48TH TURBOMACHINERY & 35TH PUMP SYMPOSIA
HOUSTON, TEXAS | SEPTEMBER 9-12, 2019
GEORGE R. BROWN CONVENTION CENTER

DYNAMICS OF MODULAR ROTORS IN HIGH SPEED CENTRIFUGAL COMPRESSORS: DESIGN, OPERATIONAL PERFORMANCE AND FIELD SERVICEABILITY

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ABSTRACT

High-speed and high-power turbomachines depend on optimal rotor-bearing systems for reliable operation. A key feature of the system is the type of rotor construction; one such type is the modular rotor. While gas turbines have extensively implemented this concept historically, there has been a steady acceptance of modular rotors in gas compressors in the past few decades. Modular rotors have demonstrated mechanical reliability, ease of re-staging and high operational availability in mission-critical operations.

This lecture presents key rotordynamic attributes of centrifugal compressors with modular rotors. Mechanical performance (synchronous/non-synchronous vibration), sensitivity to unbalance and rotordynamic stability are shown for three different machines. Computer simulation models of modular rotors are calibrated to experimental test results. Balancing techniques to achieve vibration levels per API 617 are discussed. Results from balance repeatability tests during rotor assembly are shown to result in low levels of vibration during the factory tests. Historical data on vibration from compressors operating in the field have been presented to validate the design, construction and operation. Two case studies comparing vibration characteristics of such rotors at factory tests and at site conditions are presented.

INTRODUCTION

Centrifugal-type turbo-compressors are employed extensively in the Oil & Gas industry for handling natural gas in a wide range of applications, such as gas gathering, processing, transmission, storage/withdrawal, re-injection and liquefaction to LNG. The reliability of these machines is critical to the end-user's site operations. With the recent boom in fracking and increased gas production, end-users demand improved durability requirements from OEMs. Among many different engineering attributes of these machines, the dynamics of the rotor-bearing systems play a key role in determining trouble-free performance. Modular rotors (also referred to as stacked rotors) are widely used in gas turbines for sizes ranging from 1 MW to over 100 MW. While rotors in gas turbine applications are subject to steep temperature gradients during operation from start to full load, the gas compressor rotors are exposed to higher pressures and gas densities.

Rotor construction varies widely amongst OEMs, but broadly categorized as Overhung or Beam-style designs. Within the Beam-style rotors, the assembly methods lead to two classifications: Modular rotors and Solid rotors. The modular rotors are built with multiple sections and impellers that are assembled together using a tie-bolt along the center. The Solid rotors have a single piece shaft on which the impellers are shrunk-fit. Figure 1 shows a typical modular rotor, in straight-through and back-to-back versions.

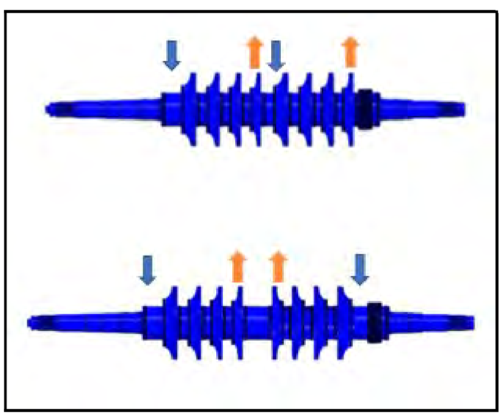

Figure 1. Modular Rotors, Straight-through and Back-to-Back versions, with direction of flow depicted

Based on historical usage, there is a large body of literature reported on the performance of solid rotor systems. The Kaybob compressor rotordynamic instability in the mid-1970's and the design methodologies used to suppress instability are well known. Smith (1975) and Fowlie and Miles (1975) showed the detrimental impact of a highly flexible rotor and the benefits of increasing rotor stiffness diameter to increase the stability threshold. Baumann et al. (1999) presented the rotordynamics of two high pressure applications and the ability of swirl-brakes to suppress vibration. The extensive operational experience with solid rotors for various applications was presented by Fulton (1985) and Memmott (1998). The comparison of measured rotordynamic performance to predictive capabilities have always been a challenge for the industry due to the physical nature of non-linear rotor-bearing systems.

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Nicholas and Kocur (2005) presented case studies to guide OEMs and end-users to employ empirical techniques for predicting the stability characteristics per the API-617 standard. Recently, Baldassere et al. (2015) employed an innovative approach to measure dynamic stability of rotors and compare with predictive capabilities.

All the above-mentioned work pertained to solid rotors, either straight-through or back-to-back configurations. The breadth of published literature on modular rotor designs is minimal. Kocur, Nicholas and Lee (2007) used a modular rotor to survey the industry's state-of-the-art on predicting rotordynamic stability and the variations in tilt-pad journal bearing and labyrinth seal coefficients. Moore and Lerche (2009) compared the analytical rotordynamic performance characteristics of a straight-through modular rotor to an equivalent solid rotor, and concluded that despite some minor differences, both rotor designs meet the criteria prescribed in API 617. The case of a high-speed stacked rotor design for a high-pressure ratio compressor was presented by Vannini (2014), who showed the performance of a stacked rotor in no-load/part-load tests with high-speed balancing. More data on this class of machine, including the effects of axial loads on the center tie-bolts, was presented by Falomi et al. (2016). The objective of this lecture is to describe certain rotordynamic attributes of modular rotors and their operational experience.

ROTOR DESIGN

The rotor design configuration used throughout this lecture is a straight-through modular rotor. Figure 2 shows the typical rotor, with various rotating components such as the stub-shafts, shrouded-impellers, rotor spacer and thrust collar, stacked and assembled together with a center tie-bolt and a tie-bolt nut. The suction stub-shaft is stacked to the first impeller and the discharge stub-shaft to the last impeller. The rotor spacer enables an identical rotor-bearing span, thereby standardizing the casing length, casing's suction and discharge flange positions and facility gas-supply piping arrangements. The materials for the stub-shafts and tie-bolt are selected to provide appropriate yield strength for the components under maximum separation loads. Proper tie-bolt and nut designs are critical to ensure a mechanically robust compressor. Pilots and interference fits provide proper seating and interface between the various components. The dowel pins between the stub-shafts and impellers enable high torque transmission; they also serve to locate the components circumferentially.

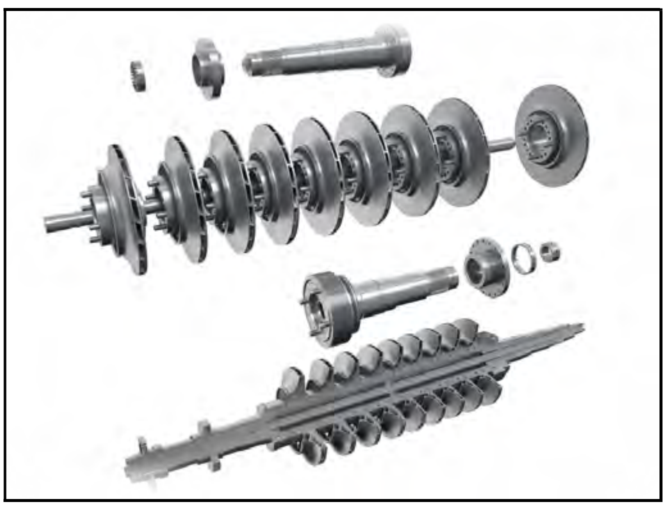
Figure 2. Rotor design of a typical modular rotor

A feature of the modular rotor design is its ability to provide adequate stiffness diameter (Figure 3) for a given impeller inlet diameter. The stiffness diameter is a key attribute for the machine's rotordynamics, with a higher diameter leading to higher shaft stiffness, thereby improving the dynamic stability threshold.

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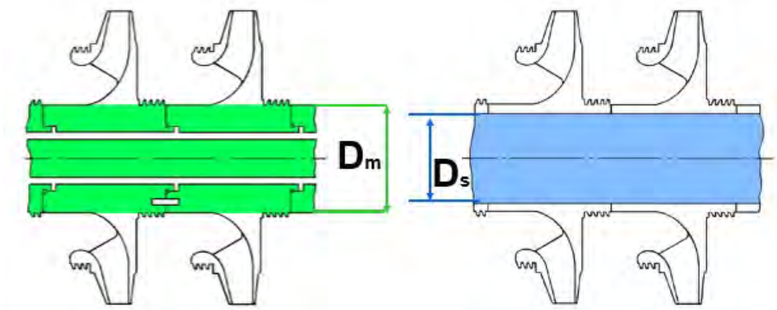

Figure 3. Stiffness diameters of a modular rotor and a solid rotor, for the same impeller inducer diameter

In addition, Finite Element Analysis, confirmed by site experience, has shown that the modular rotor concept can maintain a tighter radial fit during operation. Figures 4 and 5 show the predicted radial growth in the impeller at the rated speed in a solid rotor and a modular rotor configuration respectively. The color contours represent increasing levels of radial deflection due to centrifugal stress. Impellers designed for the modular rotor uses the radial centrifugal growth to tighten the interference fits at the pilot locations. The right-hand side male pilot shown grows radially more than the female pilot of the adjacent part, resulting in a tighter fit during operation. An example of a female pilot callout is shown on the left-hand side of Figure 5. Lower interference fits and short pilots significantly ease the assembly and disassembly of modular rotors.

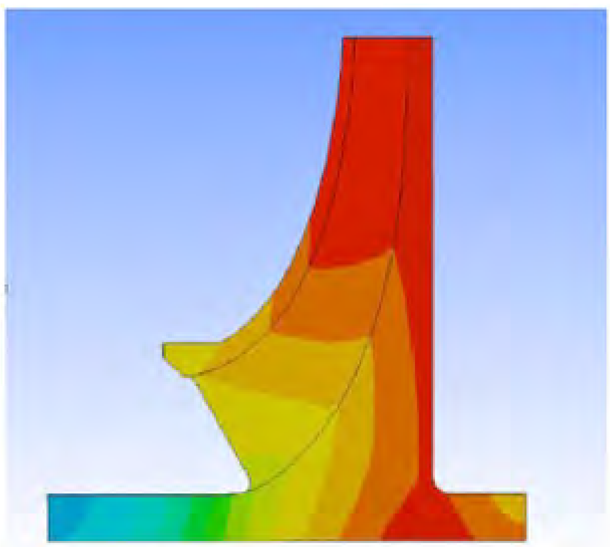

Figure 4. Radial growth in an Impeller on a solid rotor

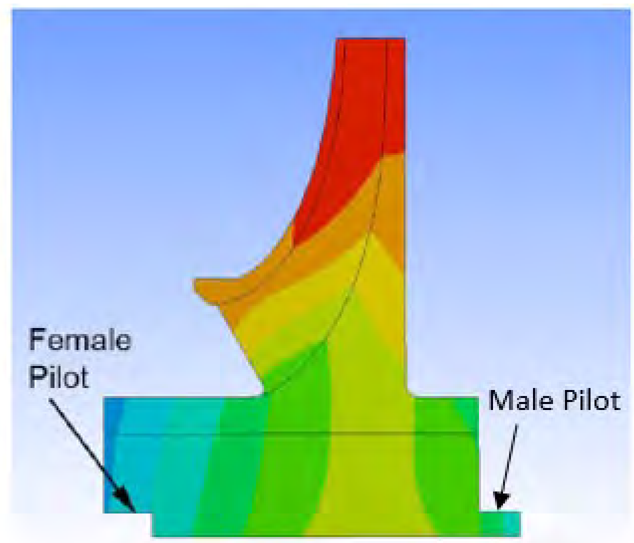

Figure 5. Radial growth in an impeller on a modular rotor

Strict guidelines on system-level design of interface pilot-fits, tie-bolt and axial clamping loads lead to adequate rotor stiffness with multiple interfaces. The diligence in design, combined with proper assembly and balancing techniques, leads to robust mechanical performance as shown in the next sections.

ROTOR RING-TESTS

Rotor ring tests (also known as modal tests) have proven to be a crucial step in modular rotor construction. Thorough ring tests and calibrating the test results to fine-tune analytical rotor models have resulted in more accurate predictions a) of rotor response during unbalance sensitivity tests and b) in estimating rotor stability threshold for site operations. The calibration of predictive capabilities using experimental determination of the free-free rotor modes have helped reduce uncertainties in rotor modeling techniques, particularly as related to:

- the interfaces between the stacked components
- effects of pilot fits on the rotor stiffness
- effects of tie-bolt, axial clamping force, and
- modeling of impeller weights, inertias and stiffness contribution if any.

Results from ring tests on a 6-stage rotor are provided below. The rotor is long, with a slenderness ratio (L/D) of 9.9, making it even more important to fine-tune the rotor model for accurate dynamic analysis. Figure 6 shows the subject rotor, along with the instrumentation used for the ring tests. Ten accelerometers are used in pre-determined axial locations along the rotor to measure modal deflections. The natural frequencies and the corresponding mode shapes are obtained from impedance response analysis of the rotor.

Figure 6: Ring test rotor - suspended in a horizontal position with ten tri-axial accelerometers

Table 1: Rotor ring test Summary

	Ring Test (Measured)	Prediction (Original)	Prediction (Calibrated with test results)	Ring Test vs Calibrated Prediction
1st Bend	196.6 Hz	193.3 Hz	195.2 Hz	-0.7 %
2nd Bend	396.9 Hz	399.7 Hz	402.6 Hz	1.4 %
3rd Bend	629.1 Hz	627.2 Hz	630.7 Hz	0.3 %

Table 1 shows the comparison of natural frequencies, measured vs predicted. The original predictions match very well with the ring test results. It should be noted that the original predictions themselves are based on prior ring tests of shorter rotors of the same compressor family. It shows the low sensitivity of the above-mentioned uncertainties to rotor slenderness ratios. Although further tuning was not necessary, one was done to match closer with the 1st bend frequency. The new calibrated predictions fall within 1.4% of the measured results for all modes considered. Figure 7 shows the mode shapes for these 3 modes.

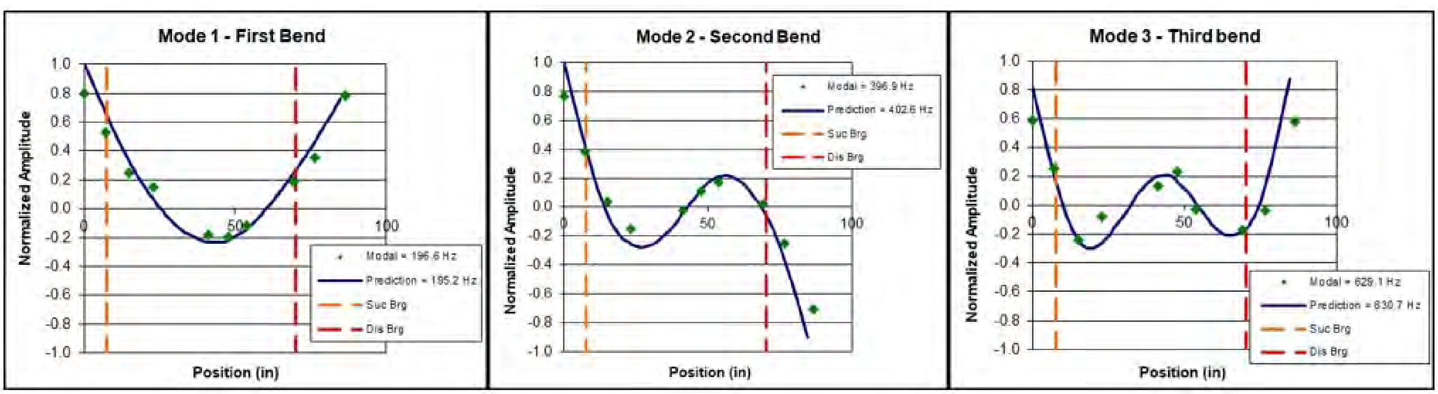

Figure 7. Ring test rotor – comparison of mode shapes – predictions vs modal tests

Modal tests are conducted on all rotor designs of new compressor models and the results are used to fine-tune analytical rotor models based on beam-element formulation. Such a model-based calibration technique avoids the necessity to conduct ring tests on every rotor at Sales phase.

RUNOUTS AND ROTOR BALANCE

Mechanical reliability of high-speed turbomachinery begins with the quality of component level manufacturing. Meeting precision, dimensional tolerances and concentricity are essential to meeting standards and operational reliability. Each rotating component is balanced and corrected for mechanical runout independently. Once assembled, the rotor's runouts (TIR) are checked for consistency. The importance of minimizing runouts cannot be overstated. Rotor runouts are measured on V-blocks along key locations of the rotor

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(correction planes) and are corrected to ensure that the built-up rotor's concentricity after assembly meets runout limits.

The final step in assembling a rotor is the balancing. With the beam style rotors supported on two bearings with span length (L) much greater than diameter (D) such that (L/D)>>1, the rotors generally display one or more lateral bending critical speeds within its operating range (from startup to Maximum Continuous Speed). It is essential to minimize the residual unbalance to traverse these critical speeds without any damage to the internal components such as the seals, bearings or impellers. The axis of rotation is the same as the axis of symmetry. The origin of unbalance is the accumulation of manufacturing eccentricity in each major plane. The net residual unbalance can result in a large radial force, further emphasizing the need for accurate balancing requirements. A rotor is balanced if its mass is uniformly distributed in each individual radial plane such that a) it displays no tendency to roll, b) the radial planes themselves are not offset with respect to the axis of rotation, and c) it will produce no (or minimal) net centrifugal force in any plane while rotating on its axis. In practice, the recommendations provided by API 617 are used for all key aspects of balancing, such as maximum residual unbalance levels, runouts, selection of balance planes and type of balancing.

Balancing techniques have evolved and improved with rotor designs that operate at high speeds. OEMs can select the best balancing choice based on the intended service and class of the rotor in question. The primary reasons for each type of balance method are summarized in Table 2.

Table 2. Choices of balancing techniques depending on rotor construction

Rotor		**Balance**	
Rotor characteristics	Rotor assembly	Balance method	Balance purpose
Robust & conservative – with good margin from critical speed interference	Precision components and light fits.	2-plane Low-speed	Final trimming
	Heavy shrink fits. Possible unbalance induced during assembly.	Multi-plane high-speed	Compensating for tolerance build-up
A compromise between rigidity and other competing goals resulting in slender flexible design	Assembled with high precision but rotor is slender & super-critical	Multi-plane high-speed	Allows rotor reach desired speeds

Progressive balancing techniques cannot be applied to modular rotors since all impellers and spacers must be in place prior to performing the tie-bolt stretch and clamping. However, extensive experience shows that the API standards on balance quality, reliability and long-term successful operation with modular rotors can be met or exceeded with a combination of stringent component level balancing, runout corrections and low-speed two-plane balancing. The following sections will highlight this observation.

It is to be noted that, once the compressors are built, these rotors are coupled to light weight couplings (with the driver) whose mass geometry and attachment type greatly influence rotor response at high speeds.

BALANCE REPEATABILITY TESTS

As discussed in the previous sections, modular compressor rotors are easy to assemble, disassemble and balance. They have good balance repeatability and predictable mass-elastic properties. The balance repeatability of two 9-stage rotors, Rotor A and Rotor B, with identical impeller configuration are presented in this section. The rotors weigh roughly about 1075 lb, with impeller tip speeds in excess of 1000 ft/s. Both rotors include the thrust collar and balance sleeve during the balance repeatability tests. Figure 8 shows the assembled Rotor A. All the rotor parts are match-marked during the first assembly to enable reassembly of the parts each time at the same relative angular orientation. For each rotor (Rotor A and Rotor B), the assembly-disassembly procedure was conducted four separate times. At the end of each assembly, runouts along the rotor were checked to ensure build compatibility and the rotor balance condition was measured.

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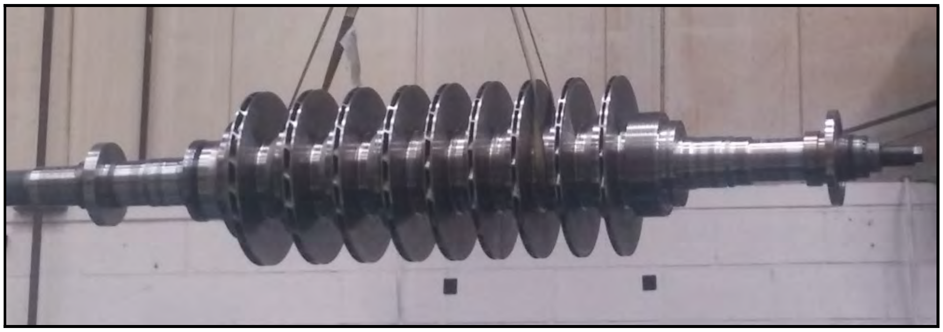

Figure 8. Fully Assembled Rotor A

Rotor A

The results from the build repeatability tests on Rotor A are shown in Figure 9 and Figure 10 for the suction and discharge sides respectively. In each figure, the green markers represent the balance measurements in oz-in. The label next to the each datapoint represents the sequence number of the rotor repeatability build. The measuring tolerance of the balance machine “X”, for the given rotor weight, has been chosen as the reference marker for the measured balance data. Multiples of this tolerance have been represented as blue circles on the chart. The dashed red line represents the residual unbalance limit at 1xAPI (4W/N oz-in) and the solid redline represents the residual unbalance limit at 4xAPI (16W/N oz-in). The charts with the balance measurements are in the Cartesian Coordinate System. The repeatability tests on Rotor A were conducted before balancing the rotor to the machine tolerance limit. This provided an as-built view of the balance repeatability of the rotor before correcting the residual unbalance. Table 3 below shows the measured balance conditions. The baseline balance condition on the suction and discharge sides are about 8X and 12X respectively. During the repeatability tests, the balance condition of the rotor did not exceed 16X in either plane. Reminder: ‘X’ implies balancing machine tolerance.

Table 3. Rotor A - Balance Repeatability Measurements Prior to Balancing

Rotor Build #	Suction		Discharge	
	Amplitude	Phase	Amplitude	Phase
	oz-in	deg	oz-in	deg
1	0.601	171	0.909	270
2	0.466	19	0.265	61
3	0.729	12	1.3	324
4	1.03	39	0.601	332
5	1.11	31	1.05	332

Rotor B

The results from the build repeatability tests on Rotor B are shown in the charts in Figure 11 and Figure 12 for the suction and discharge sides respectively. These charts are formatted in the same manner as with Rotor A. However, unlike Rotor A, repeatability tests on Rotor B were conducted after balancing the rotor to the machine tolerance limit. This is reflected in the very low unbalance in the first datapoint (label ‘1’) in the charts. During the repeatability tests, the balance condition of the rotor did not exceed 8X in either plane. Table 4 below shows the measured balance conditions for Rotor B.

Table 4. Rotor B - Balance Repeatability Measurements After Balancing

Rotor Build #	Suction		Discharge	
	Amplitude	Phase	Amplitude	Phase
	oz-in	deg	oz-in	deg
1	0.043	300	0.022	309
2	0.172	16	0.081	48
3	0.221	356	0.184	315
4	0.206	321	0.567	255
5	0.42	18	0.143	296

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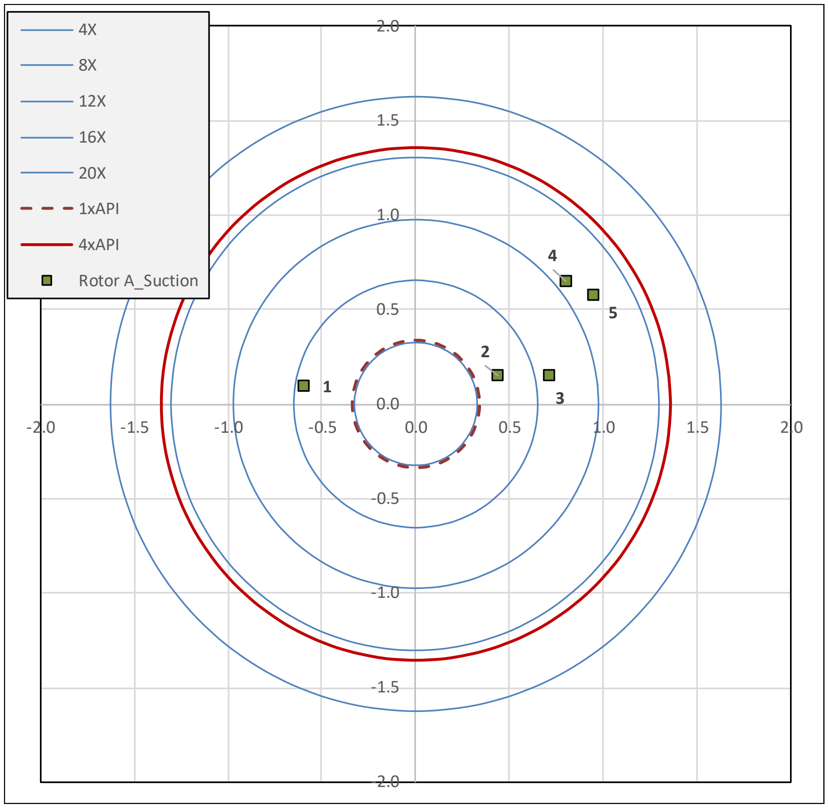

Figure 9. 'Rotor A' Build Repeatability Results at Suction End *before* balancing

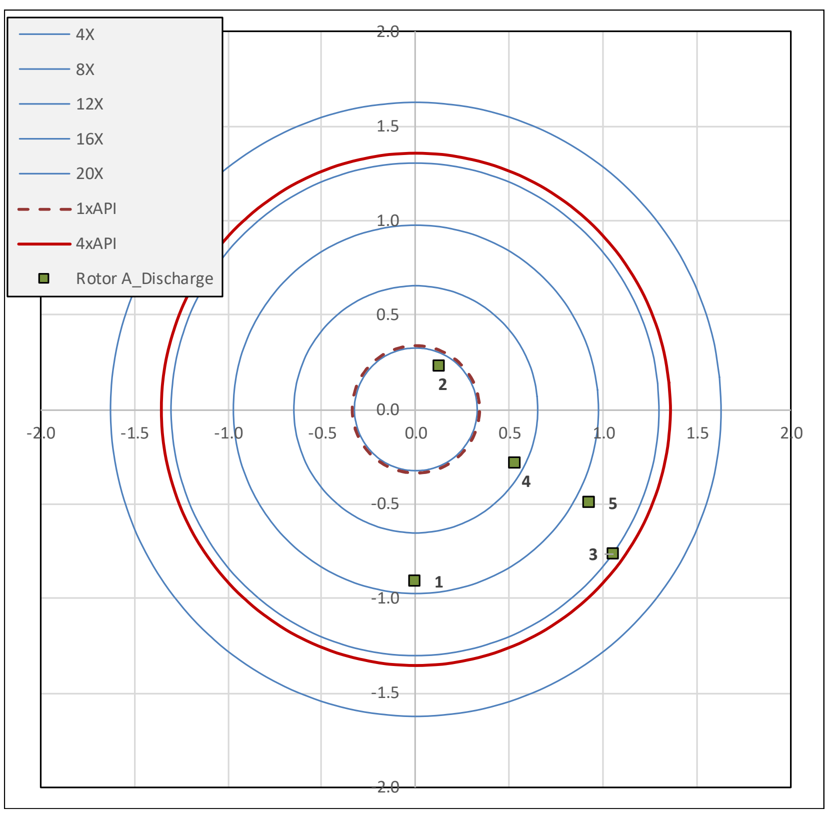

Figure 10. 'Rotor A' Build Repeatability Results at Discharge End *before* balancing

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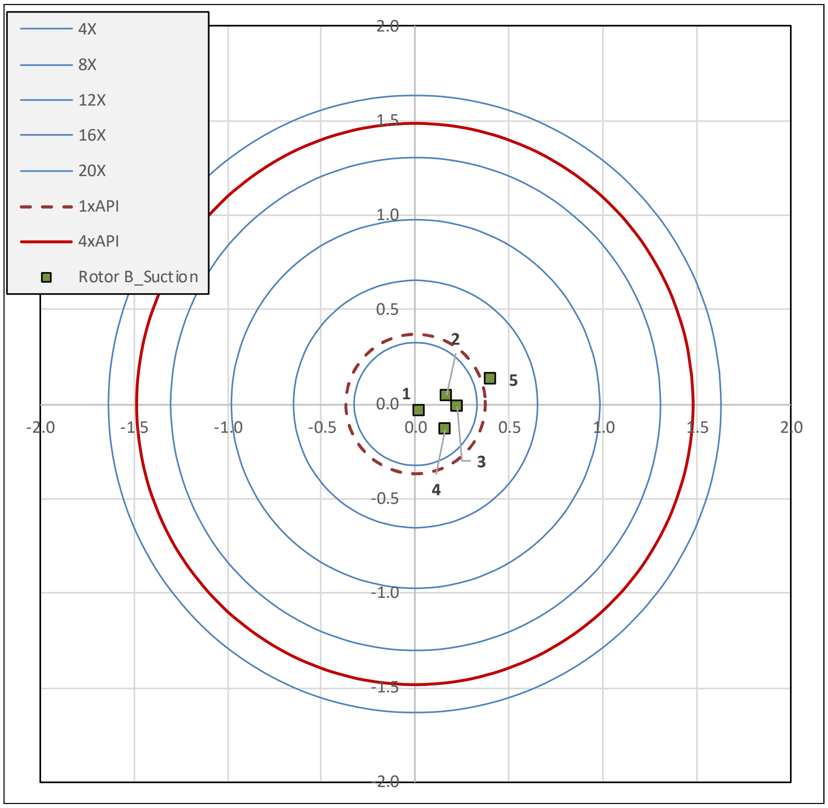

Figure 11. ‘Rotor B’ Build Repeatability Results at Suction End *after* balancing

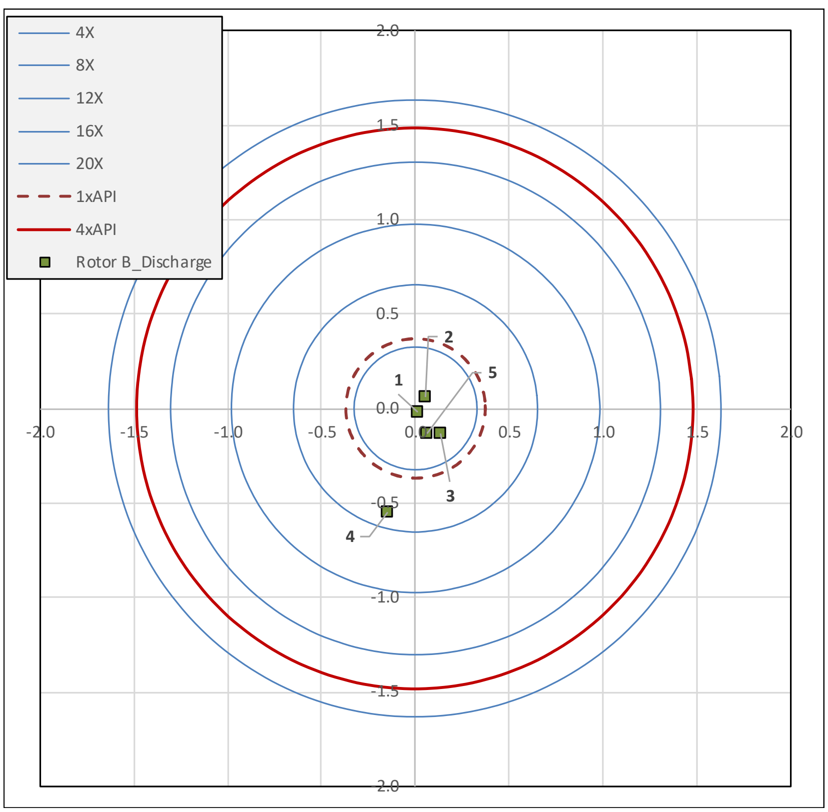

Figure 12. ‘Rotor B’ Build Repeatability Results at Discharge End *after* balancing

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ROTOR UNBALANCE SENSITIVITY TESTS

Rotor unbalance sensitivity tests are performed per API 617 requirements as part of compressor development. Trial weights are added at the shaft-ends, using provisions on the trim balance sleeve (non-driven end) and on the coupling adapter (driven end). The weights may be positioned either In-Phase or Out-of-Phase, relative to each other. The magnitude of the weights corresponded to an unbalance of 16W/N (oz-in), four times the API-recommended value. A Finite Element code was used for the prediction of rotor response and nominal bearing clearance values were used for the calculations. The results from the unbalance sensitivity test on Rotor B are shown in Figure 13 and Figure 14. Figure 13 shows the measured and predicted response at the suction and discharge ends for the first bending mode (3rd natural frequency) and Figure 14 shows the measured and predicted response at the suction and discharge ends for the conical mode (2nd natural frequency).

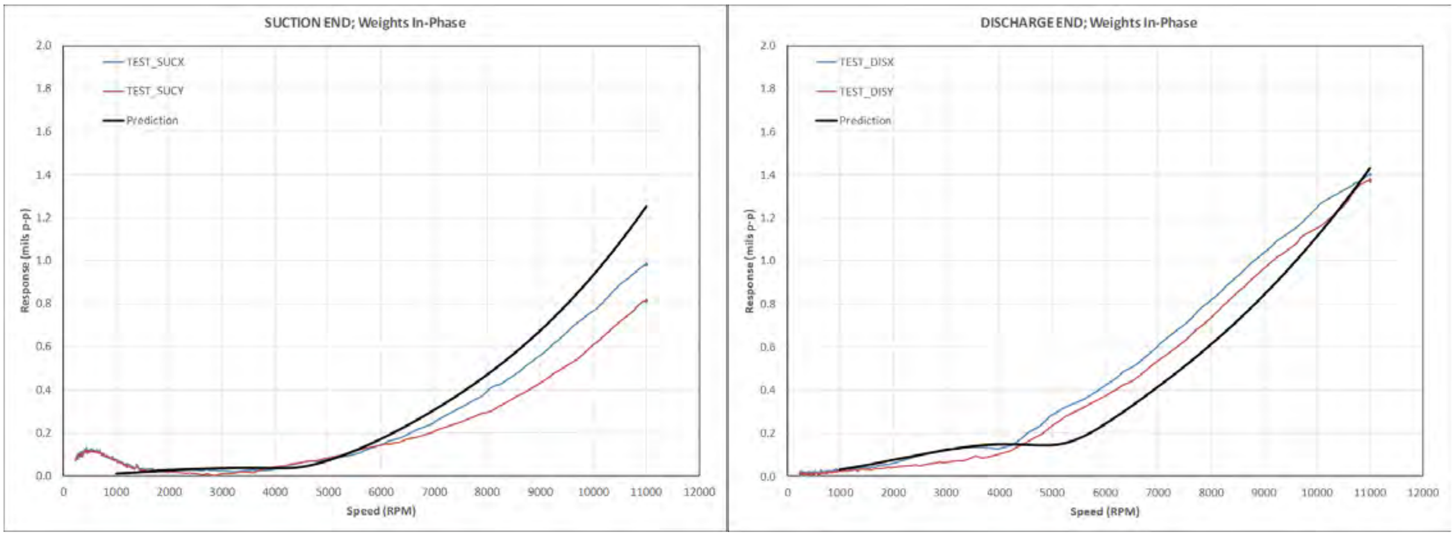

Figure 13. Unbalance Sensitivity Tests – Rotor B – Prediction vs Measurement – In-phase trial weights
(X-Scale, Speed: 0 to 12,000 rpm, Y-scale, Amplitude: 0 to 2.0 mils)

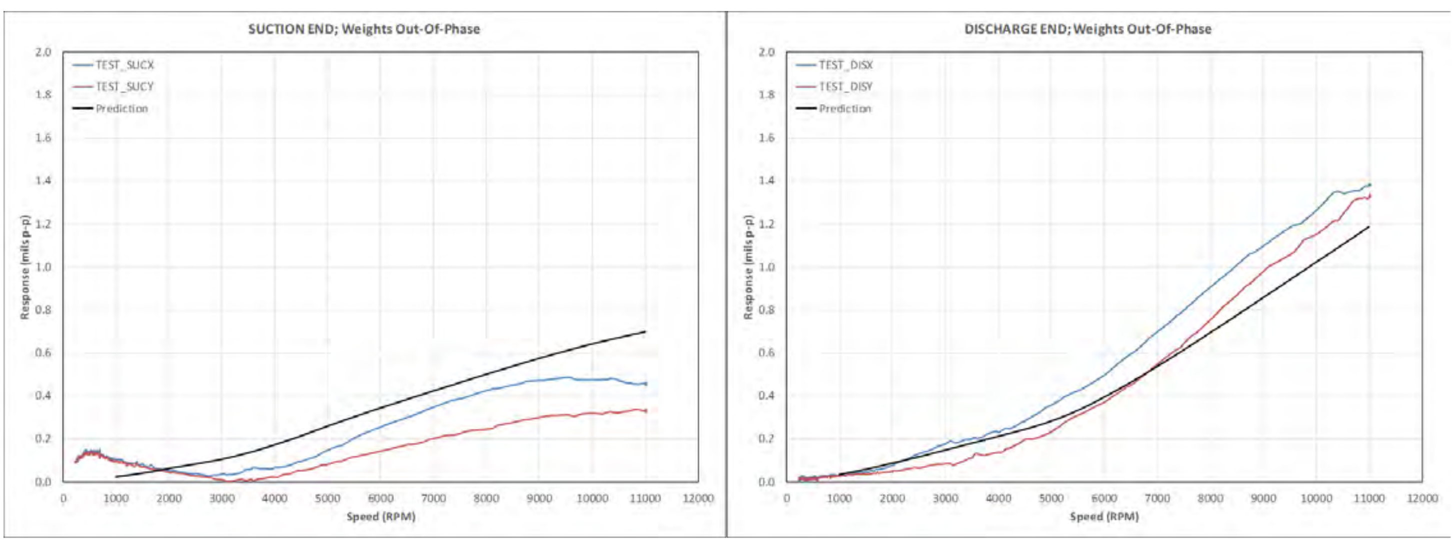

Figure 14. Unbalance Sensitivity Tests – Rotor B – Prediction vs Measurement – Out-of-phase trial weights
(X-Scale, Speed: 0 to 12,000 rpm, Y-scale, Amplitude: 0 to 2.0 mils)

The measurements match well with predictions for the In-phase trial weights (Figure 13), which excite the rotor's first bending mode that is above the maximum operating speed of the compressor. The comparison with Out-of-phase trial weights (Figure 14), which excite the rotor's conical mode, shows reasonable match with the location of the mode. The first mode is critically damped.

Experience from site installations shows that the modular rotor's sensitivity to unbalance is low, barring any serious uncorrected

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residual unbalance conditions or foreign object damage (FOD) during operations. This is elaborated further in an upcoming section.

COMPRESSOR FULL-LOAD DEVELOPMENT TESTS

Full-load development tests are conducted in a closed-loop test cell capable of 300 bar (4500 psia) Nitrogen or utility Natural gas. The driver in this test facility is a 13 MW (17,400 hp) gas turbine engine. The compressor built with Rotor B is a dual-compartment (straight-through) type. More of its attributes are provided in Table 5. The test gas is intercooled to achieve higher pressure ratios. The data shown below was taken during natural gas testing.

Table 5. Attributes of the Dual-compartment compressor with Rotor 'B', tested at full-load

Parameter	Details
Construction	Dual-compartment, Straight-through
Maximum Allowable Working Pressure	3750 psia
Nominal Pressure ratios	6-8
Number of stages tested	4 (Front section) + 5 (Back section)
Maximum Continuous Speed	13,000 rpm
Bearing size	4 inch
Bearing Length/Diameter	0.65

Figure 15 shows the bode (coast-down) plot of this rotor at no-load conditions. The response appears smooth, indicates the presence of the first critical speed around 4400 rpm, although the mode is critically damped per API.

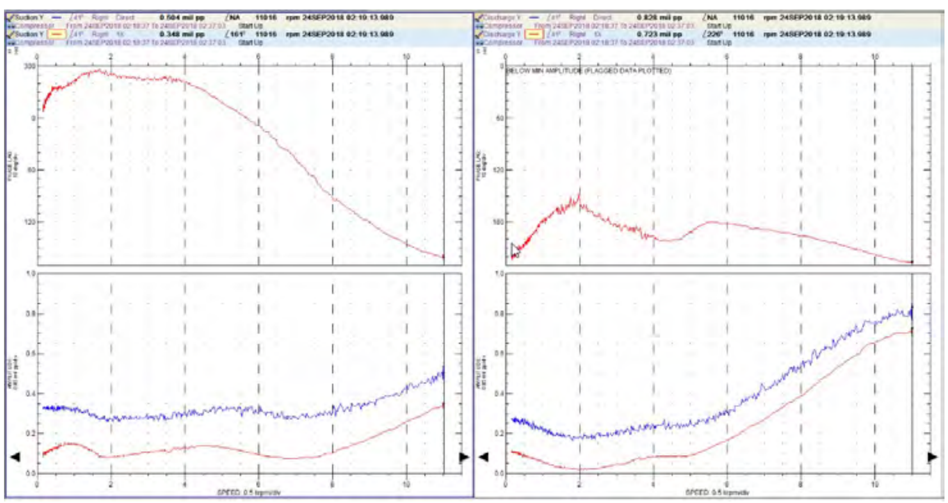

Figure 15. Coast-down plot of Rotor B from Max continuous speed, at no-load
(X-Scale: 0 to 12,000 rpm, Y-scale-Amplitude: 0 to 1.0 mils, Y-scale-Phase: 180^{o} to 300^{o})

Figure 16 shows the waterfall plots on the suction and discharge side of Rotor B at a discharge pressure of 145 bar (2100 psia) and a pressure ratio of 6.7. The rotor operating speed was 11,000 rpm. The plot shows vibration signatures for about 30-minutes as the compressor flow is throttled from the choke condition to the near-surge condition, and back to choke. As observed, the synchronous vibration is steady and acceptable with less than 1 mil p-p at both ends. No sub-synchronous vibrations can be observed. For reference, the predicted damped resonant frequency of this rotor-bearing system is 74 Hz. The logarithmic decrement predicted by rotordynamic simulation at the operating conditions, using API's recommended approach for modeling destabilizing forces, is 0.48.

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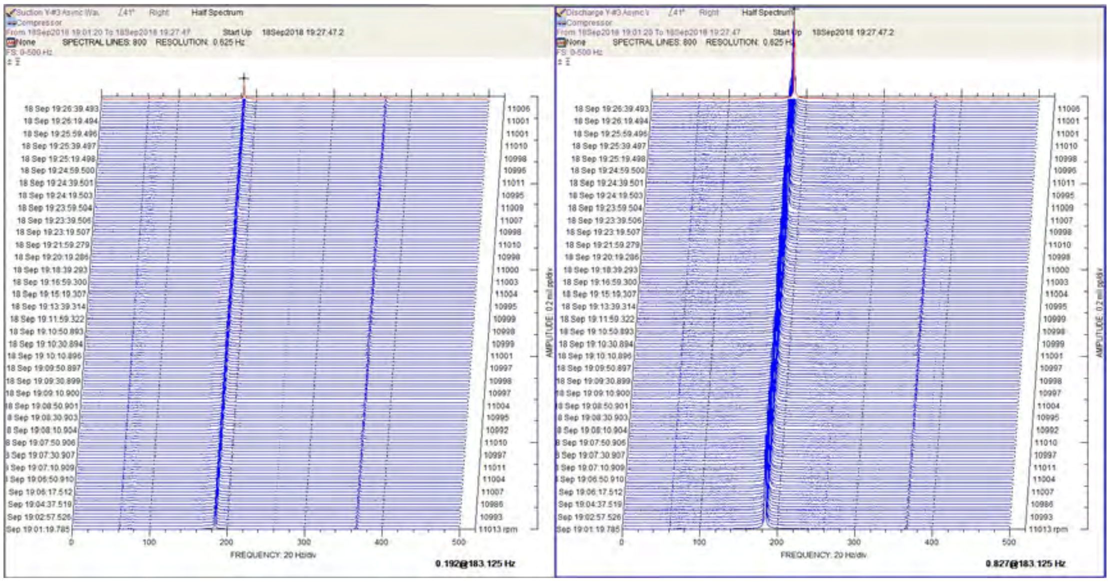

Figure 16. Waterfall plot of ‘Rotor B’ during full-load test; Discharge pressure=2100 psia; Pressure Ratio = 6.7, Speed=11,000 rpm

COMPRESSOR SITE PERFORMANCE

Modular rotors have shown reliable and consistent rotordynamic performance during operations at the end-user installations, both in upstream (on-shore/off-shore) and midstream services. A key characteristic of these rotors, as mentioned earlier, is their low sensitivity to unbalance caused possibly by dirt-buildup or minor impact from debris in the gas stream. Unanticipated incursions into surge, while strongly discouraged and actively controlled by Anti-surge systems, is tolerated well by these rotors.

The site performance of several pipeline gas transmission compressors along the US South-East corridor were monitored over a 6-month period using remote monitoring and data acquisition capabilities. All rotors in the study have between 3 to 6 impellers, and all of them have shafts with 4 inch bearings. Vibration at these six compressor installations are shown as trend plots for the suction-end (Drive-end) and discharge-end (Non-drive end) in Figures 17 and 18 respectively. As noticed, the vibration levels over the 6-month period is steady around 1.0-1.3 mils peak-to-peak, except during the stop-start cycles. The rotor vibration Alarm levels for the compressors studied are 2.5 mils peak-to-peak or higher, based on the compressor size.

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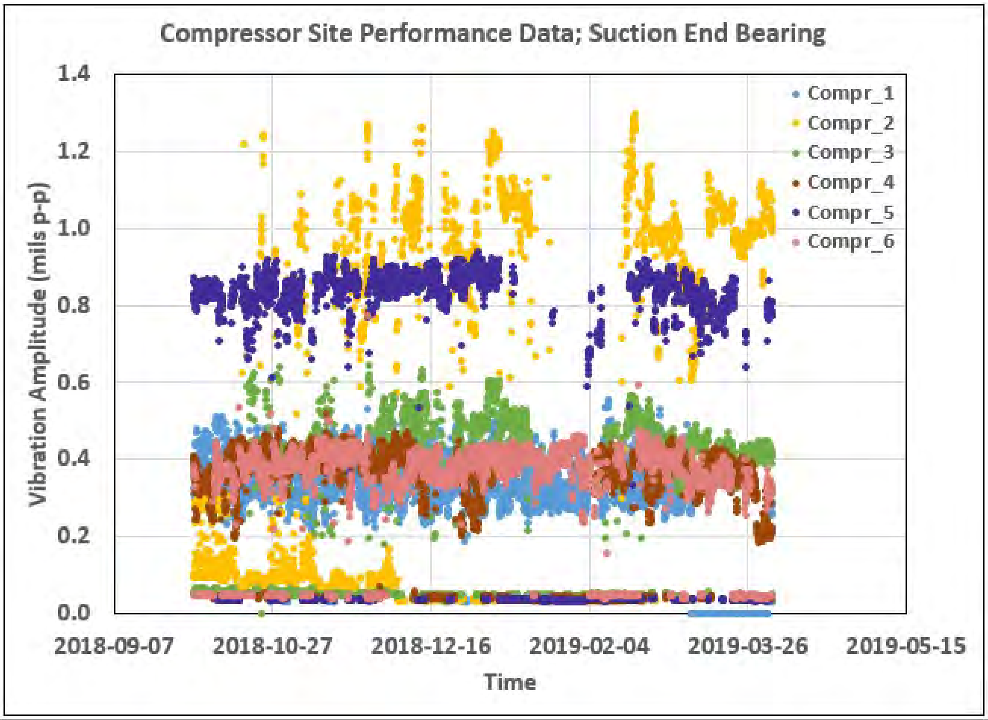

Figure 17. Site vibration trend over six-months for six different compressor installations (Suction end Bearing)

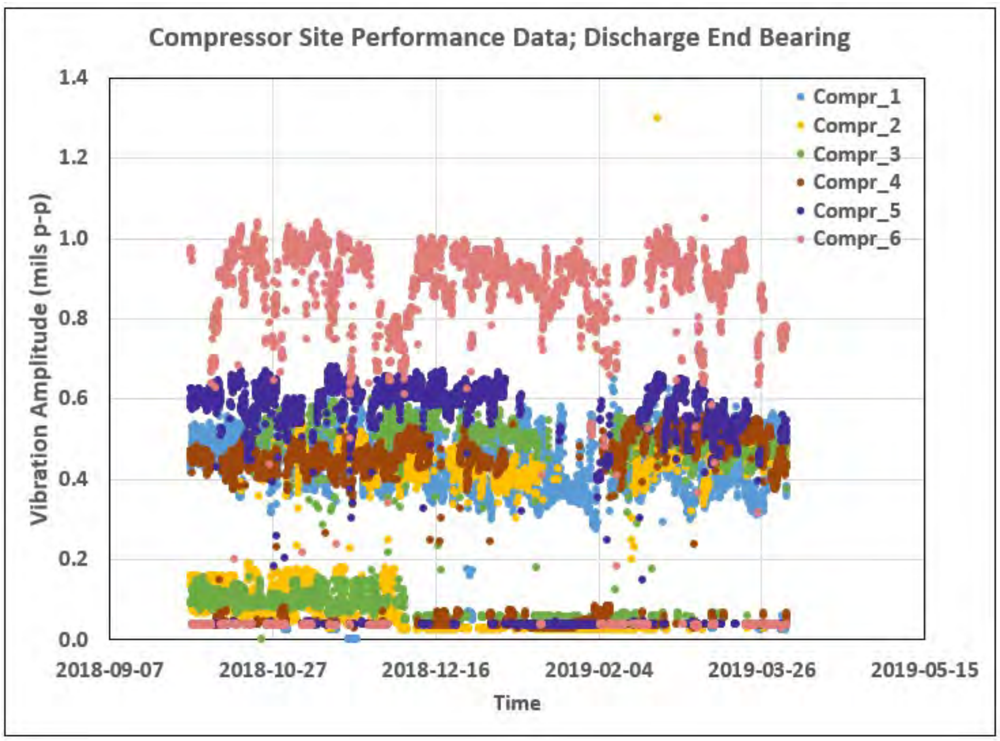

Figure 18. Site vibration trend over six-months for six different compressor installations (Discharge end Bearing)

The above plots show the consistency of vibration levels over time. To complement the same with a review of overall vibration levels across even more compressor samples, a random collection of 51 compressors operating at many sites were surveyed for vibration levels over a 6-month period. These compressors are typically used in gas processing and transmission applications. A histogram of the maximum vibration levels exhibited by these compressors over this period is shown in Figure 19.

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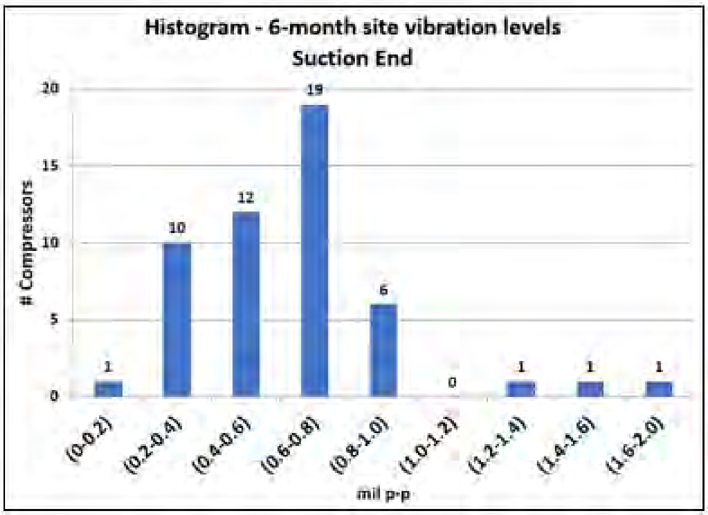

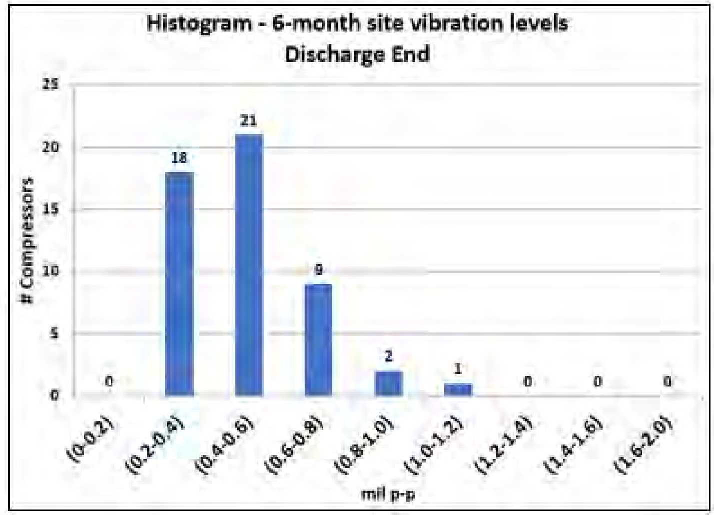

Figure 19. Histogram of site vibration survey of 51 random samples of compressors

The primary purpose of providing a histogram is not to simply show the vibration levels; rather it is provided to highlight the site performance (over time) of a small but statistically significant sample set of modular rotors. The key reasons that lead to such a result are:

- robustness of the modular rotor concept, design and construction
- quality of manufacturing and assembly
- effect of low-speed 2-plane balancing
- reliability of the compressor's auxiliary systems, including bearings, seals, lube-oil and dry-gas seal systems.

All these combined provide critical value to the end-user in maintaining trouble-free operation.

CONCLUSIONS

A discussion of key design, manufacturing and assembly features of modular rotors has been presented in this paper. Calibration of simulation tools through modal testing improves rotordynamic predictions on rotor response during Unbalance Sensitivity Tests. Build repeatability tests of a 9-stage rotor demonstrates acceptability before proceeding to final balance and assembly. Rotordynamic stability analysis of modular rotors (without including swirl brakes, damper seals or damper bearings) shows healthy logarithmic decrement levels, much higher than the minimum values recommended by API. This is also correlated with trouble-free performance of modular rotors at end-user installations for applications involving high pressures or pressure-ratios. Trends of vibration levels for a random sample of compressors during extended periods of operation are presented, showing levels within acceptance criteria.

Modular rotors possess certain unique characteristics, offering comparability in the following areas:

- stiffer rotor (higher effective shaft stiffness from increased diameter), with favorable results
- allows solid diaphragms, preventing gas leakage at the diaphragm split-lines, thereby minimizing impact on efficiency
- unbalance adequately corrected with low-speed balancing, resulting in low sensitivity to unbalance
- operational reliability and consistency at end-user installations
- easy to assemble and disassemble, helping end-users with shorter rebuild times during in-situ rotor rebuilds

NOMENCLATURE

NCR	= Rigid Bearing Critical Speed	(CPM)
NGC	= Gas Compressor Speed	(RPM)
MCS	= Maximum Continuous Speed	(RPM)
L	= Bearing Span	(L)
D	= Stiffness Diameter at Hub	(D)
L/D	= Slenderness ratio	(-)

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ACKNOWLEDGEMENTS

The authors wish to express their gratitude to Chester Lee, Aziz Fozi and Bert Edlebeck (all three retired Solar Turbines engineers), whose instrumental work on gas compressor designs forms the basis for this paper. The authors thank Solar Turbines for the permission to publish this work and Cheniere Energy for participating in the development of this paper.

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