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ABSTRACT

Full-load, full-pressure rotordynamic stability measurements were conducted on a seven-stage, back-to-back centrifugal compressor. To validate rotordynamic predictions, the rotor was excited while operating at full load and full pressure during factory testing. This was accomplished through means of a magnetic bearing, which was attached to the free end of the rotor. This device injected an asynchronous force into the rotor system to excite the first forward whirling mode. This technique measures the rotor’s logarithmic decrement (log dec), which indicates the level of stability, or damping, in the rotor. The device is designed to be nonintrusive to the original dynamics of the rotor and may be easily installed/removed on the test stand. This paper discusses the techniques used to measure the rotordynamic stability from a full-load, full-pressure test of a 6000 psi reinjection compressor. The results demonstrate the effectiveness of swirl brakes and damper seals in producing a compressor that becomes more stable as discharge pressure increases. This approach to compressor design is in stark contrast to traditional designs in which the stability degrades with increasing pressure, ultimately leading to rotordynamic instability. This technology ensures trouble-free startup and operation of these compressors in the field, minimizing risk for the end-user.

INTRODUCTION

Centrifugal compressors are used widely in the oil and gas industry in applications such as gas gathering, transmission, and reinjection. Reinjection compressors have traditionally created the greatest challenge to the rotordynamic designer due to the high pressure and power density. The high-pressure process gas produces significant excitation and reaction forces on the rotor. Improved understanding and management of the aerodynamic forces acting on the rotor system are required during the design cycle of this class of machinery. Devices such as swirl brakes and damper seals have been developed to reduce excitation forces and improve damping in the rotor to prevent rotordynamic instabilities from occurring.

Starting in the early 1970s, rotordynamic instabilities received great attention due to an unstable reinjection compressor in the North Sea (Wachel, 1975). Rotordynamic instability occurs when the forward driving forces exceed the resisting dissipation forces, which leads to self-excitation of the first whirling mode of the rotor. The result can be large subsynchronous vibration that is limited only by contact between the rotor and stator parts, often resulting in mechanical damage to the compressor. Examples of this phenomenon are widely reported in the literature including Wachel (1975), Fulton (1984), Kirk (1985), Kuzdzal, et al. (1994), and Memmott (2000a).

In traditional practice, the instability is not encountered until very late in the project cycle resulting in costly downtime and hardware changes. If full-load testing is not performed at the
rotordynamic stability. One such device is a damper seal. This is introduced at or near the rotor midspan can be very beneficial to the midspan of the rotor where they are most likely to drive the first concepts, a designer strives to keep destabilizing forces away from against the expected excitation in the system. Employing modal of the rotorbearing system’s first natural frequency is compared scrutinized for rotordynamic stability. The logarithmic decrement result is less recycle leakage and better efficiency for the BTB diameter of this seal, its leakage is relatively low. Therefore, the net seal from a rotordynamics perspective. Again, due to the low further attributes of the machine include high efficiency impellers coupled with low solidity vane diffusers (LSD). Tandem dry gas seals, which rotordynamically have insignificant radial forces, are used to seal the compressed gas from the environment. The rotor is supported with five-shoe tilt-pad bearings in series with sealed, spring supported squeeze-film dampers (SFD). For a full description of the damper bearing, refer to Kuzdzal and Hustak (1996).

The reinjection compressor has a 7500 psi (413 bar) case rating and was designed to achieve a final discharge pressure in excess of 6000 psi (413 bar) at the design point. Extensive full-load full-pressure testing was conducted at and below this pressure. The testing included injecting asynchronous forces into the rotor-bearing system using a magnetic bearing mounted on the free end of the compressor rotor.

**ROTORDYNAMIC MODELING**

Accurate rotordynamic modeling of high-pressure centrifugal compressors is crucial to the success of this class of machinery. With gas densities approaching half that of water, the prediction and management of gas forces in seals and secondary passages are required. With the improved accuracy of computational fluid dynamics (CFD), the destabilizing forces that exist in the compressor can be predicted and better managed. A rotor can be modeled using finite element techniques resulting in the general linear system of differential equations,

\[
[M]\ddot{X} + [C]\dot{X} + [K]X = F(t) \tag{1}
\]

For the homogeneous solution (free vibration), a harmonic solution is assumed as,

\[
X(t) = \bar{X} e^{it} \tag{2}
\]

The eigenvalue may be solved for and takes the form of,

\[
s = -\xi \omega_n + \omega_d i \tag{3}
\]

The real part of the eigenvalue determines the level of damping or stability, where \(\xi\) is the damping ratio. The logarithmic decrement (\(\delta\)) is another common way to state the level of damping in a system and is related to the damping ratio by,

\[
\delta = \frac{2\pi\xi}{\sqrt{1-\xi^2}} \tag{4}
\]

Notice the log dec is not defined for damping ratios (\(\xi\)) equal to and greater than one. Bearing and seal reaction forces may be modeled by linear stiffness and damping matrices:

\[
[K] = \begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix}, [C] = \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix} \tag{5}
\]

For centered annular seals, skew symmetry of these matrices exists,
To improve the stability in annular seals, the cross-coupled stiffness ($K_{xy}$) is minimized while direct damping ($C_{xx}$) is maximized.

The tilting-pad journal bearing coefficients are obtained using a bearing code (Nicholas, et al., 1979) that solves the Reynolds equation. Tilting-pad journal bearings have essentially no $K_{xy}$. In series with the tilt-pad bearings, a sealed squeeze-film damper is employed. The configuration allows optimal damping to be selected and makes the system less sensitive to journal bearing clearance tolerance. An uncavitated, 2 short-film damper solution has been found to model this class of damper well according to Lund, et al. (2001), given as,

$$C_d = \pi \mu R \left( \frac{L}{C} \right)^3$$

for a centered damper. A mechanical spring is used to offset the gravity load on the bearing allowing near-centered operation.

A special type of annular gas seal, referred to here as a damper seal, provides a positive, stabilizing effect (damping) as opposed to a negative, destabilizing one. Furthermore, the damping effect becomes more pronounced as the inlet pressure into the seal increases. Since all sources of destabilizing forces cannot be eliminated in a compressor, damper seals are important since they provide increased damping at a rate faster than the destabilizing force increases. The net result is an increase in rotor stability (log dec) with increasing discharge pressure. This behavior is in stark contrast to typical compressor designs. Traditionally, as the discharge pressure of a centrifugal compressor increases, the destabilizing forces increased while damping from the bearings remained constant. As a result, the higher the discharge pressure, the greater the risk of rotor instability. This new philosophy or goal of centrifugal compressor design is to reverse the traditional trend and provide a rotor that becomes more stable with increased pressure. Figure 2 shows a graphical view of this trend for a hypothetical compressor.

![Figure 2. Example of Traditional and New Design Philosophy for Rotor Stability.](image)

Several types of damper seals exist and are widely reported in the literature including: honeycomb seals (Memmott, 1994; Memmott, 1999; Zeidan, et al., 1993), hole pattern (Yu and Childs, 1998; Holt and Childs, 2002), and pocket type (Richards, et al., 1995). For the present study, a variation of the hole pattern seal, referred to as a damper seal, is used. Its geometry is optimized through the use of CFD analysis and test rig measurements. A three-dimensional (3D) CFD code with new boundary conditions was developed specifically for modeling the complex flows inside the damper seals. Chochua, et al. (2001), provide a thorough description of the code and methodology used. Figure 3 provides an example of the complex, recirculating flow field inside the cells of the seal. By optimizing the geometry of the holes, damping is maximized while keeping leakage to a minimum.
As previously described, the shaft-end labyrinth seal contains the ΔP generated by the first compressor section. This seal typically has a large pressure differential and is relatively long. Therefore, for maximum accuracy, the rotordynamic coefficients are calculated using 3D CFD techniques presented by Moore (2001). Figure 6 shows a vector plot for the flow field inside the 17-tooth, teeth-on-stator labyrinth seal. Swirl brakes are also used upstream of this seal to effectively block preswirl, resulting in seal coefficients that are stabilizing to the rotor system.

The excitation arising from the centrifugal impellers is estimated using a modified form of the Wachel number (Wachel and von Nimitz, 1981). After benchmarking the formulation on numerous test cases operating with different mole weight gases, the author’s company has adopted the following form and is referred to as the modal predicted aero cross-coupling (MPACC) (Memmott, 2000a; Memmott, 2000b) as shown in Equation (8). Notice, mole weight has been eliminated from the originally proposed Wachel formulation. By taking a modal sum based on the first whirling mode shape, an effective $K_{xy}$ is calculated on the midspan of the rotor allowing it to be overlaid on a passive excitation plot.

$$MPACC = 189000 \times \sum_{j=1}^{N_j} \frac{HP_j}{N \times D_j \times h_j} \left( \frac{\rho_D}{\rho_f} \right) x_j^2 \quad (8)$$

Utilizing a suite of rotordynamics software developed at the author’s company, a full rotordynamic analysis is performed. Figure 7 shows a predicted response plot (vibration amplitude versus speed) with midspan unbalance showing a well damped first critical speed and no modes in the operating speed range. No seal effects are included in this calculation in accordance with API Standard 617, Sixth Edition (1995) guidelines.

Figure 8 shows a passive excitation chart with and without the effect of the labyrinth and damper seals. The plot shows the predicted log decrement of the first forward whirling mode versus cross-coupled stiffness placed at the midspan of the rotor. The plot demonstrates how much excitation a rotor system can withstand before going unstable (negative log dec). The multiple curves show the effect of increasing discharge pressure, demonstrating a substantial increase in the threshold $K_{xy}$ required to drive the system unstable.

Overlaid on the plot is the MPACC number calculated at 6000 psi discharge pressure. The rotor stability margin (SM) is defined in Equation (9) as the ratio between the threshold $K_{xy}$ to the predicted aerodynamic excitation (MPACC). Due to the optimal damping from the squeeze-film bearings, the rotor exhibits reasonable stability margin when no seals are included, resulting in a stability margin of 1.5. When the seals are included, stability is greatly improved, increasing the stability margin to over 18.

$$SM = \frac{K_{xy}(\delta = 0)}{MPACC} \quad (9)$$
The log dec at a given pressure is represented by the intersection between the MPACC number and the log dec curve as shown in Figure 8. Note that a different MPACC number is calculated for each discharge pressure. Included in these rotor stability calculations are the bearings, impeller eye seals, division wall damper seal, and shaft end labyrinth. Figure 9 shows the resulting predictions and demonstrates stability increasing with increasing pressure. Equation (4) shows that the log dec approaches infinity as the first whirling mode approaches critical damping ($\zeta = 100$ percent). At a discharge pressure of 6000 psi, the damping ratio ($\zeta$) equals 98 percent and explains the high predicted log dec. These results will later be compared to the test measurements.

Modal testing is a well-known method for extracting system natural frequencies and logarithmic decrement. The challenge in rotor systems is supplying dynamic excitation to the rotating shaft in a noncontacting, nonintrusive manner. Impact methods have been used with some success with centrifugal pumps (Marscher and Campbell, 1998). However, the high speed and hazardous environment around compressors makes this approach undesirable.

Magnetic bearings have been successfully used in place of oil bearings in many applications. Modern magnetic bearings with laminated construction of advanced materials have excellent control and frequency response. This study utilizes a magnetic bearing on the free end of the compressor to generate dynamic forces to excite the rotor. Baumann (1999) used a similar approach on a multistage compressor by attaching a stub-shaft to the free end. Unfortunately, the added mass moved the nodal point of the lateral mode near the original journal bearing location. Therefore, the oil bearing was moved outboard of the magnetic bearing. While effective, this approach does not satisfy the goal of making nonintrusive measurements of the rotor system.

The present work attempts to satisfy the following requirements for the exciter:

1. Nonintrusive—have minimal effect on lateral modes of the original system by keeping added mass to a minimum
2. No changes in oil bearing location on the shaft
3. Installed and removed easily from machine while on the test stand
4. Have sufficient capacity to excite the rotor system

Clearly, number 1 and 4 are in conflict. However, by utilizing advanced magnetic materials with higher flux density, a powerful bearing was designed of minimal size. Use of lightweight alloys and a hollow, stiff-shaft design results in minimal addition of mass to the rotor system. The exciter assembly adds only 2.5 lb to a rotor weighing over 400 lb.

Figure 10 shows a solid model assembly of the original assembly as shipped to the client. Figure 11 shows a similar view with the magnetic bearing exciter installed. Minimal modifications of the original compressor housings were required to accommodate the magnetic bearing. This approach did not move the location of the radial oil bearings. A digital control system was custom designed for the application and allows control of excitation frequency range and magnitude. A digital tracking filter triggered by the excitation signal was used to isolate the exciter response from other frequencies in the spectrum, yielding good signal to noise ratio.
Next, the behavior of the rotor with the exciter installed is investigated analytically. Table 1 compares the predicted log dec and frequency with and without the exciter. The addition of the exciter hardware has little effect on the rotor system verifying nonintrusive measurement.

Table 1. Comparison of Predicted First Whirling Mode with and without Magnetic Bearing Exciter.

<table>
<thead>
<tr>
<th></th>
<th>Original Rotor</th>
<th>With Exciter</th>
</tr>
</thead>
<tbody>
<tr>
<td>Log Dec (1.22)</td>
<td>1.22</td>
<td>1.20</td>
</tr>
<tr>
<td>Frequency (7180)</td>
<td>7180</td>
<td>7190</td>
</tr>
</tbody>
</table>

Figure 12 shows the predicted results of an asynchronous, forward whirling excitation from the magnetic bearing while the rotor is at its operation speed (12,900 rpm). The model contained only the bearing reaction forces (no seals) and represents the low-pressure operation of the compressor to be presented later. The predicted response shows a well-damped first mode and demonstrates adequate capacity of the exciter bearing. A modal identification algorithm (MIA) uses this constant force injection sweep to excite the first whirling mode. This swept sine wave excites the first, forward whirling mode allowing the log dec and natural frequency to be identified. The magnitude of the forcing function is not important as long as it is sufficient to provide measurable amplitudes.

Figure 13 shows the high-pressure test loop used during testing in accordance with ASME PTC-10 (1997) test specification. A 15,000 hp steam turbine was used as the driver through a gearbox. The package coupling was used between the gearbox and the compressor to closely duplicate the conditions in the field. Orifice flowmeters were placed to measure the flow in each section as well as in between the sections to directly measure the division wall seal leakage. Nitrogen is used as the test gas for the full-load, full-pressure portion of this testing, including all the exciter tests.

Figure 14 shows a photograph of the compressor casing installed in the high-pressure test stand. A data acquisition system automatically acquires and displays both performance and mechanical instrumentation.

TEST RESULTS

After performance and mechanical checks were complete, the flow loop was depressurized to a minimal level (Pd2 = 140 psi), and the compressor was throttled to the design flow and speed. Since no significant gas forces would be present at this reduced pressure, this condition allowed the basic rotodynamic behavior to verifying that the modal characteristics can be accurately extracted by excitation from the free end of the rotor.

Table 2. Comparison of Predicted Exciter Results to Predicted Eigenvalue.

<table>
<thead>
<tr>
<th></th>
<th>Exciter (MIA)</th>
<th>Eigenvalue</th>
</tr>
</thead>
<tbody>
<tr>
<td>Log Dec</td>
<td>1.22</td>
<td>1.20</td>
</tr>
<tr>
<td>Frequency (7240)</td>
<td>7190</td>
<td></td>
</tr>
</tbody>
</table>

TEST SETUP

Once the exciter tests were complete, an ASME PTC-10 (1997) Class-1 test was performed to satisfy contractual requirements. This test specification requires the test gas to closely simulate the mole weight and gas properties of the field gas. This is accomplished by blending local pipeline gas with carbon dioxide and/or propane. The resulting mixture results in nearly identical gas densities, test speeds, and volume reduction as field operation. The compressor including package auxiliaries is then tested under full pressure on the hydrocarbon blend.

Figure 14 shows a photograph of the compressor casing installed in the high-pressure test stand. A data acquisition system automatically acquires and displays both performance and mechanical instrumentation.

TEST RESULTS

After performance and mechanical checks were complete, the flow loop was depressurized to a minimal level (Pd2 = 140 psi), and the compressor was throttled to the design flow and speed. Since no significant gas forces would be present at this reduced pressure, this condition allowed the basic rotodynamic behavior to
Nitrogen is incrementally added in steps of 1000 psi at the compressor discharge, while maintaining design flow and speed up to 5000 psi. At 5000 psi the compressor was discharge throttled until it reached 6000 psi. Figure 17 plots these test points on a compressor map. Exciter data were taken at each of these points resulting in frequency response plots similar to Figure 16, yet more highly damped.

Using the modal identification algorithm (MIA) presented earlier, the measured logarithmic decrement is determined and plotted in Figure 18. The log dec without the gas forces is 1.24, representing a rotor that is quite stable. As the discharge pressure increases, the log dec increases as well. This result confirms that the stabilizing gas forces (damping) increase at a faster rate than the destabilizing forces. As the discharge pressure approaches 6000 psi, the rotor is nearing critical damping ($\zeta = 1.0$), making identifying modal characteristics more difficult. The rotor demonstrated good stability up to the design pressure of 6000 psi with discharge densities exceeding 20 lb/ft$^3$.

Also shown in Figure 18 are the stability predictions from Figure 9. The results show good agreement at the low-pressure point, validating the bearing and squeeze-film damper analysis approach. The predictions show good correlation to the measured log dec as the discharge pressure is increased, though the log dec is slightly underpredicted near 4000 psi discharge pressure. As previously mentioned, the mode is approaching critical damping ($\zeta = 1.0$) as discharge pressures reach 6000 psi, resulting in high log dec values and good agreement between test and prediction.
SUMMARY AND CONCLUSIONS

Rotordynamic stability in high-pressure compressors is of concern to both the original equipment manufacture (OEM) and end user. Great strides in modeling techniques of fluid dynamic forces in seals and bearings have permitted improved understanding of their effect on stability. While accurate predictions are now possible, the need to minimize risks during field startup of equipment has necessitated improved testing techniques. Even when full-load, full-pressure testing is performed using the field gas, the margin against instability was not traditionally known. Future increases in power and pressure or mechanical wear of critical components could invite problems.

The OEM and client have three testing options for new centrifugal compressors: low-pressure Class-3 test; full-load, full-pressure inert gas; and full-pressure Class-1 hydrocarbon tests. An advanced testing method has been presented in this work that directly measures the rotordynamic stability under a variety of conditions utilizing a magnetic bearing as a dynamic exciter. This device adds value to all three testing options.

Tests at low pressure can identify potential issues with the mechanical rotor system, since little influence from the test gas exists. This testing also validates the rotorbearing models further improving confidence in the predictions. Full-load, full-pressure inert gas can simulate similar gas density, pressure, and power as the field conditions. Incrementally increasing the gas pressure inside the compressor allows the effect of aerodynamic gas forces to be directly measured. Finally, if a Class-1 hydrocarbon test is performed, the exact field conditions can be duplicated guaranteeing adequate stability. With the use of the magnetic bearing exciter, the exact stability margin can be measured.

The results presented in this paper demonstrate that a high-pressure compressor can be designed that continually increases in stability as discharge pressure is increased. This result has been accomplished by eliminating known destabilizing sources and introducing damper seals. These devices increase their damping with pressure. This result contrasts traditional designs that have a threshold pressure of instability. These measurements demonstrate that the analytical prediction tools accurately model this complex rotodynamic system.

The design of the magnetic bearing exciter accomplished its goals of performing nonintrusive measurements and allowing easy installation/removal on the test stand. Future testing is planned to further characterize rotordynamic stability at off-design operation and to improve the data reduction algorithm to characterize overdamped modes. The experience gained in this test program is supporting the development of high-pressure compressors with discharge pressures approaching 15,000 psi (1000 bar).

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$K_{xx}$, $K_{yy}$</td>
<td>Direct stiffness</td>
</tr>
<tr>
<td>$K_{xy}$, $K_{yx}$</td>
<td>Cross-coupled stiffness</td>
</tr>
<tr>
<td>$M$</td>
<td>Rotor mass matrix</td>
</tr>
<tr>
<td>$N$</td>
<td>Rotor speed (rpm)</td>
</tr>
<tr>
<td>$P_{d2}$</td>
<td>Compressor discharge pressure</td>
</tr>
<tr>
<td>$Q$</td>
<td>Compressor volume flow rate</td>
</tr>
<tr>
<td>$D_i$</td>
<td>Diameter of impeller (j)</td>
</tr>
<tr>
<td>$h_i$</td>
<td>Diffuser width of impeller (j)</td>
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<tr>
<td>$H_P$</td>
<td>Horsepower of impeller (j)</td>
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<td>$Q$</td>
<td>Compressor volume flow rate</td>
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</table>

**Figure 19. Comparison of Results with Previously Published Data.**

The most dominant component affecting rotor stability is the damper seal at the division wall. Due to its central location and high-pressure differential, it provides substantial damping to the first whirling mode. The results confirm in practice the good correlation of the damper seal code that has been demonstrated by Holt and Childs (2002). Due to the high-stage pressure differential, the impeller eye labyrinths also play an important role. The swirl brakes on these seals have transformed these components from destabilizing into a significant stabilizing feature, further contributing to the good rotor stability.

Figure 19 compares the current measured results to those previously published by Baumann (1999) for two different compressors in the same class as the present machine. Machine 1 showed an initial increase in log dec but then decreased, leading to instability (log dec < 0) near 3600 psi, due to a drop in the first natural frequency. The second case (Machine 2) presented by Baumann (1999) shows a flatter behavior by selective use of swirl brakes. These results are presented for comparison purposes only and do not necessarily reflect the current state-of-the-art for that manufacturer.

Based on the successful full-load, full-pressure nitrogen test, the MBE was removed and a Class-1, full-pressure hydrocarbon test was performed. Figure 20 shows a vibration spectrum at the design pressure demonstrating a clean spectrum with no sign of instability (subynchronous vibration).

**Figure 20. Vibration Spectrum During Class-1 Test 6000 PSI Discharge.**
\(Q_d\) = Compressor design volume flow rate
\(R\) = Squeeze-film damper radius
\(SM\) = Stability margin
\([X(t)]\) = Rotor displacement vector
\(X\) = Eigenvector (mode shape)
\(\gamma_j\) = Modal amplitude at impeller (j)
\(\delta\) = Logarithmic decrement (log dec)
\(\varepsilon\) = Squeeze-film damper eccentricity
\(\zeta\) = Damping ratio
\(\rho_d\) = Stage discharge density
\(\rho_s\) = Stage suction density
\(\mu\) = Absolute viscosity
\(\omega_n\) = Undamped natural frequency
\(\omega_d\) = Damped natural frequency

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