2nd International Symposium on Magnetic Beaing
July 12-14, 1990, Tokyo, Japan

APPLICATION AND TESTING OF MAGNETIC BEARINGS AND DRY GAS SEALING IN AXIAL INLET PROCESS COMPRESSORS

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ABSTRACT

Modern day large scale gas phase synthesis processes require a centrifugal compressor for recycling process gas around the synthesis loop, which is a very severe duty.

In order to provide for safe, reliable and efficient plant operation, the recycle compressor must be carefully designed and manufactured to provide optimum operation efficiency and minimum maintenance downtime.

Environment, noise and safety of operation are major areas of attention. The development from WET-WET to DRY-DRY technology is seen as a distinct opportunity to develop and improve machinery in these areas.

ROTOR CARTRIDGE DESIGN

The severe application for this specific compressor is the main reason for utilizing magnetic bearings outside of the process gas environment, see figures 1 and 2.

During the design phase, it was decided to carry out a number of critical engineering analyses.

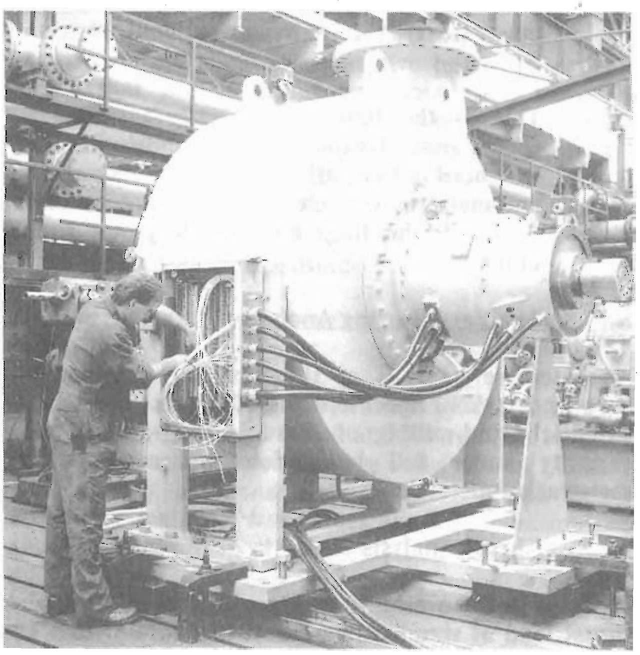

Figure 1 - DRY-DRY PV Compressor

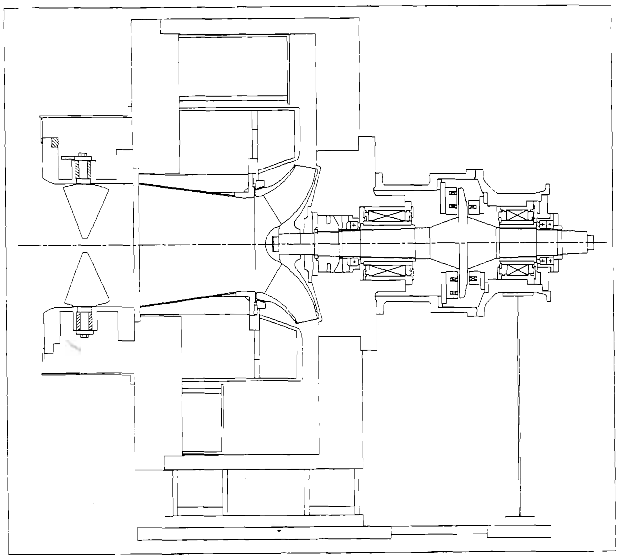

Figure 2 - Compressor Assembly

1 Dynamic Analysis

1a Undamped Critical Speed Map

Figure 3 shows the actual undamped critical speed map. This map is based on identical transfer function for both radial bearings and results in simple hardware and control circuitry.

1b Damped Response Analysis

The response analysis indicates that both the first two response modes are well damped, which is typical for overhung machines of this type with the first mode being a conical one and the second being a bouncing mode. The results show that integration of sensors of both sides of the journal bearing will make the control less noisy, particularly on the impeller side.

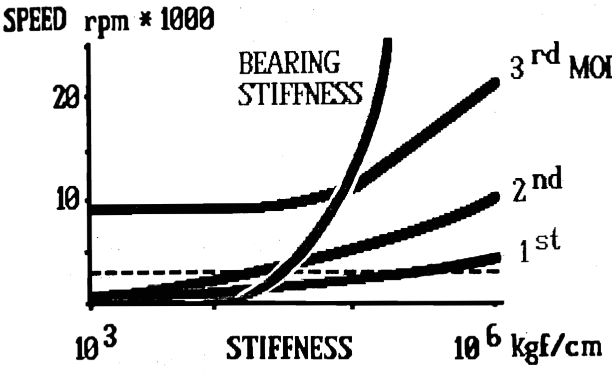

Figure 3 - Undamped Critical Speed Map

1c Stability Analysis

The stability analysis (figure 4) indicates a generous stability margin at the expected aerodynamic excitation, as would be expected of a rigid rotor configuration, even if the actual damping is only 50% of the expected damping value.

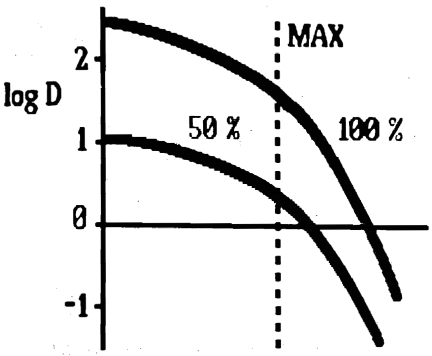

Figure 4 - Stability Analysis

2 Performance Analysis

The characteristic of the impeller for this application is its high specific speed due to high flow and low head requirements. Delaval has a vast experience in designing the mixed flow impellers for this and other applications.

3 Stress Analysis

Specific areas of design require stress analysis. As a result of these analyses, additional features are integrated in the design, see figure 5.

- Shrunk on sleeves, using hydraulic-fit on conical taper, modified to suit this specific application. These allow for operational speeds up to 7000 RPM.

- Integral thrust collar on the shaft.

- The bearing bracket for this DRY-DRY PV compressor is a barrel design.

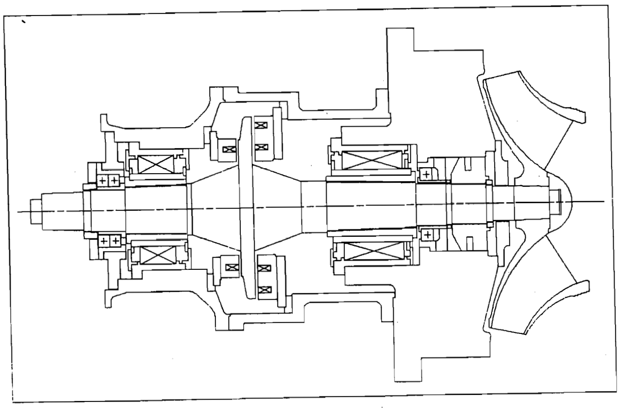

Figure 5 - Rotor Cartridge Assembly

4 Electrical Analysis

Features included in the design are:

- The speed sensor, which is modified from a magnetic type into a conventional proximity type.

- This is the first commercially installed air cooled control cabinet. Its reliability has been enhanced by utilizing fan redundancy, automatic battery check, electrical counter for full speed landings and double power supply units.

5 Auxiliary Bearing Analysis

An extensive analysis of the auxiliary bearings has been undertaken. Specific items of interest covered the potential wind-milling of these bearings, its landing capacity under full load, detection methods and registration methods.

6 Thrust Analysis

Inherent in the overhung design compressor is the high thrust load at standstill as shown in figure 6. This has been overcome by the asymmetrical magnetic thrust bearing design.

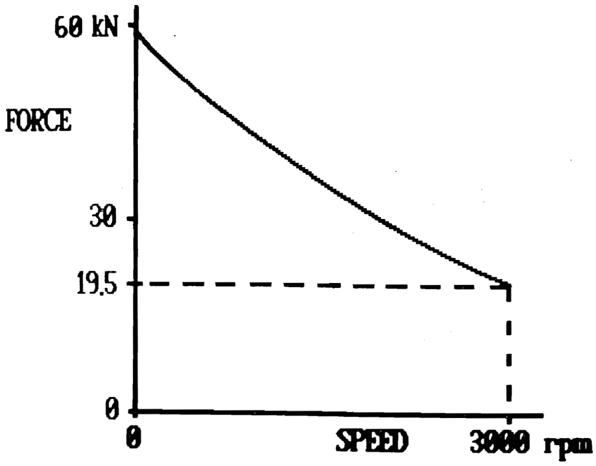

Figure 6 - Actual Force during Start-Up

7 Maintenance Analysis

Maintainability of the machine is also a major area of interest. During the design phase this has led to several improvements in the design, resulting in a backward pull-out rotor cartridge design with a barrel type bearing bracket. This cartridge incorporates all major compressor components, including: impeller, bearings and seals, resulting in a compressor which is easy to maintain, see figure 5.

TEST PROGRAM

A new concept requires exhaustive testing in order to prove that design parameters have been met. This involves more detailed testing of components and of the complete machine than are normally involved in the provision of a conventional design. Some of these tests are quite unique.

1 Dry Gas Seal Testing

Delaval has its own extensive test procedure for dry gas seals.

Each seal is full-load shop-tested by the subvendor. This seal was full-load tested at 3300 rpm, 28 barg and 160 °C. The guaranteed leakage rates are 14 l/min. in the primary seal and less than 2.5 l/min. in the secondary seal. The test values are 8.6 l/min. and less than 2.5 l/min. respectively.

2 Magnetic Bearing Testing

The manufacturer of the magnetic bearings executes specific tests during the manufacturing phase. For this compressor it was decided to carry out additional exhaustive testing in the actual compressor.

3 Rotor Excitation Free-Free Modes

Natural frequencies are calculated at 107, 220, 443 and 813 Hz. The free-free modes of the compressor rotor have been checked by exciting the rotor suspended in slings. This has been done firstly to check the engineering lateral analysis and secondly to have a starting point for the tuning of the bearing system. The measured free-free modes are in good correlation with the results of the engineering analysis.

	TEST (Hz)	CALC. (Hz)
1st mode	106.7	107.1
2nd mode	221.7	219.9
3rd mode (see fig. 7)	426.7	443.2
4th mode	805.0	812.9

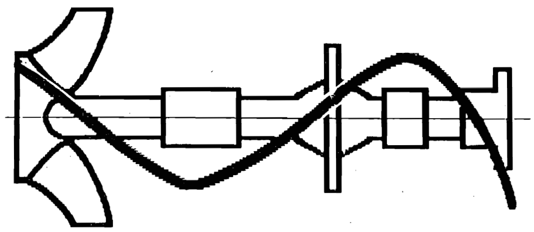

Figure 7 - Third Mode Response

4 Magnetic Bearing Tuning

The concept of the rotor layout, the sizing of the bearing and the selection of the controls has been done in the design phase. After assembly of the compressor, these three elements have to be tuned to each other in order to build a stable control loop, which will perform the normal and transient duties.

First the rotor is balanced at low speed and then the tuning starts with the information from the free-free mode excitation. Each bearing has a displacement sensor on each side. The final tuning includes both sensors on the inboard bearing and one sensor on the outboard bearing. The total tuning was finished within two weeks.

5 Static Load Testing Radial Bearing

Each radial bearing capability has been checked against measured bearing reaction force, see figure 8. The curve shows the S-shape curve for flux and force as a function of current to the bearing. The current is limited to avoid saturation of the magnets. This limit is indicated. The measured current at the reference point is in close agreement with the design value. The difference in current, as shown, is available to counterbalance the dynamic forces when the compressor is in operation.

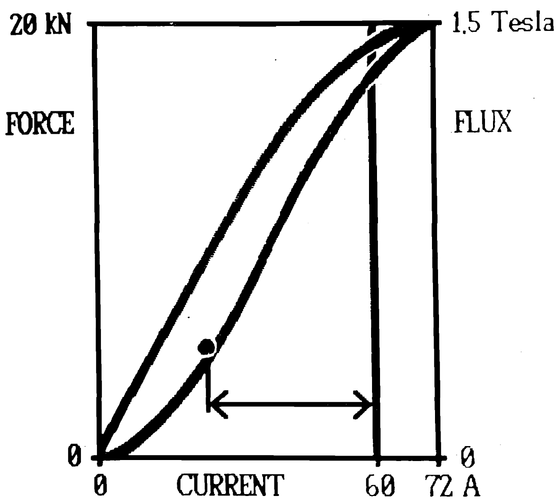

Figure 8 - Static Load Test Radial Bearing

6 Static Load Testing Axial Bearing

The axial bearing consists of three coils, which develop identical force. Two coils: Z1A & Z1B are located at NDE side and one coil: Z2 is located at DE side. The basic curve of each coil is very similar to that of the radial bearing. However, it should be pointed out that both sides of the axial bearing are pulling. An axial load on the rotor will cause a change in the current on both sides of the axial bearing such that the difference in the pulling forces of the magnets will counterbalance the axial load of the rotor, see figure 9. The axial bearing is designed for 80 kN. The difference in current between normal duty and the design point is available for dynamic forces which can occur at surge and other transient operating conditions.

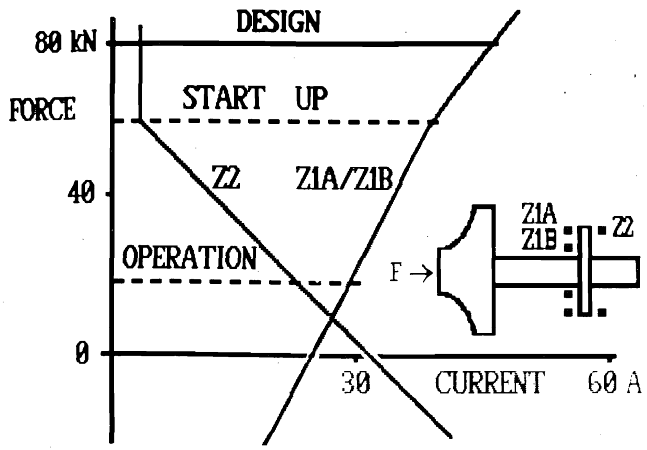

Figure 9 - Static Load Test Actual Bearing

7 Dynamic Testing Axial Bearing Capability

All pressurized overhung compressors have a high axial force at standstill. The static force at standstill for this compressor is 60 kN, which reduces to 19.5 kN at full speed, see figure 6. The bearing can handle this rapid change in force without any overshoot in the control system.

8 Simulation of Surge Condition

Surge in a centrifugal compressor creates a major upset in the axial loading of the rotor. To simulate the surge condition the compressor has been rebuilt. The impeller was replaced by a dummy with identical rotor dynamic characteristics as the original impeller. In front of the impeller a chamber has been created, which can be loaded by external pressure. By fluctuating this external pressure a dynamic load on the rotor is created, which has the same magnitude as the static load.

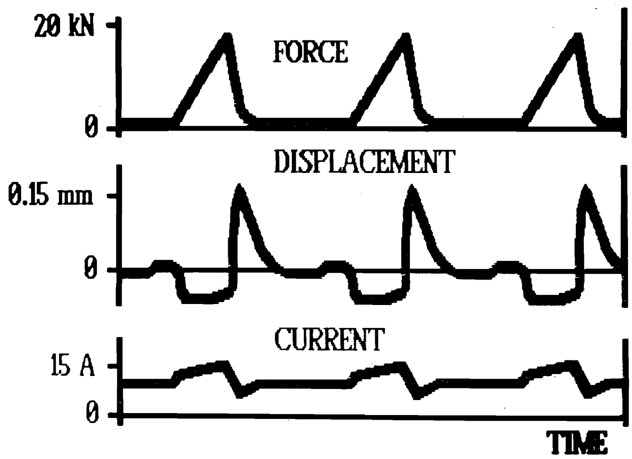

Figure 10 - Surge Simulation Test

Figure 10 shows force, displacement and current as a function of time. This unique test proves that the bearing system can handle these fluctuating loads without overshoot and well within the capability of the bearing.

9 Unbalance Response Testing

An unbalance weight corresponding to five times the API 617 value was added to the impeller. The frequency scan recorded at full running speed, shows neither subsynchronous vibrations, nor harmonics. The system has good control over the unbalanced rotor, however, the displacements are larger than calculated, indicating that stiffness of the bearing system is lower than expected.

10 Sudden Unbalance Response Testing

An important check is how the bearing system reacts in case a sudden unbalance occurs when the compressor is rotating at full speed. For this test the dummy impeller is used again with two unbalance weights, such that the rotor as a whole is perfectly balanced. Then at full speed the unbalance weight is removed.

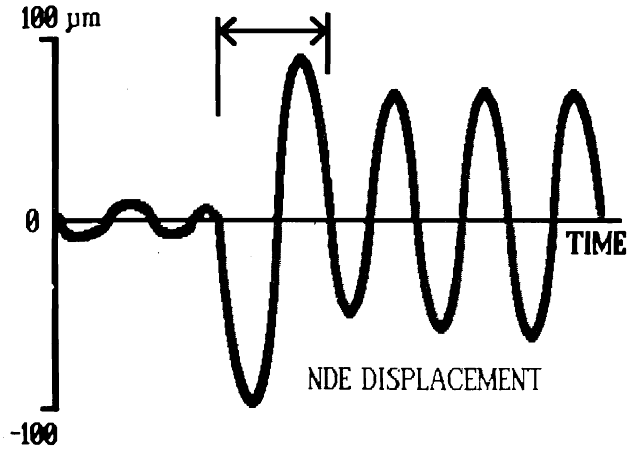

Figure 11 - Sudden Unbalance Response Test

Figure 11 shows that there is an instantaneous reaction of the bearing system on the sudden change in unbalance and it is practically demonstrated that the system can handle this situation with just a small overshoot and then come to the stable condition, within one revolution (1/50th of a second).

11 Auxiliary Bearings Non-Rotating Testing

The auxiliary bearings should not rotate when the compressor is in normal operation. To check this the end cap of the bearing bracket at the drive end side was removed. The video pictures taken, prove that the auxiliary bearing is not windmilling.

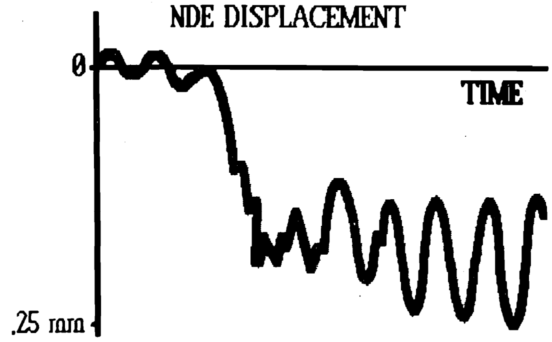

Figure 12 - Rotor Landing under Actual Load

12 Rotor Landing Under Axial Load

In general auxiliary bearings should never operate. The first line of protection is the battery back-up which allows five minutes of continuous operation without electrical power supply to the cabinet. Only if the cable between cabinet and compressor should fail, then the auxiliary bearings will safely bring the rotor to standstill. Further it is of interest to check if the auxiliary bearings can withstand one or more landings. Figure 12 shows the NDE bearing with the auxiliary bearing engaged. Due to the overhung rotor, the radial load on the DE bearing is very low. The reading of the DE bearing, figure 13, shows therefore a different behavior, which means that the rotor does not drop to its lowest position, but circulates in the full space available within the auxiliary bearing. After the landings, the bearings have been removed and extensively examined and show no signs of wear, deformation or cracks.

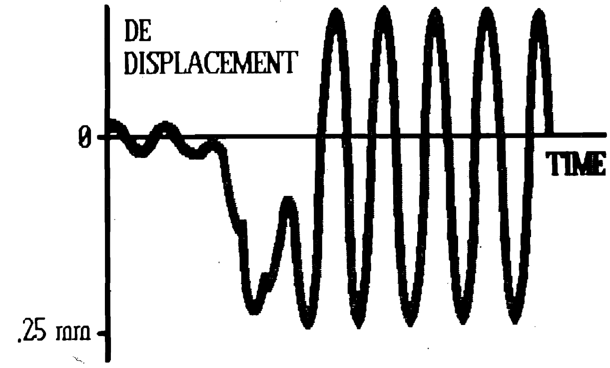

Figure 13 - Rotor Landing under Actual Load

13 System Temperature Sensitivity Testing

The heat developed by the magnets in the bearing cartridge is so low that the winding temperature without additional cooling does not exceed 60 degrees Celsius.

14 Faulty Magnet Tests

During one of the landing tests a landing has been initiated by simulating a faulty magnet. The result of this test is identical to the other landing tests.

15 Battery Capacity Test

To check the battery, the electrical supply from the control cabinet has been disconnected, whilst the compressor is in operation. The battery took over the duty without any negative effect and continued to operate for seven minutes whilst the supply voltage from the battery dropped from 110 to 106 V, thus exceeding the specified capacity by two minutes.

CONCLUSION

The results of the extensive test program on this DRY-DRY PV, as shown above, prove that active magnetic bearings combined with dry gas seals are eminently suitable for arduous continuous process duties.

We foresee that dry seal and dry bearing technology will become accepted in general practice in the near future. It will enable the centrifugal compressor designer to simplify construction, improve reliability, safety and maintainability, reduce weight and space for both onshore and offshore applications and provide an environmentally friendly concept.