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GEORGE R. BROWN CONVENTION CENTER

Application of Dynamic pressure-balanced Seals in a Multi-stage Centrifugal Compressor

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ABSTRACT

Test results for an ASME Power Test Code 10 (PTC) Type 1 test of a 4,500 psia (310 Bara) discharge pressure gas lift centrifugal compressor outfitted with dynamic pressure-balanced seals at the impeller eyes; shaft interstage and division wall locations are presented and compared to the same testing with conventional labyrinth seals. Both aerodynamic performance and rotor dynamic stability, obtained via operational modal analysis (OMA), are presented. A client's motivation, along with the design and testing of dynamic pressure-balanced (DPB) seals for turbomachinery are also presented in this paper.

With the DPB seals installed the test results indicate 2.8 percent lower power was required for the same head level across the entire range of inlet flows and pressure ratios, when compared to the same testing with conventional labyrinth seals. Rotordynamic stability, obtained via operational modal analysis (OMA), showed the dynamic pressure-balanced (DPS) seals exhibited log decs similar to standard labyrinth seals across the entire range of flows and pressure ratios. The foregoing demonstrates both the aerodynamic and mechanical/rotordynamic integrity of the dynamic pressure-balanced seals for oil and gas, turbomachinery applications.

INTRODUCTION

There are two identical centrifugal compressors, each driven by a constant speed motor through a variable speed gear box. They are used in gas lift service offshore in the U.S. Gulf of Mexico (GOM). Each compressor uses seven impellers in a back-to-back configuration, driven through the high-pressure section. The compressor employs a modular design with readily-removable internals in a single cartridge assembly from the inlet (non-drive) end. The maximum working pressure (MWP) of the casing is 5,250 psi, each impeller is 10.49 inches (266.4 mm) in diameter and rotor maximum continuous operating speed (MCOS) is 13,881 rpm (Figure 1).

Figure 1: Gas lift centrifugal compressor with 4,500 psia (310 Bara) discharge pressure.

As with all turbomachinery, sealing the fluid flow path between the rotating and stationary components is important to achieve efficient operation, as well as to minimize power, fuel consumption and emissions. Performing this seal function effectively has challenged turbomachinery designers for decades. The simplest of seal designs, the traditional labyrinth seal, has evolved in various forms to perform this function. Most labyrinth seals are simple toothed designs manufactured from a rub tolerant material such as aluminum, which mechanically deforms when contacted by the rotor. This mechanical deformation results in increased clearance

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between the stationary labyrinth and the rotor. Consequently, the increased clearance allows more gas to recirculate around the flow path, which in turn causes additional power to be applied to the rotor to maintain design flow rates and discharge pressures. If the seals are worn such that the discharge flange delivered mass flow decreases by 8%, upwards of 8% more power would be required to compensate for the worn seals. Therefore, controlling leakage is of paramount importance for reliable operation and long run times between maintenance periods.

The current state-of-the-art method for sealing flow between the rotating and stationary components consists of several types or styles of seals, such as plain stationary labyrinth seals, rotating labyrinth seals, stepped labyrinth seals, pressure activated leaf seals, finger seals and brush seals. Stationary labyrinth seals are typically manufactured from a rub tolerant material such as aluminum, but have also been made from several other materials such as bronze, monel, babbitt, and polymers such as polyetheretherketone (PEEK). Rotating labyrinth seals are usually machined from the rotating component material. As such, they have the same strength and hardness as the base metal. Therefore, they typically rub against a softer abradable material, such as polytetrafluoroethylene (PTFE) or glass-filled PTFE composites such as Fluorosint™ 500, or various types of graphite-metal matrix composites. Pressure activated leaf seals are designed to use changing pressure drop across the seal to eliminate rubs. The seal elements stay clear of the rotor surface and close to a small non-contacting steady-state running clearance as pressure is applied. Brush seals typically consist of bristles arranged like a typical brush held in place in a narrow row by a rigid backing plate. The bristles are usually angled in the direction of rotation so that any contact by the rotating component will readily cause deflection of the bristles. The bristles are usually manufactured from a superalloy material.

CLIENT MOTIVATION

The subject compressors were purchased for a gas lift service in a GOM offshore application requiring a low-flow, high-head compressor design. The required discharge pressure and flow resulted in a design that had narrow impeller passageways. The compressor was purchased with a standard internal labyrinth design. Alternative dynamic pressure-balance (DPB) seals were requested to be designed and fully tested in parallel with the standard labyrinth design for four main reasons.

The first reason for using the DPB seals was to maintain the efficiency for longer run times between service intervals. Wear of the standard labyrinths seals on this low-flow, high-head application could lead to loss of aero performance because of the large percentage of leakage across the seals with respect to the delivered mass flow. Depending on the duty and number of surge events endured by the machine, labyrinth seal degradation could warrant service intervals every 3 years. The dynamic pressure-balance seal has the potential to maintain design flow leakages longer than conventional labyrinth seals.

The second reason for using the dynamic pressure-balance seals was to increase aerodynamic efficiency. The use of the DPB seals predicted a 2.0 percent increase in efficiency due to lower leakage rates and tighter running clearances.

The third reason for using the DPB seals was the forgiving operating envelope they provide for the rotor, and the positive effect they have on the rotor dynamic stability leading to higher reliability and longevity. Standard labyrinth seals for this application would have five teeth on the impeller eye, seven teeth on the impeller interstage, and a radial clearance of 0.007 - 0.011 inches (0.177 mm- 0.279 mm) and will rub out during rotor excursions exceeding these fixed clearances. The dynamic pressure-balance seal is tolerant of rotor excursions many times that of the labyrinth fixed clearance.

Finally, and based on a field retrofit, the DPB seals are believed to have a positive effect on rotordynamics. Due to confidence gained via extensive testing by San Andrés, et. al. at Texas A&M Turbomachinery laboratory (San Andrés, 2011, 2013, 2015) the two lead authors chose to pursue installing dynamic pressure-balanced seals in a separate compressor that had rotordynamic sub-synchronous excitations in the field (Justak, 2013). In this field retrofit application, the author had the seal tested in a lab at 10,000 rpm, design delta pressure and the seal moved off center .030 inches (0.762 mm). Results showed no change in performance. Upon inspection, no signs of rubbing were observed. Next, an eccentric .006 inch (0.152 mm) rotor was installed and the same test performed. The seal shoes tracked the eccentricity and no significant change in performance was noted. Again, the inspected seal and running surface showed no signs of contact. This hydrogen recycle compressor has been running trouble-free in the field since September 2012. This demonstrates a clear maintenance advantage as the seals can accommodate internal misalignments.

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SEAL DESIGN

The dynamic pressure-balanced seal uses the flow and pressure drop associated with the fluid it is sealing to generate a force-balanced seal system with an operational clearance as tight as a few thousandths of an inch (hundredths of a millimeter) over the rotor surface. The seal is typically installed with a nominal gap that is larger than the desired running clearance. The acceleration of fluid between the seal and the rotor creates a low-pressure region that draws the seal shoes towards the rotor. As the seal surface approaches the rotor surface, features within the seal surface reduce the velocity of the fluid, and when combined with static upstream pressure results in a pressure rise that increases the radial outward force on the seal shoes. The operational clearance is achieved when this outward force is balanced with the inward force from the upstream and downstream pressures acting on the backside of the seal. Seal dimensions are tuned to achieve the desired operational clearance. This method of creating a force balance over the rotor surface facilitates a seal that maintains a set clearance during dynamic changes in pressure, temperature, centrifugal growth, and rotor eccentricity. The nature of operation is not negatively impacted by swirl ratio or surface velocity associated with high-speed operation.

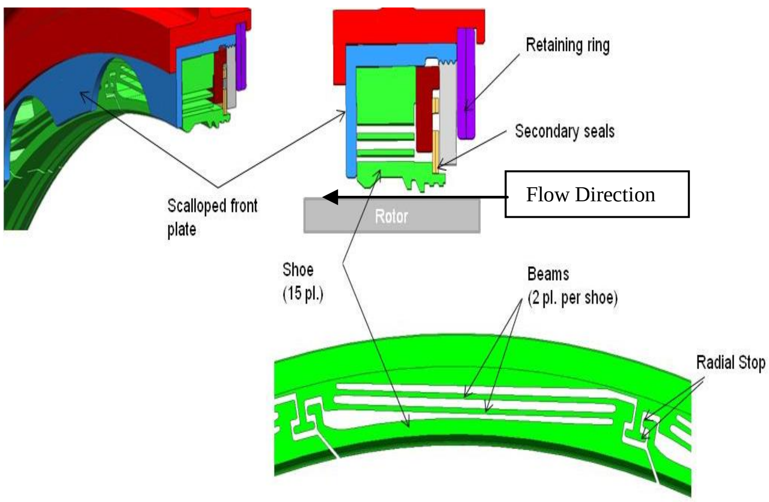

Figure 2: Cross-section of the DPB seal.

Figure 3: DPB Seal photo.

The DPB seal is made up of a circular array of shoe segments (see Figures 2 and 3). Each shoe segment acts independently and is connected to the static hardware via a spring support. The spring support is constructed of a pair of beams that support the DPB seal shoe in the radial and axial direction. The design in Figure 2 is comprised of 15 shoes around the circumference. The spring supports are design for infinite fatigue life, via traditional Goodman diagram design practices. A secondary seal limits the flow between the static and dynamic hardware components (Justak, 2009, Anderson, 2013, San Andrés, 2010, 2015). The images above call out the primary components of the DPB seal and the graph (Figure 4) provides typical analytical curves indicating an increasing net radial seal force as the seal shoe approaches the rotor surface. At the design point shown in Figure 4, the net shoe force is zero or balanced. This corresponds to a seal clearance of 0.18 mm (0.007 inches) and a predicted flow of 0.80 (unitized). A negative seal force indicates the shoe would want to move away from the rotor, opening the clearance and increasing the flow that passes through the seal. This chart is provided as an example; the actual DPB seal clearance tested was 0.102 mm (0.004 inches).

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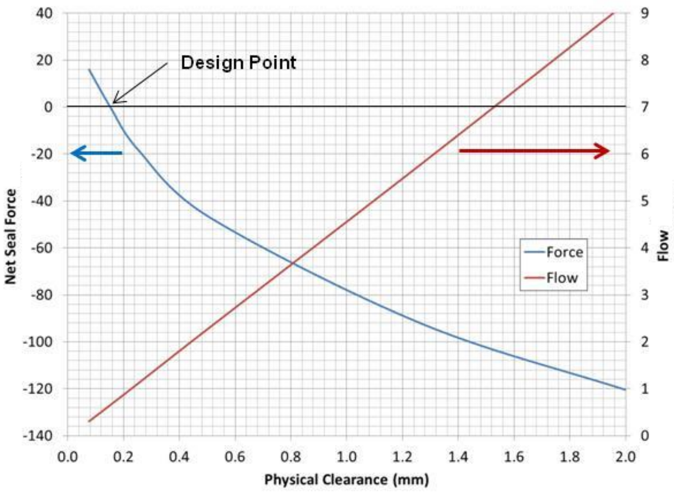

Figure 4: Shoe clearance vs. seal shoe force and shoe clearance vs. leakage flow.

MECHANICAL DESIGN CONSIDERATIONS

In order to compare the mechanical and aerodynamic performance between a compressor using traditional aluminum tooth labyrinth seals and DPB seals, the DPB seals were designed to physically fit into the same operating envelope as the traditional aluminum tooth labyrinth seals. The hook groove and dovetail groove alignment and positioning geometry of the tooth labyrinth seals, along with their anti-rotation pin features, were incorporated into the DPB seal design. Following the initial compressor testing with tooth labyrinth seals, the compressor was disassembled and the tooth labyrinth seals were removed. The DPB seals were immediately installed and the compressor was re-assembled and retested.

The axial sealing length and the axial positioning of the tooth labyrinth seals were taken into account with the design of the DPB seals to ensure that the axial position of the DPB sealing geometry would properly align with the impeller eye and not overhang the edge of the impeller eye and lose sealing effectiveness. The axial positioning of the interstage seals relative to the interstage impeller spacer is a more forgiving, as the length of the interstage impeller spacer is longer. However, to maintain a consistent design practice, the axial positioning of the tooth laby seal and the DPB seal were kept the same (Figures 5 and 6).

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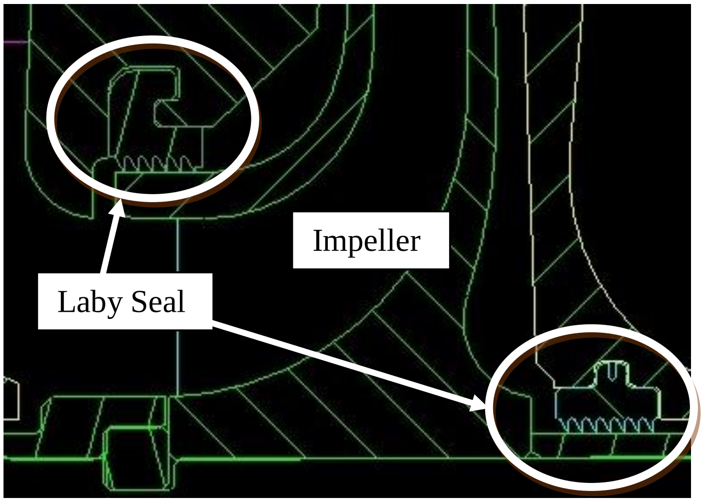

Figure 5: Traditional aluminum tooth labyrinth.

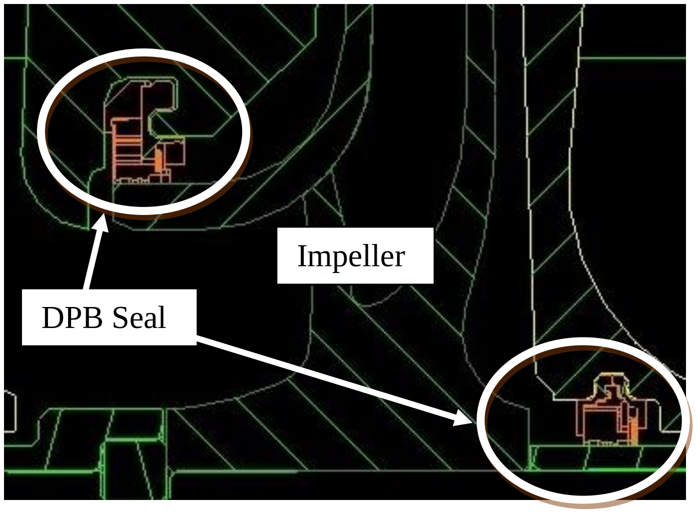

Figure 6: Dynamic pressure-balanced seals.

In addition to the physical size of the sealing envelope, other design considerations (both static and dynamic) were taken into account for the application of the DPB seals. Static considerations, such as mounted component growth (from the interference fits of the impellers and interstage spacers), rotor sag (gravitational effects on the rotor), rotor bearing drop, and concentricity at each seal location were included in the DPB (non-activated) seal clearance to ensure that the rotor (non-rotating) did not come into contact with the DPB seals. Dynamic considerations throughout the operating speed range include centrifugal growth (impeller eye and interstage spacer) and peak-to-peak vibration displacements at each seal location (Ref: American Petroleum Institute -API 617). Mechanical run-outs of the rotating components were also accounted for, so that the total rotor movement at any speed during operation would not exceed the travel limits of the DPB seal springs.

Traditional tooth labyrinth seals require the same static and dynamic considerations when setting their clearances. For this particular compressor, the typical fixed radial clearance for the aluminum tooth laby was set at a range of 0.007 – 0.011 inches (0.178 mm-0.274 mm). The DPB radial clearance was set at 0.020 inches, (0.508 mm) in the non-activated or static condition and 0.004 inches, (0.102 mm) in the activated or dynamic condition. The tighter seal clearance of the DPB seal in the activated (dynamic) condition reduced the process gas recycle flow around the impeller stages and resulted in a more efficient compressor. The pressure ratio across each of the impeller eye seals was between 1.10 and 1.25. The pressure ratio across each of the interstage seals was 1.02 to 1.05. The locations of the seals are shown in Figure 7. A picture of the compressor internals with the DPB seals installed is shown in Figure 8.

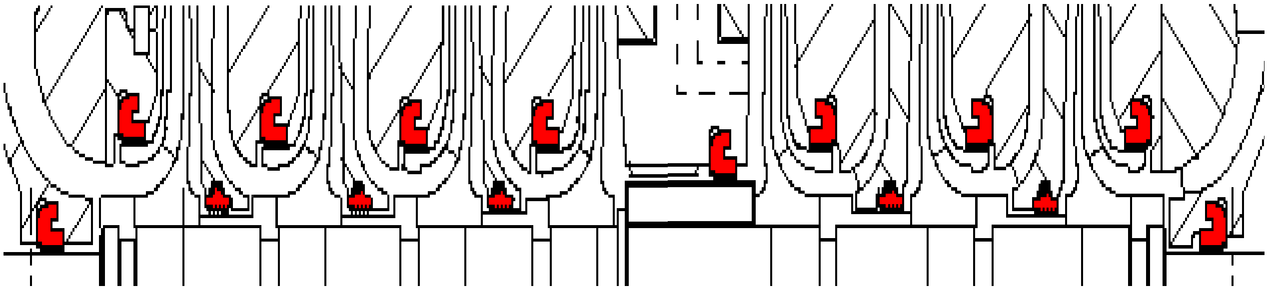

Figure 7: Location of seals that were changed during testing.

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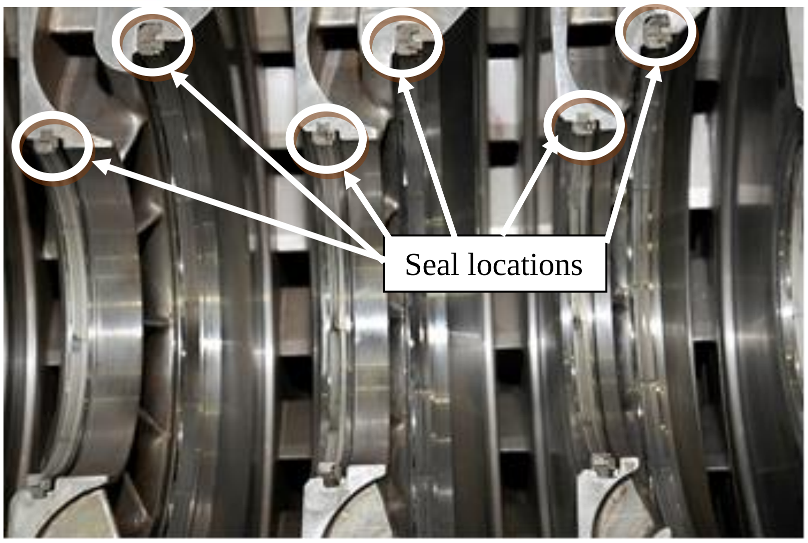

Figure 8: Dynamic pressure-balanced seals installed in unit.

AERODYNAMIC DESIGN CONSIDERATIONS

The seals are an integral part of each stage in the compressor and contribute to the overall aerodynamic and mechanical performance of the stage. This is why as conventional seals wear, the compressor operating map may shift slightly which necessitates using more power and speed for the compressor to handle the same delivered mass flow rate and pressure ratio. Eventually, the compressor must be shut down for maintenance to replace the worn seals. Most compressor designers use a nominal clearance for the stage seals in their design calculations; therefore, if the clearance is reduced from this nominal amount, the result is a reduction in the amount of recirculated flow in the stage. Hence, the impeller will do less work for the given amount of delivered mass flow; the overall efficiency of the compressor is improved and the compressor performance curve will shift to a slightly higher inlet flow rate. In some cases, there could be a slight shift in the thrust load of the compressor. In replacing conventional labyrinth seals with DPB seals, the shift of the compressor performance curve was noticeable. This change in performance could have been corrected at the design phase by a slight change to the impeller selection, but the client elected to keep the compressor configured with the original impeller and corresponding driver line-up (Figures 9 and 10). The blue curves represent the predicted aerodynamic performance with the labyrinth seals in place (with an assumed average clearance of 0.009 inches). The red curves show the predicted aerodynamic performance for the DPB seals with (0.004 inch clearances). The damper seal clearance remained unchanged throughout the testing campaign. These aerodynamic performance maps represent flange-to-flange efficiency of the entire compressor.

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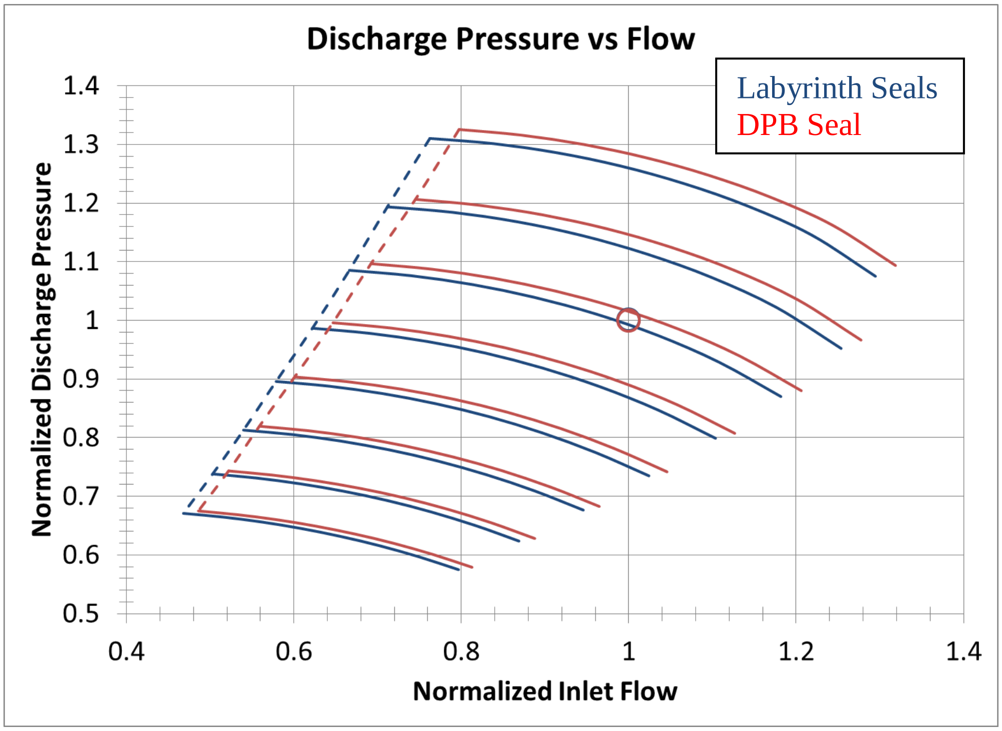

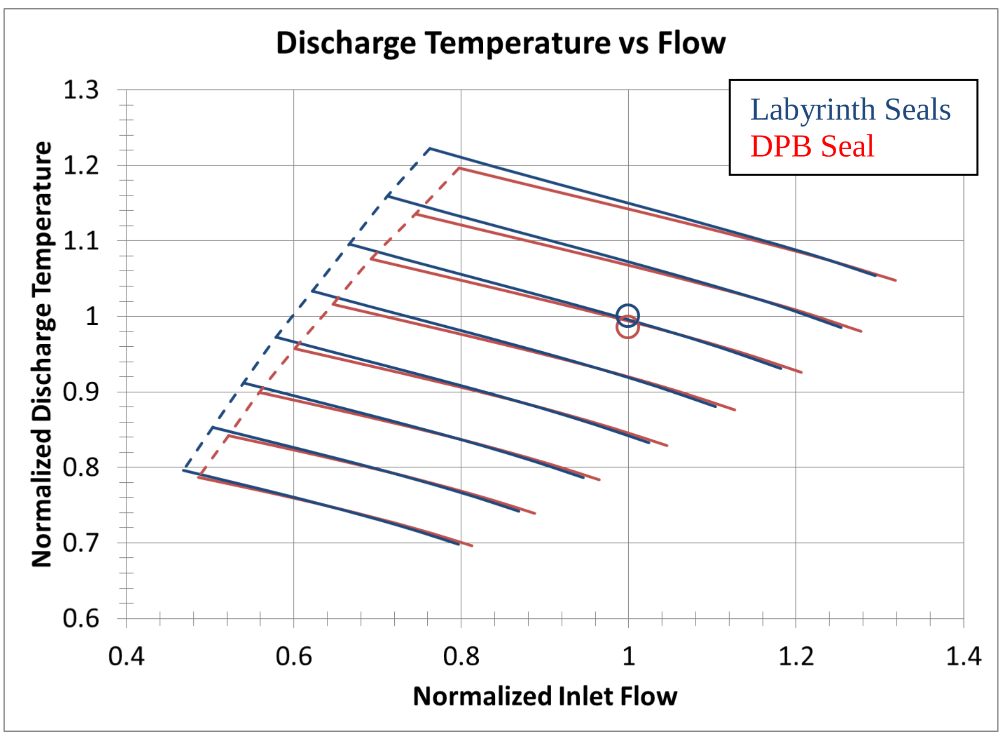

Figure 9: Discharge pressure and discharge temperature versus inlet volumetric flow, flange to flange, both sections.

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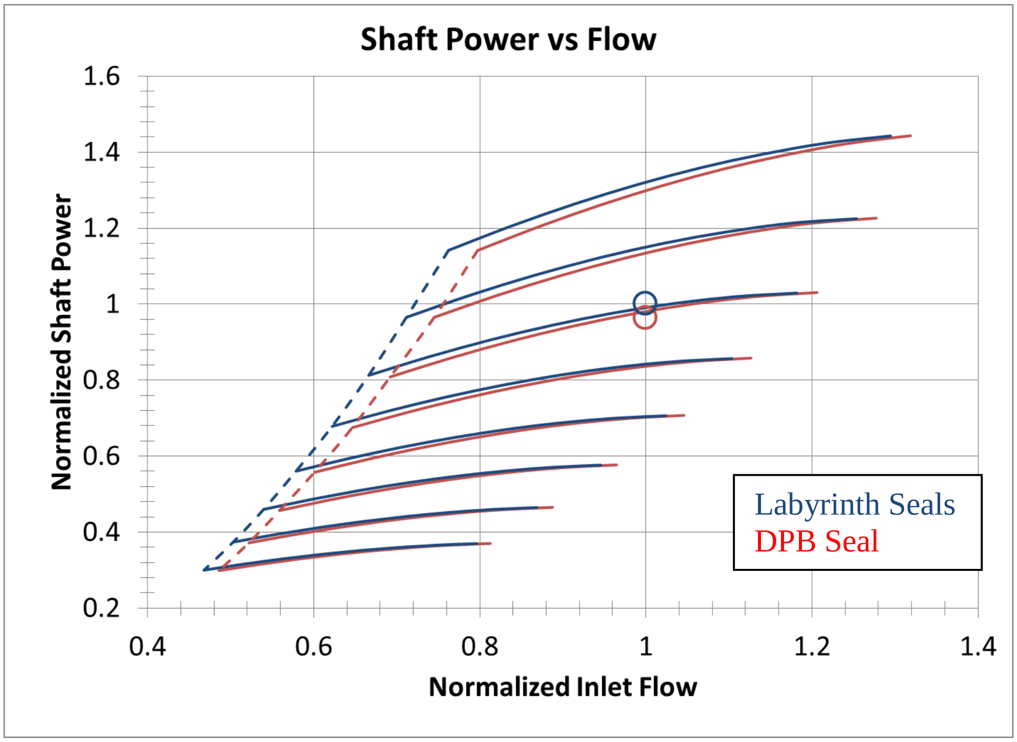

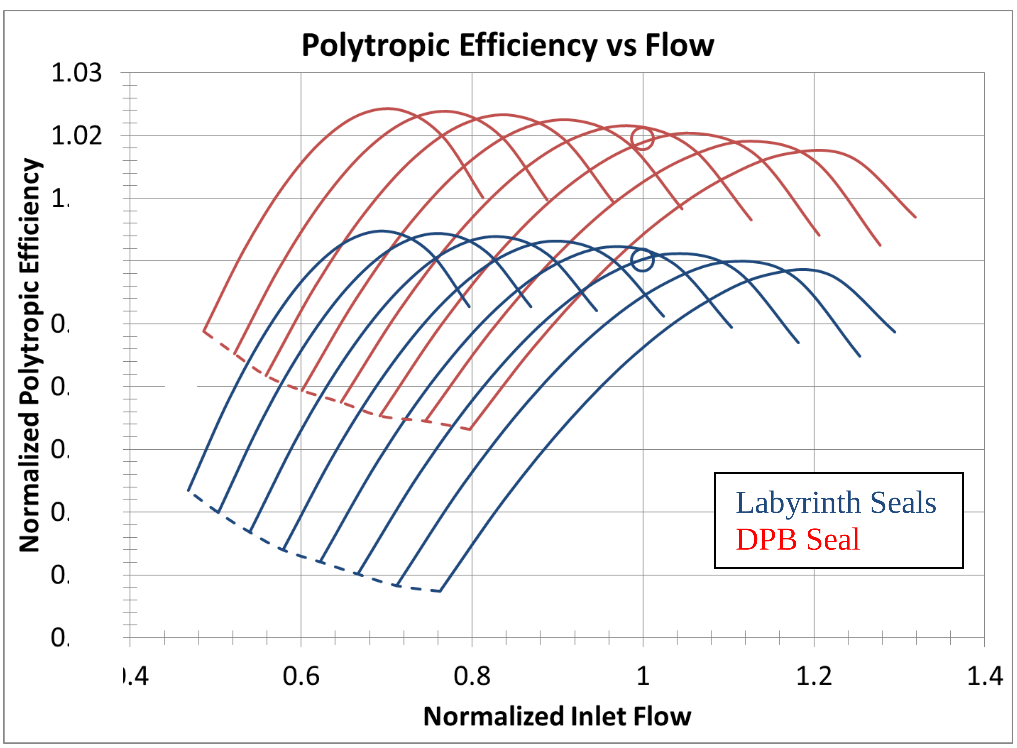

Figure 10: Shaft power and efficiency vs. inlet volumetric flow, flange to flange, both sections.

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TESTING AGENDA

In designing a comparative testing campaign, it is essential to ensure that testing is completed in a methodical fashion such that conclusive results can be drawn. This is not always possible in a development program, let alone on a production order. To that end, the testing agenda included the following steps:

1) Assemble the compressor with labyrinth seals
    a. Performance Test, ASME PTC-10 Type II – establishes a baseline aero performance
    b. Compare aero performance to prediction with labyrinth seals
    c. API mechanical run test
    d. Performance Test, ASME PTC-10 Type I, with OMA to obtain log decrement baseline effect of the labyrinth seals
    e. Evaluate mechanical and aero performance
2) Inspect and rebuild compressor with dynamic pressure-balanced (DPB) seals
    a. Performance Test, ASME PTC-10 Type II
    b. Compare aero performance to prediction with DPB seals
    c. Compare aero performance of labyrinth to DPB seal
    d. API mechanical run test
    e. Performance Test, ASME PTC-10 Type I, with OMA to obtain log decrement with DPB seals
    f. Evaluate mechanical and aero performance
3) Tear down compressor to inspect DPB seals and rotor components

The plan was successfully executed on the production order. After the testing cycle of the first unit was completed, it was mutually agreed to proceed with an API mechanical run test of the second and third (spare) bundles with the dynamic pressure-balanced seals (ref ASME PTC 10, API 617 7th) . After all of the testing was completed, the client chose to ship the equipment to the field with the dynamic pressure-balanced seals. The following sections discuss both mechanical and aerodynamic predictions, along with test results.

MECHANICAL PREDICITONS

The synchronous response and stability were calculated using the original equipment manufacturer (OEM's) rotor dynamic software suite. This tool links the centrifugal compressor aerodynamic selection and compressor modeling and the rotor dynamic programs into one cohesive engineering tool (Ramesh, 2002).

The tilt pad bearings were analyzed with the Nicholas program (Nicholas, et al, 1979). The stability analysis was done with the Lund transfer matrix program. The toothed labyrinths and the DPB seals were modeled with the two control-volume bulk flow method program by Kirk (Kirk, 1985, 1986, 1990). The hole pattern seals were modeled with the Texas A&M program by Kleynhans and Childs (Kleynhans and Childs, 1996, Holt and Childs, 2002, San Andrés and Soulas 2007, and Childs and Wade, 2003). Use of the tilt pad bearing and stability programs, especially as related to the requirements of API 617, is described by Memmott (2003).

As discussed in the introduction, the gas lift compressor is a seven-stage, back-to-back compressor with a rotor weight of 321 lbm (145 Kg), three-inch journal (non-damper type) bearings and a span-to-impeller bore ratio of 10.05:1. The unit is driven from the discharge end. Using damped eigenvalue analytical techniques with all labyrinth and damper seals included in the model, the first fundamental natural frequency is predicted between 7,900 – 8,500 CPM.

The level II stability analysis with the labyrinths, as given in Paragraph 2.6.6 of API 617 7th edition, was conducted. The OEM considers the anticipated cross-couplings from the impellers (qA) to be separate from the labyrinths cross-coupling and includes them in the level II analysis by its modal effect. For this application, the total excitation, QM, was 30,700 (lb/in) and is the modal sum of the anticipated cross coupling qA at each impeller, along with each anticipated seal excitation. This total excitation is applied at the rotor mid-span.

The log dec with the labyrinths at QM was 1.74 for minimum bearing clearance and 2.12 for maximum bearing clearance. Since the log dec with the labyrinths is > 0.1 at QM, the stability is acceptable per paragraph 2.6.6 of API 617 7th edition. The safety factor to QM with the labyrinths is 9.3 for minimum bearing clearance and 10.1 for maximum bearing clearance.

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As part of the testing campaign, operational modal analysis (OMA) was used at five different aerodynamic points across the performance map (points B, C, D, E, and F). In consideration of these five discrete points, the logarithmic decrement (log dec), again with all seals included in the model, was evaluated and predicted to be between 1.45 and 2.3. Figure 11 shows the location of the five performance points.

OMA is well-established as an effective tool to determine modal frequencies and damping in civil engineering applications that have low system level damping. Application of OMA to turbomachinery rotor shafts has been well documented in the literature by Baldassarra (2015). The technique is an output-only method that can determine modal parameters of systems based on statistical analysis of measurement sensor data without applying a known or deterministic external forcing function(s). This technique eliminates the need to apply external forces with a magnetic bearing exciter.

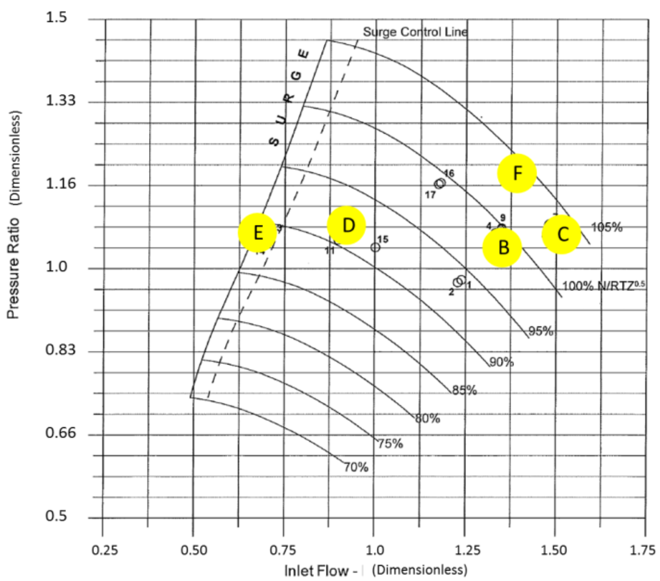

Figure 11: Dimensionless pressure ratio vs. inlet flow, flange-to-flange, both sections.

MECHANICAL TEST RESULTS

As discussed in the testing agenda section, full-load full-pressure (FLFP) Type 1 testing was conducted with both labyrinth and DPB seals. Figure 12 show a deceleration from full speed to slow roll during Type 1 testing of the Thrust End (non-drive end) vertical probe. The rotor, when outfitted with labyrinth seals, is shown in blue; when outfitted with the DPB seals shown in red. Figure 13 shows the same for the Thrust End (non-drive end) horizontal probe. Moving to the other end of the machine, the drive end, Figures14 and 15 show the same for the vertical and horizontal probe, respectively.

In comparing and contrasting the synchronous response, the results show similar rotor behavior. The location of the first natural frequency, the amplification factor of the first natural frequency and the overall rotor vibrations at full speed are similar. It is important to note: to switch from the original labyrinths to the DPB seals, the bundle is pulled from the case. This requires disconnecting the coupling, which creates an opportunity for minor changes in unbalance. To the best of the authors' knowledge, the dry gas seals and journal bearings were not disturbed during the interstage seal change-out.

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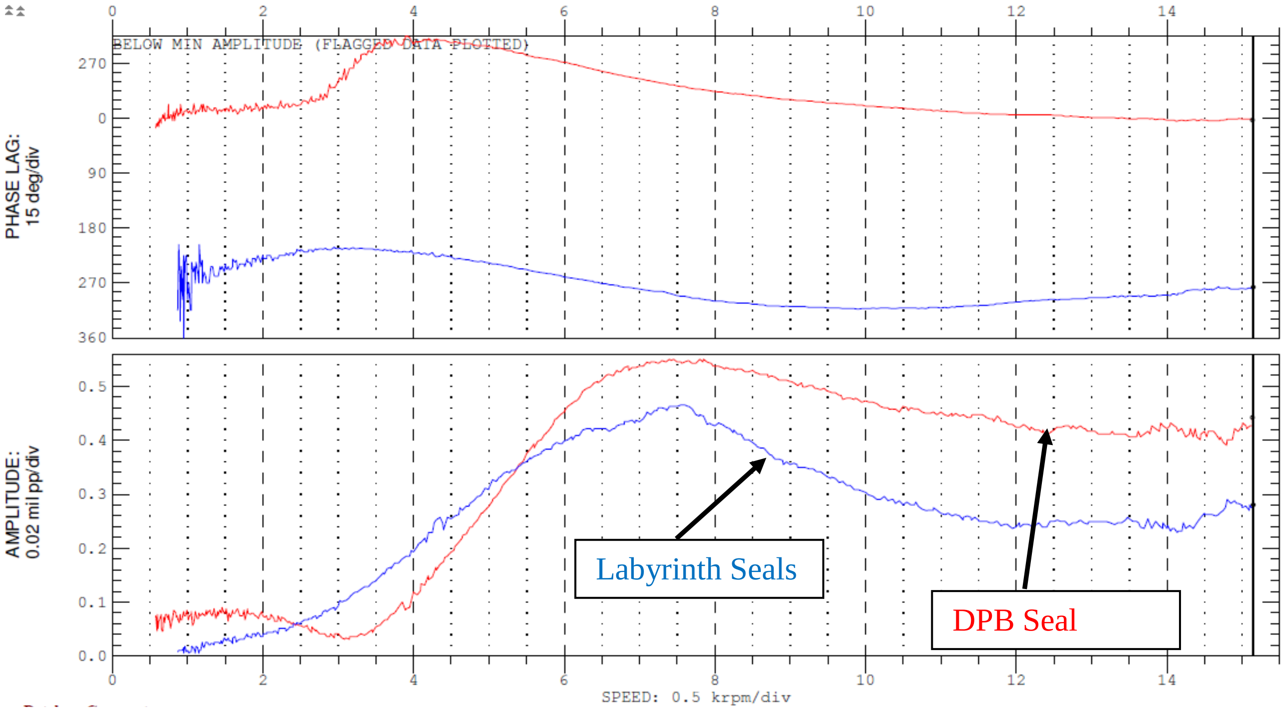

Figure 12: As tested synchronous rotor response at non-drive end vertical probe.

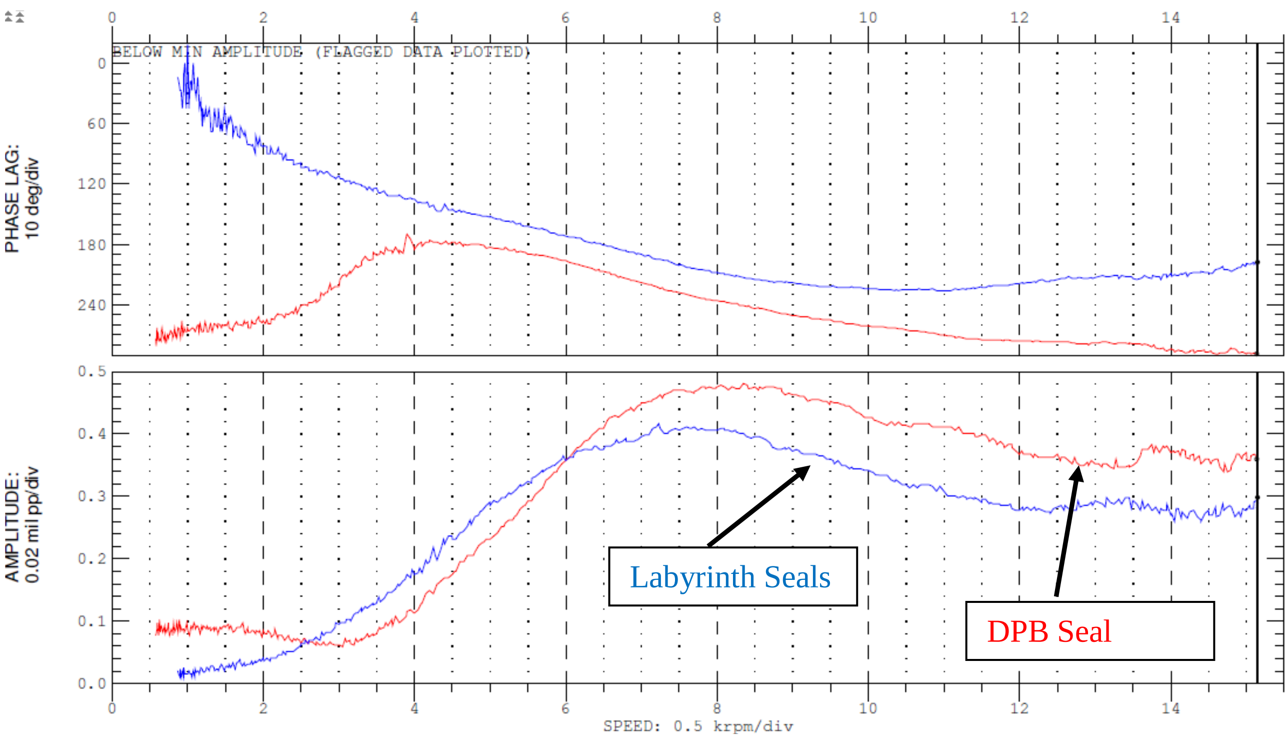

Figure 13: As tested synchronous rotor response at non-drive end horizontal probe.

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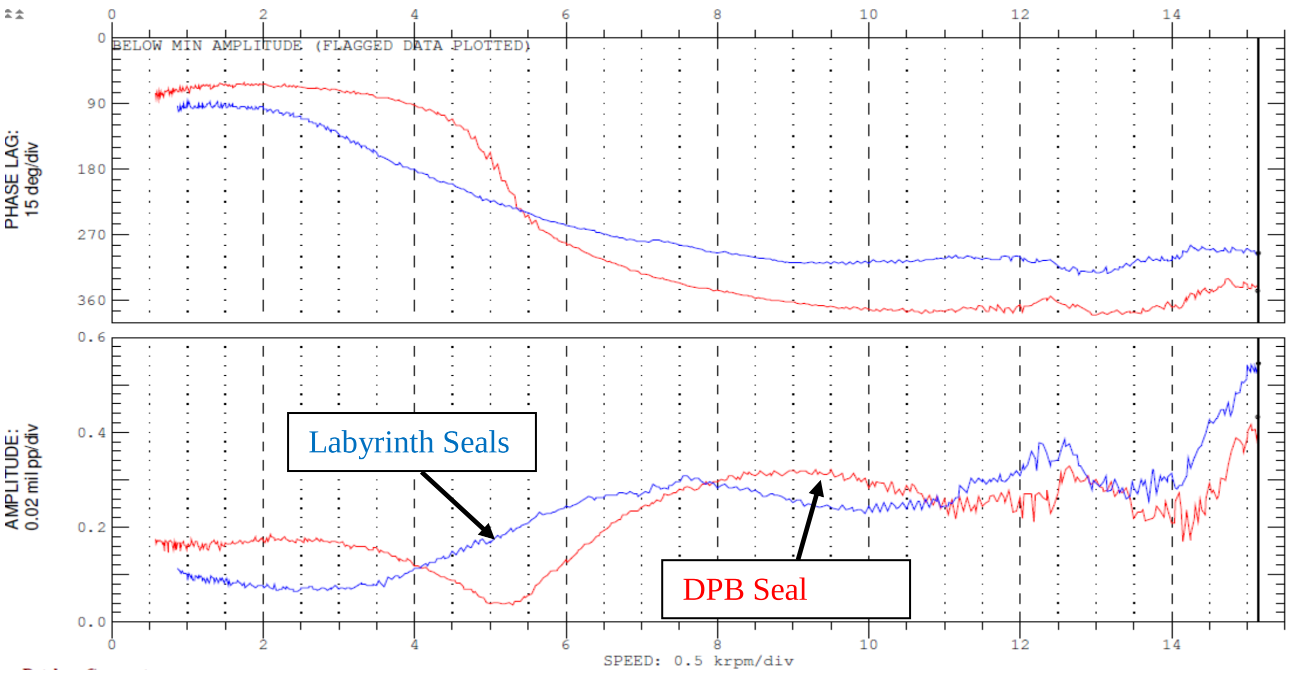

Figure 14: As tested synchronous rotor response at drive end vertical probe.

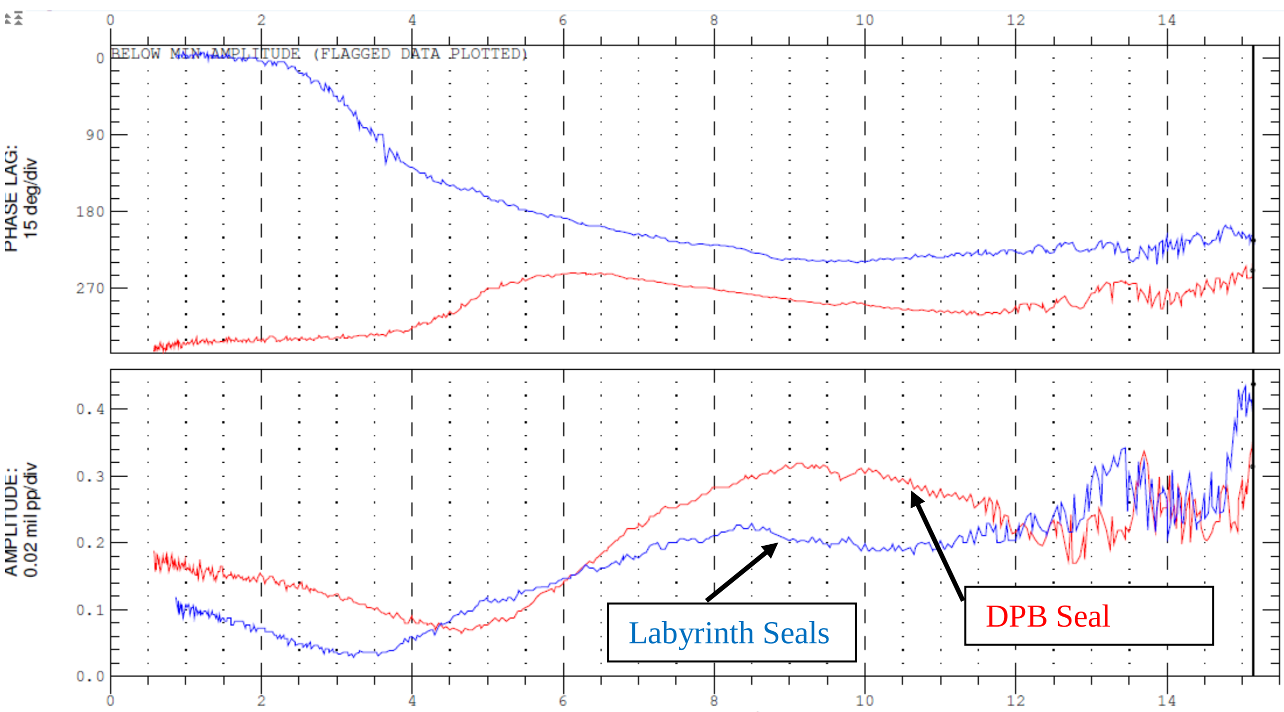

Figure 15: As tested synchronous rotor response at drive-end horizontal probe.

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OMA techniques used to extract natural frequencies, damping ratios (Cloud, 2009) and log decs during full-load full-pressure Type 1 testing with both labyrinth and DPB seals is also discussed in the testing agenda section. Data from five different aerodynamic performance points were evaluated with the intent of using the measured data to investigate and assess changes in vibration characteristics; specifically, any potential impact to rotor stability under multiple test conditions.

The OMA analysis identified natural frequencies and log dec values within the range of predicted values for the first lateral mode. Analytical model predictions for the first lateral mode at the conditions tested yielded log dec values ranging from 1.4 to 2.3. As discussed in the literature, heavily damped modes with log decs above 1.0 can be difficult to detect by OMA analysis if there is insufficient energy in the system to excite them. Nevertheless, singular value decomposition (SVD) curves and spectral results showed the presence of low-level harmonic (forced excitation) energy in the expected first mode frequency range at each test condition. The results of the OMA tests are shown in Table 1 for aerodynamic performance points B,C,D,E and F as shown on Figure 11.

		Point B			Point C			Point D			Point E			Point F		
		Predicted	Laby FLFP	DPBS FLFP	Predicted	Laby FLFP	DPBS FLFP	Predicted	Laby FLFP	DPBS FLFP	Predicted	Laby FLFP	DPBS FLFP	Predicted	Laby FLFP	DPBS FLFP
Freq. (CPM)	Min	7,936	8,622	8,700	8,016	8,610	8,460	7,954	8,115	7,584	8,033	8,760	7,980	8,165	7,278	7,248
	Max	8,414			8,532			8,448			8,511			8,670		
Log Dec	Min	1.45	1.17	1.48	1.57	1.02	1.61	1.83	1.91	1.52	1.82	1.98	1.08	1.90	1.28	1.29
	Max	1.91			1.96			2.26			2.26			2.32		
Damping Ratio	Min	0.23	0.18	0.23	0.24	0.16	0.25	0.28	0.29	0.24	0.28	0.30	0.17	0.29	0.28	0.30
	Max	0.29			0.30			0.34			0.34			.035		

Table 1: OMA test results versus prediction.

First, let's compare test results using the laby to the results using the DPB seals. At point B, C and F, the variation in natural frequency location is 1.7 percent or less. At points D and E, the variation is larger, up to 9.7 percent. Referring back to Figure 11, points D and E are toward the low-flow portion of the map. Still comparing test data and turning focus to the log decs, point F shows similar values, while points B and C show that the log dec improved with the introduction of the DPB seals. Test points D and E show the opposite trend, the log decs decreased with the introduction of the DPB seals. Again, points D and E are toward the low-flow portion of the map. Figure 9 shows how the predicted performance map (discharge pressure vs. flow) shifts to the right when the DPB seals are modeled; this is due to less internal seal leakage. With the DPB seal at low-flow, the lower log decs and larger variations of natural frequency location may be attributed to the unit being closer, or into surge. Damping ratios are included in Table 1, and as suspected, follow the same trend as the log decs.

Now, turning focus toward prediction versus test results. At point F, the predicted natural frequency is 900 CPM over predicted. This may be attributed to a few factors. First, it is well documented in the literature that the introduction of damper seals at the mid-span can have a large influence in the location of the first natural frequency (Memmott 1994). Furthermore, it is well documented that as delta pressure across the damper seal increases, so does the dynamic stiffness of the seal (Childs, 2013). At this sealing location there are two seals (orifices) in series. If the first seal (laby or DPB) is more effective, the delta pressure across the damper seal could be reduced, lowering the tested natural frequency. Secondly and more likely, the OMA process is not identifying the correct mode and cannot find the mode in the 8,500 CPM range due to the high log decs. In retrospect, with these high log decs having a known forcing function applied with a magnetic bearing exciter may prove to have less data variation.

Further work is required to determine why the frequency variation increased as the compressor was moved to the low-flow portion of the curve. Finally, a rotor system that has a measured log decrement above 1.0 at full load and full pressure is indicative of an extremely stable system. Although considerable effort was extended to minimize variability in the rotordynamic stability testing, uncertainty exists in the test data. The hole pattern seal stiffness and damping plays a dominate role in the system dynamics, clouding the effects of the labyrinth and DPB seals. The hole pattern seal increases the log decs to the level where OMA techniques may produce high levels of uncertainty. The OEM is planning on conducting additional DPB seal stiffness and damping characterization

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via dedicated laboratory rig testing where a DPB seal can be isolated with less statistical noise in the system.

In conclusion and in comparison of the OMA stabilization diagrams, modal distributions, and spectral data, they show no indications of significant differences in rotordynamic stability between the DPB seals and laby seals; both systems are highly stable.

AERODYNAMIC PREDICITONS

The aerodynamic performance prediction with the DPB seals indicate that 2.0 percent less power is required at the certified operation condition compared to the standard labyrinth seals. There was also a corresponding higher inlet flow, with essentially the entire original performance map moving to the right. The aero performance prediction accounted for the DPB seals installed at each impeller eye location, as well as at the division wall seal location, as shown in Figure 7 (Colby 2005).

AERODYNAMIC TEST RESULTS

The results of the ASME PTC-10 Type 1 aero performance test indicated that 2.8 percent less power was needed across the entire range of inlet flow rates and pressure ratios for the subject compressor (Figure 16). The solid line represents mu input (work input) to achieve the desired head rise and delivered mass flow with the labyrinth seals and the dotted line represents the same for the DPB seals. The actual test data is shown with the open diamond shapes while the lines represent a curve fit to the data points. The fact that the two lines are substantially parallel, in a strong indication that the variability and hence uncertainty between these two tests is low.

For this series of testing care was taken to conduct the testing with minimum disruption to the instrumentation. All instruments were calibrated per the ASME PTC-10 code and OEM operating procedures, and the same (non-disturbed) instruments were used to conduct both tests. Type II testing is generally recognized in the industry as having better quantitative accuracy than Type 1 testing. In this instance, with a type 1 versus type 1 test comparison, the results are more qualitative than quantitative. The hardware was the same, except for the change out from the labyrinths seals to the DPB seals. The uncertainty in the measured test data between the two tests is estimated to be +/- 2.0 percent.

The difference between the predicted performance and the actual measured performance with the DPB seals may be attributed to the division wall seal design. Recall, this design consisted of using a hybrid design for the division wall seal. The hybrid seal is a combination of a hole pattern seal in series with the DPB seal. The hole pattern was included to ensure rotordynamic stability at all conditions. Furthermore, for the hole pattern seal to be rotor dynamically effective, the differential pressure across the hole pattern seal was designed for 1,200 psid. This leaves 600 psid of the overall 1,800 psi differential pressure for the DPB seal. The consequence was that the division wall leakage was higher than predicted using the DPB seals alone. Nevertheless, as was mentioned in the preceding mechanical test results section, the measured log decrements and rotor dynamic stability were kept at conservative values, ensuring unit stability under all operating conditions.

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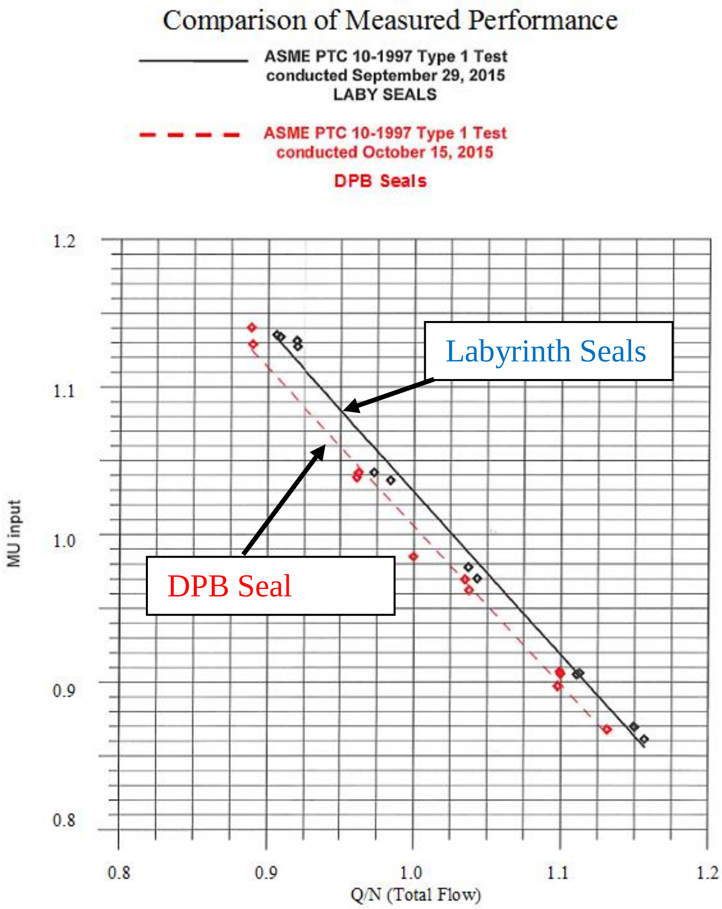

Figure 16: Full Load Full Pressure test results, work input (mu input) vs. flow divided by speed.

CONCLUSIONS

The client's motivation, along with the design and testing of dynamic pressure-balanced (DPB) seals for turbomachinery was presented in this paper. Test results for an ASME PTC-10 Type 1 test of a 4,500 psia (310 Bara) discharge pressure gas lift centrifugal compressor, outfitted first with labyrinth seals and then with dynamic pressure-balanced seals at the impeller eyes, shaft interstage, and division wall were presented. With the DPB seals installed the test results indicate 2.8 percent lower power was required for the same head level across the entire range of inlet flows and pressure ratios, when compared to the same testing with conventional labyrinth seals.

The system rotordynamic stability, obtained via operational modal analysis (OMA), showed the dynamic pressure-balanced (DPS) seals exhibited log decs similar to standard labyrinth seals across the entire range of flows and pressure ratios. This testing was conducted at full load and full pressure with a hybrid hole pattern seal at the division wall. The foregoing demonstrates both the aerodynamic and mechanical/rotordynamic integrity of the dynamic pressure-balanced seals for oil and gas, turbomachinery applications.

After all of the testing, inspection of the hardware indicated no rubbing between the dynamic pressure-balanced seals and the rotor. To that end, the client chose to ship the equipment to the field with the dynamic pressure-balanced seals. It is expected the seals will perform very well during start-and shutdown of the unit. What remains to be evaluated is performance under real-world service that includes parameters that are difficult to simulated and test.

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NOMENCLATURE

ASME	= American Society of Mechanical Engineers	(-)
CPM	= Cycles Per Minute	(1/min)
DPB	= Dynamic Pressure-balance	(-)
FLFP	= Full Load Full Pressure	(-)
GOM	= Gulf of Mexico	(-)
MCOS	= Maximum Continuous Operating Speed	(RPM)
MWP	= Maximum Working Pressure	(-)
OMA	= Operational Modal Analysis	(-)
PTC-10	= Power Test Code – 10	(-)
qA	= Impeller cross-coupling	(lb/in)
QM	= Total Cross Coupling	(lb/in)
rpm	= revolutions per minute	(1/min)
SVD	= Singular Value Decomposition	(-)

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