FULL-SCALE AERODYNAMIC AND ROTORDYNAMIC TESTING FOR LARGE CENTRIFUGAL COMPRESSORS

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ABSTRACT

This paper describes a full-scale, flexible test vehicle designed and built by the original equipment manufacturer (OEM) to validate the aerodynamic and mechanical performance of large compressors for a variety of applications. This paper provides a description of the test vehicle as well as mechanical and aerodynamic performance data gathered during testing of the vehicle.

INTRODUCTION

Demand for the most energy-efficient compressor equipment has continued to increase in recent years, driven in large part by the increase in oil and natural gas prices. In response, OEMs have focused a considerable amount of resources on developing new and improved centrifugal stages.

Advances in computational fluid dynamics (CFD) have made it possible to assess new stages or new design concepts without "cutting metal." Such analyses facilitated the study of the flow physics within the aerodynamic components, and enabled analysts to investigate the various flow effects that might be evident and to modify the configurations to eliminate any adverse flow phenomenon. In short, any flow phenomena that contributed to excess losses could be identified and addressed before "cutting metal."

However, in the OEM's experience, CFD is not sufficiently reliable to develop final performance predictions for new compressors, especially those that push the envelope of the OEM's experience. End users typically require stringent performance tolerances, and CFD is simply not up to the task. Therefore, OEMs still must rely on testing to fine-tune or optimize compressor configurations. This is especially true in heavy mole weight or high-speed applications that require impellers that operate at high peripheral or machine Mach numbers. For example, compressors in heavy mole weight service such as carbon dioxide (CO2) or propane handle gases with very low gas sonic velocities, resulting in high relative Mach numbers in the aerodynamic flow path. By their nature, such high Mach number, high-flow coefficient stages have very narrow flow maps characterized by limited choke and surge margin (Sorokes, et al., 2006; Sorokes and Kopko, 2007). Moreover, the use of fixed-speed drives in some of these applications removes flexibility that might otherwise be available to meet specified operating conditions, further increasing the challenges imposed on the OEM. Likewise, it is possible to achieve high Mach numbers even in low mole weight applications if the rotational speed is sufficiently high. Therefore, it is important that the compressor vendor and end user understand the performance characteristics of any impellers, diffusers, return channels, and/or other flow path components that are used in such equipment. The high cost associated with delays in a project schedule increase the attractiveness of design and testing methods that ensure these machines meet contractual operating requirements the first time, i.e., without needing to disassemble the compressor to modify its internal components to correct performance shortfalls.

In the past, OEMs relied on scaled model test vehicles or so-called single-stage test rigs (SSTRs) to validate new stages or components. Numerous examples of such rigs can be found in the open literature, including the work of Benvenuti (1978), Sorokes and Welch (1991), Sorokes and Welch (1992), and Sorokes and Koch (1996). The cross-section of a typical "stage-and-a-half" rig can be seen in Figure 1. While very valuable to gather validation data on new stage components, such rigs do not capture all factors that influence stage performance because they do not have a true centrifugal stage upstream; i.e., they are not multistage. Furthermore, most scaled test vehicles are much smaller than the compressors used by many clients. Therefore, so-called "size effects" must be addressed to translate the performance from the small rig to the full-scale machine. Such adjustments can lead to conflict between OEMs and end users if they do not agree on the correction factors.

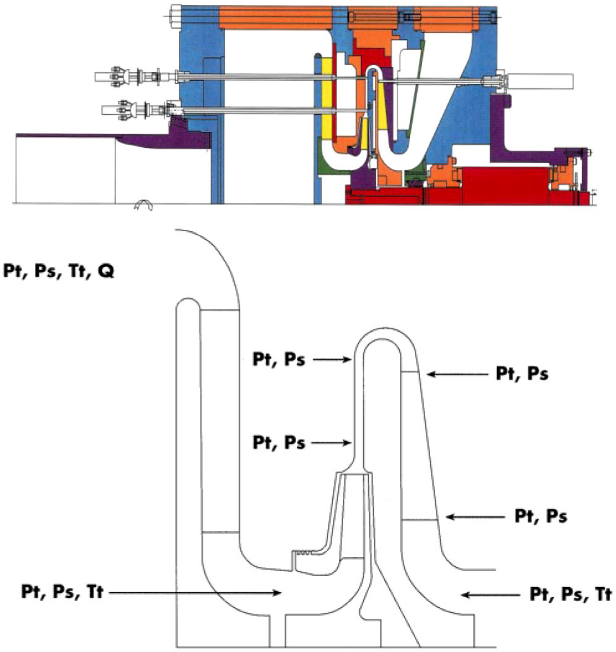

Figure 1. Cross-Section of a Single-Stage Test Rig.

In some cases, OEMs have relied on research testing in production equipment such as the work described by Sorokes, et al. (1998) and Gilarranz, et al. (2004). With limited exceptions, these test vehicles included fixed geometry in the various stationary components (i.e., inlet guide vanes, diffuser vanes, and return channel vanes). Quite often, changes in the vane setting angles were desired to improve overall stage performance or to investigate the aerodynamic or mechanical response to varying setting angles. In such cases, new components were fabricated and time was spent removing original hardware and replacing it with the new components. This significantly increased the cycle time for completion of test programs and often delayed the release of new designs into production.

The use of movable geometry is an attractive option to replace the need to change test vehicle internal components. Moving vanes to various setting angles without requiring disassembly of the test rig significantly reduces the test cycle time and maximizes the data that can be gathered from a single build.

THE NEW TEST VEHICLE

In 2005, facing demand for higher capacity and higher Mach number applications, the OEM developed a new test vehicle as part of ongoing efforts to further enhance the aerodynamic and mechanical performance of their large frame size compressors. The new test rig is a full-scale, multistage unit that can accommodate an impeller up to 60 inches (1524 mm) in diameter. The objective is to validate the performance of a full-scale compressor both aerodynamically and mechanically very early in the order execution process, possibly before the receipt of the purchase order for the new equipment. Because compressors vary significantly among plants, the test vehicle case construction had to be very flexible; i.e., it needed to be able to accommodate a variety of compressor arrangements. Similarly, the aerodynamic flow path needed to be flexible enough to determine the optimal configuration for peak production/performance.

To provide the necessary flexibility, a segmented case was developed. The compressor casing comprises a series of rings as can be seen in Figure 2. The arrangement shown is for a multisidestream compressor, so each segment includes a nozzle. However, alternative arrangements can be built with segments having no intermediate nozzles for a "straight-through" compressor or nozzles midstream for "back-to-back" or "double-flow" configurations. Further, the assembly approach allows a variety of case lengths to be created simply by changing the number and/or the length of segments/rings (see Figure 3).

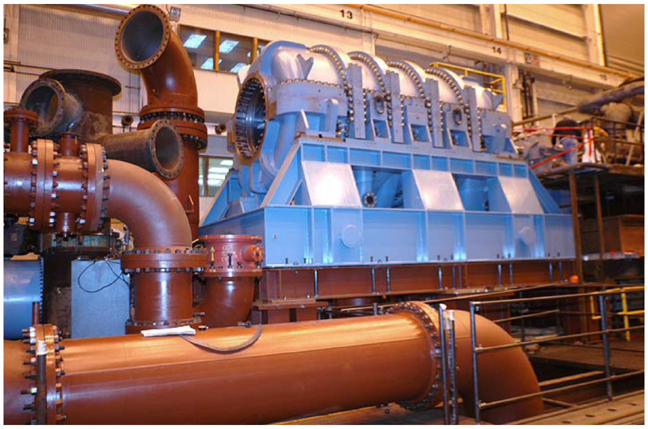

Figure 2. Segmented Casing on Test Stand.

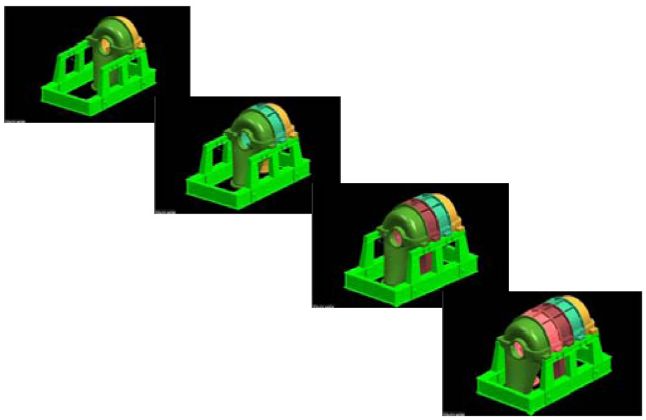

Figure 3. Schematic of Various Case Arrangements.

A significant amount of engineering time was devoted to designing the joints for the new vehicle to ensure that the necessary case rating would be achieved. Special attention was given to the bolting arrangement, particularly in the area of the four-corner joint, i.e., where the horizontal and vertical segments came together. A four-corner joint is typically very difficult to seal, so extra care was taken to make sure that these joints would seal properly. As will be discussed later, R-134 was to be used for the majority of the performance testing so no leaks could be tolerated.

Several arrangements were analyzed using finite element analysis with the key consideration being maintenance of adequate sealing at all joints at the pressure levels anticipated for the rig operating conditions. Samples of the finite element results are shown in Figure 4. As can be seen, with the bolt size and spacing specified, good contact is maintained in the joint.

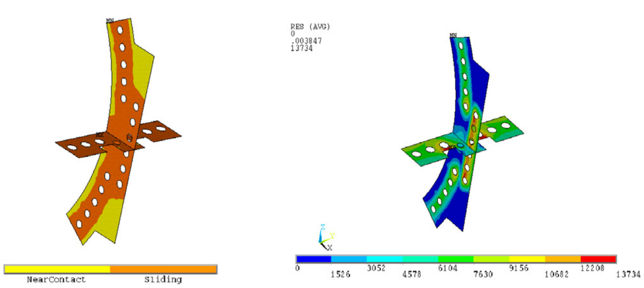

Figure 4. FEA of the Joint Arrangement.

To increase the flexibility of the compressor, movable vanes were installed in all stages. Applying movable geometry in centrifugal or axial turbomachinery is not new. There is much information in the open literature regarding its use. Examples can be found in company brochures such as those by Dresser-Roots Company (2006), as well as in technical publications such as Sorokes and Welch (1992), Ferrara, et al. (2005), and Brink (2006). However, what distinguishes this rig from prior art is that the movable geometry was applied in *every stage* of a four-stage, beam-style compressor. That is, each compressor stage was fitted with moveable vanes in the guide vanes (trailing edge) and the diffuser (the entire vane), and at the return channel or diaphragm (leading edge). A typical cross-section showing the location of the variable geometry vanes appears in Figure 5.

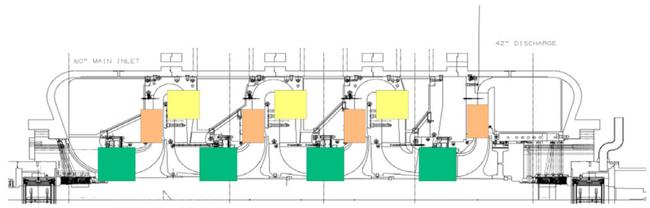

Figure 5. Movable Geometry Locations.

The moveable portions are fully adjustable between predefined minimum and maximum setting angles, providing extensive ability to tune the stage performance and obtain the optimal compromise between range and peak efficiency. The vane orientation is controlled by actuators that are mounted on the outside of the compressor casing (Figure 6) and the vane setting angles are adjusted through a series of linkages while the machine is running at load. Further information regarding the movable geometry system can be obtained by reviewing Gilarranz, et al. (2009).

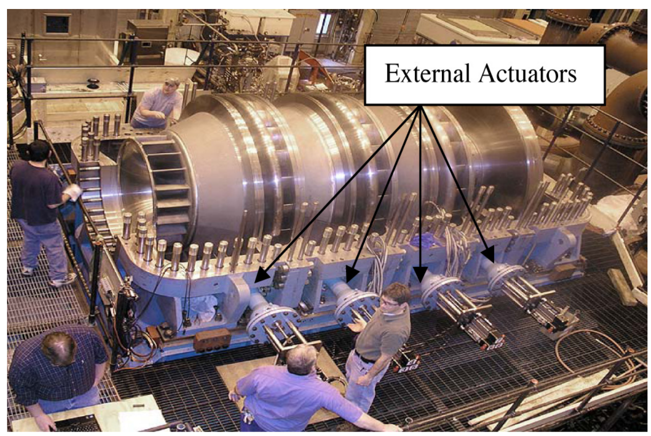

Figure 6. External Actuators.

The aerodynamic flow path was heavily instrumented to allow the measurement of the component performance in each stage. Instrumentation included total pressure probes, dynamic pressure probes, total temperature probes, and five-hole probes. Note that the five-hole probes measure static pressure, total pressure, flow angle, and flow velocity at the probe location. The work of Gilarranz, et al. (2004) provides a discussion of the use of this type of probe in industrial multistage centrifugal compressors. A schematic of the internal instrumentation layout used in each compressor stage is given in Figure 7. Note that there are five-hole probes installed upstream of each movable geometry component. These probes were used to gather flow angle data that were used to validate the optimum setting angles for the moveable geometry components.

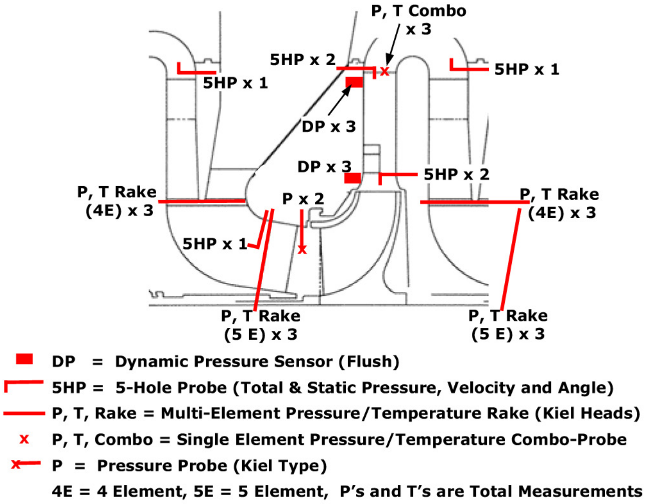

Figure 7. Typical Stage Internal Instrumentation.

It is common knowledge that peak attainable efficiency and wide operating range are, to a great extent, mutually exclusive characteristics (Sorokes, 2003). Therefore, in developing new centrifugal stages, one must be aware of the effects that guidevane, diffuser vane, return channel vane, etc., have on both range and efficiency. With the moveable geometry system, the impact of alternate vane setting angles on overall flow range and/or peak efficiency can be assessed. A matrix of vane setting angles can be acquired in a relatively short cycle time because the vanes can be actuated from position to position in seconds. These data can then be analyzed using optimization software to determine the best combination of setting angles for a given application. For example, one could acquire data at five different guidevane settings, five diffuser settings, and five return channel settings and then allow an optimization utility to determine the optimal setting angle for each to obtain the best combination of range and efficiency (Tecza, et al., 2005). In short, the test cycle time and overall project schedule are significantly decreased, and the risk associated with the application of new compressor stages will be reduced.

The rig also included instrumentation and hardware to assess the rotordynamic characteristics of the compressor including proximity probes at the bearings and midspan proximity probes to provide greater insight into the rotordynamic characteristics. The response data at the midspan probes through the first critical speed were used to validate the rotordynamics models.

Of greater significance, a magnetic bearing exciter (MBE) was used to inject asynchronous forcing functions into the rotor (see Figure 8) that was supported by two tilt-pad oil film journal bearings. The MBE did not support rotor radial loads. It was used to inject forces into the rotor at varying frequencies and magnitudes to measure the log decrement of the rotor during operation, using the same concept as previously done at the authors' company (Moore et al, 2002; Moore, 2003; Soulas and Kuzdzal, 2009). Furthermore, one of the many challenges a designer faces in the construction of a test rig is to ensure that the journal bearings support structure is significantly stiffer than the bearings. If adequate support stiffness cannot be provided to the journal bearings, there is a risk that the second critical speed of the system will encroach on the operating speed range, or for the log decrement of the first forward mode to be lower than calculated. The MBE was used not only to evaluate the log decrement of the first forward mode, but also to inject a supersynchronous forcing function to identify the location of the second critical speed. The assessment of the rotordynamic characteristics was done at various points on the performance map, at various moveable vane-setting angles and at various compressor power levels.

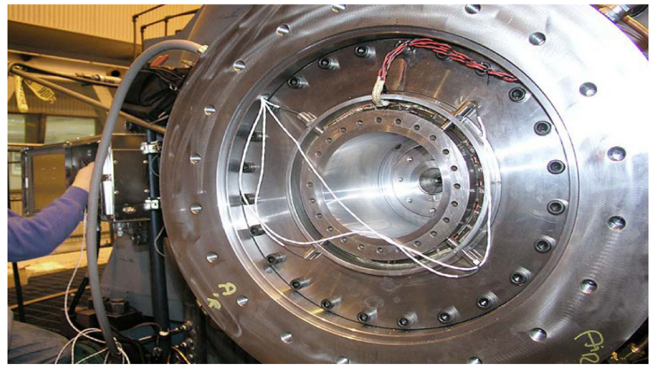

Figure 8. Magnetic Bearing Exciter.

The exciter is used in open-loop to introduce forces of varying frequencies (asynchronous excitation) to assess the response of the rotor system and demonstrate rotordynamic stability at increasing gas densities and horsepower, evaluate the log decrement of the first forward whirling mode, demonstrate correlation with analytical predictions for partial load, and extrapolate for full load field conditions. These data will be discussed later in this paper. This large exciter is capable of producing a force of 11,000 N (2470 lbf) up to 100 Hz (6000 cpm), and 4900 N (1100 lbf) up to 240 Hz (14,400 cpm).

Because the tested configuration was designed to simulate a drive-through machine in the field, a moment simulator was required for the test set-up. The MBE was designed to not only provide forcing functions, but its size and weight also were designed to simulate the moment of the field coupling. This electromagnetic radial bearing exciter includes a rotor-laminated sleeve, and a hub for adjustment on the compressor shaft end (Figure 9).

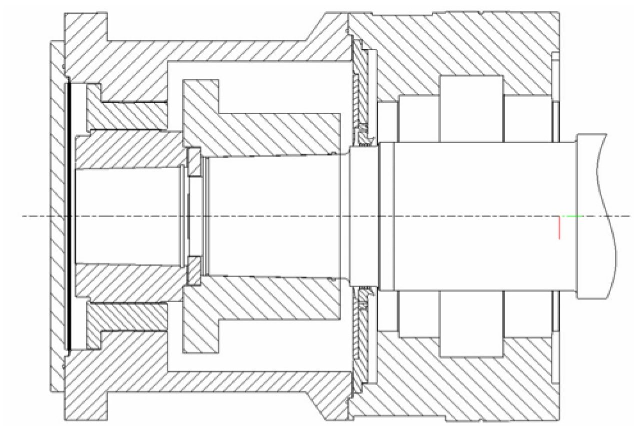

Figure 9. Large Magnetic Bearing Exciter Design.

The test rig was developed to provide detailed, high-quality data that would characterize the aerodynamic and mechanical performance of the compressor. The data will be used to demonstrate that the machine will operate as predicted and with the contractual obligations agreed upon by the OEM and end user. Furthermore, the ability to change the orientation of the moveable vanes during operation provides the capability to improve machine performance without the need to disassemble the compressor to modify the internal components. Finally, because the test rig is full scale, the OEM and end user can have a very high degree of confidence that the performance and mechanical results will be duplicated when the production units are built. In other words, the compressors that would be built for field installation would duplicate the aerodynamic flow path and the rotor configuration tested in the rig. Therefore, one should reasonably expect the mechanical and aerodynamic performance to replicate those from the test rig.

AERODYNAMIC TEST SET-UP AND RESULTS

The test compressor was installed in a closed-circuit test loop that included piping for the main inlet, sidestreams, and the compressor discharge. A loop with sidestreams allows for an increased amount of data to be collected. The compressor was driven through a gearbox by a 30,000 hp (22.37 MW) steam turbine. A listing of the key components in the test train is given in Table 1.

Table 1. Test Rig Materials.

Component	Description
Driver	22.37MW (30,000HP) Steam Turbine
Gear	Lufkin Speed Reducing Gear
Bearings	305mm (12.0") RMT in diameter
Gas Seals	343mm (13.5") John Crane
Compressor	D26A4S with 3 sidestreams
Magnetic Bearing Exciter	273mm (10.75") diameter SKF Magnetic Bearing

The aerodynamic performance testing was conducted using R-134A as a test medium. Each compressor section was tested individually at different speeds to meet the site volume reduction, Mach number and, where applicable, the flow proportioning between the core flow and the incoming sidestream flow. The testing was conducted in accordance with the ASME PTC-10 Code (1997). The only exception was the method used for flow proportioning the sidestream flows. The OEM has developed a proprietary method that requires the flow proportioning to be based on a flow function (Kolata and Colby, 1990) that ensures that the test is conducted under tighter tolerances than those specified in the ASME Code.

The test data were evaluated using real gas laws as defined by the ASME Code. The gas properties were calculated using the Lee-Kesler-Ploecker (LKP) Equation of State.

AERODYNAMIC TEST RESULTS

A tremendous amount of data was gathered on all four stages of the compressor. It is the purpose of this paper to show the rig's effectiveness as a vehicle for stage optimization.

Unlike most other test vehicles, this rig does not require any extrapolation or interpolation of the data because it is full-size and includes all aspects of the compressor flow path, i.e., upstream stages, downstream stages, sidestreams, etc. Therefore, as noted earlier, the performance data acquired from this test vehicle can rightfully be expected to duplicate that of a production compressor having the equivalent internal aerodynamic flow path.

To determine the variation in achievable performance, matrices of vane setting angles were run on each stage. The typical range of angles for the inlet guidevanes, diffuser vanes, and return channel vanes are given in Table 2. The setting angles were typically varied in five-degree increments, although intermediate angles were possible if necessary to "fine tune" the performance.

Table 2. Angle Variation on Movable Geometry.

Component	Angle Range
Inlet Guidevanes	0° ± 20°
Diffuser Vanes	Nominal ± 15°
Return Channel Vanes	Nominal ± 15°

As an illustration of the effectiveness of the movable geometry, the head coefficient for the fourth stage is given in Figure 10. The three curves in the figure represent the radial or zero-degree, five-degree, and 10-degree inlet guidevane setting angles. The "leftward shift" of the stage performance map is as expected when additional prewhirl is introduced into an impeller inlet. Note that the entire curve (including the surge and choke points) is moved to lower rates with increasing prewhirl.

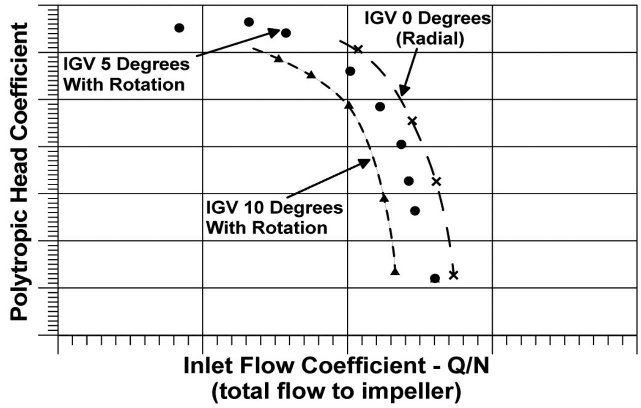

Figure 10. Effect of Movable Inlet Guidevanes on Stage Performance.

As a reminder, the data presented in Figure 10 were gathered in a single day. Without the movable geometry, it would have taken weeks or possibly months to obtain the same data because it would have been necessary to disassemble and reassemble the compressor twice after the initial test to change-out the guidevanes.

As a second example of the effect of the movable geometry, the performance for varying diffuser setting angles is shown in Figure 11. The data were gathered from Stage 2 with constant guidevane and return channel angles. As expected, the variation with diffuser setting angle is not as dramatic as with varying guidevane angle. However, adjusting the diffuser setting angle can create subtle changes in the performance characteristics.

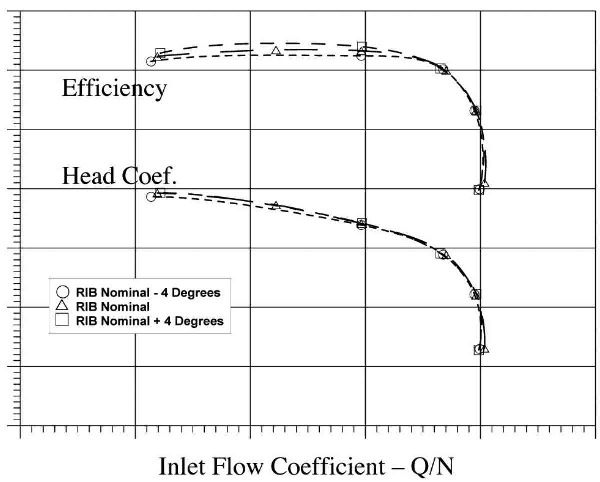

Figure 11. Effect of Movable Diffuser Vanes on Stage Performance.

Although no data are included herein, varying the return channel angle also caused small changes in the performance of Stages 1, 2, and 3. The incremental changes were smaller than those obtained for a similar change in the diffuser angle. That is, a ±10 degree change in return channel angle had less impact on the stage characteristics than a ±10 degree change in diffuser angle. Likewise, a ±10 degree change in diffuser setting angle has a significantly smaller effect than a ±10 degree change in the guidevane angle. This is expected as an inlet guidevane (IGV) directly impacts the impeller performance map and the impeller is the most critical element in a stage.

The testing demonstrated that an adjustable IGV provides the most effective method for adjusting stage performance, affecting flow range, head level, rise-to-surge, and peak efficiency. The movable diffuser vanes had less impact on the performance, primarily affecting rise-to-surge and, to some extent, surge margin and peak efficiency. Finally, the movable return channel vanes had

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the least influence, showing a small effect on the peak efficiency and overload capacity.

Another performance trend of interest was the variation in the stage characteristics with changes in the mass balance between the sidestream flow and core flow. The ASME PTC-10 Power Test Code (1997) allows the mass balance to vary by as much as ±10 percent from the design (or client) core flow to sidestream flow ratio. The OEM applies a more stringent criteria, requiring that this so-called "flow function" be maintained to within ±5 percent.

To assess the impact of varying flow functions on the stage performance, data were gathered on the fourth section, i.e., downstream of the third sidestream, with the flow function varied by ±10 percent. The results of this testing are reflected in Figure 12. As can be seen, the performance characteristics change significantly as the flow function is moved from one end of the tolerance to the other. This is a key factor for OEMs and end users to keep in mind as validation tests are conducted on new equipment.

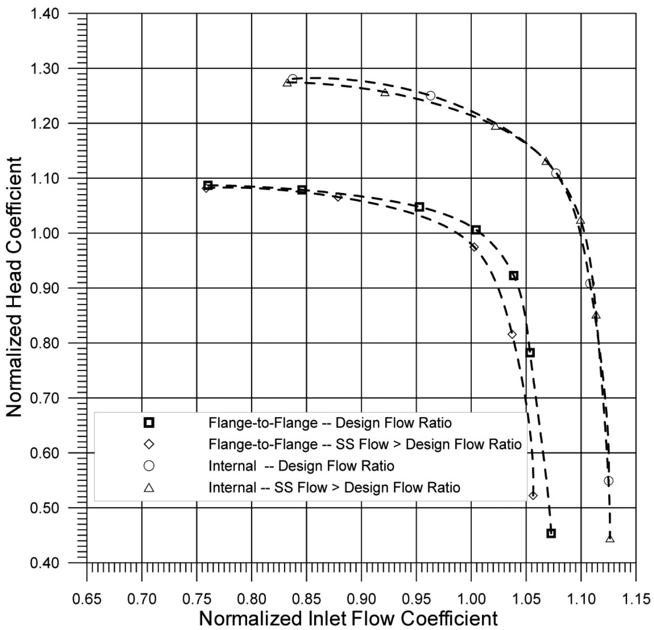

Figure 12. Sensitivity of Stage Performance to Sidestream Flow Function.

MECHANICAL TEST SET-UP AND RESULTS

For the partial-load mechanical test, the maximum available horsepower to drive the compressor was 25,610 hp (19.1 MW). For mechanical testing, the compressor was run on carbon dioxide with main inlet and sidestream flows cooled to 100°F (37.8°C). The flow capacity of each section was set between the design and overload flow coefficient on all stages. The minimum suction pressure to the test vehicle was 5 psia (0.34 bara).

At an inlet pressure of 5 psia (0.34 bara), the consumed power is about 12,200 hp (9.1 MW). At this minimum power level, the first data set was recorded by performing an MBE frequency sweep. The second power level was accomplished by increasing the mass flow through the test vehicle by increasing the suction pressure from 5 psia to 6 psia (0.34 to 0.41 bara) (at this second point, the power consumed by the unit is about 14,600 hp [10.9 MW]). Subsequent power points were attained by increasing the suction pressure in 1 psi (0.07 bar) increments until the maximum horsepower of the driver was reached, up to 10 psia (0.69 bara) suction pressure. It should be noted that the rotor speed remains constant at about 3600 rpm (full speed), while the injection force sweeps through the frequency range resulting in a peak response at the first rotor natural frequency.

The vibration response amplitude at one of the midspan proximity probes during the MBE frequency sweeps (constant excitation force), while running at 3600 rpm, for three different discharge pressures is shown in Figure 13. The maximum amplitude when sweeping through the first forward whirling mode decreases as discharge pressure increases; this is a beneficial result of the positive damping and stabilizing effect of the hole pattern damper seal on the balance piston. The MBE frequency sweep data are used to experimentally evaluate the first forward whirling mode log decrement of the system, and compare it with the analytical predictions.

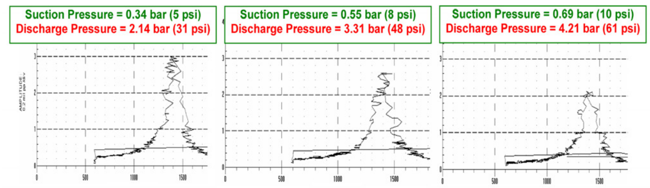

Figure 13. MBE Asynchronous Sweeps: Vibration Amplitude at Midspan Probe Versus Excitation Frequency (CPM).

In addition to rotordynamic stability, the rotor synchronous response also was assessed during the test. The rotor synchronous response at the journal bearing proximity probes during the final acceleration/deceleration at the end of the test is shown in Figure 14.

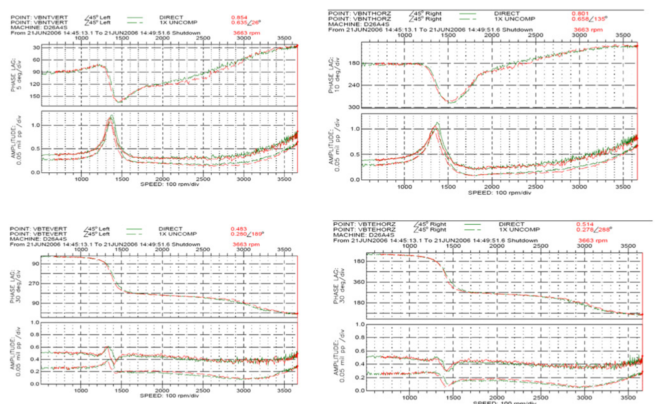

Figure 14. Final Compressor Acceleration/Deceleration (Synchronous Rotor Response).

Finally, frequency spectrum data during steady-state operation with no magnetic bearing excitation are presented in Figure 15. The vibration amplitude levels remained within acceptable levels as per API 617, Seventh Edition (2002), requirements (no high-speed balancing was performed before testing). It should be noted that all the other mechanical characteristics of the compressor (such as journal and thrust bearings temperatures) also met API 617, Seventh Edition, criteria.

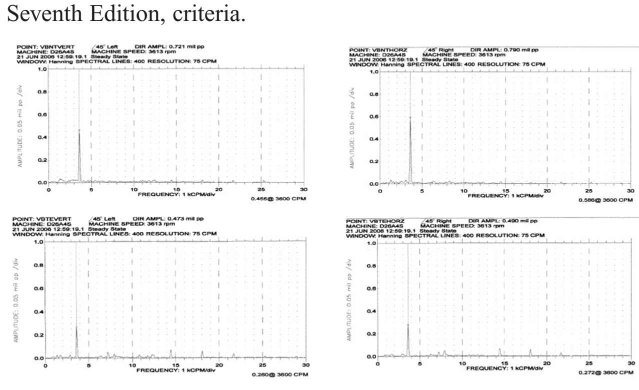

Figure 15. Compressor Vibration Spectra at Maximum Test Power (25,610 HP).

COMPARISON BETWEEN MECHANICAL TEST AND ANALYTICAL RESULTS

From a rotor synchronous response perspective, excellent correlation was found between test data and analytical predictions regarding the first critical speed location and amplification factor, as shown quantitatively in Table 3. Regarding rotor stability results, Figure 16 displays the measured logarithmic decrement values (log dec) during the MBE asynchronous sweeps with the log decrement values predicted by the analysis. Results are shown for two journal bearing configurations, offset pivot (55 percent offset) and centered pivot (i.e., 50 percent), confirming the higher log dec values predicted for the centered pivot journal bearing. For both bearing configurations, as discharge pressure and power levels increase, log dec values for both predictions and measurements also increase.

Table 3. Rotordynamics Synchronous Response: Prediction Versus Test.

Test Measure Description	Predictions[1]	Test	API 617 7th Edition Compliance
1st Critical Speed (RPM)	1,350 to 1,450	1,350 to 1,400	YES
Separation Margins (1st Critical Speed)	58% to 61%	59% to 61%	YES
Amplification Factors (1st Critical Speed)	5.1 to 14.5	6.3 to 12.8	YES

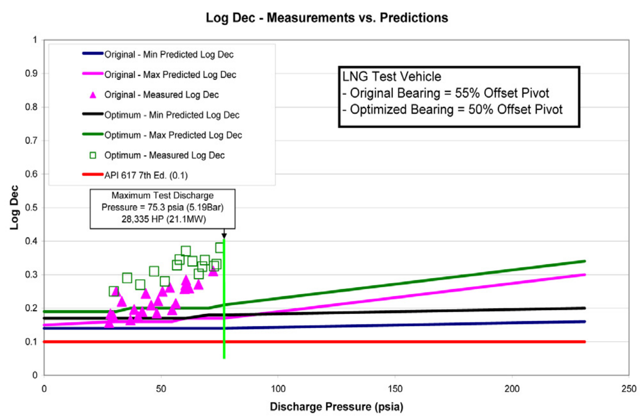

Figure 16. Compressor Log Dec Versus Discharge Pressure.

This correlation indicates that the rotor model is accurate. Furthermore, the upward trend provides both the OEM and the end user with a level of confidence that the system will be stable under full load in the field. Finally, based on the test data from this test vehicle, the analysis slightly underpredicts the log decrement and, therefore, provides conservative rotor stability predictions.

Finally, the large capacity of the MBE also was used to excite the second critical speed of the rotor-bearing system while running at 3600 rpm to confirm the separation margin with the maximum continuous speed and the amplification factor as shown in Figure 17. The measured second critical speed is approximately 5100 rpm (versus prediction of 5020 to 5170 rpm) with an amplification factor around 4 (versus prediction of 1.9 to 5.8). Acceptable separation margin to the second critical speed exists (40 percent), and the support structure provides ample stiffness to the system. Again, good correlation was observed between test data and analytical predictions, thereby concluding the demonstration of the sound rotordynamic characteristics of the test vehicle.

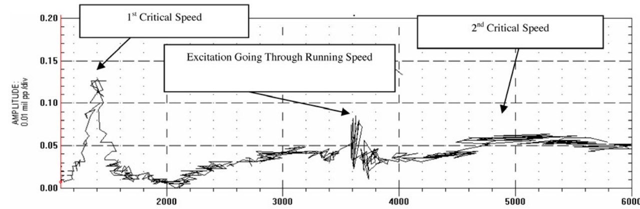

Figure 17. MBE Asynchronous Sweeps: Vibration Amplitude at Thrust-End Probe Versus Excitation Frequency (CPM).

Note that for brevity purposes, the rotor synchronous response plots and results are only shown for the 55 percent offset pivot journal bearing configuration.

All rotordynamic results showed compliance with API 617, Seventh Edition (2002), requirements—a major objective of this test.

IMPROVING DESIGN GUIDELINES

One of the real strengths of the new test vehicle lies in its ability to provide data that can be used to enhance design guidelines for high flow coefficient and high Mach number applications. As noted in the introduction, many of the guidelines applied in large-scale equipment were based on OEMs' and end users' experience. With the advent of larger plants that require higher flow coefficient and higher Mach number stages, many of these "tried and true" methods are proving to be inadequate. As mentioned, advanced analytical techniques have helped augment the lack of sound test data. However, these techniques must be calibrated against quality data. While data could be gathered from smaller scale models, this flexible test rig provides the opportunity to gather such data from a full-scale compressor.

As an example, it is commonly known that to achieve optimal performance in an impeller downstream from a mixing sidestream, the sidestream must provide relatively uniform velocity and pressure profiles to the impeller. Many factors influence the uniformity (or lack thereof) in the pressure and velocity field including:

- Upstream turning along the shroud profile in the mixing guidevane,
- Maintenance of proper area ratios between the sidestream passage and upstream return channel.
- The meridional aspect ratio of mixing guidevane (i.e., crudely, its axial length divided by its exit height).
- The vane in the guidevane.
- Additional flow turning vanes, etc.

As part of a recent test program, the OEM assessed the impact of different numbers of meridional turning vanes, as well as varying guidevane length on the guidevane exit pressure profile. A photograph of one of the arrangements is shown in Figure 18. The test program was instrumental in establishing new design guidelines to determine when such meridional vanes were required and the number of vanes needed to promote a uniform velocity and pressure field. The test data were also used to validate and calibrate the CFD methods used in the design of mixing guidevanes.

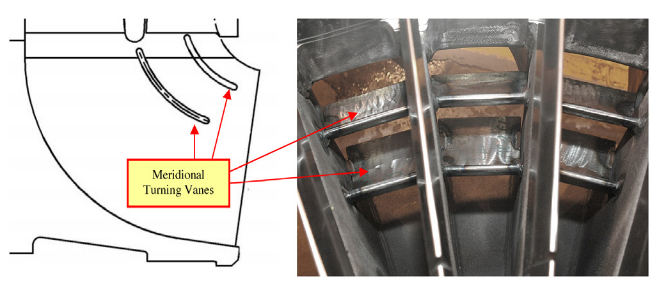

Figure 18. Guidevane with Meridional Turning Vanes.

CONCLUSIONS

This paper presented an overview of a new, full-scale test vehicle developed by the OEM for use in enhancing the technologies required for a variety of large-scale applications. Previous testing efforts during compressor development have focused on the use of small-scale test rigs or models. Data from such models or rigs have been very valuable in improving the state of the art. However, because these rigs do not include the actual upstream, downstream, and surrounding components, and because such rigs are of much smaller scale than the actual equipment, such testing cannot be relied on to provide supremely accurate results as compared to a full-scale test vehicle.

The new rig was designed to be very flexible both in its case construction and by the use of movable geometry for all primary flow path stationary components, i.e., guidevanes, diffuser vanes, and return channel vanes. By implementing such, significant time savings were possible when considering alternate vane-setting angles. In the past, such changes would require very time-consuming disassembly and reassembly of a compressor, as well as higher costs associated with building new guidevanes, diffusers, and/or return channels. The cycle time to obtain data on alternate vane setting angles was reduced from weeks to minutes. Sample data were offered to show the efficacy of the movable geometry.

In addition to the aerodynamic capabilities, this paper also described the features implemented in the rig to evaluate its rotordynamic characteristics. The value of the magnetic bearing exciter to assess the log decrement and to confirm the first and second natural frequencies was demonstrated. The data gathered from the midspan proximity probes also were used to validate the effectiveness of the damper seal with increasing discharge pressure. The results obtained confirm the accuracy of the OEM's rotordynamic modeling techniques as well as the effectiveness its rotordynamic technology.

Finally, the paper cited the ability to use the rig as a research tool to investigate or enhance the design methodologies applied for high flow coefficient and high Mach number applications. By testing or validating these concepts in a full-scale test vehicle, it is possible to avoid unpleasant surprises on the production test stand. The savings in test and rework costs as well as cycle time are significant.

In closing, the flexible test vehicle offers a tremendous opportunity to develop, investigate, and enhance the technologies necessary for today's and tomorrow's large-scale plants. Through effective use of the rig, the mechanical and aerodynamic performance of a new compressor application can be optimized, providing substantial benefits to the OEM and end user alike. Most importantly, all this can be accomplished *prior* to order placement, thus significantly reducing risk to the project schedule and enabling shorter cycle times.

NOMENCLATURE

A0	= Sonic velocity of gas in ft/sec or m/sec
MBE	= Magnetic bearing exciter
Mrel1t	= Inlet relative Mach number at the shroud leading edge
OEM	= Original equipment manufacturer
U2	= Impeller tip speed in ft/sec or m/sec

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