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SITE PERFORMANCE TESTING OF CENTRIFUGAL COMPRESSORS AND GAS TURBINE DRIVERS

Rainer Kurz
Manager, Gas Compressor Engineering
Solar Turbines Incorporated
San Diego, CA, USA

Kevin Davis
Regional Field Engineer
Solar Turbines Incorporated
San Diego, CA, USA

Roland Kaiser
Consulting Engineer
Solar Turbines Incorporated
San Diego, CA, USA

Marybeth McBain
Sr. Engineer and Sr. Project Manager
Kinder Morgan
Houston, TX, USA

Klaus Brun
Director R&D
Elliott Group
Jeannette, PA, USA

Dr. Rainer Kurz is the Manager, Gas Compressor Engineering, at Solar Turbines Incorporated in San Diego, California. He has been with Solar Turbines for 28 years. Dr. Kurz attended the Universitaet der Bundeswehr in Hamburg, Germany, where he received the degree of a Dr.-Ing. in 1991. Dr. Kurz was elected ASME Fellow in 2003. He has authored numerous publications on turbomachinery related topics, with an emphasis on centrifugal gas compressors, compressor applications, dynamic behavior, and gas turbine operation and degradation. Many of his publications were considered of archive quality and were accepted for publications in Engineering Journals. He has received several 'Best Paper' and 'Best Tutorial' Awards at the ASME TurboExpo Conferences and is the recipient of the 2013 ASME Industrial Gas Turbine Technology award. He has authored numerous publications about turbomachinery related topics, is an ASME fellow, holds 2 patents, and is a member of the Turbomachinery Symposium Advisory Committee.

Kevin Davis is a Senior Regional Field Engineer at Solar Turbines Incorporated, He is responsible for supervising, organizing, conducting and evaluating site performance tests worldwide. He has over 20 years of experience in his field.

Roland Kaiser is a Consulting Engineer at Solar Turbines Incorporated. He joined Solar Turbines in San Diego, California, in Nov. 2000. After getting acquainted with the design philosophy of Solar's compressor designs, he joined the Systems Analysis Group, where he is now supporting sales with compressor and gas turbine solutions. In 1994, he took on a Senior Design Engineer-Analyst position with Sulzer Bingham Pumps, in Portland, Oregon, which included design responsibilities for new products. In 1990, he joined the Mechanical Development Group of

Sulzer Pumps Headquarters. He joined the Acoustic & Vibration Department of Sulzer Innotec in 1987, and took on activities in the areas of active noise suppression and vibration analysis and testing. Mr. Kaiser graduated from Engineering College (HTL) in Winterthur, Switzerland with a Degree in Mechanical Engineering.

Dr. Brun is the Director of Research & Development at Elliott Group where he leads a group of over 60 professionals in the development of turbomachinery and related systems for the energy industry. His past experience includes positions in product development, engineering, project management, and executive management at Southwest Research Institute, Solar Turbines, General Electric, and Alstom. He holds nine patents, authored over 350 papers, and published three textbooks on energy systems and turbomachinery. Dr. Brun is a Fellow of the ASME and won an R&D 100 award in 2007 for his Semi-Active Valve invention. He also won the ASME Industrial Gas Turbine Award in 2016 and 11 individuals ASME Turbo Expo Best Paper awards. Dr. Brun is the chair if the 2020 Supercritical CO2 Power Cycles Symposium, past chair of the ASME-IGTI Board of Directors, the ASME Oil & Gas Applications Committee, and ASME sCO2 Power Cycle Committee. He is also a member of the API 616 Task Force, the ASME PTC-10 task force, the Asia Turbomachinery Symposiums Committee, and the Supercritical CO2 Symposium Advisory Committee. Dr. Brun is currently the Executive Correspondent of Turbomachinery International Magazine and Associate Editor of the ASME Journal of Gas Turbines for Power.

Marybeth McBain works as a Senior Engineer / Sr. Project Manager in the Kinder Morgan Midstream group. Marybeth's primary functions involve evaluating new compression projects for gas transmission, gas storage and gas processing plants as well as recommending upgrades and compressor modifications to suit changing pipeline needs. In 2018-2019, Marybeth served as the lead compression engineer for the new build-out of the 450 mile Gulf Coast Express pipeline, which included over 150,000 horsepower in both turbines and reciprocating units.
Mrs. Mcbain previously worked at Southwest Research Institute for ten years in the pulsation analysis and turbomachinery groups as well as Apache Corporation for four years as a senior facility engineer. She holds four patents mainly related to innovative pulsation control concepts and has published over thirty publications related to natural gas machinery. Her career is supported by a BSME from the University of Texas at Austin and an MSME from Georgia Tech.

ABSTRACT

While factory performance tests for compressors and gas turbines are well defined in ASME power test codes, and can be performed under well controlled conditions, site performance tests must be performed under the conditions available at site. This requires methods to bring the compressor to valid operating conditions, as well as the capability to provide steady state operating conditions. Understanding compressor controls, available recycle and cooling capability, and the impact of the process requirements are crucial for successful tests. Limitations may come from the instrumentation that can be installed based on the site installation, and the impact of limited instrumentation is discussed. Compressor drivers also may impose limitations but also offer the capability for redundant measurements of certain parameters. Due to these constraints, the appropriate calculation of test uncertainties is vital, and the concepts of test uncertainty calculations is discussed and illustrated. This paper also provides insight into the necessary instrumentation and its calibration, the requirement for obtaining an accurate gas composition, and methods to obtain representative test points, as well as troubleshooting advice and methods to verify data.

INTRODUCTION – WHY AND WHAT

All engineering analysis is fundamentally based on experimental test results. A site performance test on centrifugal compressors and their drivers is a typical example of such a task. The specific challenges of site performance tests for centrifugal compressors, driven by variable speed capable drivers such as two-shaft gas turbines or variable speed

electric motors (Figure 1) arise from the fact that the test conditions cannot be controlled to the same degree as is possible in a laboratory or a factory test setting. Test sites not only limit the amount and locations for the instrumentation, the achievable operating conditions, or the stability of the process, they also, in general, do not allow taking multiple data points at exactly the same conditions. Lastly, repeating the test with a number of identical installations is not feasible (exceptions will be discussed later). Sometimes the opportunity exists to test the same equipment in different test setups, for example if both a factory test and a site performance test are performed [1]. The purpose of a site test can be for the purpose of comparisons with factory test data, but also for a stand alone performance evaluation, or as part of a site evaluation or condition monitoring effort. In this tutorial, the primary focus is on acceptance tests. However, the described procedures are also applicable for other testing purposes.
Therefore, a practical and generally acceptable solution for site performance tests is to perform the test based on the conditions at site and account for the prevailing conditions by calculating the resulting test uncertainties. This topic has been discussed in the literature and in performance test codes [1-6].

Figure 1: Gas turbine driven centrifugal compressors

Proper testing of any physical system, given that all test variables are tightly controlled, should always yield a singular answer. In reality, test conditions are not constant and instruments readouts deviate from the 'true' physical property to be measured. In this paper, the set of agreed upon operating conditions will be referred to as 'acceptance conditions'. The set of test conditions available during the site test will be referred to as 'test conditions'.
We also need to account for the fact that in many instances the exact operating condition that was agreed upon as the condition to be verified cannot be repeated due to the conditions at site. For example, the composition of the gas or the ambient temperature might be different between acceptance conditions and test conditions. In this case, an appropriate method is necessary to either determine:

-What would the performance of the test vehicle, based on test data at the prevailing operating conditions ('test conditions'), be at the acceptance conditions, or

-What would the performance of the test vehicle be at the prevailing test conditions, based on the performance at the agreed upon conditions.

Both methods are roughly equivalent but any correction method will introduce a systematic error. This error will increase the larger the difference between the agreed upon and the as tested conditions becomes.
To address test uncertainty in field performance tests, we will introduce the necessary equations for compressor and gas turbine testing, then discuss the problem of test uncertainties, as well as how to quantify them. We will make recommendations for test procedures that can reduce test uncertainties. While uncertainties for the measuring chain starting at the end device to the final value available in digital form is very well understood, there are a few issues that need further discussion:

1- The impact of non-steady state conditions.
2- The impact of the placement and number of instruments.
3- The impact of the fact that we cannot repeat the same test. In other words, there is limited capability to directly determine bias.

Lastly, we do have to discuss methods to bring the operating points of the equipment to be tested to conditions that are relevant test conditions. This is in particular challenging if the acceptance conditions for the driver and the driven equipment are combined, for example when the acceptance criteria is a certain driver fuel consumption based on the operating point of the driven equipment. Another challenge may be lack of sufficient fuel flow or the limit in available power if the ambient temperature at the test day exceeds the temperature for the acceptance conditions.
We will also see that factory tests codes, such as ASME PTC 10 [2], ASME PTC 22 [3], or ISO 5389 [5], provide good guidelines for conducting a site test. However, they are not intended for site tests and do not address typical site limitations. The ASME PTC 10 [2] Type 1 definition for acceptable deviations between acceptance and test conditions is a valid benchmark, but site performance tests often will not fall within the defined limits, and allowance must be made for that situation.

THEORETICAL BACKGROUND

The general approach for a site test follows established test codes such as ASME PTC 10. However, these test codes are written and intended for factory tests under well controlled operating conditions. Therefore, a site test procedure will have to allow for the situation at site, that is, based on the instrumentation, installation and operating conditions available.
However, the intent for a site test is to match ASME PTC 10 Type 1 conditions as far as operating conditions at site go. This approach requires the operator of the station to be able to assure operating conditions within the allowed range of ASME PTC 10. It is also possible to conduct a test that meets the requirements for operating conditions per PTC 10 Type 2. In the context of a site test, these differences must be clearly understood. In general, the true value of a site test lies in the capability to operate the compressor at operating conditions near the intended acceptance conditions. Since the actual test conditions cannot be controlled the same way as in a factory test, the resulting uncertainties must be addressed via an uncertainty analysis.

Defining the test points
Usually, the operating points for the test are determined by the facility, and the test may not be conducted at some desired condition [7],[8]. Also, when test data are taken over time for condition monitoring purposes, the data are taken for different operating conditions. Therefore, there is a need to compare data taken or predicted at different condition. Note that the gas compressor test may serve several purposes, for example:

1. To determine the compressor performance.
2. To load the gas turbine to full load to determine the gas turbine output and full-load heat rate.
3. To verify the performance of the entire train

For site tests it is always of advantage to compare the test data with other, redundant measurements. For example, the gas turbine driver full load performance may be known from a recent factory test. If the compressor can be operated with the engine running at full load, the compressor shaft power equals the engine full load output. This engine performance can then be corrected to factory test conditions [7] and should be reasonably close to the factory test results. If the gas turbine fuel flow can be measured, a similar comparison can be made for the heat rate. If the results from the site test and the factory test are reasonably close, the confidence in the site test results is improved. Otherwise, reasons for the discrepancy should be determined. Lastly, the test uncertainties incurred by the test must be taken into consideration when reporting test results or when comparing data from different tests.

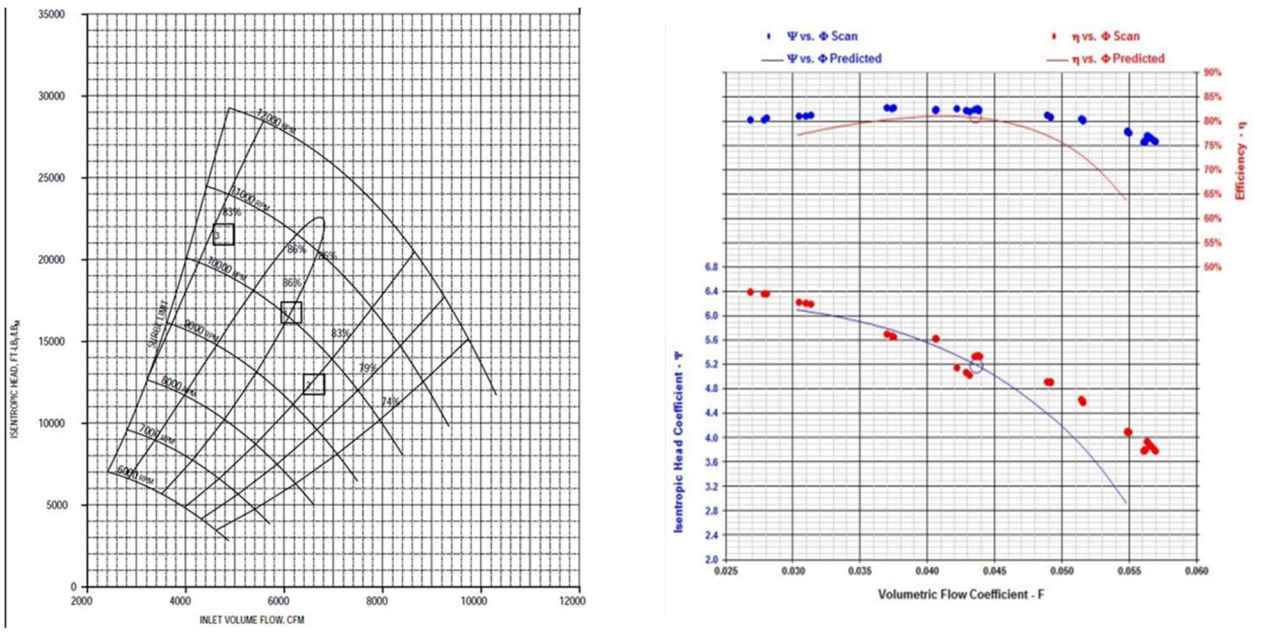

Figure 2: Dimensional and non-dimensional map.

Similarity Conditions for Gas Compressors

The goal of a site test should be to create conditions that are as close as possible to the original design conditions. The approach to a site test, therefore, is different from the approach to a factory test where a number of correction methods are used to correct performance for vastly different conditions.

To compare test data for a centrifugal compressor, it is very useful to use non-dimensional parameters for head and flow (Figure 2). Efficiency is already a non-dimensional value.

Using the Flow Coefficient:

$$\varphi = \frac{Q_s}{\frac{\pi}{4} D^2_{1,\,tip}\, u} = \frac{Q_s}{\frac{\pi^2}{4} D^3_{1,\,tip}\, N} \tag{1}$$

and Head Coefficient (isentropic or polytropic):

$$\psi^* = \frac{H^*}{\frac{u^2}{2}} = \frac{2H^*}{\left(\pi\ D_{1,tip} N\right)^2} \qquad \psi^p = \frac{H^p}{\frac{u^2}{2}} = \frac{2H^p}{\left(\pi\ D_{1,tip} N\right)^2} \tag{2}$$

eliminates the requirement to test the compressor precisely at the same speed as predicted and allows to compare data taken at (somewhat) different speeds[1]. When the volume flow ratio, that is the volume flow reduction between compressor inlet and outlet:

$$(Q_s / Q_d)_t = (Q_s / Q_d)_a \tag{3}$$

is maintained, we test the compressor with the same velocity polygons (Figure 3) at the inlet to the first stage and the exit from the last stage, thus achieving aerodynamic similarity. Figure 2 shows a typical, non-dimensional compressor map.

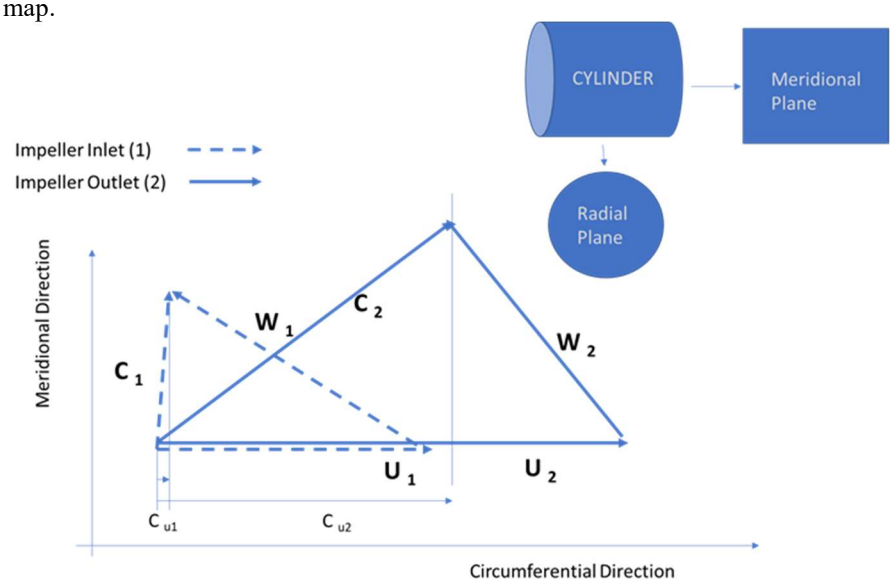

Figure 3: Velocity Polygon for a Centrifugal Compressor stage.

The other parameters that need to be maintained to accomplish similarity (although with some possible deviations) are:

Machine Mach Number:

$$Ma_u = \frac{u}{\sqrt{k_s Z_s R T_s}} = \frac{\pi\ D_{tip}\ N}{\sqrt{k_s Z_s R T_s}} \tag{4}$$

[1] The same can be accomplished by using the term Q/N and H/N^2. However, these parameters are not non-dimensional.

Machine Reynolds Number:

$$Re_u = \frac{\pi \, D_{tip} \, N \, b_{tip}}{\nu_s} \tag{5}$$

At times, only some of the similarity parameters can be brought into accordance with the desired acceptance criteria, especially when the gas composition during the test is different from the design gas. The most important parameters are head and flow coefficients, volume reduction ratio and the machine Mach number.

When keeping the flow coefficient the same as for the design case, the velocity triangles at the inlet into the first stage remain the same. Together with the head coefficient, this defines a singular operating point of the compressor as long as the fan law remains applicable. If the volume flow ratios between inlet and outlet are kept the same as for the design case, the velocity triangle at the outlet of the compressor also will be the same. Generally, this requirement involves keeping the same machine Mach number and the same average isentropic exponents over the machine (at least approximately).

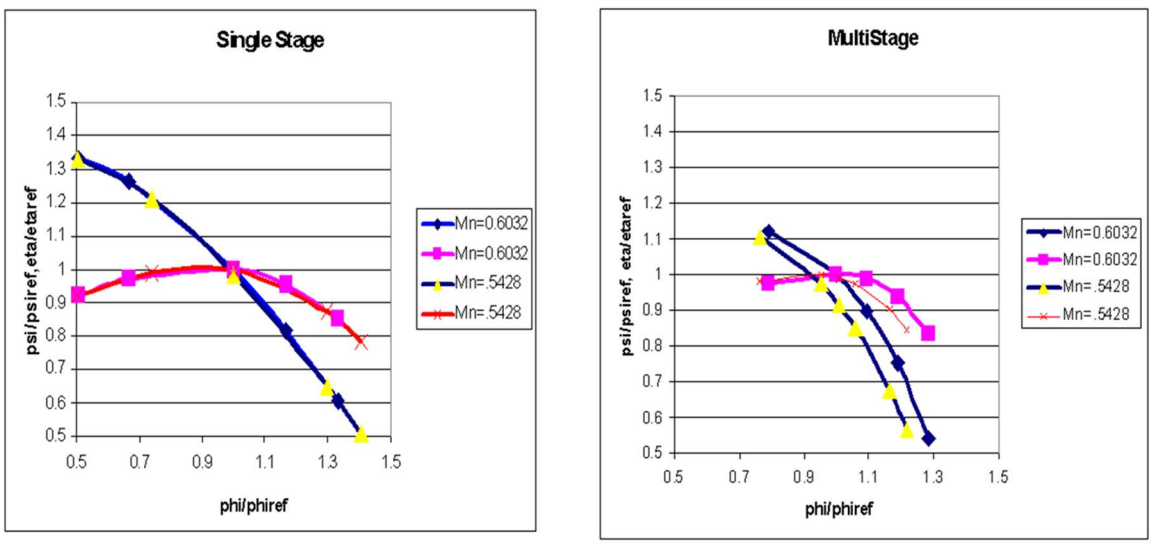

Figure 4: Limitations of the fan law

For most applications, the Reynolds number similarity is of less importance since the Reynolds numbers are relatively high and clearly in the turbulent flow regime. Additionally, the loss generation in centrifugal compressors is only partially due to skin friction effects; i.e., due to effects that are primarily governed by Reynolds numbers.

Certain deviations between design and test case for these parameters are acceptable and unavoidable. In general, as long as the deviations between test and design stay within limits as described in ASME PTC 10 [2] or ISO [5], a simple correction based on the fan law can be used. Namely, the test point must be at the same combination of φ and ψ (Eq. 1 and 2) as the design point while maintaining the volume reduction in Eq. 3. The limitations of the fan law are dictated by Mach number deviations in combination with the number of stages in the compressor (Figure 4). Pipeline compressors with usually only one or two impellers per body are typically less sensitive to deviations from the above parameters. Multistage machines show more sensitivity.

If the test conditions are considerably different from the design conditions, for example outside of the limits established in ASME PTC 10 [2], or, in more general terms, when the fan law is no longer applicable, easy corrections for Mach numbers and volume/flow ratios are not available. Often, the design programs of the compressor

manufacturer can be used to recalculate the compressor performance for the changed design conditions, that is new curves for head coefficient versus flow coefficient and efficiency vs. flow coefficient are generated for the new conditions.

ASME PTC10 [2] assumes for a Type 1 test that the test gas is almost identical to the gas for the specified acceptance conditions. In a field test, the gas composition cannot be controlled by the equipment manufacturer, and the test gas might deviate from the specified gas. In case the actual test gas deviates significantly, the compressor performance can be recalculated for the actual test gas. Deviations also occur if the gas was specified incompletely, for example, by only defining the specific gravity rather than a full gas composition.

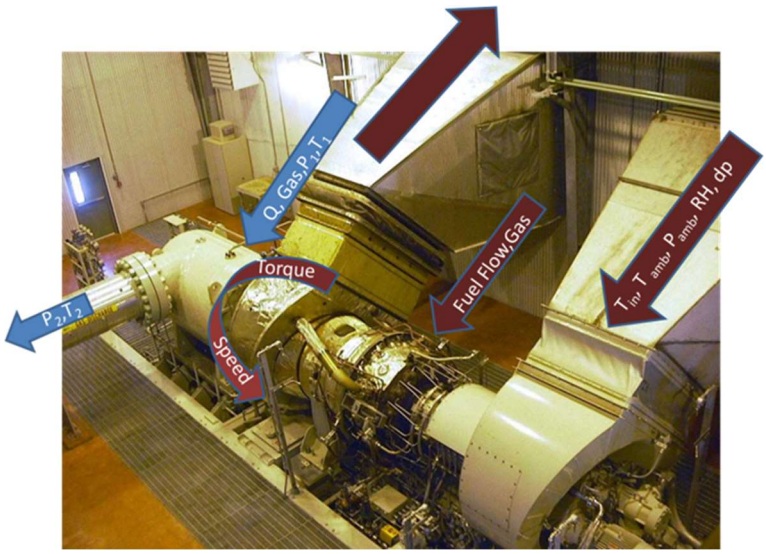

Figure 5 : Gas Turbine Measurement Needs

Gas turbine performance correction follows a different procedure, since similarity laws that can be applied do not exist. Gas turbine output power can be measured using the driven compressors. If the train consists of multiple compressors or multiple sections, all must be fully instrumented. In addition, the fuel flow into the gas turbine, the fuel composition, the power turbine speed and the ambient conditions (temperature, barometric pressure, humidity) have to be measured (Figure 5). When attempting to measure the full load power of the gas turbine, it must be assured that the engine operates at its control limits (gas producer speed, firing temperature). To determine gas turbine full load performance, the compressor operating point is of no consequence since we will not need to determine the compressor efficiency, but only its absorbed power. The correction procedure for gas turbine performance is outlined in Figure 6. The left curve shows the engine performance for the ambient conditions, power turbine speed, fuel, inlet and exhaust losses at the site test, the right curve for ambient conditions, power turbine speed, fuel, inlet and exhaust losses at the reference test. The percent difference between the test point and the performance curve is the same for both conditions.

If the fuel consumption of a gas turbine for a given compressor operating point has to be determined, the operating point for the gas turbine is typically a part load point. In that case, the compressor power consumption for said operating point has to be determined first. Then, the relative load (actual load/full load at the given ambient conditions) must be determined, and the engine has to be operated at approximately that load. Note that this procedure is tedious if followed manually, so we assume the use of a cycle deck program:
Step 1: The load at the acceptance point is calculated by dividing the power consumption of the acceptance point by the full load power at the same ambient conditions, and the same ratio of between the power turbine speed and the optimum power turbine speed at prevailing conditions (NPT/NPTOPT).

Step 2: For the prevailing ambient conditions at the test, the engine is operated at the same load (definition as in step 1), based on the reading of the torquemeter, or the power consumption of the driven compressor. After stabilization, the fuel energy flow is measured. (Note that deviations in PT speed are allowed)
Step 3: The Cycle deck is run at the same conditions (T_{amb}, p_{amb}, RH, Fuel, PT speed) as the test point in step 2. The ratio between tested fuel energy flow and calculated fuel energy flow is calculated.
Step 4: The cycle deck is run for the specified acceptance conditions. The ratio calculated in step 3 is applied to the calculated value. This yields the fuel energy consumption of the tested engine at the acceptance conditions.

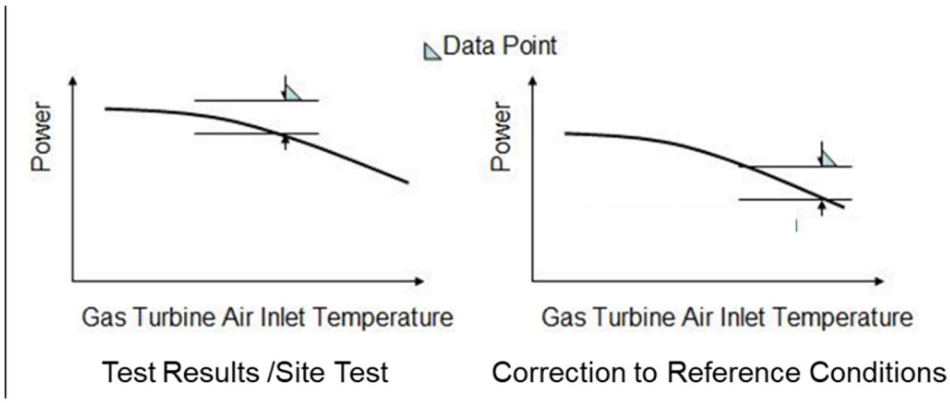

Figure 6: Correction procedure for gas turbines

Data Reduction – How to get power flow and efficiency from test data
The flow through the compressor (as well as the gas turbine fuel flow) has been measured using one of several possible flow measuring devices. If the device is a flow orifice, the relationship between the flow and the measured temperatures and pressures is as follows:

$$W = C \cdot E \cdot \frac{\pi}{4} \cdot d^2 \sqrt{2 \cdot \Delta p \cdot \rho_1} \tag{5}$$

C and E are discharge coefficients and the velocity approach factor, respectively, and d is the orifice throat diameter. The coefficients can be determined either from the orifice manufacturer's data sheets or from such codes as ASME PTC 19.5 (1971) or ISO 5167 (1980).

Other devices (venturi, pitot-type probes, etc.) have formally similar relationships between the flow and the measured pressures and temperatures. Devices that do not use the pressure differentials, such as turbine flow meters, ultrasonic flow meters and coriolis flow meters will be supplied by their respective manufacturers with appropriate methods to calculate actual flow and standard flow or mass flow. It must be noted that while the standard flow standard flow (SQ)[2] or the mass flow (W) through the flow measuring device and the compressor are identical (as long as no leaks or flow divisions are present), the actual flow will be different because the pressure and temperature at the compressor inlet will be different from the actual flow through the flow measuring device.

The knowledge of pressure and temperature at the compressor inlet nozzle enables us to calculate the actual flow (Q_s) with:

[2] Standard conditions can be 60°F and 14.70 psia, 60°F and 14.73 psia, or 15°C (59°F) and 760 mm Hg (14.7 psia). Many countries use "normal" conditions, such as 0°C (273.15 K, 32°F) and 1013.25 mbar (1 atm, 14.7 psia).

$$Q_s = \frac{W}{\rho_s} \quad or \quad Q_s = \frac{SQ \cdot \rho_{std}}{\rho_s} \tag{6}$$

The density in the above equations must be determined using an equation of state. The general relationship is:

$$\boldsymbol{\rho} = \frac{\boldsymbol{p}}{\boldsymbol{Z(p,T)} \cdot \boldsymbol{R} \cdot \boldsymbol{T}} \tag{7}$$

The compressor head (H) can be determined from the measurement of suction and discharge pressure and temperature. The relationship between the pressure, temperature and enthalpy (h) are defined by the equations of state described below.

By using the equations of state (Figure 7), the relevant enthalpies for the suction, discharge and isentropic discharge state can be computed. The isentropic head (H*) is:

$$H^* = h(p_d, \Delta s = 0) - h(p_s, T_s) \tag{8}$$

The actual head H is[3]:

$$H = h(p_d, T_d) - h(p_s, T_s) \tag{9}$$

The polytropic work follows from

$$H_{p=} \eta_p H \tag{10}$$

where η_p is calculated from an iterative process, for example as described in [9,10,11,12]. While isentropic work depends on the inlet conditions and the discharge pressure, polytropic work depends additionally on the discharge temperature.

The isentropic and polytropic efficiencies then become:

$$\eta^* = \frac{H^*}{H} \qquad \eta^p = \frac{H^p}{H} \tag{11}$$

[3] In US units, the enthalpy difference (BTU/lb) has to be multiplied by the 'mechanical equivalent of heat' (778.3 ft lb/BTU) to get the head (ft lb/lb).

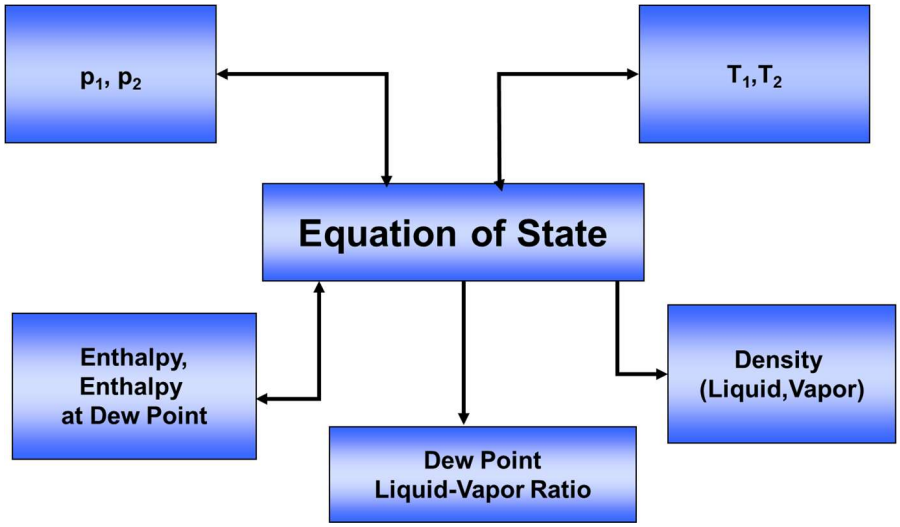

Figure 7 : How equations of state are used

It should be noted that the actual head, which determines the absorbed power, is not affected by the selection of the polytropic or isentropic process. However, the isentropic head is unambiguously defined by the user's process data (i.e., gas composition, suction pressure and temperature, discharge pressure), while the polytropic head for full definition additionally requires the compressor efficiency or discharge temperature. The comparison of the actual process with a polytropic process, as opposed to an isentropic process, has the advantage that the efficiency for an aerodynamically similar point is less dependent on the actual pressure ratio. However, it has the disadvantage that the polytropic head for a given set of operating conditions depends on the efficiency of the compressor, while the isentropic head does not. Thus, it is important to make that distinction when defining the test points.

With the flow from above, the aerodynamic or gas power of the compressor is determined to be:

$$P_g = \rho_1 Q_1 H = \frac{p_1}{Z_1 R T_1} Q_1 H \tag{12}$$

The absorbed power ('brake power') P is calculated by dividing the internal Power (Gas Power) by the mechanical efficiency η_m:

$$P = P_G/\eta_m = \frac{W}{\eta_m}\left[h(p_{t2,}T_{t2}) - h(p_{t1}, T_{t1})\right] = \frac{W}{\eta_m} \cdot \frac{H^*}{\eta^*} \tag{13}$$

After considering the mechanical efficiency (η_m) (typically around 98 to 99%), which accounts for bearing, seal and windage losses, the absorbed (or "brake") power (P) of the compressor becomes:

$$\boldsymbol{P} = \frac{\boldsymbol{P_g}}{\boldsymbol{\eta_m}} \tag{14}$$

Related to measurements of head and flow is also the determination of the surge point or the surge line. The main challenge lies in the fact that we require steady-state conditions for any of the measurements discussed herein. By

definition, surge is a non-steady condition. Even close to surge, most readings start to fluctuate. The determination of flow at surge is, thus, much more inaccurate than measurements further away from surge.

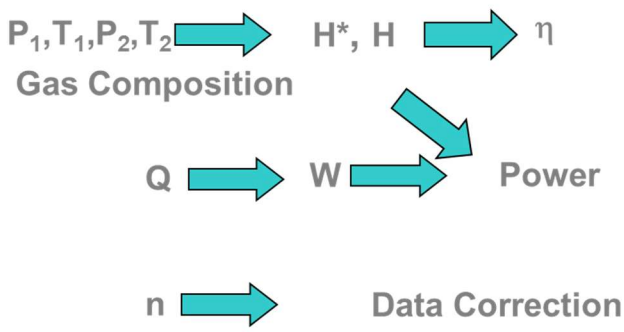

Figure 8: Data reduction for a Centrifugal Compressor.

Flow measurements can be based of a number of different inputs (such as pressure differential, or travel time of a signal in an ultrasonic flow meter), but require the knowledge of pressure and temperature at the measurement site.

Considerations for trains with multiple compressors
In trains with multiple compressors, each compressor is treated individually, both as far as pressure, temperature, flow measurements and gas compositions are concerned, but also with regards to the design points. The latter requirement is due to the fact that site conditions rarely allow both (or all three) compressors to operate at their respective design points at the same time. Therefore, their power consumption must be determined individually and later combined. If all compressors are completely instrumented, the power requirement of the train (and thus the power generated by the driver) can be determined.

Considerations for compressors with multiple sections
The particular challenge for compressors with multiple sections is to correctly separate the absorbed power for the individual sections. The difficulty arises from the fact that there can be significant mass transfer (due to leakage across the division wall) and possibly heat transfer (again, across the division wall) from section to section. It should be noted that the measurement of the overall power consumption of the compressor is not affected by these internal transfers. However, they can lead to observed efficiencies that are too high for the first section and too low for the second section, or vice versa. For compressors with n multiple sections, the absorbed power is

$$P = \frac{1}{\eta_m} \cdot \sum_{i=1}^{n} P_{G,Section\_i} \quad (15)$$

This relationship is valid, as long as all flows in and out of the system are considered. Internal leakage does not affect it.

The main difficulty in the determination of the performance of individual sections lies in the fact that the interstage leakage has an impact on the observed section performance. The interstage leakage can be determined by either

1- measuring the flow into the first section inlet, the first section discharge and the second section inlet or
2- measuring the flow into the first section inlet, and estimating the leakage flow based on theoretical considerations or factory test data.

Either method will yield the inlet flow used in the calculations above. Side stream flows have to be measured separately. Factory test codes, like ASME PTC 10 [2] can provide guidance on this topic.

Equations of State

The aero-thermodynamic performance of a gas compressor is defined by enthalpy and entropy differences, so an additional problem arises: enthalpies and entropies cannot be measured directly but have to be calculated by use of an Equation of State (EOS). The state of any fluid consisting of known components can be described by any given pair of its pressure, specific volume and temperature, and EOS approximate these relationships (Figure 7). The equations can also be used to calculate enthalpy and entropy from the condition of a gas given by a pressure and a temperature (Baehr, [13]).

The simplest EOS is the equation for a perfect gas:

$$p\,v = p/\rho = RT \tag{17}$$

$$H = h_2 - h_1 = c_p(T_2 - T_1)$$

$$H^* = c_p T_1 [(\frac{p_2}{p_1})^{\frac{k-1}{k}} - 1]$$

Real gases and in particular gas mixtures, however, display complex relationships between pressure, volume and temperature (p-v-T). EOS use semi-empirical equations to describe these relationships, in particular the deviations from perfect gas behavior:

$$\frac{p}{\rho} = Z(p,t) \cdot R \cdot T \tag{18}$$

They also allow for the calculation of properties that are derived from the p-v-T relationships, such as enthalpy (h) and entropy (s). Because EOS are semi-empirical, they might be optimized for certain facets of the gas behavior, such as liquid-vapor equilibriums and not necessarily for the typical range of temperatures and pressures in various compression applications. Because different EOS will yield different values for density, enthalpies and entropies, the EOS must be agreed upon before the test [14,15,16].

Brun et al [17] have proposed a direct enthalpy measurement method , but the method will rarely be used for site test, and also introduces some additional uncertainties. Depending on pressures and gas constituents, the frequently used EOS (RK, BWR, BWRS, LKP, RKS, PR, AGA8, GERG) provide reasonably accurate enthalpy and density calculations [18,19,20,21,22,23], but with various limitations regarding high pressures and certain gases (for example CO2 or heavy hydrocarbons). Beinecke [15] and Sandberg [24] provide thorough comparisons of Equations of State with PVT data. For the purpose of site performance tests, it is recommended to use the EOS for test data reduction that was also used for the performance prediction. This procedure is also recommended in [5] to avoid additional test uncertainties. The EOS mentioned can predict the properties of hydrocarbon mixtures quite accurately over a wide range of pressures. Still, deviations of 0.5 to 2.5% and greater in the values for gas density are common. Even more important than the compressibility factor is the calculation of the enthalpy and entropy using the EOS. EOS may differ from program to program because sometimes different mixing rules are used, different interaction parameters between the gases are assumed, or a different treatment of the ideal gas portion in the EOS is used.

Table 1. Head, Isentropic Head, Efficiency and Compressibility Factors Calculated with Different EOS [7]. Gas: 97.4% CH_4, 1.49% C_2H_6, 0.08% C_3H_8,0 .95% N_2, 0.041% CO_2; p_1 =748.2psia , p_2 = 1550.05 psia, T_1 = 100.4°F, T_2 = 215.0°F (example)

EOS	**H (ft lb_f/lb_m**	**H* (ft lb_f/lb_m)**	**η*=H*/H**	**Z1**	**Z2**
RK	43859	39019	.8896	0.9233	0.9438
LKP	43721	39284	.8985	0.9277	0.9469
BWRS	43301	39031	.9014	0.9221	0.9451
PR	43433	38463	.8856	0.9115	0.9295
Experiment (in [15])	-	-	-	0.9259	-

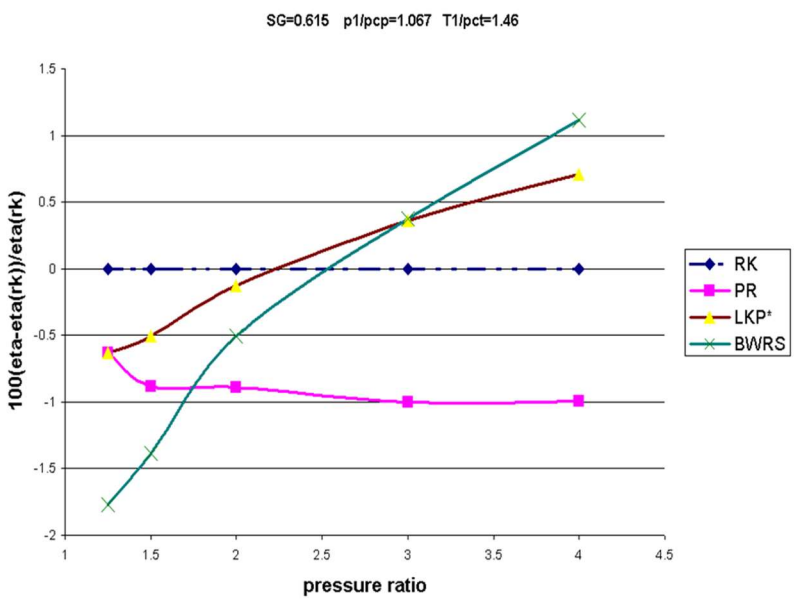

Figure 9: Efficiency calculation using different EOS for natural gas with a specific gravity of 0.615.

Because derivatives of the EOS relationships must be used to perform these calculations, the deviations can be larger than for the compressibility factor. Table 1 and Figure 9 show the effect of different EOS on the results for a given set of typical test data. In Figure 9, the isentropic efficiency was calculated based on four EOS, using the Redlich-Kwong equation as a reference. Depending on the pressure ratio, the four different EOS deliver four different results for the same measured conditions. For the calculations in the example, the following conditions were used. Suction condition was always at T_1 = 20°C (68°F) and p_1 = 50 bar (725 psia). The gas was compressed to varying end

pressures (p_2) with T_2 chosen such that the reference EOS (RK) yields 80% efficiency. The results are shown in Figure 4. Differences as high as 2% exist among the EOS models. Clearly, it cannot be concluded that a certain EOS will always lead to higher efficiency than other EOS.

Instrumentation

Both the accuracy of the instrumentation, its placement and the number of instruments have a crucial impact on the accuracy of a test. Table 2 shows a typical list of recommended instrumentation for a state-of-the-art test. The instrumentation utilized for the test should be reported in tables similar to that table. Figure 10 shows the preferred installation of instrumentation for a centrifugal compressor. It should be noted that it may be advantageous to place the pressure tabs near the flanges instead of the thermowells for the temperature measurements. This avoids the pressure loss from the thermowells to influence the results. It is acknowledged that a compressor site may not allow the installation of instrumentation as shown in Figure 10. Limitations may be due to the number of instruments, their location or the straight pipe runs that are desirable. This does not preclude a site test, but the resulting uncertainties must be considered when evaluating the results. Table 3 gives the range of achievable, end-to-end accuracies for the relevant parameters. Figure 11 outlines the required data for evaluating gas turbine performance, and Figure 5 shows typical locations of measurement points. If full load performance is to be measured it must be ensured that the gas turbine operates at its maximum load. In other words, the engine has to run at its control limits (\usually gas producer speed or firing temperature), and the output is not limited by other reasons such as low fuel gas pressure or because the compressor cannot absorb all power due to its own speed limit.

Table 2: Typical Measured Parameters and Instrumentation Quantities

Parameter	Symbol	Quantity
Gas Turbine		
Ambient Temperature	T	1
Ambient Pressure	p	1
Ambient Relative Humidity	%	1
Inlet Pressure Loss	Δp	1
Inlet Temperature	T	1
Gas Generator Speed	N	1
Volume Flow, Fuel Gas	Q	1
Composition, Fuel Gas		1
Temperature, Power Turbine	T	17
Torque, Power Turbine	T	1
Power, Power Turbine	P	1
Speed, Power Turbine	N	1
Compressor		
Volume Flow, Process Gas	Q	1
Composition, Process gas		1
Suction Pressure	p	4
Suction Temperature	T	4
Discharge Pressure	p	4
Discharge Temperature	T	4
Recycle Valve Position	%	1
Differential Pressure, Impeller Eye	Δp	1
Speed, Compressor	N	1

Table 3: In-Practice Achievable Accuracy for Measured Test Parameters [3], based on the entire measurement chain.

Measurement	Achievable Accuracy
Pressure	0.3 - 2.0% Full Scale
Temperature	0.3 - 4.0 ºC (0.5-7.5 ºF)
Flow	0.5 - 2.0% of value
Torque	0.5 - 1.5% of value
Gas composition (Density, compressibility, gas constant, specific heat, energy content)	0.2 – 3.0% of value

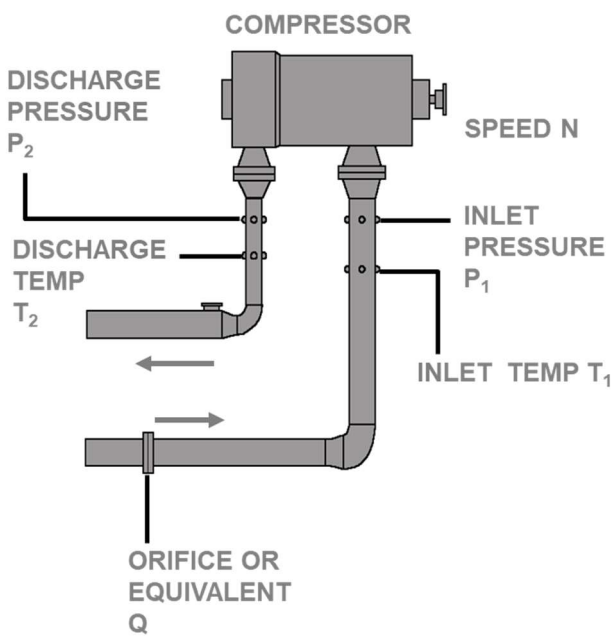

Figure 10: Probe location for a compressor test.

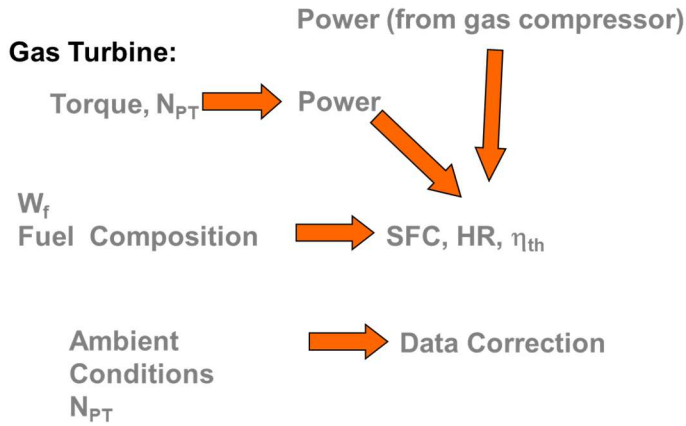

Figure 11: Measurement needs for the gas turbine

Uncertainty

Measurement uncertainty is a function of the specific measurement process used to obtain a measurement result, whether it is a simple or complex process. Measurement uncertainty analysis provides an estimate of the largest error that may reasonably be expected for the specific measurement process [1]. If the measurement process is changed, then the uncertainty analysis must be re-examined and changed as appropriate. Errors larger than the stated uncertainty should rarely occur in actual laboratory or field measurements if the uncertainty analysis has been performed correctly [25].

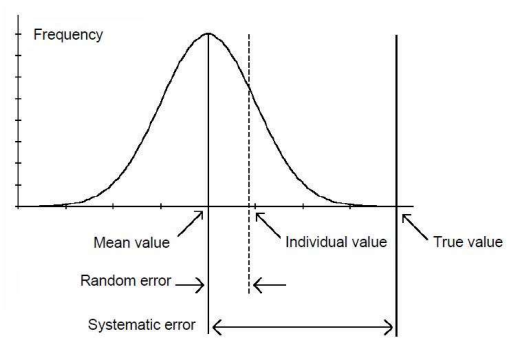

Figure 12: Simultaneously occurring Systematic and Random Errors [26]

Prior to any discussion on uncertainty, one should briefly clarify and differentiate the definitions of measurement accuracy, error, precision, linearity, bias and hysteresis (Figures 12,13):

Error is defined as the difference between measured and true value and, thus, includes all sources that contribute to any variation between a measurement chains[4] input and output.

Accuracy is simply the lack of error and it allows one to bound the range of output a measurement chain provides for a given input.

[4] We use measurement chain instead of instrument, since errors occur not just at the device, but due to the measurement location, data conversion etc.

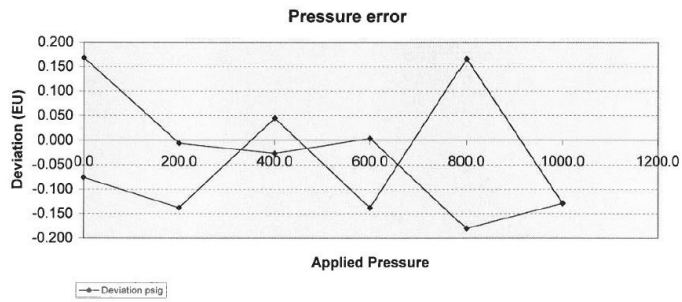

Figure 13: Hysteresis of a pressure transmitter.

Precision, linearity, hysteresis and bias are somewhat less abstract in their definition. Namely, precision defines the quality of reproducible measurements from an output reading. In other words, it is the number of significant digits a measurement chain provides with perfect accuracy. Linearity is a statistical term that is compares the deviation of a system's output to a straight-line assumption. Clearly, few physical systems behave linearly over a wide range and, thus, linearity must always be stated with an upper and lower limit. Linearity is usually determined from a statistical linear co-relation analysis with the result expressed a "k-value", where k=1.0 presents perfect linearity. Hysteresis has nothing to do with an instrument's accuracy degradation over time, but rather refers to the instrument (or system's) output dependency on directionality of the input. In most cases hysteresis is defined as the maximum difference in instrument reading for a given input value when the value is approached first with increasing then decreasing input signals. Hysteresis is often caused by energy absorption in the elements of the measuring instrument or system.

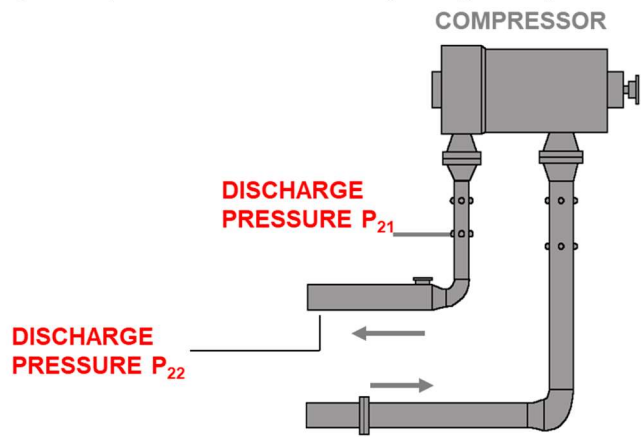

Figure 15 : Bias errors

Two fundamental types of errors must be distinguished:
-Random Error (Precision Error): Repeated measurements of a given performance parameter do not and are not expected to agree exactly. There are always numerous small effects which cause random scatter of the measured data. There is inaccuracy of the measurement of the control parameter (i.e. T_{rit}) due to its random error. There is also the variation with time in either the performance parameter or the control parameter (also called the "set point").
-Bias Error (Fixed Error): There is a systematic deviation of a measurement chain's output from a fixed input. It is the result from several individual bias errors. The bias errors must be estimated and included in the uncertainty analysis. Bias can be a complex functional form over the chain's operational range, but in many cases it is just the consistent over or under reading of input data. A constant offset is the simplest example of bias. Since the bias must be estimated, the estimate itself has an uncertainty. Thus, we can treat the bias error also as a normally distributed uncertainty.

An example is the measurement of the compressor discharge pressure (Figure 15): A Bias error arises by measuring the compressor discharge pressure as p_{22} because the elbow upstream of the location would create a

pressure loss. This bias can be corrected by calculating this pressure loss. However, both a single measurement for p_{21} or p_{22} is inaccurate since the pressure distribution across the pipe diameter is generally non uniform and not known. This bias error can be improved, but not eliminated entirely, by the use of multiple probes.

Test uncertainty calculations distinguish between random and systematic errors. It is worthwhile to estimate the level of random errors present for typical site performance tests. For this purpose, a set of test data is evaluated in [1]. The calculated test uncertainty for systematic errors in isentropic head, actual flow and absorbed power was on average 2.3%, 2.5% and 2.6% respectively. The data was taken as follows: After steady state for a given operating point was established, one data point per second was taken for ten seconds. This was repeated five times. Then, the compressor was moved to a new operating point, and the process was repeated. The set of data contains 86 of the 10 second data sets. Within each of these 10 second data sets, the standard deviation for isentropic head, actual flow and absorbed power were calculated. While the standard deviation for isentropic head was always below 0.02%, the standard deviation for flow and power could be as high as 0.3%. In other words, in all cases the impact was at least one order of magnitude smaller than the impact of the expected systematic test uncertainty.
This leads to the observation that for well conducted site performance tests, random uncertainties are much smaller than systematic uncertainties. Based on observations of a large number of different site tests, this seems to be generally true.

All of the above are factors contribute to, but are fundamentally different from, the definition of measurement uncertainty. Uncertainty does not refer to a single instrument's accuracy but evaluates the complete range of possible test results for a particular test condition. As previously stated, no test can be performed with all variables fixed, such that each input into the test system is a range rather than a point. Consequently, the measured output from the system must also be a range rather than a point and must account for all possible input combinations of all input variables.

It is important to understand that if the input ranges to the system are defined as statistical bounds, such as 95% confidence intervals, then the output from the uncertainty analysis will also present the same 95% confidence interval statistical bounds. Similarly, if the inputs are absolute errors of measurements, then the uncertainty analysis will also yield absolute errors. Therefore, whatever the type of uncertainty range for the input variables is, that will be the type of uncertainty range for the result. Consistent application and definitions of the input variable's uncertainty ranges is critically important in any uncertainty analysis.

Furthermore, prior to determining a test uncertainty it is important to know whether the measured variables in the test are independent or dependent as this determines the method of uncertainty calculation that must be employed. For almost all real measurement scenarios there is some physical dependency between the input variables; therefore, unless one is absolutely certain that all measured and given system inputs are independent, it is safer to opt for the more conservative assumption of measurement dependence. For example, if in an experiment to determine gas density one measures both temperature and pressure in a closed container, the uncertainty analysis must assume measurement dependency as the pressure and temperature are related to each other via an EOS. On the other hand, if one wanted to measure the total area of a room by measuring its length and width, one can assume that the measurement variables are independent. Clearly, the length of the room has no direct physical relationship to the width of the room. Another, not quite so obvious example is if one wanted to determine a vehicle's speed by measuring its tire radius and its wheel's angular velocity. Although it appears at first glance that tire radius and wheel angular velocity are independent dimensional measurements, closer inspection of the physical system shows that the tire's angular speed is related to the angular speed via the physical connection of the engine and transmission. In this case an uncertainty analysis method for dependent measured variables must be employed. Thus, as the determination whether an experiment's measured variables are interdependent directly establishes the uncertainty analysis method that must be employed, a thorough physical understanding of the measured system is imperative.

Factory testing of a gas turbine is almost always "zero tolerance testing" in which the minimum factory acceptance value includes any test uncertainty. In other words, the manufacturer includes the test uncertainty in the performance rating. Since the manufacturer can control the test conditions, he can control the uncertainties. So the margin from

nominal to minimum acceptance includes not only the manufacturing variations but also the impact of the measurement accuracy of the standard factory test.

Field performance testing has some inherent additional uncertainty, which does not exist in factory testing. The airflow of the gas turbine cannot be measured accurately in the field. Therefore, the actual combustion control temperature cannot be determined with accuracy so there is uncertainty whether the gas turbine is operating at the rated turbine inlet temperature. Direct measurement of the output power of mechanical drives is possible if a torque measuring device is used with the coupling, otherwise the test must use a heat balance around the driven equipment to determine the output power. Properly calibrated and selected instrumentation is the primary requirement for obtaining satisfactory field test data.

Table 4: Impact of Unsteady Test Conditions per ISO5389-1992 [5]

Fluctuation in absorbed power about the mean value (%)	**Added Uncertainty (%)**
2	0
3	0.5
4	1
5	2

Accuracy levels that can be achieved in practical site tests for relevant measurement chains based on specific end devices can be found in Table 3, and [6] [7] and [8]. Table 1 shows a typical list of recommended instrumentation for a state-of-the-art test.
Site tests sometimes have to be performed without the steady state operating conditions that are achieved in factory tests. While steady state conditions are desired, the following table gives an approximate increase of test uncertainties for absorbed power (ISO 5389-1992,[5]). Practical experience shows that deviations due to unsteady operation are underestimated by the data given in [5]. Any fluctuation in power higher than about 0.5% will add to the uncertainty of the results, especially for tests involving gas compressors (Table 4).
Based on the uncertainty of the measured parameters, the test uncertainty for the results must be determined. The description in this tutorial follows the ASME PTC 19.1 [3] Taylor Series Method (TSM) for error propagation. Because the PTC 10 code [2] uses an iterative method for the calculation of polytropic work and polytropic efficiency, the partial differentials in the TSM must be replaced by finite differences. The method described is also efficient for the use of isentropic calculations. In addition, the use of EOS results can be accommodated by the method.
In the given example, it is assumed that the uncertainties for the temperature and pressure measurements are known, and the process gas is 100% Methane, so the gas composition is not subject to uncertainties.
This example does not consider the uncertainty associated to the EOS that is used to predict the thermodynamic properties of the gas. As shown in the work of Sandberg [11,24], Beinecke et al.[15] and Kumar et al.[14], the ability to accurately determine the thermodynamic properties of the gas, and hence the performance of the compressor, can be influenced by the EOS that is selected for the evaluation. Special care is required during the test planning phase when selecting the appropriate EOS.
The relevant uncertainties that were used for pressures and temperatures include the systematic uncertainties of the entire measurement chain and are determined according ASME PTC19.1 with a 95% confidence interval. It is further assumed for this example that the systematic errors in temperature and pressure are not correlated. ASME PTC 19.1 also provides guidance for correlated systematic errors. These methods are also discussed in [8,27,28, 29].

The sample case uses the following conditions:

Sample Case

P1	psia	1000
P2	psia	2000
T1	degF	80

T2 degF 195

Gas Composition (Mol %)
Methane 100
The systematic uncertainties for the measurement chains for temperatures and pressures with a 95% confidence interval used for the example are:
T1: 1.0°F
T2: 1.0°F
P1: 5.0 psi
P2: 10.0psi
The sensitivity of the polytropic efficiency and polytropic work have been calculated using a perturbation analysis in which the expressions for these performance parameters are evaluated using the nominal values for the measured variables (inlet and discharge pressures and temperatures), and subsequently evaluated by implementing a perturbation (perturbation = nominal value + systematic uncertainty) in each one of the measured variables while keeping the other variables at their nominal value (Tables 5 and x6 below).

The total systematic uncertainty is calculated as:

$$Abs.Uncert = \sqrt{\sum(\Delta_i)^2} \qquad (19)$$

where Δ_i corresponds to the difference between the value of the performance parameter (polytropic efficiency or polytropic work) evaluated under nominal conditions and the value for that parameter evaluated under the perturbed condition for variable *i*. Here, variable *i* represents p_1, p_2, T_1 and T_2.

Tables 5 and 6 show the sample calculations for polytropic efficiency and polytropic work, respectively.

Table 5: Sample calculation of the systematic uncertainty for polytropic efficiency

parameter	uncertainty	nominal value	P1	P2	T1	T2
P1 (psia)	5	1000	995	1000	1000	1000
P2 (psia)	10	2000	2000	2010	2000	2000
T1 (deg F)	1	80	80	80	79	80
T2 (deg F)	1	195	195	195	195	196
Poly Effy		0.8205	0.8287	0.8285	0.8101	0.8113
delta squared $(\Delta_i)^2$			6.724E-05	6.400E-05	10.82E-05	8.464E-05
abs. uncert		0.018001				

Table 6: Sample calculation of systematic uncertainty for polytropic work

parameter	uncertainty	nominal value	P1	P2	T1	T2
P1 (psia)	5	1000	995	1000	1000	1000
P2 (psia)	10	2000	2000	2010	2000	2000
T1 (deg F)	1	80	80	100	79	80
T2 (deg F)	1	195	195	195	195	196
Poly Head		36132.8	36402.4	36390.4	36084.3	36172.9
delta squared $(\Delta_i)^2$			77673.69	72576.36	70.56	92.16
abs. uncert		378.1563				
rel. uncert		0.010655				

The results give a systematic uncertainty for polytropic efficiency $b_x = 0.018001$ and for polytropic work $b_x = 378.1563$ ft lb_f/lb_m (or 1.0466%).

ASME PTC 19.1 requires the user to treat systematic and random uncertainties separately. For the purpose of this example, we will assume a random uncertainty for head and efficiency that would be found for a test with multiple data points for each test point based on the analysis of the scatter of the test results, per ASME PTC 19.1 to be s_x= 0.003 for polytropic efficiency and $s_x = 40.1$ ft lb_f/lb_m for polytropic work.

The expanded uncertainty U_x then becomes (ASME PTC 19.1, Sect.5) for the polytropic efficiency

$$U_x = \sqrt{0.018001^2 + (2 * 0.003)^2)} = 0.018249 \tag{20}$$

and for the polytropic work

$$U_x = \sqrt{(378.1563 \ \frac{ft\ lb_f}{lb_m})^2 + (2 * 40.1 \frac{ft\ lb_f}{lb_m})^2} = 380.2765 \ \frac{ft\ lb_f}{lb_m} \tag{21}$$

with a confidence of 95%.
The results would then be reported as follows:
Polytropic efficiency $\eta_p = 0.821$+/-0.018
Polytropic work: H_p=36132.8 ft lb_f/lb_m +/- 380.3 ft lb_f/lb_m

Since many calculations involve multiple parameters (Figure 16), the resulting test uncertainty is not just a linear range but must be expressed by an uncertainty ellipse.

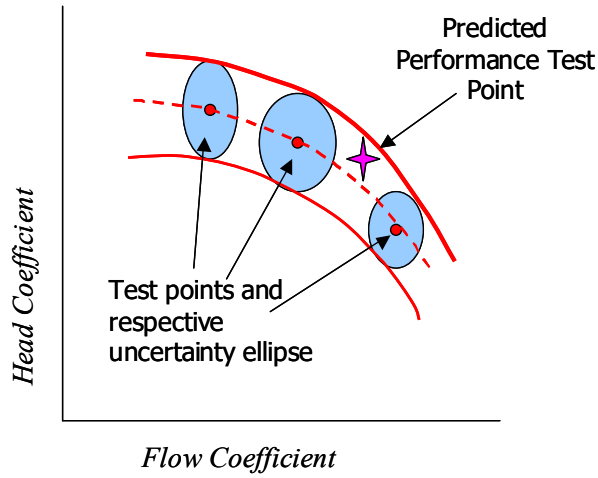

Figure 16: Example of Test Uncertainty Range

It must be noted that that there is no single 'test uncertainty' even for a given test and test setup since the test uncertainty will change with the operating conditions (Figure 17). In the depicted example for a centrifugal compressor, the higher head cases have a lower test uncertainty due to a higher temperature and pressure rise. The relative errors from the temperature and pressure measurements are therefore reduced.

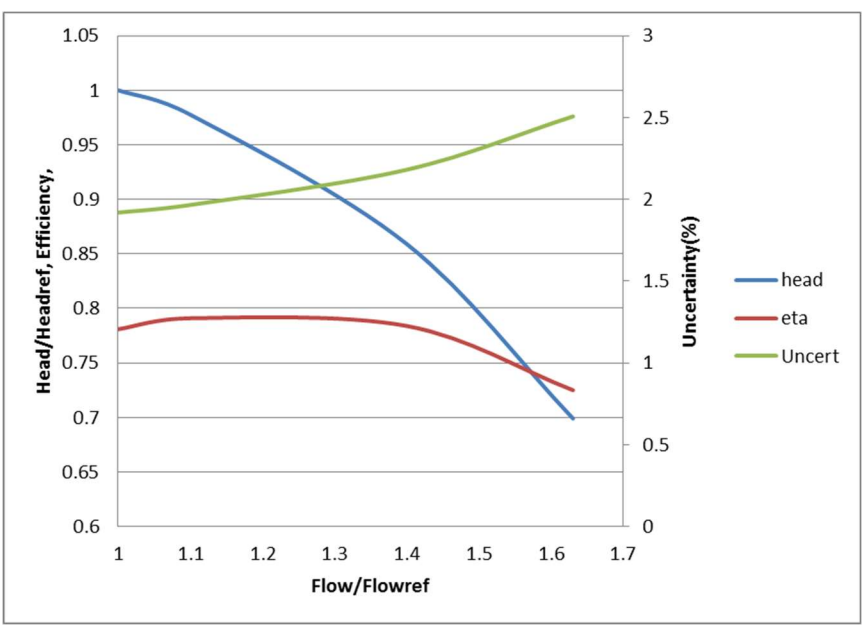

Figure 17: Test uncertainty for different operating conditions.

EXECUTION AND INTERPRETATION

A meeting between the test engineer and the organization requiring the test to discuss test procedures and the situation on site should be conducted in advance of the performance test. Where possible, the site P&ID, Site Layout and Mechanical Installation Drawings diagrams are to be available. Also, if test results from previous tests of the equipment, be it from factory tests or previous site tests exist, they should be available. Agreement must be reached on the test procedure, safety requirements and responsibilities during the test, availability of necessary operating conditions, and the acceptance conditions.

Installation of Test Equipment

Provisions are to be made during the construction phase of the gas compressor station to allow for the installation of the necessary instrumentation such as thermowells and pressure taps. If those instruments are not part of the permanent installation, block and bleed valves should be installed ahead of the pressure sensing device to allow change out during engine and/or compressor operation. It is acknowledged that field tests are often performed on existing machines with no provisions as described. It can be extremely difficult to get the necessary instrumentation taps installed in the permanent piping. Assuming the site allows for the necessary flow, temperature and pressure measurements, a site test can be performed, but the resulting test uncertainties have to be considered when interpreting the data.

There should be sufficient distance of straight pipe between measuring positions on the pipe and elements such as elbows, valves, reducers and/or diffusers. The distance between gas compressor flanges and elbows (or a reducing transition upstream of the compressor) should be at least three pipe-inside diameters. If an expanding transition is located upstream of the gas compressor, there should be at least six diameters distance. The distance between the orifice plate and upstream elbows and valves should be at least ten inside pipe diameters of straight pipe. Downstream of the orifice should be at least five diameters distance.

Inlet and discharge pressures and temperatures shall be measured at agreed points. Figures 9 and 10 show a typical arrangement. Instruments must be calibrated to a reference, and calibration certificates for all test instrumentation used for the performance test should be available.

Factory Performance Test Data

If performance data from the factory test (or other, previously conducted site test) of the compressor is available, it is actually data from an independent test, using a different measurement chain. Deviations between this data and the data from the site test gives valuable insight in the test accuracy of both the factory test and the site performance test.

Site Test Conditions

It is recommended that three complete speed lines be tested in order to fully validate the compressor performance. Test points should be taken starting at the high flow side of the curve, gradually reducing the flow. However, process conditions do not always allow for the realization of three complete speed lines. If conditions do not permit testing of three speed lines, then the test should concentrate on the design point. For each test point, data should be taken during a 10-minute interval. At least three sets of data shall be taken. All data readings for one test point shall be scanned at the same instant. For each individual acceptance point, a number of points, bracing the specified point, shall be taken and averaged.

Before readings are taken for any individual test point, steady state operating conditions must be achieved. Steady state is achieved if all of the following apply during the 10-minute interval:

-Operating speed constant within five rpm

-Fluctuations of the efficiency reading no larger than +/- 0.5 points from average, while head and actual flow remain within +/- 0.5% from average, respectively. This is significantly lower than the limits in other specifications ([14],[16]), but it is achievable in practice.

-If the compressor test point requires full load, or if the compressor is used to determine the full load power of the driving gas turbine (where applicable), it must be heat soaked for several hours (depending on the size and design of the gas turbine) to avoid drift. In case of drift, adjustments to maintain the allowable deviations can be made.

Measurement Philosophy

Where several independent instruments are used to measure a pressure or temperature value, the value of that pressure or temperature used for the evaluation will be the arithmetic average of the individual instrument's readings scanned at the same instant. Where four independent instruments are used to measure a temperature or pressure value, and one recorded observation is inconsistent due to measurement error, its value will be dismissed and the value of the measurement are determined from the average of the other three. Where fewer than four independent measurement devices are used, all values shall be used and averaged to determine the measurement value. An attempt shall be made (depending on the actual conditions), to test at five or more operating points on the same speed line ranging from choke to as close to surge as the conditions allow. The acceptance point shall be bracketed by two nearby test data points.

Preparation for Site Tests

The test should be conducted when the station conditions allow operation at or near the engine design load and conditions allow operation near the gas compressor design conditions. These requirements can be waived for condition monitoring tests.

The operating conditions for conducting the test must be agreed upon prior to the commencement of the test. The guaranteed design point and predicted performance maps should be included in the test agenda. If the inlet test conditions cannot be obtained as originally stated, for example if Machine Mach Number or the correct flow coefficient cannot be met, the prediction of the acceptance point and the performance maps should be based on the new test conditions.

In order to achieve a variety of operating points, preferably on the same speed line, the station must have the capability to move operating conditions. This can be accomplished by throttling on the suction or discharge side, recycling flow through cooled recycle lines, or loading and unloading other compressors on the same station.

The test conditions should be as close as is reasonably possible to the acceptance conditions. If these conditions cannot be achieved, an alternate test point can be defined, and performance for the point will be recalculated using appropriate performance software. In this case, no further corrections for Mach or Reynolds numbers are necessary. The alternate test point shall have the same non-dimensional flow φ and non-dimensional isentropic head ψ as the original guarantee point.

As the gas composition has a decisive influence on the test results, it is recommended to take gas samples as a minimum at the beginning and end of the test. In situations where the gas composition fluctuates, samples should be taken for each test point. Continuous gas monitoring, for example with an on-line gas chromatograph has the advantage that changes in gas composition are available while the test is conducted, thus allowing better interpretations of the results.

Installation of Test Equipment

Provisions be made during the construction phase of the gas compressor station to allow for the installation of the necessary instrumentation such as thermowells and pressure taps. If those instruments are not part of the permanent installation, block and bleed valves should be installed ahead of the pressure sensing device to allow change out during engine and/or compressor operation. Further details are addressed in earlier sections. A site test can be conducted with a reduced amount of instruments, but will have increased tests uncertainties.

Pre-Test Checkout

-The test engineer shall verify that the unit has been proven suitable for continuous operation.

-Test Engineer shall note if a gas compressor start-up strainer is installed in the inlet pipe. The customer will verify that it is clean, either by use of a differential pressure gauge, direct inspection or by borescope inspection. However, it is desirable to have the strainer removed prior to the performance test.

-The gas generator compressor of the gas turbine shall be cleaned immediately prior to the performance test using approved detergent wash procedures with water or detergent and water (if applicable). The test engineer shall determine to what extent the engine compressor shall be cleaned. Engine performance with a fouled compressor shall be deemed invalid.
-Sufficient gas must be available for proper operation of the gas compressor.
-All instrumentation, where applicable, shall be calibrated in the range in which it will be operated during the test, either at site or by an approved calibration facility.
-All RTDs used in the test shall use spring load type fittings, or when necessary, the thermowells will be serviced with oil or other approved heat transfer material.
-If a large portion of the thermowell is exposed to the atmosphere, the area around the exposed portion shall be insulated to preclude the ambient air from affecting the temperature reading.
-Where pressure taps involve a tubing run, the tubing shall be checked for leaks. When piping is experiencing vibration from flow disturbances, flex hose is to be used to connect the transmitters to the pressure tap points.
-The proper number of capable personnel shall be available to ensure that all the data can be recorded in a reasonable amount of time.

OPERATING THE COMPRESSOR

One of the key challenges at a site test is the task to bring the compressor to a relevant operating point. Many sites allow recycle operation and have a station cooler that is sized appropriately. In that case, recycle valves and the engine power setting can be used to set the compressor operating point. In many instances it makes sense to lock the power turbine speed (if the driver is a gas turbine), so data for a full or partial speed line can be gathered. Also, the fuel used and gas lost from leakage have to be compensated, for example using a load valve. This is often an iterative process and might be time consuming since equilibrium is necessary. The principles are generally the same as used in a factory closed loop facility.
If the loop entails multiple units, locking the speed of the tested unit and forcing the pressure ratio by operation of a parallel unit may be attempted. If the station has to continue to pump gas into the system, it may not be possible to achieve a relevant operating point for the compressor. It may then be necessary to operate the compressor at the prevailing conditions and use a compressor map to determine the performance relative to the map. This method will require collaboration between the OEM and operator. Using test points in or near choke or close to surge should be avoided, unless these are conditions that have to be verified.

INTERPRETATION OF TEST DATA

Site tests, if properly conducted, can provide repeatable and accurate results. If the test data deviate from the predictions or from other test data, the reasons must be explored. Assuming the test data are reduced correctly, it must be determined whether the test conditions were close enough to the conditions for the prediction. Otherwise, effects due to different Mach numbers or different volume flow ratios Q_1/Q_2 may be responsible for the deviations. In such cases it is always helpful to repeat the prediction procedure for the actual test conditions.

A key consideration is the level of test uncertainty achieved in the test. Only deviations from expected results that are outside the test uncertainty range are significant. If the test point does not meet the prediction, but a test uncertainty ellipse (Figure 16) drawn around it still covers the prediction, the test results don't contradict the prediction. The uncertainty ellipse expresses the fact that not only the measured power is subject to test uncertainties, but also the ambient temperature.

If performance data from the factory test (or from another previously conducted site test) of the compressor or the gas turbine are available, it is actually data from an independent test, using a different measurement chain. Deviations between this data and the data from the site test gives valuable insight in the test accuracy of both the factory test and the site performance test, but deviations within the range of test uncertainties must be accepted. When

comparing field test results with factory tests, the influence of test uncertainties in both tests must be considered. Whatever factory test results are available, they can be used for comparison and verification purposes. Engine performance can then be corrected to factory test conditions, using the method outlined in Figure 6, and should be reasonably close to the factory test results. If the gas turbine fuel flow can be measured, a similar comparison can be made for the heat rate. If the results from the site test and the factory test are reasonably close, the confidence in the site test results is improved. Otherwise, reasons for the discrepancy should be determined. Whatever the deviation might be, it is best if it can be detected, discussed and possibly corrected during the test. This is one of the reasons why qualified personnel from both the user and the manufacturer need to attend the test.

Another necessary step for the compressor is to compare the whole measured curve with the predicted curve. For compressors, it might be found that the head versus flow curves have just shifted horizontally, which possibly points to an incorrect flow measurement. If some points of the curve match the predictions and other do not match, variations of the gas composition during the test could be the cause.

For gas turbine power, it is helpful to use two different, independent measurements. The power can be measured by using the compressor gas power. If a torque meter is installed, it can be used to gain an independent power measurement. Comparing the results with corrected factory test data is another way.
It is also recommended to thoroughly clean the air compressor prior to the test: 3% and more engine power has been recovered after cleaning the air compressor. For site tests it is advantageous to compare the test data with other, redundant measurements. For example, the gas turbine driver full load performance may be known from a recent factory test. If the compressor can be operated with the engine running at full load, the compressor shaft power equals. the engine full load output

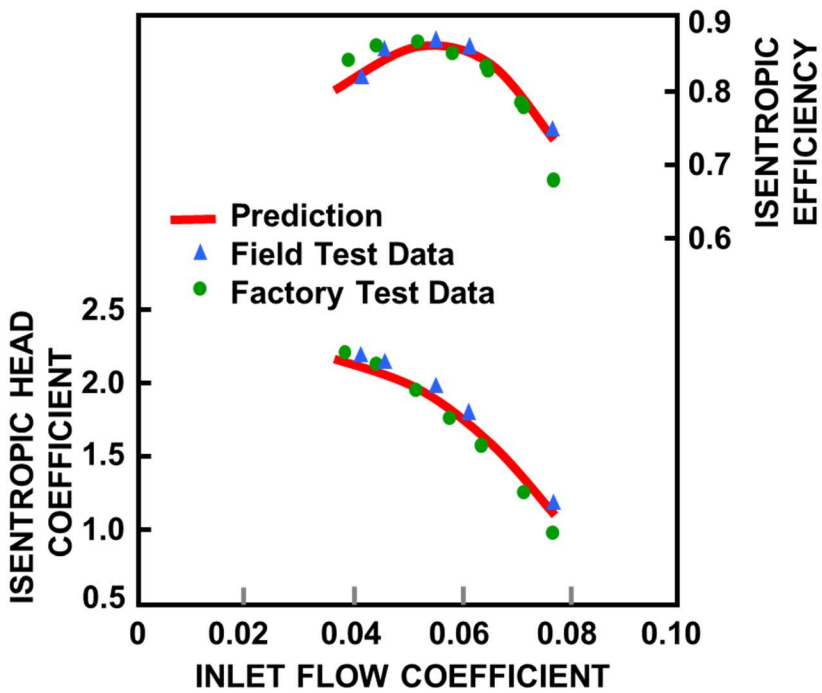

Figure 18: Factory and site test data for the same compressor

Figure 18 shows that site test, factory tests and performance predictions can and should fall closely together. If there are discrepancies, the test conditions and methodologies should be analyzed to find the reason for the discrepancy. These discrepancies can also indicate damage to the equipment or fouled components, especially for machines that

have been in service for some time,.

Figure 19 shows the results of a well-executed site test versus original predictions. The results in Figures 20 and 21 indicate problems with the test; in the first case due to unsteady process and in the second due to the unsteady change of gas compositions. In either case, we see significant variations of individual test points for essentially the same operating condition.

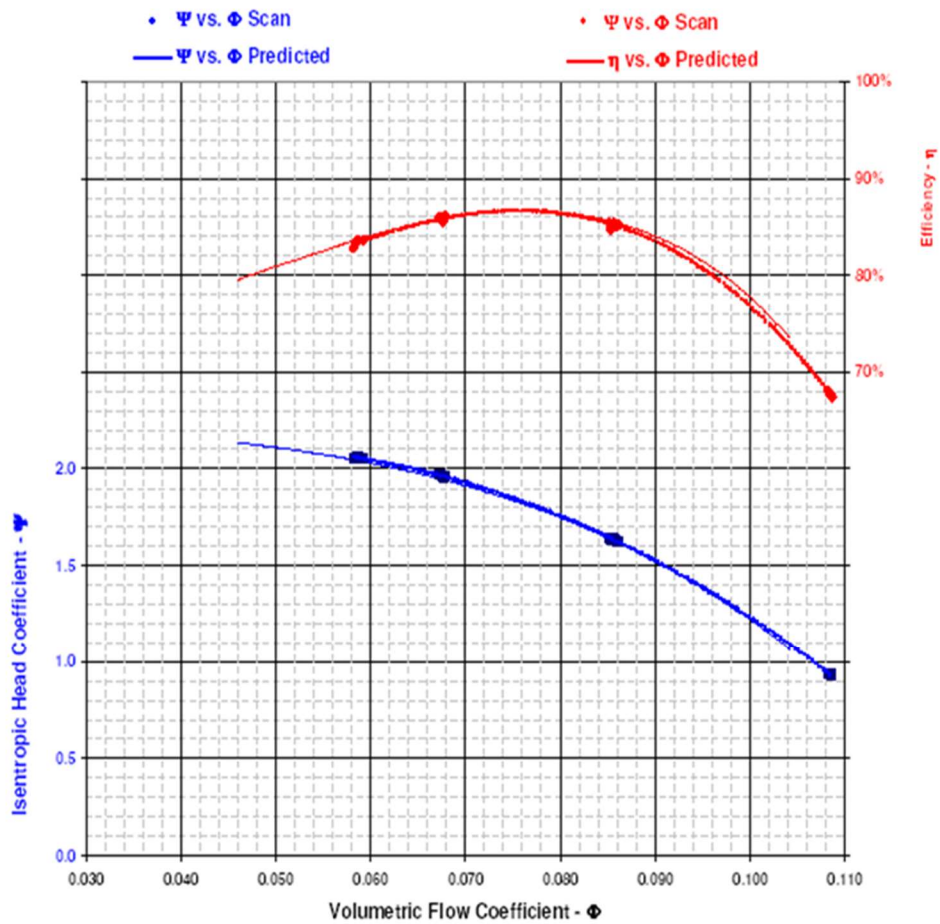

Figure 19: Results of a well execute site test: Individual test points form a tight pattern.

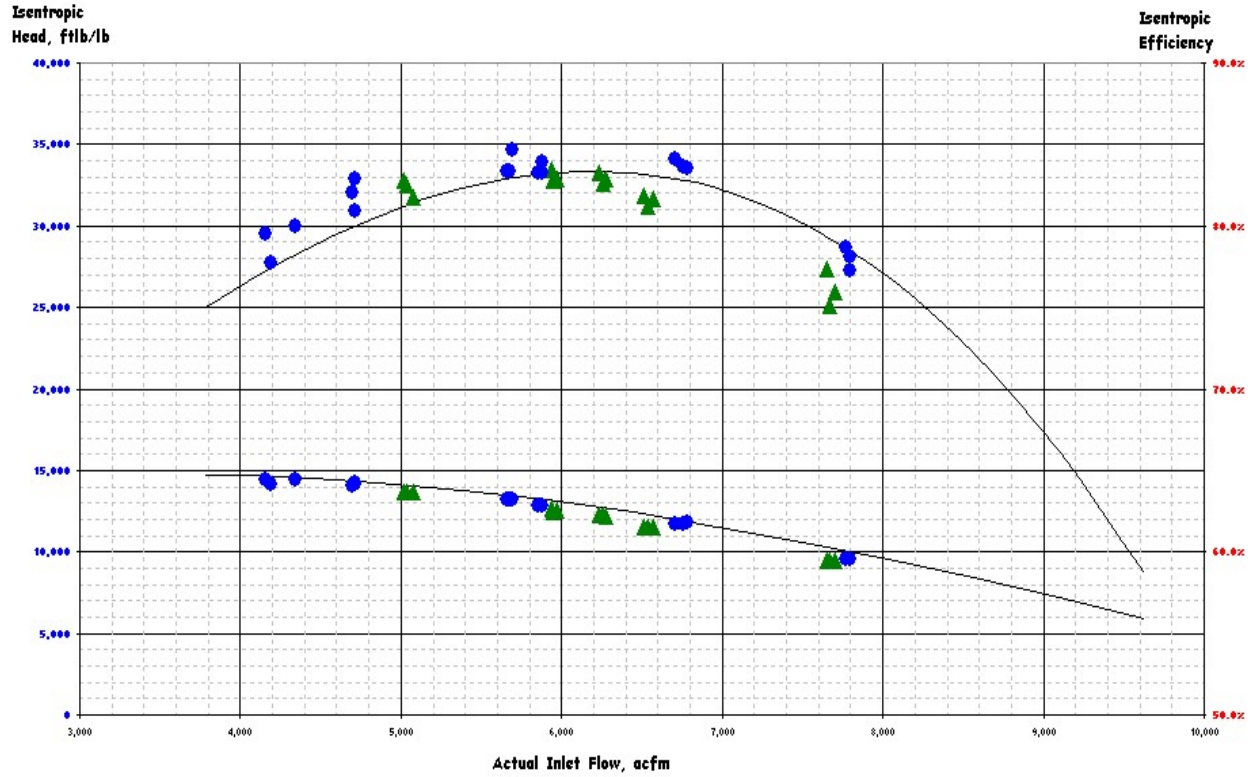

Figure 20: Impact of somewhat unsteady conditions (symbols denote two separate site tests on the same compressor)

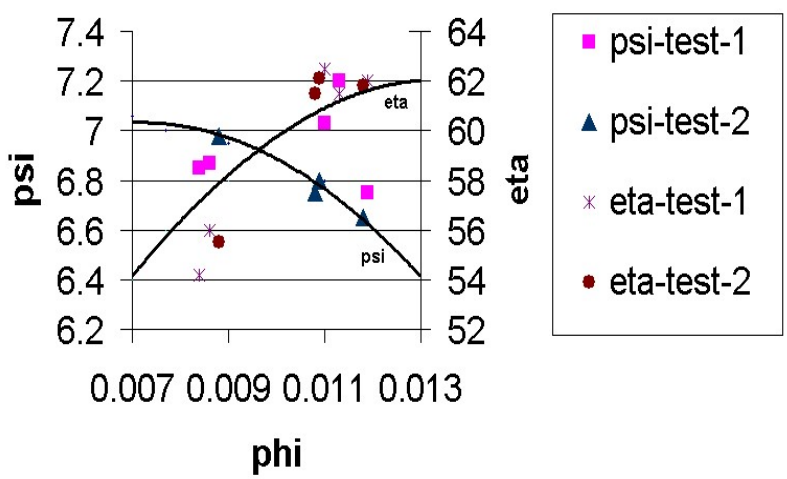

Figure 21: Test with significantly unsteady conditions, in this case due to fluctuation in gas composition. Test 1 showed strong fluctuations in gas composition, while the conditions for Test 2 were less severe.

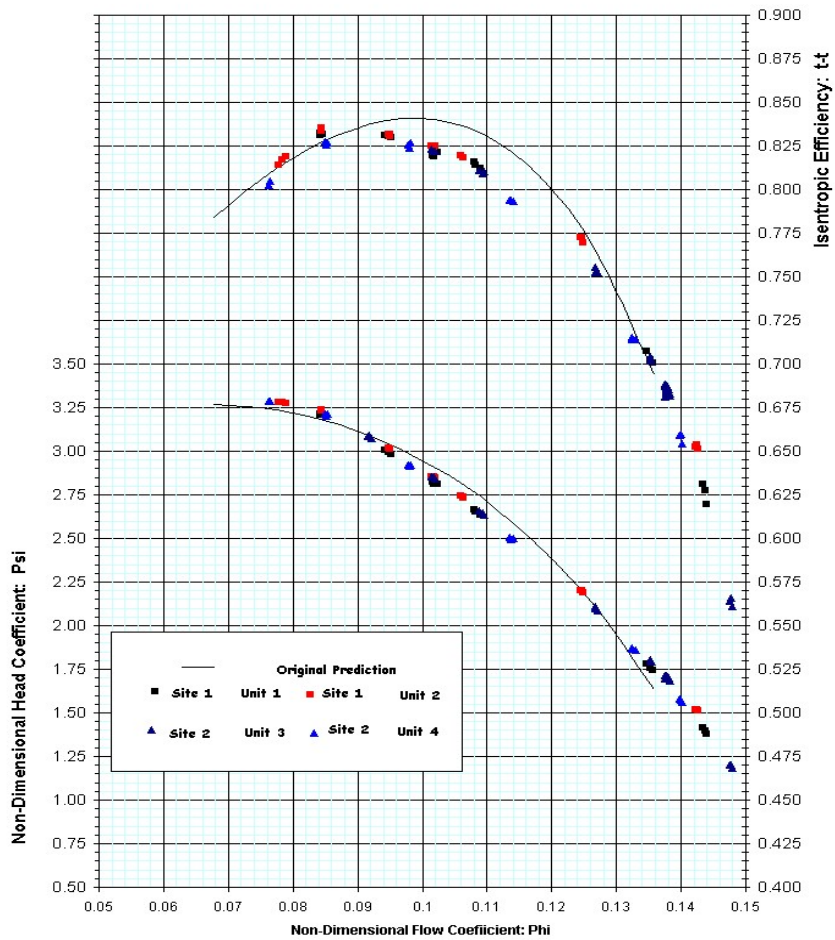

Figure 22: Repeatability. Test on four identical compressors in two different compressor stations.

Figure 22 provides an example for the repeatability that can be achieved in a site test: four identical compressors, installed at two different sites were tested successfully. The close correlation between individual data points indicates the high quality of the test, as well as the repeatable performance of the compressors.

The example shown in Figure 23 shows the results of a full load site test for a two-shaft gas turbine driving a centrifugal compressor [30]. Since the pipeline operating conditions made it possible to operate the engine at full load, several data points for full load engine power output and engine thermal efficiency were taken.
For these test points, the compressor operating point was not important because the compressor served strictly as load for the gas turbine. Gas turbine output power was measured both with the installed torque meter and by using the compressor absorbed power. Initial measurement had shown significant discrepancies of 4.4% between the two measurements. The differences were traced to a leaking valve and the calibration coefficient for the flow metering. After the corrections, the two independent results were nearly identical and also matched well with the expected engine fuel consumption. For a range of ambient temperatures, both power and efficiency showed acceptable values. Engine test data was taken at two different ambient temperatures (18.9°C and 28.6°C, respectively). The fact that relative

power and relative efficiency (Figure 23) are almost the same confirms the correction methodology outlined in Figure 5. Had the numbers been different in excess of the test uncertainty, this method would have to be questioned.

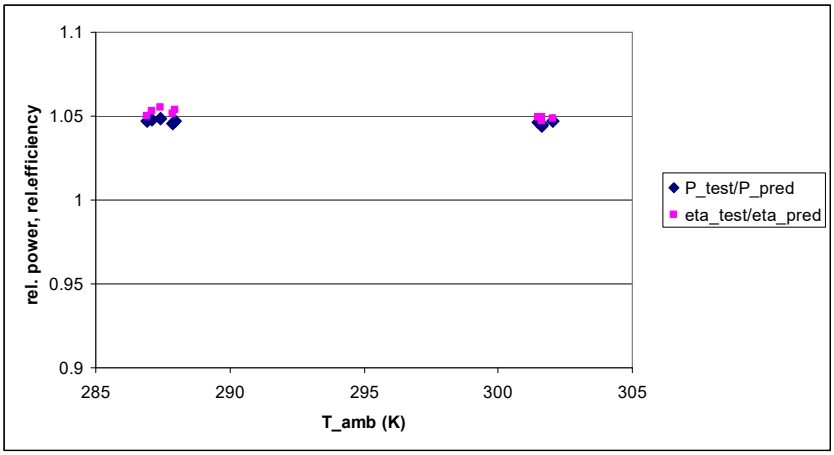

Figure 23: Engine output power and thermal efficiency at full load, compared with predicted values [30].

During the test of the compressor, a large number of data points were taken. Stable data was gathered for three different speeds and additionally, data points at the surge line were gathered. Comparison of the predicted data and data from the field test found the compressor well within the acceptable performance parameters of efficiency, range and flow at surge (Figure 24). The best efficiency was at lower flows than predicted, but the prediction of the surge line was confirmed in the test. The shape of the speed lines over the entire flow range and speed range also coincides well with the prediction. The surge line verification in site tests is challenging, since the operator wants to avoid actually surging the machine. For this particular test, the good correlation with prediction allowed to bring the compressor very close to the predicted surge line without surging it.

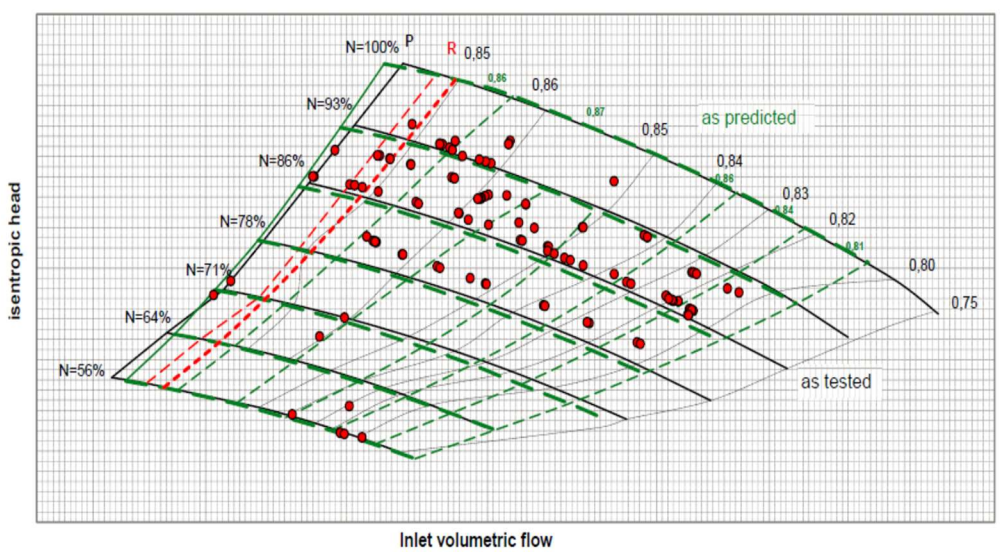

Figure 24: Comprehensive site test, showing test points as well the as tested and predicted performance of the compressor [30].

Some problems in site tests can be addressed as follows:

-Determine the shape of the head-flow and flow efficiency curves and compare them with predictions. If the curves are just shifted to the left or right, the flow measurement is suspect. If some points of the curve match the predictions and others do not match, variations of the gas composition during the test could be the cause.

-Additional evidence comes from a comparison between compressor absorbed power and expected driver available power: Determine the absorbed power and compare it with the expected power from the driver. For a gas turbine, full load factory test data is usually available. The compressor should be operated at a point that requires the gas turbine to operate at full load. The absorbed compressor power should be close to the factory tested gas turbine power (corrected to the site test conditions regarding ambient conditions and power turbine speed), assuming the gas turbine is in new and clean condition. For electric motor driven compressors, the motor, gearbox and VFD efficiencies can be used to compare the measured electric power consumption to the absorbed compressor power.

If the compressor test point requires full load, the driving gas turbine (where applicable) must be heat soaked to avoid drift. The time requirement to achieve heat soaking should be provided by the gas turbine manufacturer [30]. As a rule of thumb, one hour is required for smaller engines (below, say 8000hp), while larger engines may require two or more hours.

COMMISSIONING PHASE ISSUES REGARDING SITE PERFORMANCE TESTS

In performance testing of turbine-driven centrifugal compressors, the operators often desire a site performance verification test in the first 90 days of operation to verify the unit is operating as designed and to ensure that power consumption and efficiency meets the OEM quoted design points.

However, in the first 30 days of operation, many commissioning type issues may arise due to various issues:

- Schedules tend to be very tight and not all contractors are working on the same deadlines (pipeline contractors vs facility contractors, for example).
- Pipeline construction may be ongoing (hydrotesting and valve greasing operations could influence gas quality)
- Gas producers may not be fully ready to flow gas into the pipeline at full flow (hence, pipeline flow rates are lower than designed at first)
- Not all aspects of the compressor station are fully ready (for example, utility power, other parallel units).

The following issues should be considered in planning and designing the compressor station to assure performance testing complications do not arise during commissioning phase field tests.
The first issue that will cause lower compressor efficiency is gas quality. Entrained oil/valve grease from contractor commissioning activities or residual water as a result of insufficient drying out after the hydro-tests tend to be major issues and difficult to control, especially with multiple contractors. The operator and primary contractor should work together to allow for sufficient drying of the pipe, but often times the schedule can get compressed or temperature effects can compromise the interior of the pipe surface (humidity effects or condensed liquids, for instance). If the pipe is not sufficiently dry or if other oil or particulate matter passes through the compressor blades due to these first pipeline flows through the station, the compressor blades may become coated. In general, the first pipeline flows through a system (even outside the station side gates) will have more possible liquid carryover from initial valve greasing (or over greasing) or the pipeline construction hydrotesting beyond the station. It is important to verify the gas composition and ensure the compressor and its direct piping is clean and meets the original designed gas components. Also keeping an eye on all filters (DP and inspection ports/liquid gages) for the primary coalescing filters on the compressor station, fuel filters and dry gas seal system filters is imperative during commissioning.

Secondly, the compressor station may not be operating in the initial design window due to other system effects. Other stations or units on the pipeline may still be in the commissioning phase which may cause pressures to be low. Gas flows may also be low if not all upstream entries are fully flowing yet or gas flows may be excessively high for one unit if another unit at the same station is not commissioned yet (for parallel operation). This may cause the instrumentation and the compressor to experience off-design conditions which may lead to higher uncertainties. In addition, RTDs and thermowells must be verified as to the proper insertion length since high velocities (if gas flows are higher than design) may lead to a Strouhal vortex-shedding resonance (see API 14.1 [32] Section 6.4.1 for recommended insertion lengths for various diameters of RTD). This can become an issue for both the discharge temperature probes operating at high pipeline flow rates and the dry gas seal sample probes used to collect the discharge side gas for the compressor dry gas seal system. All lengths were shortened to less than 6.0" (for 0.75" and 1" diameter) in the case a specific midstream pipeline system to assure the resonance condition was not possible at high pipeline flow velocities.

Table 7: Maximum Recommended insertion probe length for RTDs, thermowells and dry gas seal sample probes [31]

Probe Outer Diameter inches (cm)	Recommended Max Probe Length, inches (cm)
0.25 (0.64)	2.00 (5.08)
0.375 (0.95)	3.25 (8.26)
0.5 (1.27)	4.25 (10.80)
0.75 (1.91)	6.50(16.51)

Third, often the compressor station will progress faster than the related pipeline construction. The turbine-driven compressor OEM will be ready to commission the units but cannot move forward with commissioning or performance tests due to lack of gas flows on the pipeline (since the pipeline may not be fully ready or flow rates may be so low that full performance verification is not possible yet). However, sufficient gas is often made available to purge in the station. In this case, if the compressor station is designed with a full sized recycle valve downstream of the station's process gas cooler, it will be possible to recirculate all the gas required to meet the compressor's design window without having to open the pipeline side gate valves, meaning essentially the compressor can be commissioned independent of the pipeline construction schedule (or potential delays). This gives the OEM running the performance test more flexibility as well to achieve many operating points on the map by simply tuning the recycle valve to the flow rates desired. If the opposite occurs and the station's cooled recycle valve is not sized for the maximum possible compressor flow, then it will not be possible run these tests until the pipeline is flowing full flow. The commissioning performance tests will then also require gas control and pipeline schedule considerations to change or alter flow rates to verify the entire operating map. In addition, a full-sized recycle valve placed downstream of the gas coolers, in addition to the unitized surge control valve, will allow more flexibility on start-up and ensure sufficient gas is recycled for keeping the unit gas cool during testing.
Finally, full-sized station back-up generators will ensure the units can be commissioned and started without having to wait on utility power. Often commissioning delays occur due to issues with the utility power coming online or poor power quality from the utility, especially in remote areas. If power blips or low power quality issues occur, full-sized generators will also keep the station online so that performance testing can be carried out regardless of utility power issues. If the station is designed with this philosophy, the operator should work closely with the generator and MCC supply vendors to ensure the generators and switchgear can seamlessly switch to generator power in the event of utility power loss. In addition, some controls changes may be required of the turbine driven compressor OEM to extend power outage windows to 60 or 120 seconds to give the generators sufficient time to come up to speed online and not shut down the turbine units, including timers on the lube oil coolers and turbine enclosure fans for how long they are permitted to be down during the switch to generator power.

CONCLUSIONS

Field performance testing has been identified as an important part of projects involving gas turbine driven compressor sets.

For any site test, the goal should be to create conditions that are as close as possible to the original design conditions. To compare test data for a centrifugal compressor, it is useful to use non-dimensional parameters for head and flow. Efficiency is already a non-dimensional value. Since the operation of a gas turbine is dependent on the site ambient conditions and these conditions cannot be influenced, correcting gas turbine test data for significant differences between test conditions and design conditions is inevitable.

The authors have stressed the importance of a correct and thorough preparation of such tests, which are conducted in a wide variety of working environments. Because testing in a commercial environment requires a sound balance between economics and necessary accuracy, the concepts of test uncertainty have been presented and ways to optimize the performance test have been described.

To determine the correct results from field testing, steady-state conditions and adequate instrumentation are necessary. An overview on the necessary instrumentation was given. While the tutorial focuses on acceptance tests, the procedures and recommendations also apply to any other site test, for example for condition monitoring purposes. The tutorial describes instrumentation requirements that reduce test uncertainties. Site tests can be conducted with less onerous requirements, as long as the impact of test uncertainties in the use of the results is considered.

The concepts for the calculation of efficiency, power, fuel flow, capacity, and head for an installation and how to reduce and correct test data have been introduced.

The most critical success factor is to achieve a cognizant agreement between the responsible parties prior to the test on how to conduct and evaluate the test. Well before the test, an analysis should be performed that identifies the sources of measurement errors and aims to improve those instruments that have a significant impact on the overall uncertainty. The goal is to perform the best possible test within the constraints of a production site.

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The authors want to thank Richard Huntington and Jose Gilarranz for their support of the section on uncertainty calculations.

NOMENCLATURE

A	Area
c_p	Specific heat at constant pressure
γ	Ratio of specific heats
η	Efficiency
h	Enthalpy
H	Head
HR	Heat rate
k	Isentropic Exponent
LHV	Lower Heating Value
Ma	Mach Number
MW	Molecular Weight
N	Speed
p	Pressure
P	Power
Q	Volumetric Flow
q	Fuel Heating Value
R	Gas constant
ρ	Density
T	Temperature
τ	Torque

W	Mass Flow
Z	Compressibility Factor

Abbreviations

BWRS	Benedict-Webb-Rubin-Starling
EOS	Equation of State
LKP	Lee-Kesler-Ploecker
PR	Peng-Robinson
PT	Power Turbine
RK	Redlich-Kwong
SRK	Soave-Redlich-Kwong

Subscripts

amb	Ambient
d	Discharge
f	Fuel

superscripts

*	isentropic
p	polytropic

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