions of the bearing manufacturer. Rotorbearing behavior was also close to the rotordynamic predictions. Independent predictions of the rotor behavior during a drop incident were confirmed on test. Thus further validation of the numerical tools used to predict rotorbearing performance was obtained from the test program.

Finally, the concept of a train using active thrust control, without balance piston, was shown to be feasible for residual thrust levels much higher than normally encountered in a conventional high-pressure train. In particular, the strenuous test conditions showed that the thrust controller could accommodate the rapid transients, which occur during trip. This confirms the overall power savings that can be achieved by eliminating the balance piston losses on such high-pressure machinery.

COMMISSIONING IN ALASKA

After the equipment was received in Alaska, it was installed, grouted, and aligned. All piping and vessels were thoroughly cleaned and the system then closed and pressure tested. The VFD, all control valves, instruments, and controls were functionally checked statically.

To assure successful long-term operation, a startup/test plan was developed. The following phases and goals for the testing were established:

● *Cleanup Run(s)—*

• Debug first part of start sequence to allow machine operation on 21 molecular weight lean gas.

• Verify stability of magnetic bearings and VFD with operation at a loading of about 3900 kW.

• Circulate system in complete recycle for sufficient time to verify cleanliness of suction strainers.

• Verify surge control settings and adjust as necessary.

• Verify proper functioning of dry gas seal system, motor, drive, etc.

• Confirm aerothermal performance data for the compressors.

• Monitor alignment and vibrations to assure acceptable mechanical performance.

● *Mechanical Run(s)—*

• Debug remainder of start sequence to allow machine operation on blended gas between 27 and 32 molecular weight.

• Verify all other aspects of machine operation (as listed above under the cleanup runs) at a load of approximately 7000 kW.

The most difficult part of commissioning the equipment proved to be debugging the startup sequence and tuning the various process controllers. The compressors are designed to start operation in full recycle on the 21 molecular weight lean gas. During this period of operation, the compressors are in full recycle and the process system is warmed up. After reaching the specified temperature, a mixing valve is opened in the inlet piping that blends a proper proportion of a 36 molecular weight gas with the 21 molecular weight lean gas to achieve the required composition for optimum miscible injection.

The blending was particularly sensitive to valve tuning and system dynamics. Much effort was required to successfully start the equipment without tripping. Early attempts saw very rapid pressure and compositional changes within the compression system that ultimately resulted in the machine tripping on high axial rotor movements.

Another factor also leading to trips on axial position was the sequencing of a control valve that admitted gas to the LP balance piston. The intent of this valve was to bias the axial thrust of the rotor when at normal operating conditions. The quarter-turn valve was originally designed to switch open at 173 barg (2500 psig) discharge to provide an optimum control range for the active thrust

controller. This action along with system transient response as the rich gas was blended proved troublesome. Eventually a solution was to change the valve to a control valve and modify the pressure at which the valve was opened.

Other aspects of machinery performance, vibration, stability, and aerothermal performance were found to be completely acceptable.

The machine was in service at design conditions for about two weeks when a short occurred within the stator windings of the motor. An internal shroud, made from copper alloy, was positioned around the rotor short-circuit end ring area. It had broken loose and contacted the stator windings causing the short. Figure 27 is a picture of the bearing shroud that had become loose. The shroud had also contacted the rotor, but did only minor damage to the rotor and was easily corrected by grinding and polishing.
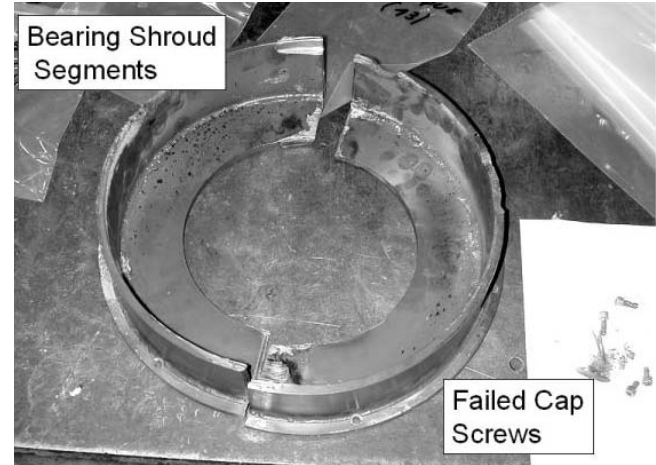

Figure 27. Loose Bearing Shroud.

This shroud was an axial cylindrical extension of the motor bearing housing and was attached on this bearing housing. Figure 28 is a picture of the motor bearing housing and rotor indicating where the bearing shroud was attached. Inspections of the failed pieces revealed that the cap screws that held the shroud in place had broken or backed out, allowing the shroud to fall into the rotor and into the stator windings. The flanged area of the bearing housing and the shroud were also pitted from arcing.
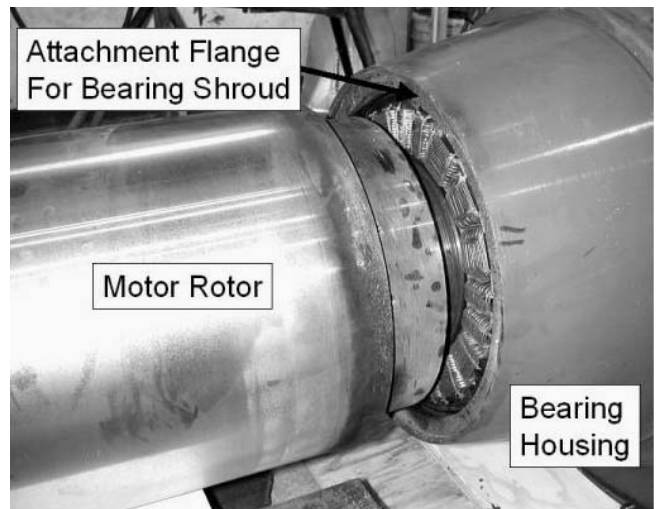

Figure 28. Motor Bearing Housing and Rotor.

This shroud around the rotor short-circuit ring area had been added uniquely for this job, assuming it was necessary to protect

this rotor short-circuit ring area from the stator magnetic fields. This modification had been done during the string tests in Zurich. Based on more recent successful experience without this shroud being installed, it became obvious that this shroud was unnecessary and not recommended. The changes include:

● Removal of the axial shroud around the short-circuit ring area.

● Tinning the copper parts of the bearing housing to protect them from corrosion rather than painting them.

● Bonding the two shroud segments together and to the bearing housing.

● Changing the fixation screws from steel to brass.

At this writing, May 2001, the stator is being rewound at the manufacturing facility in Belgium.

CONCLUSIONS

Thorough testing helped develop and convincingly demonstrated the following features in a high-pressure centrifugal compressor:

● Active thrust control with a fail-safe feature

● Magnetic bearings on a multicasing rigidly coupled rotor

● A unique combination of speed and power in a variable frequency drive and induction motor

The degree of testing required to confirm the new design features was well justified by the number of problems found and corrected. We believe corrections at the operating site during commissioning would have been much more expensive than corrections at the vendor's works. The problem with the motor internal shroud took about two weeks running at design conditions to develop, and thus was not discoverable on a short term test in the vendor's works.

The software for the variable frequency drive showed promising flexibility and useful diagnostic capability. Use of a software simulator to thoroughly exercise the drive software prior to tuning the drive will be considered for future applications.

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