Performance and mechanical running tests of centrifugal compressors - I

May 26 2014 - Neetin Ghaisas, Fluor Canada

Radially split type compressor set up for shop tests

Usually contractors and users include performance test in the base scope of the purchased compressor. However during bid conditioning, the Engineers tend to give less attention to the evaluated bidders’ shop test capabilities. Frequently this significant parameter is not assessed until after receipt of the compressor manufacturer’s test procedure. There are instances when performance test is not carried out in accordance with ASME PTC-10 test Code because one or more limits in the Code are violated. Therefore it is important that the user and the supplier mutually agree with the scope of shop performance test and discuss practical limitations of the test facility. Ideally this activity should be completed before issuing a purchase order for compressor(s). As a minimum, the following parameters should be included in the review.

-Will a closed loop performance test be carried out with compressor inlet pressure below atmospheric pressure, to reduce power requirement? This situation may arise in the case of high power compressors when the shop driver is limited in its output.
-Is a suitably rated torque measuring device available in the test shop to determine compressor shaft power?
-Instead, if heat balance method is used to calculate shaft power input, request compressor casing heat loss calculation from the supplier. It should include convection and radiation heat losses. Also, the location of compressor casing and ambient temperature measurement sensors should be stated in this calculation procedure. The importance of sensor location is explained in the next bulleted item.
-The ambient temperature should be measured in the vicinity of the compressor for determination of the true temperature difference. This \( \Delta T \) is the lost work input and it affects the calculated polytropic efficiency. Test Power = Gas Power + Heat Loss. Compressor discharge temperature should be corrected by adding the heat loss due to radiation.
-Is balance piston leakage estimated, or measured during the test and subtracted from the inlet weight flowrate to compressor?
· Are the exchangers (usually of shell-and-tube type) in the shop test loop adequate for the maximum heat load generated during the shop performance test? These exchangers include the ones in lube oil and when applicable, in inter-stage services.

· What facility is available in the shop to purge the test loop with an inert gas prior to introducing the test gas?

· What gas sampling method is used for the test? Gas sampling must be carried out before each test measurement point.

· How is the control of gas mixing achieved when a volumetric mixture of test gases are used?

· On-line oxygen analyzer is required to monitor percent of air in the test loop (measurement of impurity in the test gas). Analyzer’s output shall be fed to data reduction computer program, to adjust the properties of the test gas. This is important, because change in molecular weight of the test gas will affect its density and the calculated compressor head. Note: ASME PTC-10 test Code requires that composition of test gas in a closed loop test must be continuously monitored to avoid formation of combustible mixtures.

· What is the maximum allowable working pressure of piping and equipment inside the test loop and is it adequate for the derived shop test conditions?

· What safety devices are installed in the test loop to protect from accidental overpressure?

· Which devices in the test loop detect and alarm abnormal conditions like loss of coolant to shop shell-and-tube heat exchangers?

· How is the lube oil cooling duty at full load established from no-load, full-speed mechanical running test?

· If a complete unit test is purchased, is the supplier’s shop equipped to use the contract lube oil system?

· What automated data acquisition and analysis systems are installed in the test facility to produce vibration response plots such as Bode’, Orbit, Polar, Amplitude vs. Frequency, and Waterfall Spectra? Can the supplier provide electronic copies of plots with the test report for user’s future reference?

· The shop data acquisition and processing system should be capable of adjusting the calculated test speed at the guarantee point with respect to the actual gas inlet temperature prevailing on the day of the test. Note: The inlet temperature used in the calculation of test speed can be different from the actual compressor inlet temperature.

· Similarly, the data processing system should be able to correct the calculated shaft power input based on the actual inlet pressure measured during the test. The specific volume ratio is affected by the pressure ratio. Specific volume ratio in each stage should be within the allowable departure in Table 3.2 in ASME PTC-10 test Code (within 95 to 105 percent of design values). From the perspective of similarity of flow when converting the test results to the specified or guarantee conditions, the volume flow coefficients, impeller tip Mach Numbers and Reynolds Number must be within the deviations permissible by the test Code.

**TYPES OF PERFORMANCE TESTS**

ASME PTC-10 test Code includes two types of tests. Type 1 test is carried out with specified (or design) gas at guarantee point operating conditions. Allowable deviations in the operating conditions and operating parameters are found in Tables 3.1 and 3.2 in the Code. Type 1 test is conducted when specified gas is readily available such as for air compressors.

Type 2 test allows the use of a substitute gas when testing with specified (design) gas is impractical. This test is used when design gases are hydrocarbon mixtures which are difficult to be duplicated.
under test conditions. Also note that some of the design gases are either toxic, or explosive, or highly flammable.

The permissible deviations from the specified operating parameters are shown in Table 3.2 in the Code. Substitute gas can be either a single gas or a gas mixture which closely resembles thermodynamic properties of the specified gas. Performance calculation methods may use either ideal or real gas laws. Table 3.3 in the Code includes the limits of departure from ideal gas laws for specified and test gases.

TEST GASES
Carbon dioxide, mixture of helium-nitrogen, propane, R134a, and natural gas are generally used as test gases in closed loop testing. Air is used in open loop testing but it should never be used in a closed loop testing because of potential for explosion if lubricating oil comes in contact and contaminites air.

Low molecular weight test gases are not considered for compressors in high molecular weight gas applications since the required test speed may exceed the maximum continuous speed of the compressor. Refinery recycle gas has molecular weight less than 10. Centrifugal compressors in this service are usually tested with a mixture of helium and nitrogen. Compressors in high molecular weight applications such as wet gas, and chlorine are typically tested with either R134a, or carbon dioxide.

The three factors that require particular attention when determining equivalency of the test and specified conditions are volume ratio, Machine Reynolds Number, and Machine Mach Number.

- Volume ratios of the test gas and the specified gas must be matched in accordance with the limits in Table 3.2 in the Code to ascertain that volume flow in all stages of the compressor will be equivalent during performance test. Volume ratio is directly proportional to molecular weight, speed, and compressor’s head but bears an inverse relationship with compressibility Z, inlet temperature, and K–value of the gas (ratio of specific heats C_p/C_v).
- Machine Reynolds Number is related to friction losses in the compressor gas passage. A correction in the test efficiency is required by the Code if Machine Reynolds Numbers at the test and specified conditions differ. Allowable Reynolds Number departures are shown in Figure 3.5 in the Code.
- The Machine Mach Number is the ratio of the velocity at the outlet of the first stage impeller to the acoustic velocity at compressor inlet conditions. It is a measure of the maximum compressor capacity and therefore associated with stonewall effect. Allowable Machine Mach Number departures are represented in Figure-3.3 in the Code. Ideally, the test gas should have the same or if not possible, a higher K–value than that of the specified gas to achieve close proximity to design Machine Mach Number.

SEQUENCE OF SHOP TESTS
Compressor performance test can be conducted either before, or after the mechanical running test. If a compressor was first subjected to mechanical running test but failed the performance test, the time, resources, and efforts expended in arranging the mechanical running test would be deemed wasted. From this viewpoint, some users prefer to run performance test first, followed by mechanical running test. If a spare rotor was purchased, the performance tested rotor, in this situation, will likely not be installed in the casing during final assembly in shop.

It is a common practice that compressor manufacturers carry out factory internal test prior to customer (or user) witnessed test. Therefore, many users specify mechanical running test of spare rotor first, followed by mechanical running test and performance test of the job or main rotor. In this arrangement, the performance tested rotor is usually left in the compressor casing at delivery from the shop. The order of tests shown below is only typical and can vary, depending on the scope and mutual agreement between the user and the compressor manufacturer. The requirements of mechanical running test and performance test are included in API Standard 617.

1. Mechanical running test under vacuum, with spare rotor.
2. Vary lube oil supply pressure and temperature in the second-half (after two hours) of mechanical running test with spare rotor.
3. Spare rotor change with the main rotor. During this activity, visually inspect bearings, seals and internal parts.
4. Mechanical running test under vacuum, with main rotor.
5. Vary lube oil supply pressure and temperature in the second half (after two hours) of mechanical running test with main rotor.
6. Shop verification of unbalance response test on main rotor at the end of four-hour mechanical running test.

7. Sound pressure level check during mechanical running test on main rotor.

8. ASME PTC-10 performance test with main rotor.


10. Assembled compressor gas leakage test.

FULL-SPEED, FULL-LOAD TEST

Complete Unit Test is included in API Standard 617 under the section ‘Optional Tests’. Frequently, it is called as string test. Many users stipulate full-speed, full-load mechanical running test for compressors in offshore installations, large liquefied natural gas (LNG), re-injection, and ethylene plants. This is partly due to the fact that output of these facilities has been increasing over the years, thus requiring larger frames/bigger casings. Some of the important considerations for specifying a full-speed, full-load test for compressors are presented in this section.

- Largest frame sizes for the compressor, (gear if furnished), driver, and auxiliary systems being used in a given application. Note: users should check if these frames are installed and operating successfully under analogous conditions at other locations.
- Compressor with multiple operating conditions that have direct impact on the plant's output.
- Shorter plant commissioning schedule resulting in far lesser time to adjust the performance of critical machines in the field. In such situation, users may want to pay more money upfront for complete unit test than having to spend much more in rectifications at site, if something were to go wrong.
- Known rotordynamic issues or concerns arising from stability analysis of the compressor.
- Compressor with multiple side loads.
- Compressor fitted with honeycomb seals, shunt holes, or swirl brakes. These features may require more exhaustive testing than conventional performance test and mechanical running test.
- Ability of the test shop to conduct complete unit test.
- Logistics of transporting gear and auxiliary equipment to the test shop. Does the project schedule allow for additional time duration for transportation?

If a complete unit test is specified for compressors installed in identical, multiple production trains, it may be carried out on the compressor(s) in only one train.
TEST PROCEDURE DOCUMENTATION

The test Procedure should be detailed enough to serve as the single source of reference when conducting performance and mechanical running tests. Ideally, it should include the following information.

- Test Sequence and overall schedule
- Type, location, quantity, and accuracy of instrumentation used for test. Their current calibration records should be available.
- Mechanical running test program including graphical presentation of test steps in speed versus time form.
- Schematic of mechanical running test arrangement. An example is shown in Figure 1.
- Mechanical running test measurements and sampling rate. The following list is typical and will vary, depending on the specific application.

1. Lube oil pressure at each supply point, psig (kpag)
2. Lube oil supply temperature, deg F (deg C)
3. Lube oil flow at each bearing, gpm (l/min)
4. Lube oil drain temperature at each return point, deg F (deg C)
5. Bearing metal temperature, each bearing, deg F (deg C)
6. Speed, rpm
7. Inlet vacuum, psia (kpa)
8. Inlet temperature, deg F (deg C)
9. Filtered and unfiltered shaft vibrations, mils peak-to-peak (microns peak-to-peak)
10. Axil displacement of rotor, mils (microns)
11. Buffer gas supply pressure to dry gas seals, psig (kpag)
12. Sweep of frequencies at maximum continuous speed and recording of vibration amplitude versus frequency range
13. Unfiltered and filtered vibration amplitude and phase angle versus speed plots during coastdown.
14. Determination of first lateral critical speed
15. Slow roll run-out
- Mechanical running test acceptance criteria.
- Performance test procedure.
- Schematic of performance test arrangement.
- Tabulation of test and specified (or design) conditions.
- Adjustment of test conditions to contract or design conditions

HOW THE TESTS ARE DONE

A radially split type compressor is set up for shop tests. Various spectral plots are generated during mechanical running test of the compressor. The compressor is installed on baseplate after test.

A graphical representation of the vectorial subtraction of vibration levels between mechanical run test and rotor unbalance response test, at each speed, is used to verify if the compressor rotor's response to known unbalance at all speeds was within the predicted levels at minimum/maximum radial bearing clearances. This Amplitude versus Speed Chart includes vibration level lines for:

1. Unbalance weight Influence (unbalance response - Mechanical Run Test vectorial subtraction)
2. Mechanical Run Test - Steady State conditions

3. Unbalance Response

4. Predicted - maximum radial bearing clearance and Predicted - minimum radial bearing clearance.

HOW TESTING AFFECTS PLANT REVENUE

More often, compressor's flow and discharge pressure are directly related to production level and in turn, to the revenues of process plants. In many cases, compressors also influence performance guarantees of new process facilities. While shop testing is expensive, it is justified because any performance deficiencies noticed at site after start-up can cost a lot more money and time to rectify.

CODE PROCEDURES

Most users specify ASME PTC-10 power test Code based performance test for their purchased compressor(s).

The Code procedures are used as the basis to predict compressor performance. It is usually either not possible or is impractical to test compressors with specified or design gas. Therefore, suitable test gases are used for performance testing. These gases are either single gases or gas mixtures. The three most significant criteria used in the selection of test gas are Volume ratio, Mach number, and Reynolds number.

Mechanical running tests serve the following objectives:

1. Confirm that transient and steady-state rotor vibrations are within specified limits and check that potential troublesome or discrete frequencies are not present in the vibration plots.

2. Demonstrate mechanical integrity of the compressor.


4. Check damped critical speed(s) and compare with calculated values shown in lateral critical speed analysis.

5. Assess quality of compressor assembly; in specific, cleanliness of lubrication passages, the internal clearances, unit alignment, and freedom from leakage across casing joints and shaft end labyrinth.

6. Coupling hub fit-up and contact with shaft.

7. System response to variations in lube oil supply pressure and temperature.

8. Provide baseline readings for bearing metal temperatures, bearing oil return temperature, bearing oil flow rates, shaft seal primary vent flow (assuming that dry gas seals are installed), balance piston flow, axial position of rotor, and unit noise level.
Shop test of centrifugal compressors

June 7 2011 - Amin Almasi

Centrifugal compressor purchasers and end-users realize the value and payback of accurate and proper shop performance test and shop mechanical run test. Shop test is a necessary step for centrifugal compressor future reliability and trouble free operation. ASME-PTC-10 type 1 performance test should be done wherever possible. Otherwise arrangement and details of ASME-PTC-10 type 2 performance test, as close to specified operating condition as possible, should be fixed before machine order. Three case studies are discussed.

PERFORMANCE TEST PROCEDURE

Centrifugal compressor performance tests are described in ASME-PTC-10 and API 617. With reference to ASME-PTC-10, there are two different types of performance tests. A type 1 performance test is actually a shop performance test in anticipated site condition. It is conducted with same gas as site (same gas with molecular weight deviation below 2%). Generally pressures, temperatures, compressor speed and capacity permissible deviations are below around 4-8%. Type 2 performance test is completely different. It permits the use of a substitute test gas and accepts extensive deviations between test and specified operating conditions. There are only a few limits on some essential gas dynamic parameters of test conditions (compare to specified operating conditions). Specific volume ratio and flow coefficient should be within around 5% deviations.

There are some limits on machine Mach number and machine Reynolds number. The test speed, capacity, mass flow, pressures, temperatures, power, etc are often totally different from the specified operating condition speed. In Type 2 test, a suitable gas is identified which does not lead to excessive power or discharge temperature and is readily and cheaply available. Substitute gas such as air, nitrogen, CO2, CO2/He mixes, fuel gas, etc are used. Safe operating speed, critical speeds, maximum allowable pressures, allowable temperatures and other machine limits are considered in test condition selection. ASME-PTC-10 Type 2 allows considerable deviations in test conditions.

For example a natural gas (MW=16) centrifugal compressor can be performance tested (Type 2) using CO2 (MW=44) with around half inlet flow, approximately 20% mass flow, around 50% speed, approximately 6% gas power and much less pressures (less than 10%) compare to specified operating conditions. There is always question regarding accuracy and usefulness of type 2 test. The flow patterns of centrifugal compressors are complex. This complex performance is function of main fluid characteristics such as volume ratio, flow coefficient, machine Mach number and machine Reynolds number. Concept behind type 2 performance test is performing a test with different gas and flow details, where as main fluid characteristics are within certain limits (let say volume ratio, flow coefficient within around 5%, machine Mach number within around 0.1 deviation and machine Reynolds number within 0.1 to 100 times) and use available mechanics of fluid knowledge and formulations to estimate machine performance in specified operating conditions. For example the friction aspect of compressor performance is affected by the machine Reynolds number.

In type 2 test, test Reynolds number is different compare to specified operating condition but it is still within certain limits to keep governing friction formulations the same (same model and flow regime). Based on theory, a modification (or let say correction) to the test results is applied based on available
gas dynamic knowledge to estimate the friction effects of compressor performance in specified operating condition. All correction formulations are available in ASME-PTC-10 for estimation. Some engineers believe that ASME-PTC-10 type 2 test is not an actual performance test but it is a laboratory-type fluid test on real machine to confirm some fluid dynamic characteristics. ASME-PTC-10 type 1 test is always preferred. If ASME-PTC-10 type 1 is not possible (for example real gas can not be supplied and used in shop), it is necessary to plan for a type 2 test with test conditions (gas molecular weight, speed, capacity, power, pressures, etc) as close as possible to specified operating conditions. Arrangement and details of type 2 tests should be fixed in bidding stage and before machine order. Vendors always prefer simplest and cheapest arrangement for type 2 test (and code ASME-PTC-10 type 2 allows this). Unfortunately type 2 test gas selection and arrangement are generally discussed after order or even near test time. It usually causes considerable change order costs or purely laboratory-type test on real machine.

Type 2 test can be useful if gas mixture that closely approximate the job gas used and pressures, compressor speed, capacity, power, etc matched as close as possible to specified operating conditions (within ASME-PTC-10 limits and as close as practical to operating conditions). This properly arranged type 2 test may give useful prediction on future machine behavior such as operation close to surge point, some types of aerodynamic excitations, effects of seal on dynamic behavior, etc. These effects are of serious concern particularly in high pressure applications.

For performance test whether type 1 or type 2 is used, following limits should be confirmed (in case of type 2, based on estimation):

1- Head and capacity: zero negative tolerance.

2- Power: not exceed 104% of predicted power (sometimes when plant efficiency is very important, lower limits, as low as 2%, may be adopted and agreed before machine order).

3- Surge: stable operation at near calculated surge (let say around 8% above calculated surge flow).

Based API 617 and refer to type 2 test procedures, test speed may be completely different with specified machine speed. It is not permitted for ASME-PTC-10 type 1 test (in type 1 test only 2% speed deviation is allowed). Complete unit test is always recommended (if practical). All train components including compressor casings, gear units, driver, and all auxiliaries are tested all together. If complete unit test is not possible, a tandem test is useful. In tandem test, usually shop driver, and shop oil system are used. A separate auxiliary test may be performed. Torsional vibration measurements are recommended to evaluate torsional calculations. It is necessary for complex multi-machine casings including gear units with variable speed driver.

**MECHANICAL RUN TEST**

Job seals and bearings are generally employed in mechanical run test. Sometimes additional heat dissipation elements (such as specially designed fins) are required to avoid overheating particularly in low pressure tests. Pressurized run test is always preferred, except special cases (such as cases when under vacuum tests are required). Extensive measurements (speeds, pressures, temperatures, oil and seal flows, bearing metal temperatures, vibration data, seal gas data, etc) are required for mechanical running test. Generally as far as practically possible, all contract equipment and systems
are used for the test. If using job coupling is not practical, test can be implemented with simulator. Overhung moments should be simulated within suitable margin of job coupling (preferable below 5%, maximum deviation 10%). Test facilitates should be capable of continuously monitoring, displaying, and recording vibration, vibration spectra, Bode plots, and shaft orbits. Shop mechanical run test procedure is straightforward. The machine is started and speed increases to the maximum continuous speed. Operation continues until bearing metal temperatures, oil temperatures and shaft vibrations are stabilized. Then machine is operated 15 minute at trip speed. After that, 4 hours operation at maximum continuous speed. Main focus is on vibration. Usually vibration measurement covers a range of 0.25 – 8 times of operated speeds.

Based on experience, high frequency ranges (usually above 1500 Hz or 90,000 rpm) do not contribute to the judgement of the mechanical performance. The frequency analysis recorded must not show significant amplitudes at frequencies other than running speed or twice running speed. Shop verification of the unbalanced response analysis is usually required. Each spare rotor needs a mechanical running test. Other important goal is verification of lateral critical speeds. The locations of all critical speeds below the trip speed are confirmed on the test stand. Actual critical speeds are expected within 5% of analytical values. Peak responses amplitudes should not exceed the analytical values. During test the lube oil temperature rise through the bearing is not expected to exceeding 30°C.

Vibration readings and bearing temperatures at the end of the four-hour run should be essentially the same as those recorded at the beginning of the four-hour test. Real-time vibration data (from startup to shutdown) are recorded and delivered to end-user. All hydrodynamic bearings are removed, inspected, and reassembled after the mechanical running test.

Removal and inspection of dry gas seal is not recommended. Dry seal gases (particularly cartridge type) may require that the seal be returned to the seal manufacturer if removed for inspection. It is recommended to remove oil seal (mechanical oil film type shaft end seals which very rarely used today) for inspection after test. Minor scuffs and scratches may occur on the bearings. Subsequent minor cosmetic repairs of these parts do not justify repetition of the test. If melting or smearing, overheating or distinct wear occurs in the babbit of bearing shoes, these parts should be replaced. The cause of the defect must be investigated and eliminated, and the mechanical run test should be repeated. After run test, compressor casing is gas leakage tested to evaluate joints and seals. Assembled compressor is tested to maximum operating pressure for minimum of 30 minute (inert gas MW - Molecular Weight, less than job gas MW, helium for low MW gas or nitrogen or refrigerant gas for high MW gas). Please consider following recommendations regarding optional tests:

1- Varying lube oil conditions (oil pressure and temperature at minimum and maximum values) is an API 617 optional test but it is strongly recommended.

2- Noise level test is API 617 optional test. It is only required for large machine when high noise is expected.

3- Post test inspection of casing internal is not recommended due to advantage of delivery of proven run and pressure tested compressor.
SITE CONSIDERATIONS
Care should be taken when designing the team that represents equipment purchaser for shop tests. Some of individuals, who attend shop test, need to be worked at site for commissioning and at least one for long term operation.

There are many notes about necessity of vendor representative involvement in compressor piping check and alignment. Based on experience it is useful that vendor representative observes a check of the piping but it is unnecessary for the vendor representative to be present during the initial alignment check or to check alignment at the operating temperature.

As soon as possible a site performance test should be conducted on each compressor (after installation and pre-commissioning). The performance is checked based on ASME-PTC-10 type 1 test. It should be planned in advance. Test procedure is necessary and some extra instruments and temporary provisions may be required. The test should confirm, as a minimum, that the compressor can meet the guarantees of flow, pressure and power and that the other points on the curve are within +/- 5% of the shop performance test curve.

Site performance tests may also be implemented during operation (for example after several years of operation), to identified extent of degradation.

5 CASE STUDIES
The first case study is presented for type 2 performance test of a large centrifugal compressor for hydrogen service. Table 1 shows performance test data for this centrifugal compressor.

Table 1 Type 2 Performance Test Data for Hydrogen Centrifugal Compressor (first case study).

<table>
<thead>
<tr>
<th>Specified Operating Condition</th>
<th>Type 2 Test Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas Molecular Weight</td>
<td>~4</td>
</tr>
<tr>
<td>Inlet Volume Flow (m³/h)</td>
<td>6200</td>
</tr>
<tr>
<td>Volume Ratio</td>
<td>~1.12</td>
</tr>
<tr>
<td>Machine Mach Number</td>
<td>0.241</td>
</tr>
<tr>
<td>Machine Reynolds Number</td>
<td>~8x10⁶</td>
</tr>
<tr>
<td>Inlet Pressure (Barg)</td>
<td>150</td>
</tr>
<tr>
<td>Discharge Pressure (Barg)</td>
<td>185</td>
</tr>
<tr>
<td>Power (MW)</td>
<td>~6</td>
</tr>
<tr>
<td>Speed (rpm)</td>
<td>9950</td>
</tr>
</tbody>
</table>

Based on experience, proposed test conditions of this case study are better than many type 2 tests performed in industry. Test gas (gas mixture with molecular weight 6) and test speed are not so far from operating conditions. But improvements can be done for more realistic type 2 performance test such as
using closely matched gas mixture (molecular weight~4), increasing test gas pressures (while keeping parameters within ASME-PTC-10 limits), closely match isentropic index, etc. These modifications can help to better match test power, capacity and compressor dynamic behavior to specified operating conditions. In this case, because it was not negotiated in bidding stage, agreement could not be reached for improvements and performance test was done based on data of Table 1. This case study shows importance of negotiation of performance test details before machine order.

The second case study is presented for type 2 performance test of medium pressure natural gas compressors. Table 2 shows proposed performance test plan for this machine. It is a good example of considerable differences between test and specified operating conditions. As a rule of thumb, test speed within 20% limits of machine speed is always preferred. For this test, test speed is around 60% of machine operating speed. But for this purchase order, several identical compressors are required and these type 2 performance tests are applied for subsequent units after first unit successful type 1 performance test. This arrangement is optimum and acceptable.

Table 2 Type 2 Performance Test Data for Medium Pressure Natural Gas Centrifugal Compressor (second case study).

<table>
<thead>
<tr>
<th></th>
<th>Specified Operating Condition, Natural Gas</th>
<th>Type 2 Test Condition with CO2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas Molecular Weight</td>
<td>~16.3</td>
<td>~44</td>
</tr>
<tr>
<td>Inlet Volume Flow (m3/h)</td>
<td>~20,000</td>
<td>~12,000</td>
</tr>
<tr>
<td>Volume Ratio</td>
<td>~2.6</td>
<td>~2.6</td>
</tr>
<tr>
<td>Machine Mach Number</td>
<td>0.614</td>
<td>0.605</td>
</tr>
<tr>
<td>Machine Reynolds Number</td>
<td>~6.4x106</td>
<td>~9.8x106</td>
</tr>
<tr>
<td>Inlet Pressure (Barg)</td>
<td>12</td>
<td>0.7</td>
</tr>
<tr>
<td>Discharge Pressure (Barg)</td>
<td>46</td>
<td>4.8</td>
</tr>
<tr>
<td>Power (MW)</td>
<td>~13</td>
<td>~1</td>
</tr>
<tr>
<td>Speed (rpm)</td>
<td>~10,000</td>
<td>~6,000</td>
</tr>
</tbody>
</table>

Third case study is presented for a high pressure natural gas compressor. Test speed is around 80% of specified operating speed. Improvement still can be done for a more realistic type 2 shop test using gas mixture to closely match gas molecular weight. Again modification proposals were not successful because of commercial issues. Test was implemented based on data of Table 3. Comparison with site performance test shows acceptable results.
Table 3 Type 2 Performance Test Data for High Pressure Natural Gas Centrifugal Compressor (third case study).

<table>
<thead>
<tr>
<th>Specified Operating Condition, Natural Gas</th>
<th>Type 2 Test Condition with Nitrogen (N2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas Molecular Weight ~16</td>
<td>~28</td>
</tr>
<tr>
<td>Inlet Volume Flow (m³/h) ~5500</td>
<td>~4500</td>
</tr>
<tr>
<td>Volume Ratio ~1.8</td>
<td>~1.8</td>
</tr>
<tr>
<td>Machine Mach Number 0.47</td>
<td>0.48</td>
</tr>
<tr>
<td>Machine Reynolds Number ~10x10⁶</td>
<td>~1.3x10⁶</td>
</tr>
<tr>
<td>Inlet Pressure (Barg) 40</td>
<td>8</td>
</tr>
<tr>
<td>Discharge Pressure (Barg) 100</td>
<td>16</td>
</tr>
<tr>
<td>Power (MW) ~10</td>
<td>~2</td>
</tr>
<tr>
<td>Speed (rpm) 11,800</td>
<td>9,400</td>
</tr>
</tbody>
</table>

Notation
MW Molecular Weight

ASME American Society of Mechanical Engineers