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A COMPARISON OF TYPE 1 VERSUS TYPE 2 TESTING – RECENT EXPERIENCES TESTING A HIGH PRESSURE, RE-INJECTION CENTRIFUGAL COMPRESSOR

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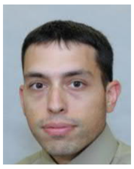
Jim is a member of ASME and the ASME Turbomachinery Committee. He has authored or co-authored more than 50 technical papers and has instructed seminars and tutorials at Texas A&M and Dresser-Rand. He currently holds several U.S. patents and has other patents pending. He was elected an ASME Fellow in 2008 and was also selected as a Dresser-Rand Engineering Fellow in 2015.

John A. Kocur, Jr. is a Machinery Engineer in the Equipment Engineering Division at ExxonMobil Research & Engineering in The Woodlands, TX. In his current capacity, he provides support to the downstream, upstream and chemical business lines within ExxonMobil with expertise on vibrations, rotor/thermo dynamics, failure analysis and health monitoring of rotating equipment. Prior to joining EMRE, he held the position of Manager of Product Engineering and Testing at Siemens Demag Delaval Turbomachinery. There, Dr. Kocur directed the development, research, design, engineering and testing of the compressor and steam turbine product lines. He has also held positions with Pratt & Whitney and Amoco Corporation.

Dr. Kocur received his BSME (1978), MSME (1982) and Ph.D. (1991) from the University of Virginia and an MBA (1981) from Tulane University. He has sat as a committee chairman for NASA Lewis, is a member of ASME and holds a patent on hydrostatic bearing design. Currently, he holds positions within API as 617 chair, 684 chair, Standard Paragraph chair and Subcommittee on Mechanical Equipment (SOME) steering committee member and serves on the advisory committee for the Texas A&M Turbomachinery Symposium.

Gary M. Colby started his career at Dresser-Rand in 1973 as a Performance Computation Technician after two years at Alfred State College in NY, studying mechanical technology. Over his 40 years of experience in centrifugal compressor performance, Gary split his career between the prediction of thermodynamic performance and evaluation of centrifugal compressor performance. Most of his work has been in developing test methods to improve the similarity between in-shop testing and specified performance. Gary retired from Dresser-Rand, A Siemens Business in 2016.

Gary has published several technical papers on full load testing methods, both inert gas and hydrocarbon gas ASME PTC Type 1 test. Mr. Colby has presented Tutorials, been a Short Course Speaker and Discussion Group leader on the topic of centrifugal compressor performance and testing for the Texas A&M Turbomachinery Symposium.



Nathan Keim was a Test Engineering Supervisor with Siemens GP O&G, in Olean, New York until August 2019. He was responsible for developing test methods to meet objectives for production testing and analytical aerodynamic performance analysis of centrifugal compressors. He has 10 years of engineering experience, all related to compressor operation and testing per industry guidelines.

Mr. Keim received a A.S. degree (Electro-Mechanical Engineering Tech, 2006) from Alfred State College, a B.S. degree (Electro-Mechanical Engineering Tech, 2010) from Rochester Institute of Technology, and is pursuing a Masters in Manufacturing Leadership from RIT. He is also a member of ASME.

ABSTRACT

End users and Original Equipment Manufacturers (OEMs) are faced with decisions regarding that type of performance testing they will require to demonstrate that centrifugal compressors will meet the agreed-upon requirements. This lecture paper details lessons learned by an end user and an OEM during Type 1 and Type 2 testing of a high pressure (>8,000psia, 550bar), natural gas, re-injection compressor. The objective is to give readers insight into some considerations they might overlook when choosing between Type 1 and Type 2 testing.

INTRODUCTION

To test or not to test? That was, and is, the question. Prior to the 1980s, OEMs built and shipped compressors without subjecting many of them to any type of aerodynamic testing. Most turbomachines were mechanically tested but the costs, complexities associated with and the lack of demand by clients, caused in-house aero-performance tests to be somewhat rare. However, as competition between OEMs escalated, revenue loss due to production shortfalls increased and end users began installing equipment in more remote and/or limited access locations; i.e., an oil platform in the North Sea, it became increasingly more important to confirm the aerodynamics of the compression systems before shipping to site.

By the mid-1980s, the number of aerodynamic and mechanical shop tests was nearly equal. These performance tests typically involved running the compressor at aerodynamically equivalent conditions to match the volume reduction and Mach number through the compressor. As the demand increased for higher and higher discharge pressures and greater power densities in smaller and fewer compressors, faults were detected in the field not identified by the mechanical or "equivalent" performance tests being performed in the OEM's shop. These high pressure/high power density compressors were experiencing rotordynamic (instability) and aerodynamic (stall) complications more repeatedly than those intended for petrochemical service. Thus, demand grew for more stringent and/or demanding in-house testing. End users understood the critical nature of each component in the compression train (including any auxiliary equipment) and needed assurances that everything was operating as expected BEFORE it left the OEM's facilities. Their desire was to perform a test at the OEM's facility that would, as closely as possible, replicate the operating conditions the equipment would experience in the field.

In response, OEMs and the turbomachinery community as a whole developed a wide range of test programs and facilities used to demonstrate that the equipment meets the end users' requirements. These options ranged from volume reduction, inert gas tests conducted using small open or closed loops (typically called Type 2[1] testing) to full load, full pressure, full power testing (typically labeled Type 1 testing) with the contract drivers, gears, auxiliaries, complex piping systems, etc. Each type of test offers different levels of insight into the compression system's aerodynamic and mechanical behavior.

[1] PTC 10, "Performance Test Code on Compressors and Exhausters", The American Society of Mechanical Engineers, ASME International, 1997.

The end user selects the type or types of testing they require on their compression equipment to mitigate risks and ensure project success. These decisions weigh the cost, schedule time, and delivery ramifications versus the benefits obtained from demonstrating acceptable operation under the different test conditions.

This lecture paper details lessons learned by an end user and an OEM during Type 1 and Type 2 testing of a high pressure (>8,000psia, 550bar), natural gas, re-injection compressor. It is hoped this paper will give readers insight into some considerations they might overlook when choosing between Type 1 and Type 2 testing.

OVERVIEW OF TYPE 2 AND TYPE 1 TESTING

Type 2 testing is the more frequent approach chosen by end users and used by OEMs, so Type 2 will be addressed first. It could also be noted that a Type 1 test is nothing more than a highly specialized Type 2 test. As an aside, nearly all research or stage validation testing done by OEMs also adhere to Type 2 guidelines and procedures.

Type 2 Testing

Type 2 low pressure inert gas performance testing is based upon the similitude between the field conditions volume reduction and volume reduction at the shop test conditions. Such tests are typically conducted using readily available gases such as nitrogen, helium-nitrogen mixtures, carbon dioxide, and R-134A refrigerants; the latter typically being used for high Mach Number, high volume reduction applications.

The selection of test gas or gases, operating speed, etc. are made to, as closely as possible, match the volume reduction the compressor will experience in the field. Therefore, it is important to understand what is meant by volume reduction and the parameters that affect it. Volume reduction is a critical parameter when assessing the performance of a multi-stage compressor because it has a cascading effect through the machine. That is, the volume reduction of stage 1 will determine the volumetric flow into stage 2. Stage two's volume reduction sets the flow into stage 3, etc. The combined volume reductions of the individual stages then determine the overall volume reduction of the compressor.

Volume Reduction

Several factors influence a compressor stage's volume reduction. These include the polytropic head, the efficiency, the gas density, and the gas properties. In each stage, the impeller increases both the static and total pressure. The stationary components further increase the static pressure. The total pressure drops somewhat downstream of the impeller exit due to losses in the stationary components but is still much higher than at the impeller inlet. The increase in pressure results in reduction in the volumetric flow. The volume reduction of the stage is then the ratio of discharge volume to inlet volume.

Equations 1 and 2 below provide the relationship between head and pressure ratio.

$$Head = Z_1 T_1 R \left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{n-1/n} - 1\right] \qquad (1)$$

$$\left(\frac{n-1}{n}\right) = \left(\frac{k-1}{k\eta}\right) \text{ for an ideal gas} \qquad (2)$$

Where: R = 1545 / Molecular weight in U.S. customary units

As pressure ratio increases, the head increases. The reverse is also true. An impeller or stage designed to achieve higher head will provide higher pressure ratio than will a lower head design. Therefore, it logically follows that a higher head stage will provide higher volume reduction than a low head stage. One will also note that if head is to be held constant and the molecular weight is reduced, the pressure ratio must also be reduced, leading to a reduction in volume reduction. Or stating this a different way, as the molecular weight of the gas decreases for a given set of inlet conditions, the volume reduction reduces.

The impact of other factors on the volume reduction are provided in Table 1 below.

Table 1. Effect on Volume Reduction for Changes in Operating Parameters.

Variable	If Variable	Volume Reduction
Head	Increases Decreases	Increases Decreases
Mole Weight	Increases Decreases	Increases Decreases
Inlet Temperature	Increases Decreases	Decreases Increases
Inlet Compressibility	Increases Decreases	Decreases Increases
Isentropic Exponent (k)	Increases Decreases	Decreases Increases
Inlet Pressure	Increases Decreases	No Effect

Note that the change in inlet pressure has no effect on the volume reduction of the stage, though changes in pressure can impact the gas isentropic exponent (k) and the compressibility factor (z).

The impact of volume reduction is illustrated in Figure 1 below. The three curves are the individual stage performance maps for a three-stage compressor. At the design condition, represented by the solid blue vertical line, each stage operates near the center of their performance map (which typically coincides with the best efficiency point). If the molecular weight of the gas is increased, the volume reduction of the stages will increase (as noted in Table 1). The first stage will continue to operate at the same location but due to the higher volume reduction, the second stage operating condition moves to a lower flow rate (see the dashed black line). The third stage will move even further to the left because of the combined increase in volume reduction of the first two stages (see the dashed black line on the 3rd curve).

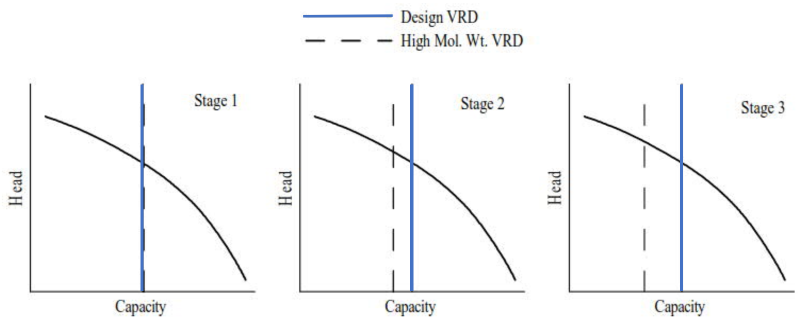


Figure 1. Stage Curve Example for a Three-Stage Compressor.

If we now assume that the lighter molecular weight gas would be used in the field and the heavier mole weight gas was being used for testing, the volume reduction of the compressor on test must be reduced to design level to yield the proper relative performance to the design condition. Assuming no other test gas is available to better match the field volume reduction, the reduced volume reduction must be achieved by lowering the head of the compressor relative to design. Therefore, the overall head level during the shop test will be lower than the head level at the field conditions and the test versus design polytropic head must be compared using the polytropic head coefficient.

The equation for polytropic head was given in Equation 1. The polytropic head coefficient, which is a function of the geometry of the impeller, the associated stationary components, the capacity being passed, the speed, and the inlet Mach number, can also be determined using Equation 3 below:

$$\mu = \frac{Head}{{U_2}^2} \qquad (3)$$

Where U2 = impeller tip speed in ft/sec or meters/second

Since the geometry of the impellers, stationary components, etc. cannot be changed and because it is assumed no more appropriate test gas can be found, the head must be reduced by decreasing the compressor operating speed.

Stage head coefficient and efficiency also vary with machine Mach number, U_2/A_0, which is the impeller tip speed divided by the inlet sonic velocity of the gas. The tip speed is a function of the impeller diameter and rotational speed while the sonic velocity is dependent on the k-value, gas constant, temperature and gravitational constant.

$$M_{U2} = {}^{U_2}/_{A_0} = \frac{U_2}{\sqrt{kgzRT}} \qquad (4)$$

The variation with U_2/A_0 is illustrated in Figure X which provides a compressor stage performance map for a fixed impeller geometry operating at different Machine Mach Numbers. The differences in the head coefficient and efficiency levels at the varying Mach numbers. Therefore, it is critically important to conduct the performance tests at Mach Numbers close to the design Mach Numbers for each stage in the compressor. The ASME PTC-10 (1997) code provides guidance on the allowable deviations between the design and test Mach Number values. This is found in Figure 3.3 of the Code, which is also provided in Figure X below. Note that the allowable deviation becomes smaller as the Mach number increases. Also note that the Code only addresses the first stage machine Mach number as the remaining stages should have similar deviations if the proper volume reduction is achieved.

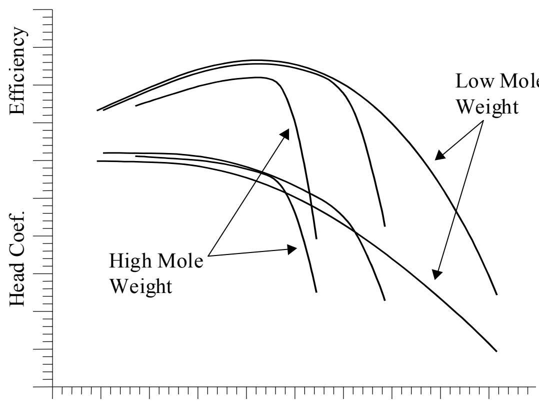


Figure 2. Typical Stage Curve at Three Mach Numbers.

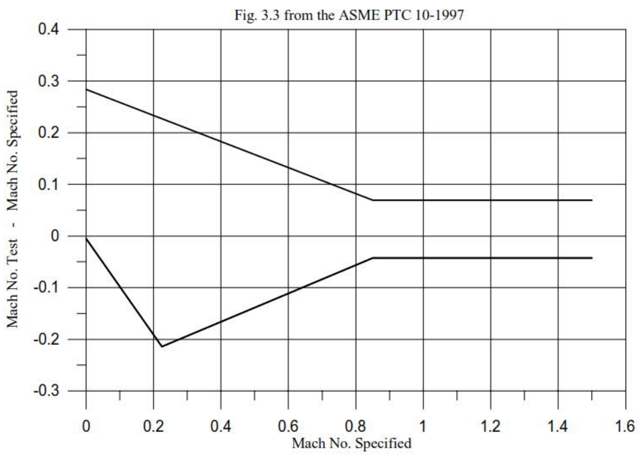


Figure 3. Allowable Test Mach Number Departure from Design.
(Courtesy ASME)

Determining Test Conditions

In setting the performance test conditions, the OEM can typically vary inlet temperature, inlet pressure, compressor operating speed and the test gas. The inlet temperature will be a function of the cooling capacity available on the OEM's facilities. The variation in volume reduction caused by the difference between test and field inlet temperature must be corrected via the test speed. As noted earlier, inlet pressure has no effect on volume reduction except for its impact on gas properties. However, inlet pressure does have a significant effect as it directly influences the amount of mass compressed and the mass flow determines the power requirement. Inlet pressure also impacts the test Reynolds number.

For those interested in following the steps used in specifying a Type 2 test, a sample compressor application is presented in Colby (2005 - pages 149 through 151).

There are inherent errors in Type 2 testing because of assumptions made when calculating the test conditions. The speed calculation assumes a straight line polytropic path between inlet and discharge points on the pressure-enthalpy diagram when, in fact, the path is slightly curved. As noted, there are also allowable differences in the Mach numbers between the test and field condition and it is possible the deviations are somewhat higher in latter stages of a multi-stage machine. The compressibility factor (z) change is also greater in field conditions (typically higher pressure) than in the lower pressure test conditions. These all cause slight variations in the individual stage volume reduction between test and field conditions.

To evaluate the overall impact, the compressor performance at the test condition is calculated on a stage by stage basis. The results are converted to head coefficient (μ), efficiency, and flow coefficient. This test performance curve is plotted against the predicted curve at field conditions to determine if the test conditions are adequate. The end user should always request a copy of this set of comparative curves to ensure the test meets their objectives.

Type 1 Performance Testing

A Type 1 test per the ASME code is conducted with a gas that is as close as possible to the gas that will be compressed in the field. The testing will also be done at or very near the field operating conditions. Deviations from the specified gas conditions are subject to limitations given in Tables 3.1 and 3.2 of the Code. As a result, the Type 1 test results will accurately reflect the field performance levels and one should expect little or no difference between the performance measured during Type 1 test and the field performance.

In setting up the test, the same principles apply to the Type 1 test as to the Type 2 test described earlier. If it is possible to conduct the test using the actual field gas and field inlet conditions, then clearly there would be no deviations between the field and test conditions. However, this is very rare. In most instances the inlet temperature at the specified condition cannot be achieved due to limits in the cooling capacity of the OEMs test stand. It might also be impossible to exactly match the field gas. The test can be conducted on a different gas, but the test gas must have a k-value close to the field gas to ensure that the test and field Mach numbers are equal. This also ensures that the thermodynamic conversion of the work input to the gas produces the same pressure ratio.

Most OEMs have a local gas supply that is close to pure methane. Blending this gas with carbon dioxide, propane, or other gases can be done to achieve the specified molecular weight and k-value. If the test inlet temperature is greater than the specified inlet temperature, the test gas molecular weight must be higher than specified to offset the temperature change and any associated change in the compressibility factor. The higher mole weight mixture must still have a mixture k value close to that of the specified gas. The gas constant, R (1545/mole weight), inlet temperature, and inlet compressibility factor into the calculation of volume reduction and machine Mach number as shown by Equations 1 and 4. If a blended gas has the same k value and the product of compressibility times gas constant times inlet temperature; i.e., zRT; is maintained at the inlet, the performance will be the same as on the specified gas. While the ASME PTC-10(1997) (Table 3.2) allows a much greater tolerance, experience has shown that the inlet zRT product and inlet pressure should be maintained within ±2 percent of the specified value.

As with the Type 2 testing, comparable plots of compressor performance maps showing polytropic head, efficiency, pressure ratio, and power can be produced for the specified gas and the planned test gas. These plots, again, will demonstrate how the compressor performance operating on the test gas blend and conditions compares to the compressor performance under the specified gas conditions. These curves should be provided for review before beginning the test.

The number of points and their position on the overall performance map must be agreed upon by the end user and the OEM before beginning a Type 1 test and must be dependent upon the objective of the test. Experience has shown that this objective is not always well-defined during the proposal stages of a project. It is strongly recommended that such discussions occur as early as possible in the project. This will ensure that the test stand loop design, test conditions and data acquisition system will satisfy all of the objectives.

Motivations for doing a Type 1 Test

One of the primary drivers for requesting Type 1 testing is the desire to avoid costly time-consuming trouble-shooting efforts in remote or difficult to access locations and the associated loss of production. For example, if the equipment is to be installed on an oil platform in the North Sea or other similar, limited access location or if the geopolitical circumstances make travel to the site difficult or even life-threatening, there would be a very strong desire to "get all of the bugs out" before the compressors are shipped from the OEM. Further, the production loss and delay associated with making changes to equipment in the field are an order of magnitude larger than the cost to make similar changes at the OEM facility. Therefore, users must weigh the cost of the Type 1 tests against the potential production loss arising from the risks faced. Because the Type 1 test exposes the compressors and its auxiliaries to near field conditions, there is greater likelihood that any abnormalities in the performance (mechanical or aerodynamic) will be uncovered during Type 1 testing than during a Type 2 test.

The "real world" factors that impacted the aero and mechanical behavior during the Type 1 and Type 2 testing

There are numerous factors that could impact the aerodynamic and mechanical behavior of turbomachinery between a Type 1 and Type 2 test. Nearly all of these are tied either directly or indirectly to the higher pressure and aerodynamic loading experienced on the Type 1 test. Primary among these are the deflections or relative movement of parts caused by these factors. As a result, there can be subtle differences in the primary and secondary flow path geometries under Type 1 and Type 2 conditions.

For clarity, in this paper, a compressor's primary flow path consists of the flow passages where the primary process flow passes. This would include the inlet nozzle, inlet plenum, inlet guides, impellers, diffusers, return bends, return channels, sidestreams, volutes/collectors and discharge nozzles (see Figure 4). The secondary flow path is comprised of any passages where the gas might travel that is not part of the primary flow path; such as recesses around impeller, the horizontal split, gas seal passages, passages around the balance piston, the gap between any inner bundle and the casing, etc. (see Figure 5).

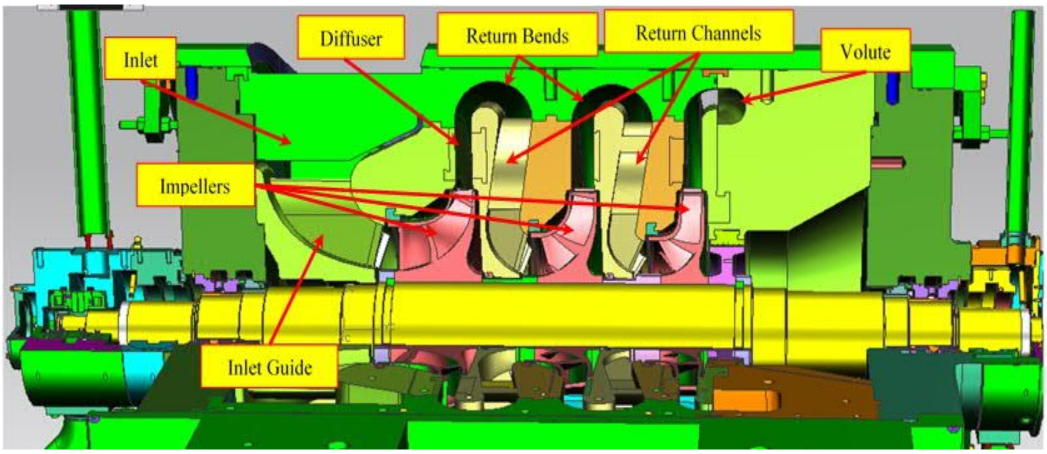


Figure 4 – Cross-Section Showing Primary Flow Path Components

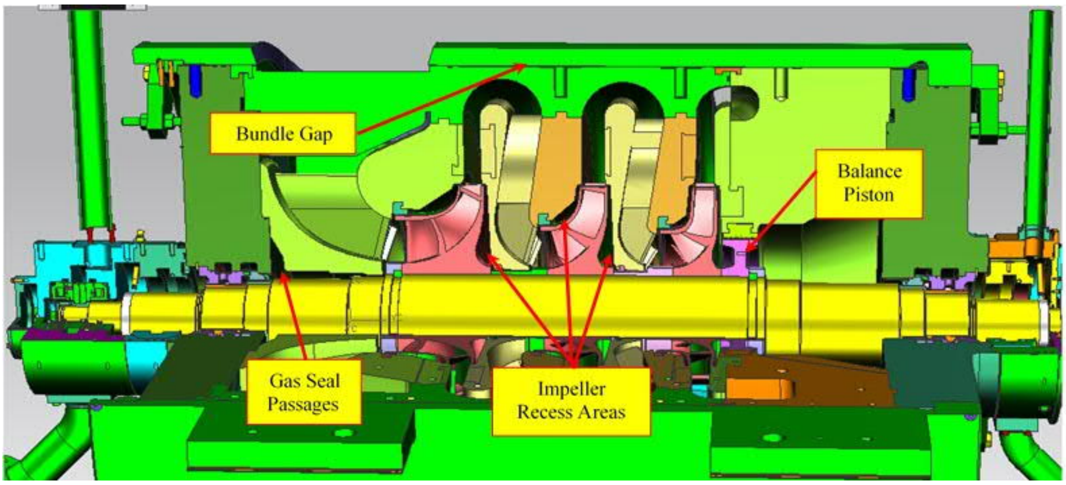


Figure 5 – Cross-Section Showing Secondary Flow Paths

One of the more common geometric changes that can occur during a high-pressure test is bundle deflection due to tolerance stack-up. In a typical centrifugal compressor, the highest pressure occurs in the last stage diffusers. This is true for either a straight-through or back-to-back arrangement. As a result, the forces on the sidewalls of the last stage diffuser will cause the diffuser walls to move apart as illustrated in Figure 6. Because the last stage diffuser is also typically the narrowest in a compressor, its behavior is the most sensitive to such axial deflections.

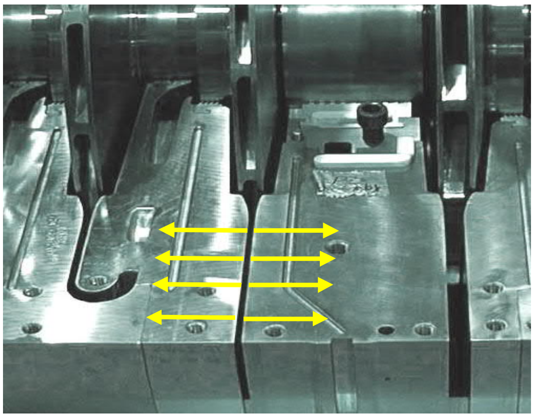


Figure 6 – Last Stage Diffuser Deflection at High Pressure

The amount of movement or deflection will depend on the machining tolerances on the bundle or stage components and fits as well as any material deflections that might occur due to the pressure forces. In most circumstances, assuming parts were machined and assembled correctly, maximum deflections will be on the order of tens of thousandths of inches (or tenths of millimeters). Such deflections might be of little consequence in most compressors. However, if the diffuser passages become very narrow; as is often the case with high-pressure, reinjection compressors; tens of thousandths of inches can be a large percentage of the diffuser width. For example, if the diffuser width required for a stage is 0.070” (1.78mm) and the deflection due to tolerance stack-up is 0.020” (0.51mm), the diffuser width will increase nearly 30% during the Type 1 testing. Therefore, it is possible for these deflections to cause the onset of diffuser rotating stall.

Much has been written on the subject of diffuser rotating stall, so no detailed discussion will be provided herein. Those seeking more information on the subject can review the Turbomachinery Symposium tutorial published by Sorokes *et al* (2018) or the works of Frigne *et al*, Jansen, Kammer *et al*, Senoo *et al*, etc. that are listed in the reference section of this paper. Nearly all these published works point to a critical flow angle in the diffuser and state that if the diffuser flow angle becomes more tangential than this critical angle, diffuser rotating stall will occur. The diffuser flow angle is a function of the diffuser width. An increase in the diffuser width causes a decrease in the radial velocity component and the diffuser flow angle to become more tangential (see Figure 7). If the increase in width is sufficiently large, it can cause the diffuser flow angle to move from an acceptable to an unacceptable level. Bringing this to its logical conclusion, if the deflections do not occur during the lower pressure Type 2 test but do occur during the high-pressure type 1 test, it is possible that the compressor will experience diffuser rotating stall during the Type 1 but not during the Type 2 testing.

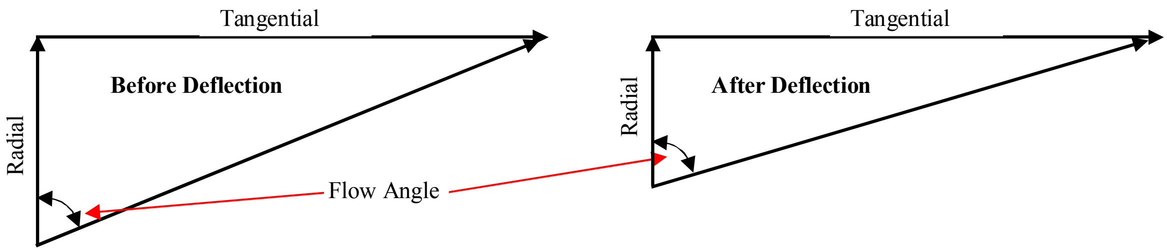


Figure 7 – Change in Diffuser Flow Angle due to Bundle Deflection

While on the subject of rotating stall, another situation can lead to differences in the onset of a form of rotating stall between a Type 1 and Type 2 testing. In this case, the cause is the misalignment of the impeller and diffuser flow passages. As noted in Sorokes et al (2018), a misalignment of the impeller exit opening and the diffuser entrance opening (see Figure 8) can cause subsynchronous radial vibrations in the subsynchronous range. The most common cause for this misalignment is the improper installation of the compressor rotor during assembly. However, rotor axial movement during operation in combination with deflections of stationary component deflection (i.e., diffusers) due to pressure and thermal effects can cause components to move from properly to improperly aligned. Again, because the forces within the compressor under Type 2 conditions are lower than during Type 1 testing, the misalignment might not occur during Type 2 but could occur during Type 1 tests.

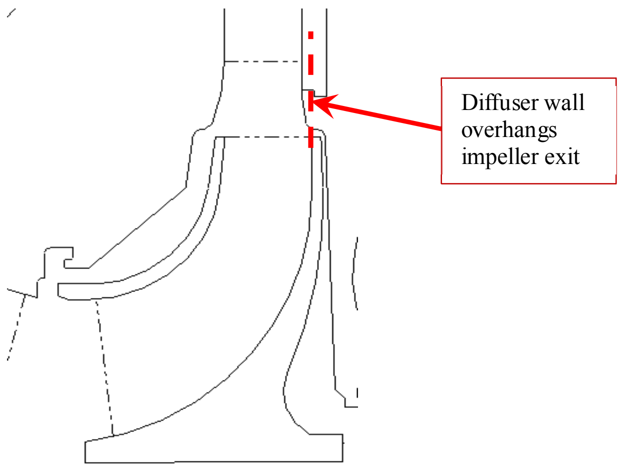


Figure 8 – Impeller – Diffuser Misalignment

The above situations could be further exacerbated if proper consideration is not given to possible variations in the thermal expansion of materials in the compressor assembly. If parts within the assembly grow at different rates at elevated temperatures, the variation in thermal growth could lead to rotor/stator misalignment or other misalignments in the compressor flow passages. Again, these could result in difference in the aerodynamic or mechanical behavior of the compressor between Type 1 and Type 2 testing.

It is also possible that the phenomena described above do occur during the Type 2 testing but that the aero-mechanical forces associated with the phenomena are significantly smaller because of the lower gas densities and/or pressures during such testing.

Returning to rotating stall, the so-called rotating stall “cells” are actually pockets of non-uniformity in the circumferential pressure field in the diffuser or impeller. The magnitudes of these non-uniformities tend to be a percentage of the pressure level of the core circumferential pressure field. For example, if the core pressure field is at “X” psia, the magnitude of a stall-related non-uniformity might be in the range of 1% - 5% of "X" (or greater). [NOTE: The percentages are offered for illustration purposes. The percentage will vary depending on the number of stall "cells" and other factors. However, OEM author has observed pressure non-uniformities in this range during stall phenomena.] Now consider a Type 1 test operating at 6000 psia discharge pressure versus a Type 1 test operating at a discharge pressure of 600 psia. Assuming the pressure non-uniformity is 3% of the core pressure, the magnitude would be an order of magnitude higher on the Type 1; i.e., 180 psi in the Type 1 v. 18 psi on the Type 2. Therefore, it might be possible to overlook a response due to stall on the Type 2 test whereas the subsynchronous amplitudes might be more noticeable during Type 1 testing.

To illustrate this point, the data shown in Figures 9 through 11. The data shown in Figure 9 was gathered while establishing the minimum stable flow rate on the Type 2 testing. The OEM will establish the "surge" line based on several criteria; one being the apparent onset of stall. Note that the subsynchronous amplitude level at 23 Hz is below 0.1 mils under the Type 2 conditions. Also note that the running or synchronous speed is at 140Hz for the Type 2 testing. NOTE: The scale in this plot is expanded to accentuate the subsynchronous amplitudes. Under normal circumstances, the y-axis scale would be set to 1.0 mils during a Type 2 test and the subsynchronous "spikes" would likely be ignored as "hash" or noise in the signal.

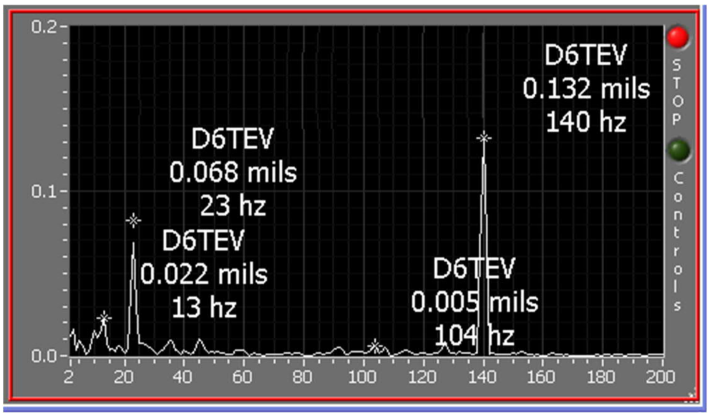


Figure 9 – Frequency Spectra at Onset of Stall During Type 2 Test

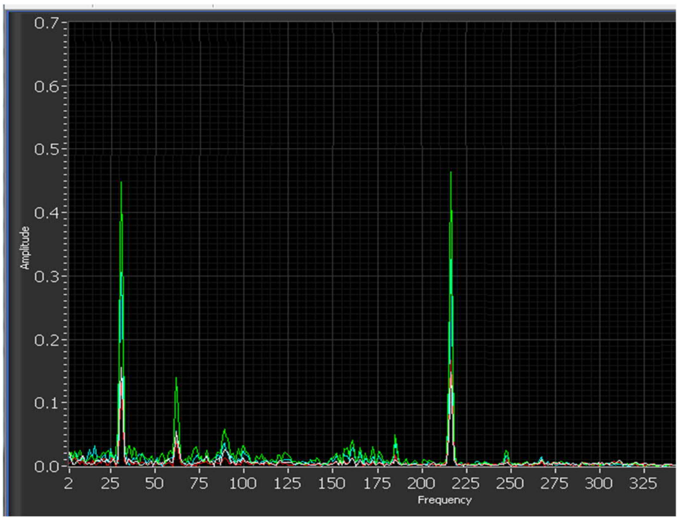


Figure 10 – Onset of Diffuser Stall During Type 1 Test

The vibration amplitudes at the onset of diffuser stall during the Type 1 testing is notably higher, as can be seen in Figure 10. Under Type 1 conditions, the fundamental subsynchronous frequency is at 31 Hz and has an amplitude in excess of 0.4 mils as compared to less than 0.1 mils under the Type 2 conditions. This was clearly visible to those witnessing the testing. Note also that the running speed on Type 1 is 216 Hz v. 140 Hz during the type 2 test. The Type 2 test was run using nitrogen while the Type 1 used a lighter molecular weight gas mixture whose properties more closely matched those of the field gas. Therefore, the Type 2 test was run at a lower speed to properly match the volume reduction. One might note that the shift in subsynchronous frequency from the Type 2 to Type 1 is not the same as the speed shift. This is due to a combination of the diffuser width changes due to bundle deflection and the possible Reynolds effects due to the change in gas properties and operating conditions (i.e., pressures and temperatures).

The compressor also exhibited the response shown in Figure 11 prior to the onset of stall. This response was attributed to a misalignment that occurred between one or more of the impellers and the diffuser openings due to bundle deflections that occurred during the Type 1 test that did not occur at the lower Type 2 pressures. No similar response was observed during the Type 2 testing.

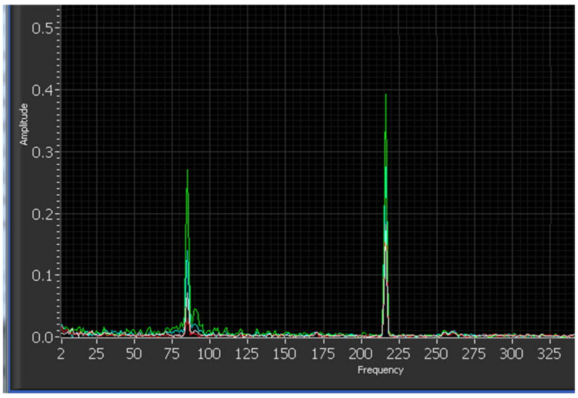


Figure 11 – Frequency Spectra Associated with Impeller – Diffuser Misalignment

The compressor being tested was unacceptable because of the vibration amplitudes caused by the diffuser stall and misalignment issues. Further, were the surge control line moved to avoid entering the portion of the curve where the subsynchronous vibrations occurred, the compressor would have had insufficient range. Therefore, modifications were made to eliminate the overlap and premature stall issues. After these changes were made, the Type 1 test was re-run and the unit was accepted.

The compressor in question is a good example of how issues might be discovered during a Type 1 test that were not observed during Type 2 conditions.

Another potential source for differences in aero-mechanical forces is the discharge volute or collector. The volute or collector gather the flow exiting the last stage diffuser and guide it to the compressor discharge nozzle. As the flow is captured around the circumference of the compressor, it eventually reaches a point where it either stays in the volute / collector or exits into the discharge nozzle, Figure 12. The dividing feature is often called the volute/collector "tongue" (or "cut-water"). It is well-established that the "tongue" creates a disturbance in the circumferential pressure field in the last stage diffuser (Borer *et al*, Flathers, Hagelstein *et al,* Sorokes *et al*) and that this pressure non-uniformity can have an impact on the mechanical and aerodynamic performance of the compressor. It can manifest itself aerodynamically as a reduction on efficiency and/or operating range. The circumferentially integrated pressure field also yields an unbalanced radial force that causes additional loads on the radial bearings. Failure to account for this latter effect can result in unacceptable performance of the rotor system.

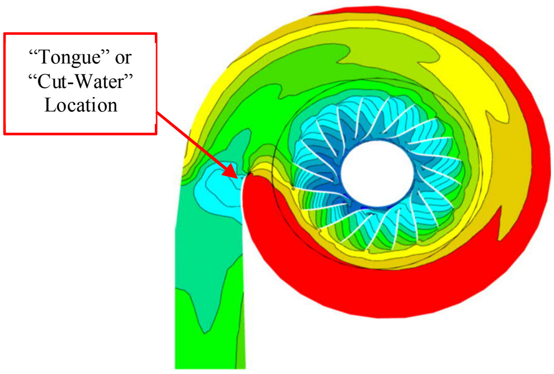


Figure 12 – CFD Simulation of Volute-Induced Circumferential Non-Uniformity

Like the rotating stall example described above, the magnitude of the volute-induced non-uniformity is a function of the pressure in the discharge volute or collector. Unlike the rotating stall example, the volute-induced force is stationary for any fixed operating condition. Regardless, one should expect much higher radial forces during a Type 1 than during a Type 2 test. It is possible that the forces are insufficient to cause any rotordynamic issues during Type 2 but could cause load problems for the bearings and other mechanical components during a Type 1 test of the same compressor.

One other consideration related to the differences in forces between the two types of testing, the aero-mechanical forces due to interactions of rotating and stationary components will increase with increased operating pressure or gas density. This is especially true for the interaction of impellers with vaned diffuser and of greater concern when the diffuser vanes are in close proximity to the exit of the impeller. At off-design conditions, pressure field between the impeller blades and diffuser vanes becomes more disturbed due to incidence effects on the diffuser vanes. As a result, the amplitude of the forces acting on the impeller blades increase. Should said higher forces align with an impeller natural frequency, it could put the mechanical integrity of the impeller at risk during the Type 1 testing.

Other Mechanical Considerations

As noted, Type 1 testing exposes the internal components to the deflections arising from pressure and temperature gradients at operating conditions. This distortion or expansion of diffusers can lead to stall even with low solidity diffuser vanes installed especially in latter stages where the designed diffuser width may be less than 0.15” inches. With low solidity vanes, care must be taken in the design to prevent diffuser widening from permitting the flow to bypass the vanes, Figure 13.

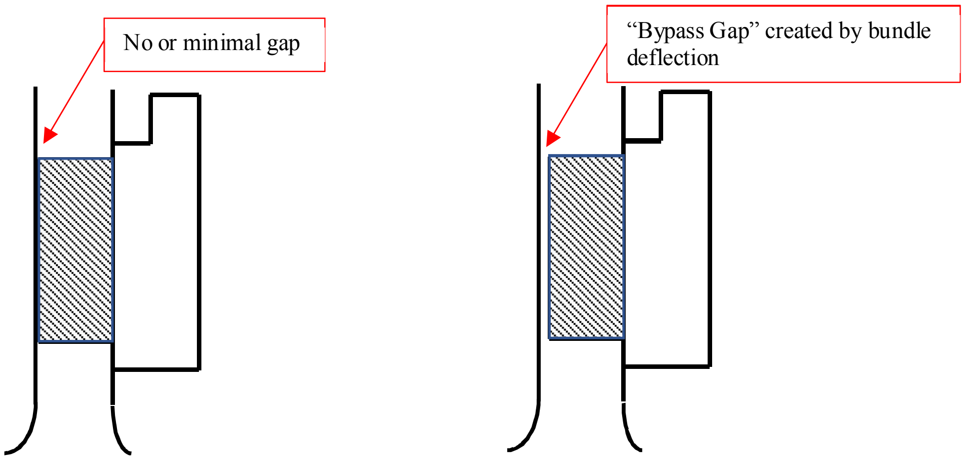


Figure 13 – Low Solidity Diffuser Vanes – Shroud-Side Gap With & Without Deflection

While we focus on diffuser passages deflections magnitudes approaching 0.1”, there are areas within the compressor where thousands of an inch are critical. One such location is the center seal or balance piston. These components are critical in determining the rotordynamic behavior (discussed later), thrust forces and internal flow in the compressor. It is easy to understand or appreciate the impact of opening the clearance of the center or balance piston seal on the leakage or thrust balance. In back-to-back injection compressors, such as the one referenced in this paper, the center seal leakage can equal 10% of the design flow. For straight thru designs, the balance piston leakage can exceed 20% of the design flow. With clearance ratios of 1.5 to 2.0‰, deflections can alter the clearance by half. In process compressors where the seal leakage is roughly 1.5% of design flow, increasing the seal leakage by 50% will not present a significant impact on compressor efficiency. In high pressure injection compressors, this will decrease efficiency by 5-10% due to the increased flow in the compressor.

Somewhat counter-intuitive is that decreasing the center or balance piston seal clearance can also create problems. Obviously, efficiency increases are to be expected with a reduction in seal clearance as is the higher potential for rubbing due to rotor sag, vibrations and rotor position within the stator. In the absence of a Type 1 test (or even surge testing during a Type 1 test), unexpected or early surging of the compressor is also possible. Compressor operators monitor the suction flow to determine product delivery (ignoring for now the recycle flow.) This does not include the center or balance piston leakage that is also passing through specific process sections in the compressor. Type 2 testing determines the actual surge or minimum stable flow location relative to the internal flow due to the minimization of the center seal/balance piston flow. The predicted seal leakage flows at operating conditions are then added to determine the surge onset flow as determined by the suction flange flow monitored in the field. This assumes there are no changes in the leakage rates between the Type 2 test and field conditions other than those due to pressure or gas characteristics. However, if deflections occur that cause a change in the seal clearance between the Type 2 and field conditions, the leakage rates will be impacted. This exact phenomenon occurred during the Type 1 test of the injection compressor. Center seal clearances were reduced during the Type 1 test decreasing the leakage rates by 50%.

Let's examine the impact of reducing the seal leakage by 50% by looking at the "apparent" performance map as determined by the Type 2 testing, Figure 14. For the injection, back-to-back compressor tested, the center or division wall seal leakage was approximately 10% of the rated mass flow. This leakage rate is roughly 20% of the flow at predicted surge. With the reduction in seal leakage, the "apparent" onset of surge has moved by 10%. It now appears (as measured by the suction flange flow) that surge is occurring at the surge control line (SCL) as determined by the Type 2 test. In the absence of a Type 1 test, this would have only been experienced during operation. With surge protection based on the suction flange flow including a predicted center seal leakage, the compressor would experience surging in the field if operations at the SCL was needed. With the reluctance to surge test these compressors at operating conditions in the field, this can create unexpected problems by restricting operation or requiring additional shutdowns to reset the surge control line. Type 1 testing can identify this risk if the compressor operation is varied from stonewall to the Type 2 determined surge control line relative to the actual suction flange flow.

Given the reluctance to surge test these compressors at operating conditions, this can create unexpected problems by restricting operation or requiring additional shutdowns to reset the surge control line in the field. Type 1 testing can identify this risk if the compressor operation is varied from stonewall to the Type 2 determined surge control line.

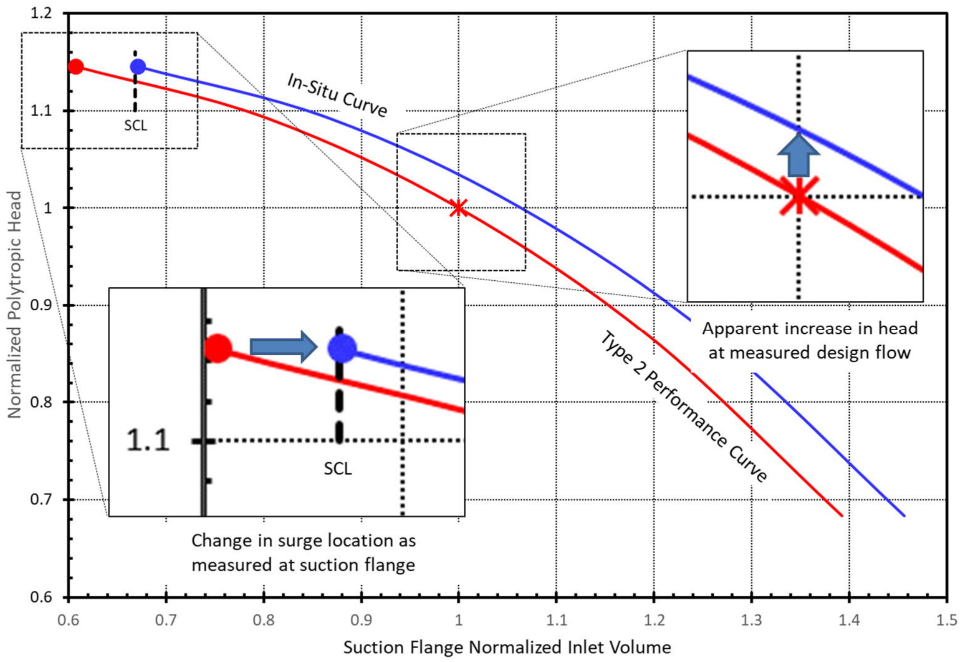


Figure 14 Apparent Performance Map Change Due to Center Seal Leakage Reduction

Another mechanical consideration highlighting the differences between Type 1 and Type 2 testing is the load the thrust bearing is required to support. During Type 2 testing, minimal thrust loads are developed that may represent a small fraction of the load experienced during operation. Additionally, since the thrust load represents only a small portion of the aerodynamic axial force of the flow path, typically 5-10%, small errors in the aerodynamic thrust force calculation can overwhelm the thrust bearing's capability leading to high bearing temperatures, potential bearing failure and compressor inoperability. This is further aggravated by the complexity of calculating the aerodynamic thrust force for high pressure compressors, Bidaut, Y. and Dessibourg, 2016. Mechanical testing where little or no aerodynamic forces are present or Type 2 testing where reduced speeds and pressures are typically used, generate only a fraction of the thrust load on the bearing. This provides little or no confirmation that the thrust bearing will be capable of handling the thrust loads developed over the operating envelop. Type 1 testing provides the possibility of testing the compressor over the operating range while measuring the thrust bearing temperature and thrust load itself through the use of load sensors in the bearing.

Rotordynamic Considerations

The differences with respect to the rotordynamic behavior are stark when comparing the Type 1 test to the Type 2 or more appropriately the mechanical test. During the mechanical test, critical speed location and unbalance response of the rotor and confirmation to rotordynamic predictions are intended and required. However, no indication of the stability of the compressor in its in-situ operation is possible. Mechanical testing is typically performed in a vacuum or other reduced power consumption configurations to permit shop drivers the ability to get the rotor to the required test speeds (for similar reasons as to why high-speed balancing is done in a vacuum.) During the mechanical test, the typical API 617 high pressure compressor rotor experiences effects from the radial tilt pad bearings (nearly infinite whirl frequency ratio), the dry gas seals (basically neutral with respect to stability) and the rotor assembly. Type 1 testing adds the dynamics of the impeller secondary flow and pressure driven behavior of the internal seals; impeller eye and hub, casing, balance piston and/or center. These constitute the major potentially destabilizing influences on the rotor lateral behavior.

The previous discussion regarding internal deflections focused on the increase or decrease in the seal clearance. However, a change in clearance profile, the slope for example, has been shown in literature to have a dramatic effect on the behavior of hole pattern or honeycomb seals (Nielsen, K., et al., 2015, Childs, D. and Wade, J., 2004, Alverez, D., 2006, Kocur, J. and Hayles, G., 2004). In these compressors, deflection of the division or end wall needs to be known within thousands of an inch to control the stator portion of center or balance piston seal. Controlling the change in angular position of the axial surface of the seal to within 0.01° is necessary to define the seal's behavior. A 0.02° change in angular position is enough to change a 10" seal length from 0.002" convergent to 0.002" divergent. This can alter these seal's direct stiffness from positive to negative. These deflections and their impact on the seal behavior is only demonstrated during the Type 1 test. The mechanical test under vacuum or significantly reduced load will not produce the deflections nor the pressure driven behavior needed to determine the seal's in-situ impact on lateral behavior.

The rotordynamic behavior of the back-to-back compressor tested with respect to stability shown a marked difference between unpressurized (rotor + bearings) and fully loaded (rotor + bearings + labyrinth seals + hole pattern seal). This is largely due to the behavior of the hole pattern seal on the division wall. When properly designed, hole pattern seals can dominate the rotor stability. At mechanical testing, the rotor was predicted by the OEM to have a minimum log dec value of ≈0.8. During Type 1 testing, the minimum log dec was predicted to be 2.3. The dramatic increase was due solely to the hole pattern seal. While stability testing was not done, the vibration plots did not show sub-synchronous vibrations associated with self-excitation of the 1st forward lateral mode as predicted.

When or why Type 2 testing might be sufficient

Type 2 testing certainly has its place in the arsenal of test approaches that can be appropriate for given applications. One obvious situation where a Type 2 test would be sufficient is if the compressors will operate at relatively low pressure and temperature in the field; i.e., with discharge conditions similar to those used in a Type 2 test. For example, the first body of a multi-body train might operate at sufficiently low pressures and temperatures so as to not be subjected to the pressure-related or thermal deflections and/or aero-mechanical forces experienced by higher pressure applications. Therefore, the Type 2 test conditions should be sufficient to assess both the mechanical and aerodynamic performance of that particular compressors. A Type 2 test might also be deemed sufficient if the compressor is an exact duplicate of a compressor whose aero-mechanical performance was previously validated via a Type 1 test. Finally, the OEM and end user might mutually agree to a Type 2 test if it can be adequately demonstrated via analysis and prior experience that there is minimal risk for any mechanical or aerodynamic issues.

Can a Type 2 test be designed to minimize risks?

OEMs can certainly design Type 2 test that will help identify issues that might arise at higher pressure, temperature or horsepower conditions. One obvious way is to conduct the testing at as high a discharge pressure as possible without exceeding the limits of the test loop (i.e., piping, coolers, valves, instrumentation, etc.). This might require the use of more expensive piping and the driver power consumption (and associated utility costs) will also increase due to the higher power required to operating the compressor at elevated pressures. However, the incremental cost associated with higher pressure piping and more power consumption on a Type 2 test pale in comparison to the costs associated with setting up and running a Type 1 test. Therefore, if a potential problem can be identified and investigated via a Type 2 test, it is still more cost-effective to design a slightly more expensive Type 2 test than to absorb the costs and cycle times associated with a full Type 1.

It is also possible to adjust the components in a Type 2 test so that they mimic the geometries that will occur at field conditions. For example, as described above, it is well-known that, because of machining tolerance stack-up and thermal growth, the width of the last stage diffuser in a high-pressure compressor will increase. It is possible to estimate the amount of deflection that could occur and then install devices to force the diffuser to be at the wider condition during the Type 2, low pressure test. As a result, the diffuser flow angles would be in the same range as they would be under field conditions. Therefore, if the wider width allows the flow angle to exceed the critical angle necessary for diffuser rotating stall, it should be evident during the Type 2 testing.

Similarly, as discussed earlier, in a back-to-back compressor arrangement, there will be some amount of deflection in the division wall between the low pressure and higher-pressure sections of the compressor. As with the last stage diffuser, it is possible to assess the deflections that could occur in the division wall and, more importantly, any deflection that might occur in the divisional wall seal that is mounted in the division wall. Again, it would be possible to adjust the mounting system or to the seal itself so that the clearance replicates field conditions under the lower pressure / temperature Type 2 conditions. Therefore, if the clearance would increase at higher pressure conditions, the impact of the increased leakages would be evident on the Type 2 test.

One other approach that can be used to minimize risk via a Type 2 test is to increase the amount and types of instrumentation that are installed in the compressor. Though it is typically "frowned upon" to install internal instrumentation in a "client's compressor", such instrumentation can be very important in identifying performance or mechanical anomalies that will cause issues in the field. In particular, dynamic pressure transducers installed in critical diffusers can sense rotating pressure phenomenon; a.k.a. rotating stall; even at low pressures; recognizing that the amplitude of the response will be dependent on the operating pressures. That is, the magnitude of pressure pulsations and associated vibration amplitudes will be lower than would be experienced on Type 1 test.

For high pressure centrifugal compressors, mechanical running test (MRT) becomes limited in its ability to mitigate risks that are faced by the application especially concerning rotordynamics. Internal deflections due to pressure loading are not significant during the MRT as compared to operation. Potential seal clearance changes will not be experienced. Rotordynamic behavior, especially rotor stability, is influenced only by the rotor, bearings and oil seals (if configured) during the MRT. All dynamics effects from pressure driven components are basically non-existent.

Type 1 vs. Type 2/MRT: Advantages and Disadvantages

From an OEM's perspective, the most obvious advantages of Type 2/MRT testing are the reduced cost, cycle time and resource requirements. Compressors can be installed relatively quickly on existing test stands and connected to pre-piped test loops, existing control systems and available drivers (electric motors or steam turbines). A limited number of test personnel are able to complete the necessary aerodynamic and mechanical testing and the standard data sheets and test curves can be quickly assembled for the client. As a rough approximation, it will take on the order 30% - 80% less time and cost to prepare a nominally-sized straight-thru compressor for Type 2 testing versus the setup for a Type 1 test. Granted, the time needed to set up and run the test will be dependent on the size and complexity of the compressor, or train, if required. For example, it takes much more time to prepare for and run a compressor that has multiple sidestreams than it does to set up and test a straight-thru compressor. However, this increases the cost of both types of testing.

A Type 2 test is also a more controlled test than a Type 1 test. Typically, more instruments are used during Type 2 testing than during Type 1. During Type 2 testing, probes are often installed directly into the gas stream to measure total pressure and temperature directly. Type 1 tests utilize static pressure measurements and static temperature measurements via thermowells installed in the loop piping. Type 2 tests are also typically conducted on a single gas constituent and in pressure and temperature ranges at which gas properties are well-documented. Finally, Type 2 test conditions allow easier manipulation of test variables and test setup for troubleshooting purposes during operation, due to the use of inert gas instead of a hydrocarbon.

The advantages of Type 2 testing to the project are the reduced project cost and schedule. The reduced complexity of the Type 2 test in comparison to the Type 1 test also reduces schedule risks due to testing delays at the OEM shop. The OEM performs two orders of magnitude more Type 2 tests than Type 1. Test loops, shop drivers, lube systems and instrumentation are readily available and significant experience exists with conducting Type 2 tests. Obviously both the experience and risk get strained when either an extended Type 2 or Type 1 test is conducted.

The disadvantages to the Type 2 test from an OEM standpoint are much the same as for the end user, i.e. the Type 2 test might not identify aerodynamic or mechanical performance issues due to the effects previously described. The disadvantages of this to the OEM are: (1) dissatisfied clients and likely loss of future sales to said clients; (2) being forced to investigate and resolve the issues in the field rather than in their own facilities; (3) additional warranty costs; (4) other expenses related to re-establishing reputation or working relationship with clients, etc. For the end user, lost revenue from production delays, resulting from correcting aero/rotor dynamic problems in the field, can easily exceed the cost of the Type 1 test (or the equipment in its entirety.)

Regardless of how the Type 2 is modified or altered, several risks may remain unaddressed. Many of the mechanical risks being mitigated (diffuser geometries / seal clearance) by mimicking operation during Type 2 by altering the geometric dimensions are based on analytical predictions. Philosophically speaking, this is what the testing is supposed to validate. So, the uncertainty remains regarding whether the analytic predictions are accurate.

Stability testing is now possible during any type of test, Pettinato, B., et al., 2010. With injection compressor applications, the risk of instability can be somewhat mitigated by performing a stability testing during the MRT or extended Type 2 test. However, in comparison to some process compressors where low shaft bending stiffness or high bearing loads may be the key factors in determining susceptibility, injection compressors are typically the opposite (i.e. high bending stiffness and low bearing loads). The susceptibility factor with these applications are the internal seals and their pressure driven behavior. With the acknowledgement that predictive tools continue to improve, Kocur, et al., 2007, presented an indication that variability in the prediction of gas labyrinth seal dynamic behavior among analysts is significantly greater when compared to tilt pad bearing behavior.

One final disadvantage of the Type 2 test concerns the thrust load. As mentioned, thrust calculations in high pressure centrifugal compressors are not to be taken lightly. With significant flow related pressure profile complexities on the impeller external surfaces, it does not take much of a calculation error to overload the thrust bearing (the thrust bearing load is $\approx$5% of the aerodynamic impeller axial force.) Thrust load/pressure profiles during a Type 2 are more representative of a low-pressure compressor and do not address the risks associated with injection applications. The possibility of overload or high temperature operation of the thrust bearing due to inaccurate thrust calculations is not eliminated in a Type 2 test.

The most obvious disadvantages to Type 1 testing are the time and cost to design, assemble and test the compressor at near field conditions. It is difficult to approximate the cost differential between Type 1 and Type 2 because of the wide range of compressor applications, processes and control systems. However, one could reasonably assume that Type 1 test costs will be on the order of ten times the cost for Type 2 testing or greater. Likewise, the cycle time to set up for a Type 1 test (i.e., build the more complex loop, possibly install the client driver, implement the full control system, etc.) will be much greater than for a Type 2 test. Again, the schedule difference is difficult to approximate because it depends heavily on the complexities and details of the testing to be performed. There are also increased personnel requirements during a Type 1 test because of its more complex nature. Further, there are increased safety precautions related to high pressure testing with, at times, highly volatile gas mixtures. Most of these costs are passed along to the client who specifies the type of testing that must be conducted. Should problems be encountered during the testing or should the compressors and their auxiliary systems not meet the client requirements, costs to resolve the problems (i.e., additional analytical work, rework efforts, new part procurement, etc.) and to re-run the tests are absorbed by the OEM while the end user faces schedule and startup delays with the potential for lost or reduced production.

One might argue that the biggest advantage of Type 1 testing to the OEM is reduced warranty costs and to the end user is further ensuring that "right the first time" is achieved at startup in the field. Like the end user, the OEM would certainly prefer to discover any mechanical or aero-performance issues with the compressor and/or its auxiliary systems before the equipment is shipped to the field. As described above, if such are found after the equipment is at site, resolution costs on both sides escalate significantly.

Finally, one advantage to both parties of any type of testing is that they gain further insight into the mechanical and aerodynamic behavior of their products. For the OEM, this can lead to improvement/evolution of their design. For the end user, improved equipment health monitoring and risk-based maintenance. It has been said that "One good test is worth a thousand expert opinions." (Original source unknown) and some have added "… or a thousand CFD simulations." The bottom line is that there is still no replacement for high quality test data when it comes to validating or calibrating a design, analysis and prediction system. The long-term value of such data cannot be overstated.

CONCLUDING REMARKS

The paper presented a comparison of Type 1 and Type 2 testing of centrifugal compressors and has highlighted some of the critical differences between as well as the advantages and disadvantages of such tests. A review of the considerations that influence the test set-up and operating conditions required was offered and the importance of matching the volume reduction between the field and test conditions was emphasized. Discussions were offered on the potential aerodynamic and mechanical performance variations that could occur because of differences in geometry, operating conditions and/or gas properties between the two tests. References were made to recent performance testing conducted at the OEM facility that illustrate some of these differences. Recommendations were offered on how Type 2 tests could be conducted to bring the results closer to what would occur during Type 1 testing.

Ultimately, the choice as to which types of testing to conduct as well as the objective of the testing must be based on an extensive dialog between the end user, the process engineers, and the OEM. There is no greater waste of time and money than running a test and taking data that is not necessary apart from *not* doing the testing necessary to identify potential performance issues before the equipment leaves the OEM facility. It is hoped that the information and experiences described in this paper will help the reader make more informed decisions.

NOMENCLATURE

MW = Molecular weight, mols
T = Temperature, degrees Fahrenheit
Z = Compressibility factor, dimensionless
P = Pressure, pounds force per square inch
n = Polytropic volume exponent, dimensionless
k = Isentropic exponent
U = Tip speed, feet per second
μ = Polytropic head coefficient, dimensionless
A0 = Sonic velocity, feet per second
g = Gravitational constant, feet per second squared
V = Specific volume, cubic feet per pound mass
N = Speed, revolutions per minute
Q = Capacity, cubic feet per minute
GHP = Gas horsepower
Mass = Mass flow rate, pounds per minute
D = Diameter, inches
S = Radial clearance, inches
Density = Pounds mass per cubic foot
η = Efficiency

Subscripts:
1 = Inlet
2 = Discharge

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