PREDICTING AND VERIFYING RERATED COMPRESSOR PERFORMANCE THROUGH SINGLE STAGE SCALE MODEL AND LIMITED FIELD TESTING

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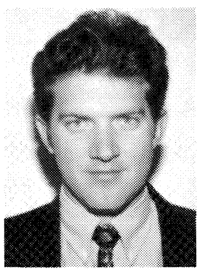

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ABSTRACT

Performance verification of centrifugal compressor rerates creates a unique challenge to turbomachinery users. It is not often practical to perform flange-to-flange performance tests on machinery already installed in the field, as the redundant instrumentation necessary to conduct a reliable test is rarely present onsite. In addition, rerated units are typically commissioned during a turnaround where time constraints do not allow the luxury of a test. In some cases, it may be possible to remove a relatively small compressor body from the site and shop test it. However, in almost all circumstances the cost associated with doing so is prohibitively expensive. This leaves the user faced with the dilemma of being uncertain of a rerated compressor's performance until it is actually run with the new process.

A solution is proposed to the problem of performance verification for rerated centrifugal compressors. Single stage scale model testing of the aerodynamic elements that comprise a compressor stage is a proven method used by original equipment manufacturers in the development of new components. This practice may also be used to find the performance of aerodynamic elements intended to be used in compressor rerates. If the test proves successful, the design can be incorporated with a high degree of confidence that the equipment will operate as expected. After the rerate is complete, the performance can be verified through limited field testing.

The details a single stage testing program as well as the independent field performance verification of four very different compressors are presented. Two of the units under consideration were single body. The other unit consisted of a two-body compressor train. The compressors under consideration were originally supplied by three different original equipment manufactures. The full scale hardware consisted of impellers ranging in size from 17.0 in to 43.5 in in diameter. The scale factors used to model the aerodynamic stage ranged from 0.81 to 0.25. A total of seven different stages were single stage scale model tested between the three compressors.

The limited field testing used only the existing instrumentation previously installed on the compressor trains. After the field variables were taken into account, it was found that the predicted performance was in good agreement with actual compressor operations.

INTRODUCTION

Performance testing of special purpose equipment has always been critical to the success of projects at a major oil company. Although performance is usually guaranteed by the vendor, there is no way to be certain that the equipment will perform as designed until it is operated. It has been the user's experience that roughly 30 percent of new equipment purchased needs some form of modification to meet the guaranteed performance. While many of these are only minor mechanical adjustments, sometimes major changes are required. If the problems are not discovered until after the equipment is shipped to the field, or worse yet, after startup, the costs to repair them rise exponentially over catching them in the vendor's shop. In a worst case scenario, the repairs may not be feasible until the next shutdown, so that the equipment is forced to operate at reduced levels for extended periods of time. Such a situation could have severe financial impact on both the user and the vendor.

In order to comply with API 617 [1] requirements, centrifugal compressor performance is guaranteed by the vendor to deliver the normal head at the normal capacity without a negative tolerance. Initially, the values of head that a compressor will produce at different flows are calculated with sophisticated computer programs that analyze the geometry of the gas passages internal to the compressor. Even though these calculations are reasonably accurate, they can be refined further by using previous general experimental results to quantify boundary effects, loss coefficients, etc. But even so, extreme confidence cannot be placed in the computer programs' ability to predict how real equipment will behave.

In illustration, three different companies using state-of-the-art computer codes analyzed the same compressor stage, and all three arrived at different results. These companies include the OEM, a well known third party consultant and a rerate vendor. Eventually, a scale model was built and tested to PTC-10 standards, and the results are shown in Figure 1. While it is pleasing to note that all three erred conservatively, only one firm's prediction fell within a few points of the actual head coefficient at the design condition. And only one other firm accurately predicted the shape of the curve.

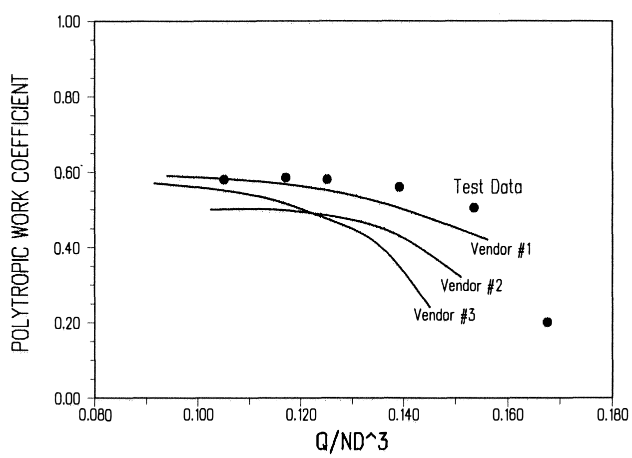

Figure 1. Comparison of Computer Simulations.

Because of the inability of computer codes to completely model a compressor's performance in advance, and also to cover for any manufacturing irregularities, it is standard practice for many companies to performance test *new* special purpose compressors. This is straightforward as the vendor normally has a test stand in his shop, complete with driver, piping, coolers and instrumentation. However, if aerodynamic modifications are made to a compressor that is already in service, it is economically unjustifiable to remove the machine to a test facility for thorough testing. But the need still remains to ensure its performance before startup. For a centrifugal compressor, this need can be met through single stage scale model testing.

THEORY OF SINGLE STAGE SCALE MODEL PERFORMANCE TESTING

Background of Scale Model Testing

The concept of scaling aerodynamic elements of centrifugal compressors is not new to the field of compressor design. It is the primary method used by original equipment manufacturers to develop new compressor lineups from existing machine designs. A compressor lineup consists of a given frame (casing) size and an associated family of impeller designs. A family of impellers is typically defined as a number of impellers with identical aerodynamic blade and disc flow path geometry (Figure 2).

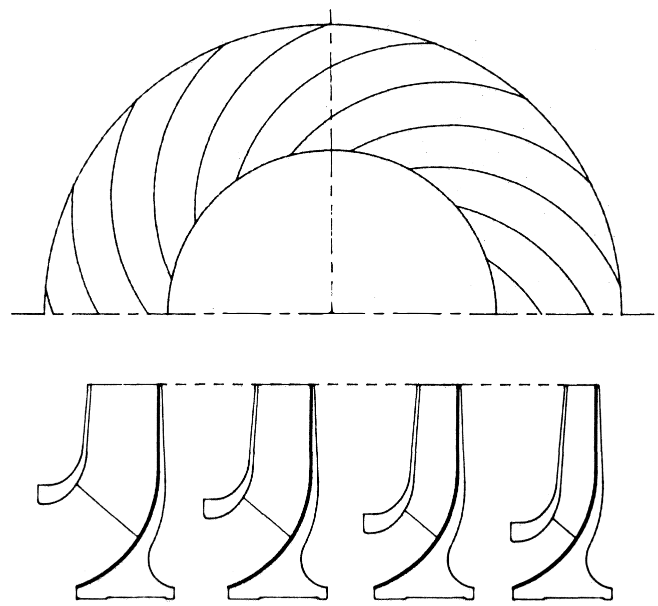

Figure 2. Typical Impeller Family.

The process of scaling an established set of impeller designs allows a manufacturer to vary the flow capacity and head of a given lineup with a minimum of effort. This is done by scaling an existing lineup to establish a new full-size model of the impeller and stage geometry being developed. This model is then tested in its as-to-be manufactured size to determine its performance.

The practice presented herein borrows from that theme. However, it differs significantly in that it scales the stage hardware into a group of aerodynamically similar components that will fit into a fixed-size test rig, much the same ways as scale models of airplanes are tested in fixed-size wind tunnels. This permits testing of nearly an infinite number of unique designs while meeting the constraints of a commercially driven production schedule.

ASME PTC-10 Requirements

The performance of each stage is conducted in accordance with the ASME Power Test Code 10 (PTC-10) [2] for compressors and exhausters. The test equivalent speed is calculated per section 4.51 of PTC-10. Deviations from the actual code are presented below:

- Performance is calculated on an individual stage basis rather than inlet to discharge as addressed in the code.
- Driver torque, electric motor input, cooling water inlet temperature, cooling water flowrate, and line voltage will not be measured as they are not used in any of the performance verification calculations.
- Modern electronic instrumentation with an automatic data acquisition system are used instead of gauges, a mercurial barometer, and potentiometer type thermocouple readout device.
- Efficiency is determined by heat balance.
- Enthalpy is calculated using the Redlich-Kwong equation of state for the test gas.

• Gas specific gravity is not monitored due to the fact that pure gases are used for the model test. Gas samples are taken before and after the test to verify gas composition.

Instrument calibration consists of the following:

• Thermocouples are made of premium grade single melt of wire. Samples of each batch are calibrated against an N.I.S.T. traceable RTD prior to the test. Online calibration verification is made using eight sample thermocouples in an elevated temperature bath along with three sample thermocouples in an ice point reference.

• The barometric pressure device is calibrated on an annual basis. Comparisons with the local branch of the National Weather Service station (located approximately one mile from the test site) are made within four hours of each test.

• Pressure transducers are continuously calibrated at atmosphere and full scale pressure. N.I.S.T. traceable digital gages are used to measure full scale pressure and communicate with the computer.

Data fluctuations during the test conform to Table 2 in the test code. Automatic averaging and exception indication are used during the test to highlight any over-limit deviations. Flow is measured with an ASME long radius flow nozzle. Test results are calculated per Class III procedures using the Redlich-Kwong equation of state to calculate enthalpy.

During the actual test, all data points are displayed on line (real time) against the predicted stage nondimensional performance curves. The test consists of five equally spaced data points taken along the design equivalent speed line. The range of the points is from overload (choke flow) to surge. The final data point is taken at approximately five percent higher in flow after the actual surge flow is established. Two additional speed lines of five points are taken at 105 percent and 95 percent of design equivalents speed. The alternate speed lines give information as to the behavior of the stage with variations in tip relative mach number.

Scaling of Components

Two commonly used terms to describe the flow and head coefficients are the quantity constant and μ_{poly}, respectively. These terms are defined as:

$$\text{Quantity Constant} = \frac{Q}{N\,D^3} \qquad (1)$$

where:

Q = Inlet Volume Flow (ICFM)
N = Speed (RPM)
D = Impeller Diameter (Ft), and

$$\mu_{poly} = \frac{H_{poly}\, g}{U_2^2}$$

where:

H_{poly} = Polytropic Head Rise (Ft-lb/lbm)
g = gravitational constant (32.2 ft/sec)
U_2 = Impeller Tip Speed (Ft/Sec).

In order for the scaled hardware to be aerodynamically similar to the full scale hardware, both must have identical values for quantity constant and μ_{poly}. These requirements are met by maintaining identical inlet and outlet blade and flow path geometries between the scaled and full-sized hardware. While angular dimensions remain unchanged, linear dimensions vary with the scale factor, area dimensions vary with the scale factor squared and volume dimensions vary with the scale factor cubed. The design point is calculated on the basis that the speed increases as the inverse of the scale factor and the flow decreases as the square of the scale factor.

Selection of the Test Gas

The selection of the test gas is dictated by the restrictions imposed by PTC-10. The code dictates that the volume ratio and impeller tip relative mach number of the test case must match the actual machine conditions within plus or minus five percent. The preferred test gases are carbon dioxide or nitrogen. In some cases, when simulating heavier process gases, it becomes necessary to use a heavy refrigerant in the test loop.

SINGLE STAGE TEST PROCEDURE AND RESULTS

Description of Test Rig Hardware and Data Acquisition System

The test rig used throughout the studies presented in this paper is capable of testing scale models of a single centrifugal compressor stage. A stage in a centrifugal compressor consists of an inlet, an impeller, a diffuser (either vaned or vaneless), a crossover section, a return channel and a discharge (Figure 3). The scale models are based upon unique impeller designs. The rig can accept up to a 15 in diameter impeller. A 200 horsepower variable speed electric motor drives the rig through a speed increasing gear to a maximum speed of 20,000 rpm. Testing is performed in a closed loop (Figure 4) with an inert gas to properly match the stage volume ratios as defined by PTC-10.

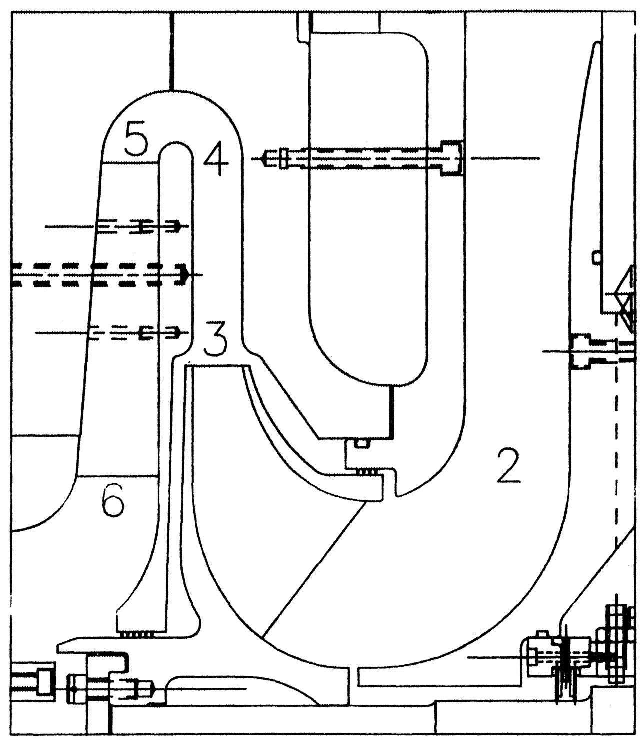

Figure 3. Stage Instrumentation Locations.

The single stage test rig described here is used primarily to test actual stages in the course of hardware contracts, so it was designed for rapid assembly and disassembly (Figure 4). Thus, the aerodynamacist and test engineer can quickly redesign and interchange hardware. The basic rig design also follows a modular concept, where each of the key components is manufactured as a

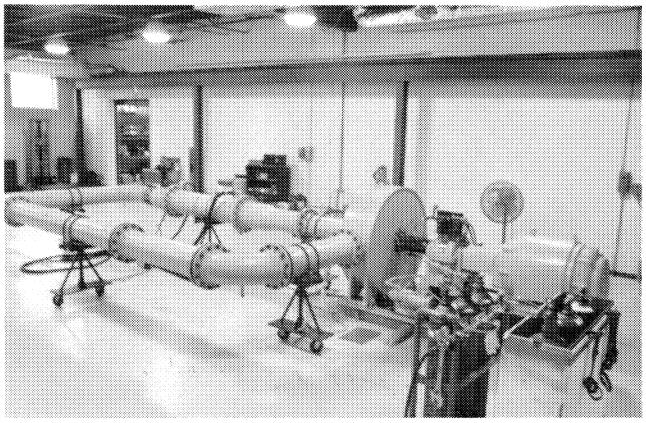

Figure 4. Single Stage Test Rig.

separate element. This allows testing on variations of any one basic design. For instance, if the results of the test indicate that the diffuser passage is mismatched to the impeller design, a new diffuser plate may be quickly manufactured, installed and the stage retested.

Parameters used to Scale Test Hardware

The challenge of scaling aerodynamic components for use in the test rig arises from two areas. First, the scale factors, along with the test gas and operating speed, must yield an acceptable aerodynamic similarity to the full size hardware. Second, the scaled down hardware must fit within the physical confines of the single stage test rig. In the initial design phases of the test rig, a decision was reached to develop a system capable of testing impellers up to a maximum of fifteen in in diameter. Having fixed this parameter, a casing was designed and built to allow adequate diffuser ratio for such an impeller. Other limitations affecting the scale factors are the 200 available horsepower and 20,000 rpm speed limitation of the electric motor driver. Test speed is further influenced by the centrifugal stress levels in the brazed aluminum impellers and the rotordynamics of the overhung rotor supported on antifriction bearings.

Model tests will produce performance results identical to full sized hardware if geometric, kinematic, and dynamic similarity are maintained. Geometrical similarity means that linear dimension ratios are identical everywhere. Kinematic similarity means that velocity ratios are the same throughout the stage. And dynamic similarity means that the ratio of forces is the same everywhere.

Test Loop and Data Acquisition System

In order to accurately measure component performance, several flowpath stations must be defined, with enough measurements taken at each station to make the data statistically significant (Figure 3). Stations two and six are the measurement planes for overall stage performance with four each total pressure and temperature probes and four static pressure taps at each station. For determining component performance (impeller, diffuser, return channel), total and static pressures are also measured at other locations along the flowpath.

The necessity of capturing such a large volume of measurements results in the need for a data acquisition system (DAS) which is capable of measuring over 250 steady state pressures, temperatures and miscellaneous channels. A fully automated scanning system is used for this purpose (Figure 6).

The DAS employs online pressure and temperature calibrated standards to ensure system accuracy even as ambient conditions vary. The pressure standards are used on every scan to recalibrate

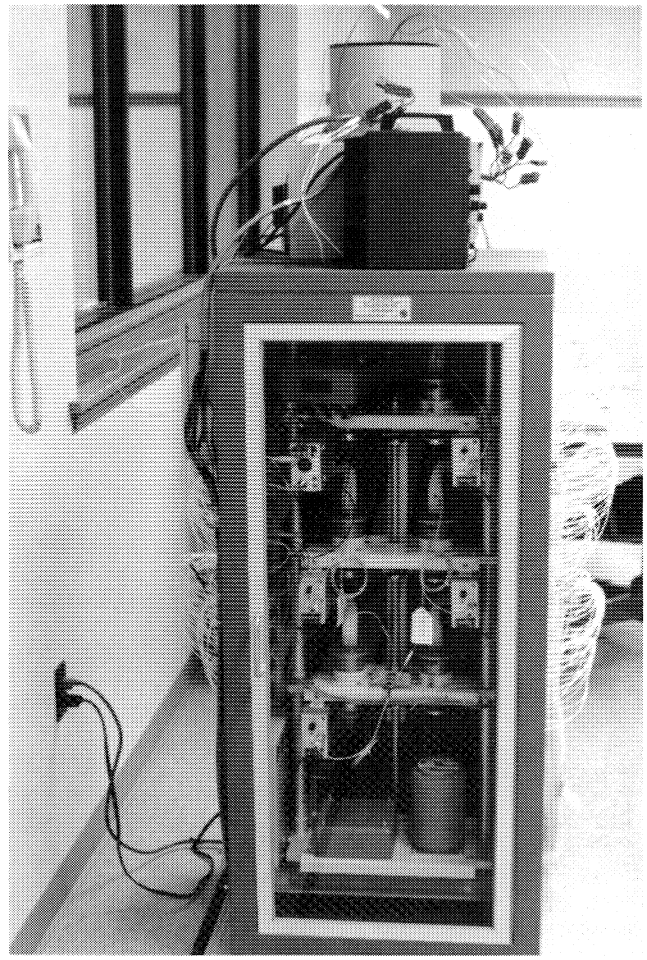

Figure 5. Test Rig Data Acquisition System.

the pressure transducers, while the temperature standard is used continuously to verify the accuracy of the temperature calibration. In addition, total temperature probes are corrected for the recovery factor based on a calibration method which includes mach number correction. ASME flow nozzle weight flow calculations, overall stage performance, individual component performance, impeller 2-D tip calculations with Reynold's number corrections and corrections to specified conditions (including gas composition) are also performed for each scan. In addition, probable uncertainties for both the primary and selected measurements are calculated.

The entire process of scanning 250 channels, applying the online calibration standards and collating the filtered station values and standard deviations is completed in approximately 15 sec. The results are presented in real time by plotting each point on the input stage characteristic performance curve. The remainder of the data may be presented on one of four possible display pages.

Application of Test Results to Overall Performance Model

A proprietary stage-by-stage computer performance program is used to develop all of the compressor designs presented here. The program requires that the designer input performance parameters such as flow, polytropic head, characteristic curve shape and polytropic efficiency for each stage. These parameters are initially based upon the empirical and theoretical experience of the designer. Once the designer finalizes the performance parameters required to meet the contract specifications, the aerodynamicist sets out to design a stage that will meet those parameters. The aerodynamicist uses his experience and any one or a combination of proprietary and commercially available design programs to develop the final design of the inlet, impeller, diffuser, and return channel.

The results of the single stage test serve to verify or modify the performance parameters selected by the designer. After the test, these new parameters are entered into the stage-by-stage computer model which calculates overall compressor performance. Since it is not practical to test all of the stages for a given compressor, the test results are extrapolated to cover other stages in the same impeller family. Usually, the results are used as an exact multiplier for all of the initial parameters. However, in some cases it is necessary to interpret the data with respect with other empirical factors not directly associated with the test to develop the appropriate modifications.

The end result is a new stage-by-stage model for the design in question. The contract requirements are then run with the new ratings to determine the final predicted performance. If necessary, stage designs are modified and retested until the desired results are achieved. In ideal cases, all of this work is completed prior to the manufacture of the full scale hardware.

RESULTS OF SCALE MODEL TESTS

Intercooled Wet Gas Compressor

The first test ever conducted in the rig was an investigation of several new stages proposed for use in an intercooled wet gas compressor. The compressor initially consisted of six stages of compression in a 3/3 split between the first and second sections. The original design used two dimensional blade shapes in all stages. The goal of the rerate was to achieve a 14 percent increase in flow with an accompanying three percent increase in discharge pressure. In the first section, the existing first and second stages were removed and replaced with newly designed, three dimensional bladed impellers with new diaphragms. The existing third stage impeller was retained. The existing fourth and fifth stage impellers were also retained. However, a new fourth stage return channel was designed in an effort to gain additional stage efficiency. The rerated compressor became a five stage configuration with a 3/2 split between the first and second sections (Figure 6).

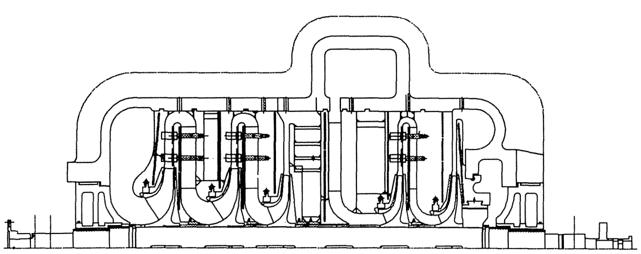

Figure 6. Rerated Intercooled Wet Gas Compressor Configuration.

Scale model testing was conducted on the new first and second stage hardware configurations. A scale factor of 0.36 was used for both stages. For the first stage, this resulted in a test impeller diameter of 10.80 in *vs* 30.00 in for the proposed full size component (Figure 7). Additional tests were conducted on the fourth stage with both old and new return channel designs. The graphic results from the first stage test are shown in Figure 8. The curves are the original predictions, while each mark indicates a test point. A comparison between the two shows that the actual performance is greater than predicted over the entire operating envelope. Results of the other tests for this compressor are not shown as they are very similar to Figure 8. The details of the four tests performed

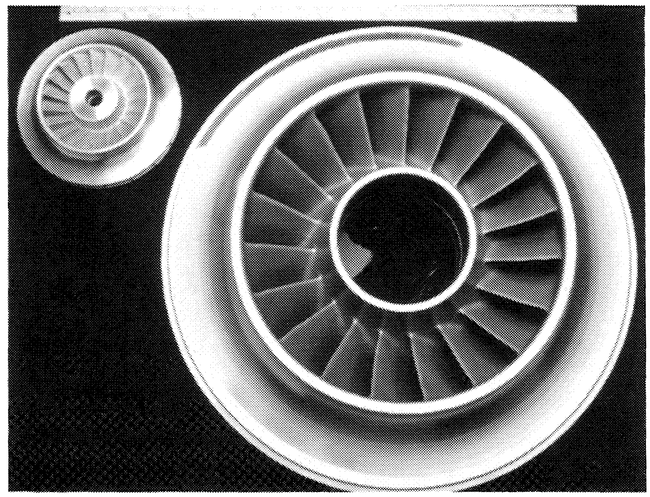

Figure 7. Comparison of Model Test Impeller with Full Size Impeller.

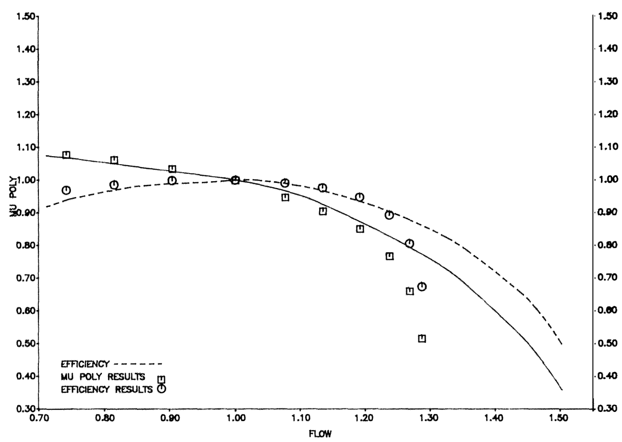

Figure 8. Intercooled Wet Gas Compressor Single Stage Test Results.

for this rerate are listed in Table 1, as are the values for μ_{poly} and η_{poly} at each design point. As can be seen, each stage demonstrated higher than predicted levels of polytropic work and efficiency.

The data obtained from the model tests were reduced to the necessary parameters of flow, polytropic head, polytropic efficiency, and curve shape. The results were loaded into the computer performance model developed for the rerated compressor, which indicated that the compressor would operate at a lower speed than originally predicted with a significant increase in efficiency. Based upon these results, a set of final performance curves and data sheets were developed.

Two-Body Wet Gas Compressor

The second testing program presented in this paper involves the rerate of a two-body wet gas compressor train. The existing machines consisted of two casings containing three stage each. The goal of the rerate was to increase the mass flow through the train by 20 percent. The proposed solution involved replacing the first stage of the first casing (Figure 9). The only other change involved replacing the existing prerotation guide vane ahead of the second stage of the second casing with a radial bladed guide vane.

Scale model testing was conducted on the new first stage hardware configuration and on the second casing's second stage to verify the changes in the guide vanes. Like the earlier tests, the

Table 1. Intercooled Wet Gas Compressor Comparison of Predicted Vs Tested Performance

INTERCOOLED WET GAS COMPRESSOR				
Stage	1st	2nd	4th-Old Return	4th-New Return
Design O.D.	30.0	30.0	27.5	27.5
Test O.D.	10.8	10.8	9.9	9.9
Scale Factor	0.36	0.36	0.36	0.36
Design Gas	Wet Gas	Wet Gas	Wet Gas	Wet Gas
Test Gas	Freon 22	Freon 22	Freon 22	Freon 22
Design Mach #	0.878	0.848	0.783	.783
Test Mach #	0.899	0.872	0.822	.822
Design μ_{poly}	0.502	0.536	0.594	.594
Test μ_{poly}	0.554	0.605	0.615	.631
% Difference	110%	113%	104%	106%
Design η_{poly}	0.75	0.78	0.742	0.742
Test η_{poly}	0.83	0.84	0.780	0.793
% Difference	111%	108%	105%	107%

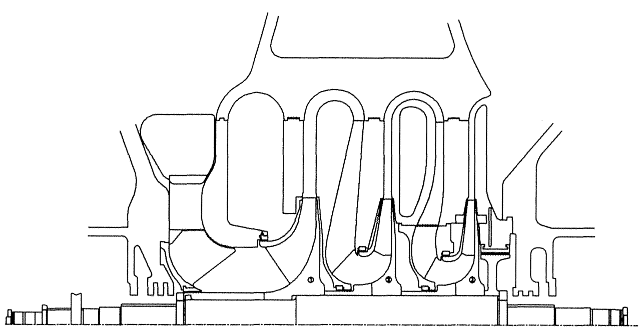

Figure 9. Rerated First Secton of Two-Body Wet Gas Compressor Configuration.

actual performance was significantly better than predicted. The graphic results for the first section's first stage are shown in Figure 10. The detailed results are presented in Table 2.

Intercooled Light Ends Compressor

The final testing program presented in this paper involved the rerate of an intercooled light ends compressor. The machine

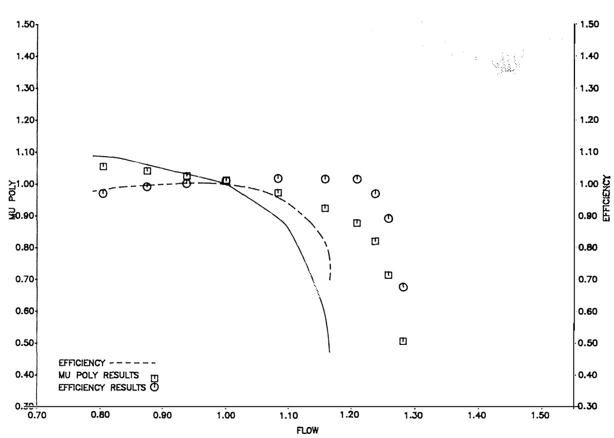

Figure 10. Two Body Wet Gas Compressor Single Stage Test Results.

Table 2. Two-Body Wet Gas Compressor Comparison of Predicted Vs Tested Performance

TWO-BODY WET GAS COMPRESSOR		
Stage	1st	2nd-New I.G.V.
Design O.D.	43.5	43.0
Test O.D.	10.8	14.19
Scale Factor	0.25	0.33
Design Gas	Wet Gas	Wet Gas
Test Gas	Freon 114	Freon 12
Design Mach #	1.018	0.962
Test Mach #	0.991	0.961
Design μ_{poly}	0.519	0.496
Test μ_{poly}	0.577	0.586
% Difference	111%	118%
Design η_{poly}	0.787	0.731
Test η_{poly}	0.831	0.769
% Difference	106%	105%

originally consisted of eight stages of compression in a 4/4 split. The goal of the rerate was to increase the mass flow through the train by almost 100 percent. The proposed solution involved replacing nearly all of the aerodynamic components in the compressor. The rerated design would configure the compressor with six stages in a 3/3 split (Figure 11). All of the new impellers would have two dimensional bladings.

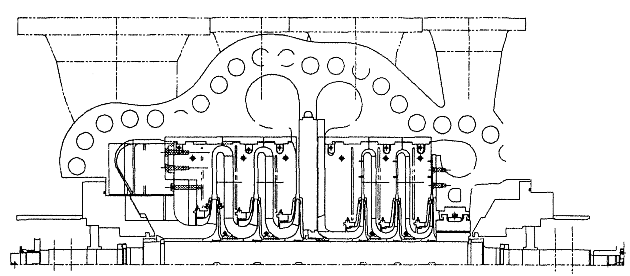

Figure 11. Rerated Light Ends Compressor Configuration.

The new design consisted of two different families of impellers. The first section consisted entirely of one design, while an alternate design made up the second section. It was decided to performance test a model of the first section's first stage and the last stage of the second section. Due to the fact that the sixth stage discharges into a volute section rather than a diffuser return channel, it was configured with a volute. However, the machine was online and a dimensional inspection of the existing volute was not possible, so a conservatively designed volute was developed and installed into the rig for that test.

The first stage performed much better than predicted, but the sixth stage did not perform well at all. The sixth stage's performance was well below that predicted for all of the operating range, as is shown in Figure 12. The details of the two tests are shown in Table 3. The lower performance was initially attributed to the volute configuration. In order to fully investigate this possibility,

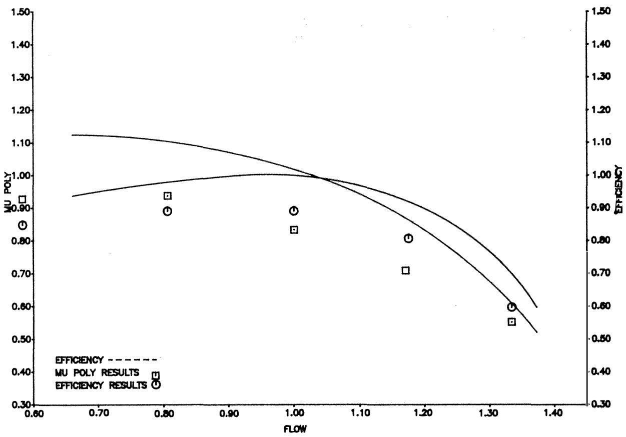

Figure 12. Light Ends Compressor Single Stage Test Results.

a conventional return channel was designed and built around the sixth stage impeller. For all practical purposes, the results of the stage with the return channel were identical to the results obtained with the volute.

The data obtained from the model test were reduced to the necessary parameters of flow, polytropic head, polytropic efficiency, and curve shape. The results were loaded into the computer performance model developed for the rerated compressor. The tests indicated that the compressor would still meet the customer's specified operating conditions. Based upon the new computer performance model, a set of final performance curves and data sheets were developed. These were to become the basis for evaluation of the later field performance test of the finished compressor.

REQUIREMENTS FOR A LIMITED FIELD PERFORMANCE TEST

Measurement Requirements

After a successful rerate of a compressor, the user will often wish to verify its performance after startup. However, a full-scale PTC-10 field test is usually too expensive to be a consideration, as this involves redundant instrumentation, careful positioning of the sensors, control of the process gas and a host of other details. It is often preferable to just use the existing instrumentation in its current location and take into account the field variables when evaluating the accuracy of the results.

Table 3. Intercooled Light Ends Compressor Comparison of Predicted Vs *Tested Performance*

INTERCOOLED LIGHT ENDS COMPRESSOR		
Stage	1st	6th
Design O.D.	17.00	17.00
Test O.D.	12.75	13.77
Scale Factor	0.75	0.81
Design Gas	Wet Gas	Wet Gas
Test Gas	Freon 22	Freon 22
Design Mach #	0.739	0.700
Test Mach #	0.742	0.709
Design μ_{poly}	0.528	0.483
Test μ_{poly}	0.552	0.398
% Difference	105%	82%
Design η_{poly}	0.790	0.731
Test η_{poly}	0.855	0.769
% Difference	108%	90%

The instrumentation requirements for the limited field performance test are very simple. The equations require only the inlet and discharge temperatures and pressure, flow through the compressor and the gas composition. The sensors should be as close a possible to the compressor flanges, but not immediately downstream of an elbow or bend in the pipe. The flowmeter can be either upstream or downstream of the compressor and should read the total flow through the machine, before any sidestream enters or exits the pipe. A standard configuration for the compressor that meets the minimum field test requirements is outlined in Figure 13.

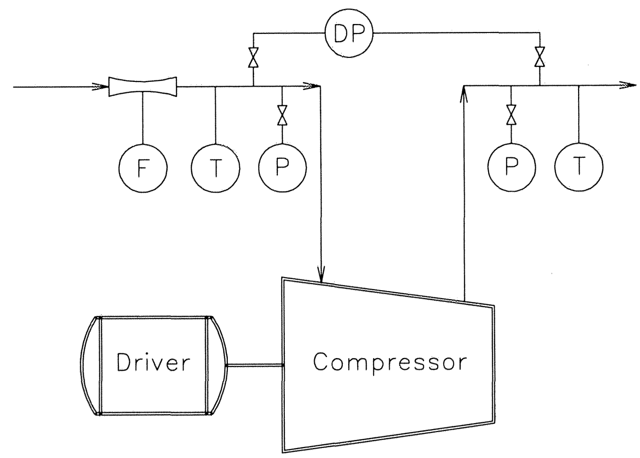

Figure 13. Basic Field Test Instrumentation Setup.

This simple setup does not use redundant instrumentation, so if a single pressure or temperature transducer is reading incorrectly, the calculations will be thrown off. However, it is a common practice to measure the differential pressure across the compressor independently of the inlet and discharge pressure. By comparing the sum of the inlet and differential pressures to the discharge pressure, one can verify the accuracy of pressure measurements. Well-calibrated meters should compare within ± 0.5 to 1.0 percent. This is not normally done with temperature measurements, so special care should be taken to properly test and calibrate these transducers during shutdowns and turnarounds.

It is also very important to configure and calibrate the flowmeters to be as accurate as possible. Calculating the flow from an orifice- or venturi-type flowmeter is a science unto itself as many different variables are used in the equations. Even if the setup is good and the calculations done correctly, a flowmeter could be inaccurate if the gas composition changes significantly. Most flowmeters are set to measure a gas with the specific gravity outlined in the data sheets. So if the actual specific gravity is different or varies with the process, the reading will not reflect the actual flow. The following equation can be used to correct the indicated value [3].

$$Q_a = Q_m \sqrt{\frac{SG_{ds}}{SG_a}} \qquad (2)$$

where:

Q_a = actual flow

Q_m = measured flow

SG_{ds} = specific gravity from data sheets
SG_a = actual specific gravity.

Gas Analysis

The most crucial factor in an good field performance test is obtaining an accurate gas analysis. For machines that run on air or on other well-defined gases this is not a problem. But for the many compressors that are used in a mixed hydrocarbon service, the gas composition may change according to the feed inputs into the unit, such as running with a different crude slate, or it may change according to cyclic process conditions, such as the wet gas compressor on a delayed coking unit. The information needed from the gas analysis is the mole weight of the gas and the compressibility. The specific heat ratio, or k value, is also important when determining the efficiency. However, the efficiency equation is extremely sensitive to the temperature measurements, so a small error in a temperature reading changes the calculated efficiency by several points. And although efficiency is important, it mainly impacts the horsepower requirement. So efficiency often becomes a secondary consideration as long as the driver is capable of meeting the compressor's demands.

Performance Equations

The following equations are found many publications [4] and are presented again as a matter of convenience to the reader. These equations were used to verify the performance of the rerated compressors in normal operation.

Conversion of Flow

A typical English unit for flow in a process system is MSCFD. All the compressor sees is the actual flow, which is in ACFM. To convert to from MSCFD to ACFM, use the following equation:

$$Q\ (SCFM) = Q\ (MSCFH) * \frac{P_s T_1\ 1000}{P_1 T_s\ 60} \tag{3}$$

where:

P_s and T_s = standard pressure and temperature, respectively
P_1 and T_1 = actual inlet pressure and temperature, respectively

Polytropic Constant Relationship

The polytropic constant, n, is used several times in the head equation in the following relationship, (n-1)/n. (n-1)/n can be found either from the gas specific heat ratio and the compressor efficiency, or directly from the pressure and temperature measurements using the following equation:

$$\frac{(n-1)}{n} = \frac{\ln \frac{P_2}{P_1}}{\ln \frac{T_2}{T_1}} \tag{4}$$

where:

P_1 and T_1 = absolute inlet pressure and temperature, respectively.
P_2 and T_2 = absolute discharge pressure and temperature, respectively.

Polytropic Head Equation

Finally, the polytropic head can be calculated using the previous information and the following equation:

$$H_{poly} = Z_m \frac{1545}{MW_m} T_1 \frac{\frac{P_2}{P_1}^{\frac{n-1}{n}} - 1}{\frac{n-1}{n}} \tag{5}$$

where:

Z_m = measured compressibility
MW_m = measured molecular weight
P_1 and T_1 = absolute inlet pressure and temperature, respectively
P_2 = absolute discharge pressure

The performance of the compressor is verified by comparing the head and flow at a certain time against the performance curves generated with the stage-by-stage performance model. The accuracy of the calculations depends completely upon the quality of the measurements, but with good data, the points should fall within 5.0 percent of the predicted performance.

RESULTS OF FIELD VERIFICATION TESTS

Intercooled Wet Gas Compressor, Variable Speed

The test for the intercooled wet gas compressor consisted of six points taken at ten minute intervals for an hour. An oversight was made in that gas samples were not taken during the test period. However, earlier samples had shown that the gas composition was nearly constant with time with the molecular weights approximately five percent below those listed on the data sheets. The results for both sections are shown in Figure 14 and Figure 15. For the first section, the actual performance appears to lie above the anticipated values, while for the second section the points fall on or just below the predictions.

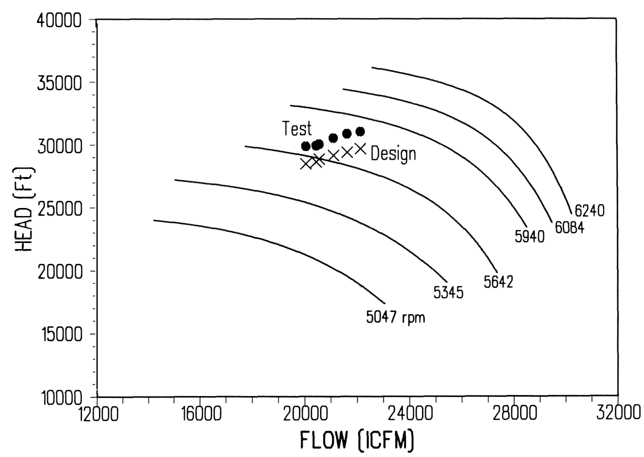

Figure 14. Intercooled Wet Gas Compressor, Section One Field Test Results.

Two-body Wet Gas Compressor, Variable Speed

The test for the two-body compressor involved only one point which was taken coincident with the gas sample. The primary flow measurement had to be corrected as the reading was inconsistent with the pressure ratio and with other secondary flow measurements. Once this was done, the measured performance compared well with the curve calibrated with the single stage rest results, as shown in Figure 16 and Figure 17.

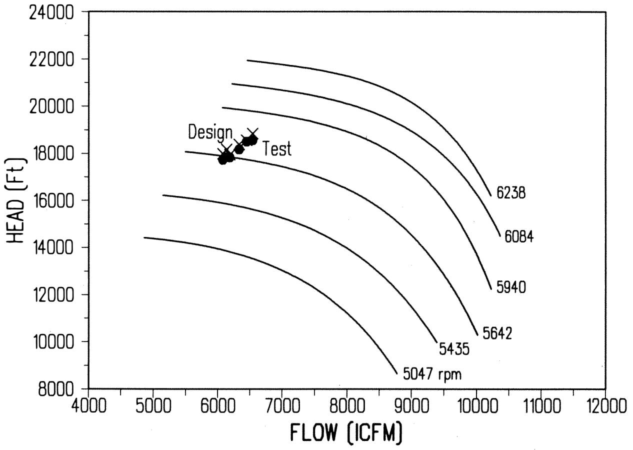

Figure 15. Intercooled Wet Gas compressor, Section Two Field Test Results.

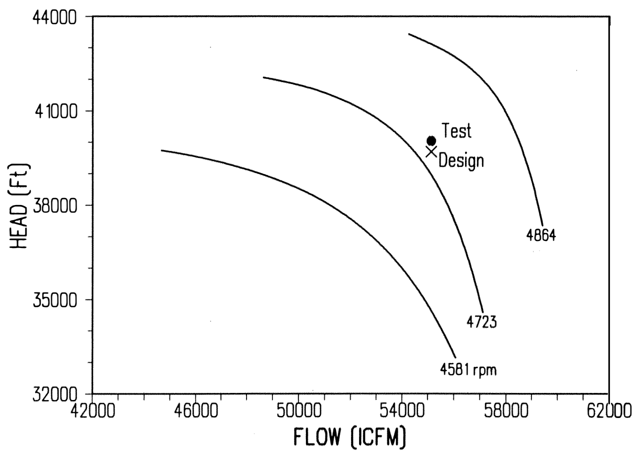

Figure 16. Two-body Wet Gas Compressor, Section One Field Test Results.

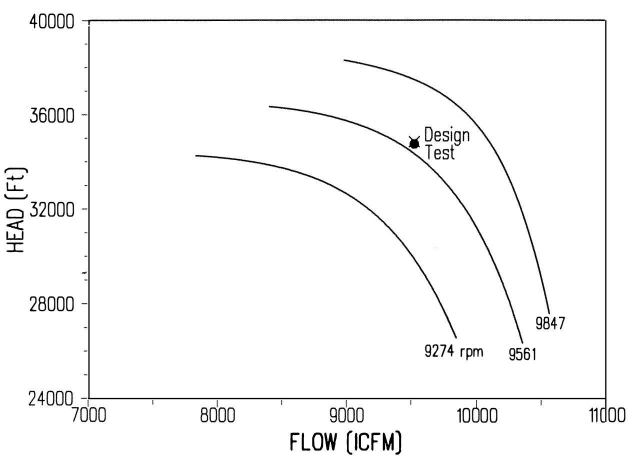

Figure 17. Two-Body Wet Gas Compressor, Section Two Field Test Results.

Intercooled Light Ends Compressor, Constant Speed

A great amount of difficulty was encountered when trying to determine the properties of the gas flowing through this compressor. Part of this came from the variable nature of the service (this compressor was used in a delayed coking unit), where the molecular weight of the gas varies by as much a five points over a 12 hr cycle. In the second place, the quality of the sampling procedure was suspect as there appeared to be a high degree of inconsistency among the results. And finally, the actual gas composition was very different from that originally listed on the data sheets, which caused additional complications.

To overcome these problems, an attempt was made to correlate the gas sample mole weight data with one of the process variables. A satisfactory relationship was found between the mole weight and the flow into the first section. This is shown in Figure 18. Armed with this information, the compressor performance could be calculated for each hourly data point, regardless if a sample was taken or not. The results of these calculations are shown in Figure 19 and Figure 20 for sections one and two, respectively. In Figure 19, the first section performance is very close to the predicted. However, in Figure 20, the performance is quite a bit above the performance curve, even more than can be explained from the measurement tolerances. The reason is that the calculated second section molecular weight is too low, which forces the head value to be artificially inflated The accuracy of the second section calculations will improve as the molecular weight *vs* flow model is further refined.

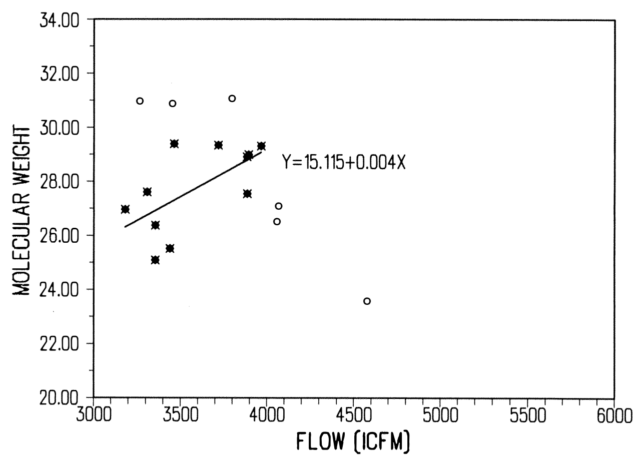

Figure 18. Intercooled Light Ends Compressor, Mole Weight Vs. *Section One Flow.*

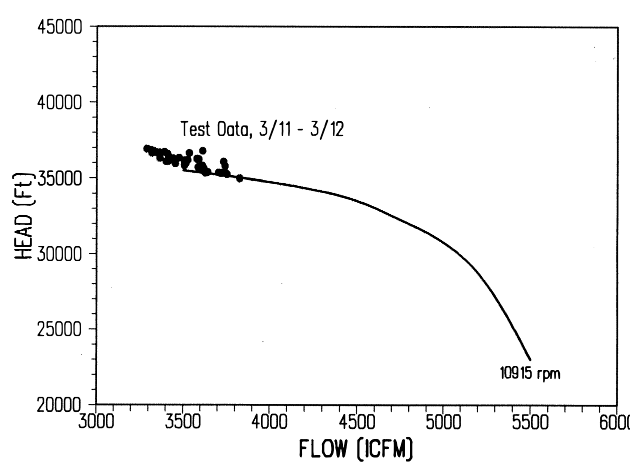

Figure 19. Intercooled Light Ends Compressor, Section One Field Test Results.

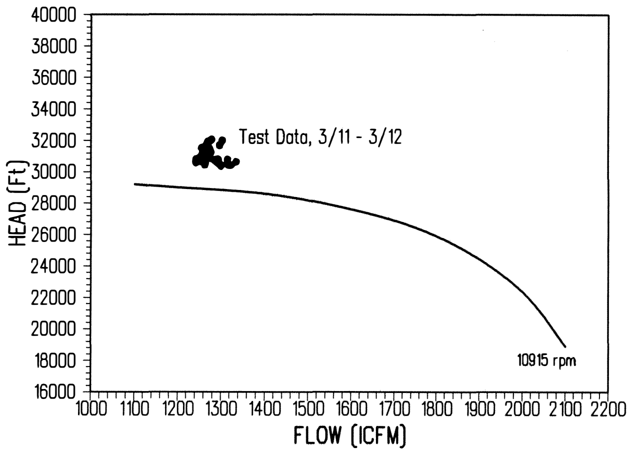

Figure 20. Intercooled Light Ends Compressor, Section Two Field Test Results.

CONCLUSIONS

The results of the scale model testing confirm that, even with the use of state of the art computer aerodynamic modelling software, perfect prediction of compressor performance is not yet possible. Although none of the results of the test deviated significantly from the initial predictions, enough error has been discovered to warrant slight changes to the stage designs prior to manufacture of full scale hardware. Until analytical prediction capabilities improve to the point of being entirely accurate in all situations, single stage testing will remain the only method of ensuring guaranteed performance of stage hardware prior to manufacture.

Single stage scale model testing coupled with limited field testing has also proven to be a cost-effective method of verifying the overall performance of rerated compressors. In each case presented here, the overall compressor performance was demonstrated to meet or exceed the predicted values (which had been calculated with the single stage test results). Used in combination, these two methods offer a degree of confidence that the rerated compressor will meet its performance requirements before it is built, and also offer a way to verify and track its performance after it comes online.

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