FIELD PERFORMANCE TESTING OF GAS TURBINE DRIVEN COMPRESSOR SETS

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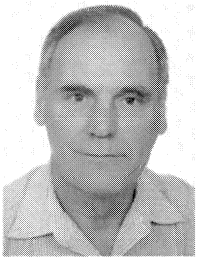

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ABSTRACT

Field testing of gas turbine compressor packages requires the accurate determination of efficiency, flow, head, power, and fuel flow in sometimes less than ideal working environments. Nonetheless, field test results have significant implication for the compressor and gas turbine manufacturers and their customers. This paper discusses field testing of gas turbine driven compressors and measurement uncertainties one can expect when following appropriate test guidelines.

A compressor field testing procedure that reduces measurement inaccuracies and maintains cost efficiency is outlined herein. Such field tests provide the user with valuable operation and maintenance data and the manufacturer with information complementary to the data gathered through factory testing.

The paper addresses the issues of planning and organization of field tests, necessary instrumentation, data reduction, data correction, test uncertainty, and the interpretation of test data. Applicable test codes and their relevance for field testing are reviewed.

INTRODUCTION

Gas turbines are used to drive gas compressors in a variety of applications, such as in pipelines, on offshore applications, in storage applications, and many others. Field performance testing of gas turbine packages is becoming increasingly frequent because economic pressures demand that the efficiency, power, fuel flow, capacity, and head of an installation be verified to assure a project's return on investment. However, during the field tests, an accurate determination of the performance of the packageor its components is often difficult because of working environments that are not optimized for testing. Nonetheless, field test results may have significant financial implications for the compressor and gas turbine manufacturers and their customers. They may be the basis of future decisions on plant modifications or extensions, or may serve as baseline data for monitoring purposes. Field tests also provide the operator and the equipment manufacturers with information complementary to the data collected during factory testing. Thus, for the end user and the manufacturer, an accurate determination of the package field performance is critical. This paper discusses problems and challenges related to the field performance testing of gas turbine driven compressor sets.

Testing procedures that reduce measurement inaccuracies and maintain cost efficiency are outlined herein. Special attention is given to the preparation and organization of the test. This paper also addresses the issues of necessary instrumentation and the interpretation of test data. Applicable test codes and their relevance for field testing are mentioned. The appropriate use of portable computers, which have introduced new powerful tools for analysis, data acquisition, and data reduction, even in remote locations, is also discussed.

To conduct a field test as economically as possible, methods to identify the test parameters that create the most significant influence on the performance uncertainties are presented and suggestions are provided on how to optimize their accuracy. The effect of different equations of state on the calculated performance is discussed.

Furthermore, issues are discussed that are often encountered in preparing and conducting field performance tests, including:

- Planning, administration, and execution of the field test.
- Instrumentation and measurement accuracies.
- Data reduction and correction (similarity laws).
- Equations of state.
- Test uncertainties.
- Evaluation of test results.

Tests on other subjects such as vibration, emissions, and control systems are not covered within the scope of this paper. Although this paper does not specifically address electric motor driven compressors, most of the content can be applied to this type of application, too.

PLANNING FOR FIELD PERFORMANCE TESTS

Challenges

The challenges of field tests arise not only in applying the laws of physics and engineering that govern the behavior of turbomachinery, but also in the depth of preparation and the organization of the necessary tools, conditions, and personnel required to conduct the tests and analyze the results. A typical field installation is shown in Figure 1.

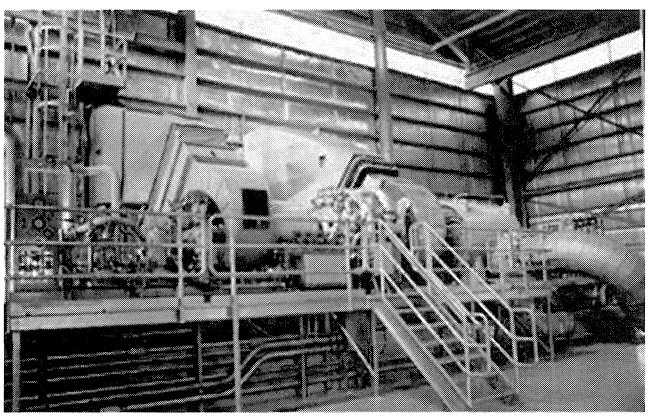

Figure 1. Typical Installation of a Compressor Set.

Governing Codes

The American Society of Mechanical Engineers (ASME), the International Organization for Standardization (ISO), and the Verein Deutscher Ingenieure (VDI) have issued specifications covering thermodynamic calculation methods, instruments, site preparation, and the reporting of turbomachinery test results in various degrees of detail (ASME Power Text Codes (PTC) 10 (1997), 19.1 (1985), and 22 (1997), ISO 2314 (1989), VDI 2045 (1993), and 2048 (1978) are examples of these standards). The authors will discuss some of the most prevalent codes in APPENDIX A.

General Conditions

Compliance with such specifications, as listed above, is a relatively easy matter in a factory environment where facilities are designed specifically for testing; qualified support personnel, instrumentation, and calibration laboratories are available; and real-time online computers routinely monitor the test progress. This is usually not the case at actual installation sites designed for commercial operation of turbomachinery. Site performance tests generally require concerted planning and execution, including development of a unique test agenda prepared jointly by the manufacturer and the equipment end-user. Such an agenda should communicate the field conditions and equipment layout, list the instruments to be used and their location, describe the method of operation and the pressure and temperature limits of the facility, and specify any deviations from normal operation that may be necessary to conduct the test. It also should describe the methods of data reduction, of determining the test uncertainties, and the acceptance criteria. The items of such a test agenda can and should be discussed in a very early stage of the project.

Preparations also include discussions on available operating conditions and operational limitations. In many cases, a specified operating point can only be maintained for a limited period of time (for example, because the pipeline operation depends on the tested package) or at fixed ambient conditions (if the necessary gas turbine power is only available on cold days).

Because the installation of instrumentation is part of the overall station design, the requirements need to be communicated early. Details, such as the necessary immersion depth of thermowells, as well as more onerous items, such as determining what flow measurement to use (taking into account the tradeoffs between pressure losses, efficiency, and cost), have to be agreed upon.

The selection and calibration of the test instrumentation are extremely important. Generally, the instruments supplied for monitoring and protection of the packages are not accurate enough to achieve the small uncertainty margins necessary for a field test. This is mainly due to the necessarily more stringent calibration requirements for a field test. Whenever possible, laboratory quality instrumentation should be installed for the tests. The accuracy of the instruments and the calibration procedure should be such that the measurement uncertainties can be eliminated from future discussions regarding the performance of the unit. The requirement for special instrumentation is especially important for field tests of compressor sets with a low pressure ratio.

Test Uncertainties and Building Tolerances

Test uncertainties need to be clearly distinguished from building tolerances. Building tolerances cover the inevitable manufacturing tolerances and the uncertainties of the performance predictions. The actual machine that is installed on the test stand will differ in its actual performance from the predicted performance by the building tolerances. Building tolerances are entirely the responsibility of the manufacturer.

Test uncertainties, on the other hand, are an expression of the uncertainty of the measuring and testing process. For example, a machine tested with 84 percent efficiency may have an actual efficiency somewhere between 82 percent and 86 percent, assuming two percent test uncertainties.

The test uncertainty is basically a measurement of the quality of the test. An increased test uncertainty increases the risk of failing the test if the turbomachine is actually performing better than theacceptance level, but it reduces the risk of failing if the machine performance is lower than the acceptance level. Because it is normal practice to use a lower performance than predicted as an acceptance criterion, it is in the interest of the manufacturer as well as the user to test as accurately as possible (Figure 2).

Agenda Prior to the Test

The customer and the manufacturer should agree upon and document the parameters of interest for the test, as well as the criteria (minimums and maximums) for acceptance. Gas turbine power and fuel flow, and gas compressor efficiency generally are the primary parameters, while compressor flow range and surge margins are examples of other common performance parameters.

In a very early stage of the project, discussions should be started about necessary instrumentation and the site preparation to allow for the installation of the required test instrumentation, such as flow metering runs, thermowells, and pressure taps. Inevitable shutdowns and the effect of the test on production need to be addressed. In this phase, the tradeoff between various options of installing instrumentation and the effect on conducting the test can be evaluated.

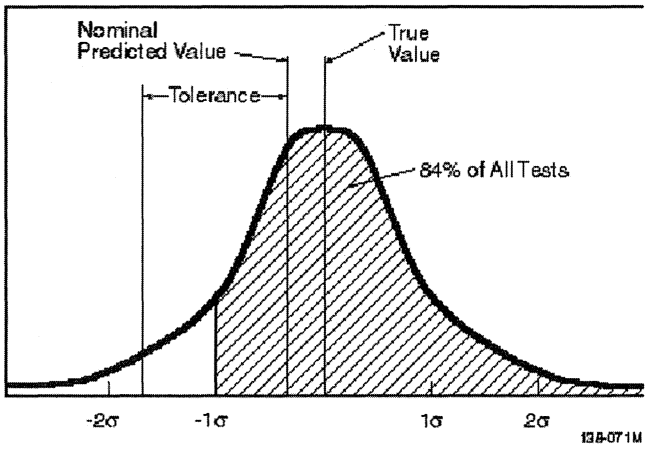

Figure 2. Test Uncertainty.

MEASUREMENT REQUIREMENTS

In the previous section, the physical properties that need to be fed into the equations for data reduction were mentioned. In the following section, the requirements for determining these properties are discussed. Table 1 outlines the necessary instrumentation for making these measurements.

Table 1. Required Instrumentation.

Turbine: Ambient temperature, T_{amb} Ambient pressure, p_{amb} Power Turbine Speed, $_{N.P.T.}$ Fuel Flow, W_f Fuel Gas Composition, - Inlet and Exhaust pressure Loss, dp_i, dp_e Turbine Inlet Temperature, T_1 Torque, τ ,* Airflow W* Internal temperatures and pressures, T_2,T_5,T_7,P_2*
Compressor: Suction Temperature, T_s Suction pressure, p_s Inlet Flow, Q, W Gas Composition, - Discharge Pressure, p_d Discharge Temperature, T_d
* optional

Instrumentation for Compressor Sets

Figure 3 shows the typical locations for instrumentation in the gas turbine. Figure 4 shows the practical requirements for temperature, flow, and pressure measurements in a gas compressor. The presented configuration, essentially taken from PTC 10 (1997), differs somewhat from the VDI 2045 (1993) recommendations. PTC 10 recommends four pressure taps and four temperature stations for inlet and discharge. It now also follows the good practice to rotate the temperature measurements by 45 degrees from the pressure measurements. The authors recommend following the PTC 10 requirement, because in most applications the measurement stations are relatively close to elbows. Also, it adds the chance to exclude one of the measurements if it deviates from the other three. This might save test time that would have to be spent if only two probes were used. Because the maximum error due to flow nonuniformities is in the order of the dynamic pressure, the decision has to be made based on the magnitude of that pressure relative to the total pressure. Having several pressure and temperature stations at the inlet and outlet also gives the chance to verify the results for each measuring chain, including the transducers and digital voltmeters. The time saved by having this capability often outweighs the extra cost for the equipment. Figure 5 shows a gas compressor with test instrumentation, ready for the field performance test.

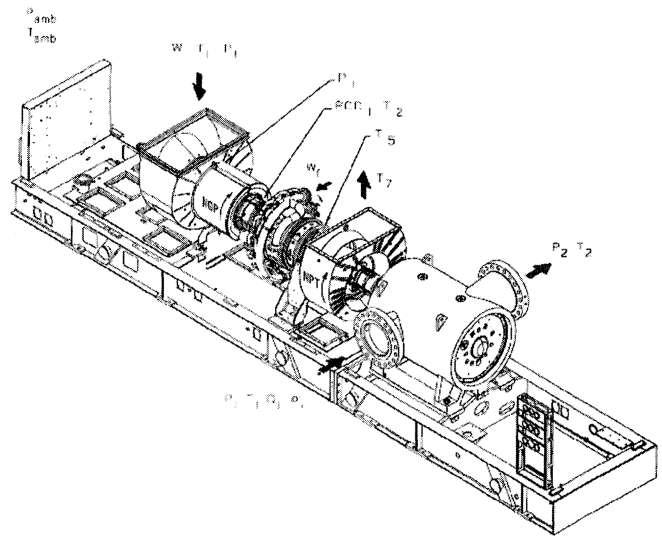

Figure 3. Instrumentation Locations for the Gas Turbine Package.

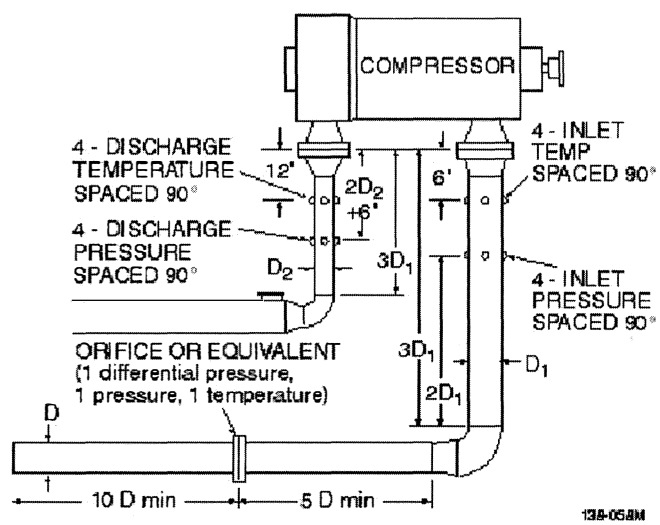

Figure 4. Instrumentation for a Gas Compressor Performance Test.

Compressors with discharge volutes typically create a relatively nonuniform pressure and especially temperature distribution at the exit. For these machines, tests should not be performed with less than three pressure and temperature stations on the discharge side.

It must be noted that tests are possible with less than the recommended pressure and temperature measurements. Often, the test has to be conducted with whatever instrumentation can be installed. The resulting additional uncertainties have to be taken into consideration, and have to be evaluated on a case-by-case basis.

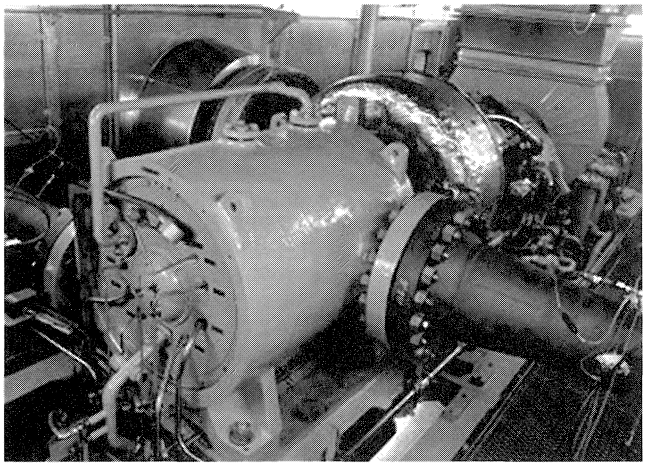

Figure 5. Compressor Set Ready for Field Test.

The flow measurement devices, such as orifice plates, nozzles, pitot type probes, or venturi nozzles, require runs of a certain straight length. Typical requirements are shown in Table 2. Because there may be a considerable distance between the flow measurement device and the compressor inlet, the gas temperature and pressure need to be measured at the flow measuring device, in addition to the measurement locations close to the compressor inlet. Fuel flow measurement lines also need temperature and pressure measurements close to the flow measuring device.

Table 2. Straight Length (in Pipe Diameters) Required by ISO 5167.

β	On upstream (inlet) side of the primary device							On downstream (outlet) side
	Single 90° bend or tee (flow from one branch only)	Two or more 90° bends in the same plane	Two or more 90° bends in different planes	Reducer (2D to D over a length of 1.5D to 3D)	Expander (0.5D to D over a length of 1D to 2D)	Globe valve fully open	Gate valve fully open	All fittings included in this table
≤0.20	10 (6)	14 (7)	34 (17)	5	16 (8)	18 (9)	12 (6)	4 (2)
0.25	10 (6)	14 (7)	34 (17)	5	16 (8)	18 (9)	12 (6)	4 (2)
0.30	10 (6)	16 (8)	34 (17)	5	16 (8)	18 (9)	12 (6)	5 (2.5)
0.35	12 (6)	16 (8)	36 (18)	5	16 (8)	18 (9)	12 (6)	5 (2.5)
0.40	14 (7)	18 (9)	36 (18)	5	16 (8)	20 (10)	12 (6)	6 (3)
0.45	14 (7)	18 (9)	38 (19)	5	17 (9)	20 (10)	12 (6)	6 (3)
0.50	14 (7)	20 (10)	40 (20)	6 (5)	18 (9)	22 (11)	12 (6)	6 (3)
0.55	16 (8)	22 (11)	44 (22)	8 (5)	20 (10)	24 (12)	14 (7)	6 (3)
0.60	18 (9)	26 (13)	48 (24)	9 (5)	22 (11)	26 (13)	14 (7)	7 (3.5)
0.65	22 (11)	32 (16)	54 (27)	11 (6)	25 (13)	28 (14)	16 (8)	7 (3.5)
0.70	28 (14)	36 (18)	62 (31)	14 (7)	30 (15)	32 (16)	20 (10)	7 (3.5)
0.75	36 (18)	42 (21)	70 (35)	22 (11)	38 (19)	36 (18)	24 (12)	8 (4)
0.80	46 (23)	50 (25)	80 (40)	30 (15)	54 (27)	44 (22)	30 (15)	8 (4)

	Fittings	Minimum upstream (inlet) straight length required
For all β values	Abrupt symmetrical reduction having a diameter ratio ≥ 0.5	30 (15)
	Thermometer pocket or well of diameter ≤ 0.03 D	5 (3)
	Thermometer pocket or well of diameter between 0.03 D and 0.13 D	20 (10)

Note: The unbracketed values are "zero additional uncertainty" values; the bracketed values are ± 0.5% additional uncertainty values. All straight lengths are expressed as multiples of the pipe diameter D.

The arrangement of pressure and temperature devices in the ASME PTC 10 (1997) code on the discharge side (looking downstream: first temperature devices, then pressure tabs) is practical. The distances of the devices from the nozzles can be handled more flexibly. However, certain maximum distances from the nozzle should not be exceeded, the pressure tabs and thermowells should be offset by 45 degrees (to avoid the influence of wakes from the thermowells) and no bends or other obstructions should be between the compressor nozzle and the sensing elements. Pressure tabs should not be placed at low points of the pipeline or in a six o'clock position. They may fill with liquids, which would distort the reading.

Use of Package Instrumentation

The use of package instrumentation leads to a considerably lower accuracy compared to tests with dedicated test instrumentation, especially due to higher calibration standards for the test instrumentation.

Package instrumentation is normally selected to allow for sufficient accuracy for trending. For trending purposes, the absolute accuracy of a measurement is not important, but the difference from certain baselines is. Package displays usually do not consider changes in gas composition. Furthermore, dedicated test instrumentation is calibrated on a regular basis and maintained continuously.

Instrumentation Tolerances

When considering instrumentation tolerances, the whole measuring chain needs to be taken into account. The instrument, such as the resistance temperature device (RTD), thermocouple, or pressure transducer, has a certain measuring tolerance. Yet the overall error is also influenced by the location of the instrument (flow measurements with insufficient straight runs), the way the instrument is installed (thermocouples in thermowells without heat conductive paste or insufficient immersion depth), potential reading errors (especially if gauges are used), or the accuracy of the digital voltmeter, and the calibration quality. The following are typical instrumentation tolerances:

- Pressure, percent 0.5 to 2.0
- Temperature 0.5°F to 4°F
- Flow, percent 0.5 to 2.0
- Gas composition, percent 1.0 to 5.0
- Torque, percent 1.0 to 1.5
- Equation of state, percent 0.2 to 2.5

Pressures

The pressure in a pipe, as well as in the different gas turbine locations, is typically measured using wall taps, i.e., we measure the static pressure. The actual measurement device can be a transmitter, transducer, a deadweight tester, a U-tube, a Bourdon type gauge, or others. In order to accommodate electronic data acquisition, transducers or transmitters have to be used.

In a typical field test, accuracies of 0.5 percent to 2.0 percent can be achieved. While somewhat higher accuracies are reported under lab conditions, they depend on test conditions that are not feasible under field conditions.

Note that a significant portion of the measurement error comes from the areas of installation. The error will also depend on the uniformity of the flow at the measuring location. Shaw (1960) shows the dependency of the static pressure error on the quality of the pressure taps. Wall taps need to be exactly perpendicular and flush to the surface. No burrs or slag are acceptable. The authors recently tested a machine where the pressure taps were cut into the pipe with a cutting torch. The resulting spread of the four pressure readings was almost 2.0 percent of the static pressure. The ratio of dynamic to static pressure for most applications will be below one percent, because the pipe diameters (and the compressor nozzle diameters) are selected to avoid high flow velocities.

VDI 2045 (1993) assumes 0.2 percent error of full scale for transducers and gauges. It should be emphasized that the instruments should be selected in a way that the measured values are normally in the upper 25 percent of full scale. Liquid columns can be more precise, but the precision depends largely on the accuracy of reading the scales. Because field tests tend to last many hours, any system that avoids human error in reading instruments is preferred.

While many codes require the measurement of static pressures and dynamic pressures, it is typically as accurate and less complicated to correct the dynamic component of the pressure using the flow velocity in the pipe, which is known through the volume flow measurement. The total pressure can also be measured directly using Kiel probes.

Pressures such as ambient pressures are measured using calibrated barometers or absolute pressure transducers.

Temperatures

Temperatures in pipelines can be measured using RTDs, thermocouples, or thermistors. Thermometers can also be used. However, they do not allow electronic data acquisition. All devices are usually inserted into thermowells. These thermowells have to be inserted sufficiently deep into the pipe to avoid the influence of the pipe wall.

Thermocouples may be used for temperature measurements preferably above 200°F. Below this temperature, the signal from the thermocouple can become very low. Thermocouples are available in different types that create the highest output signal in given temperature ranges (type "E" (Chromel-Constantan) between 200°F to 1400°F; above 1400°F, type "K" (Chromel-Alumel)). The thermocouple measures differences in temperature between the measurement location and a reference point ("cold junction"). The greater this difference, the higher the electromotive force (EMF) voltage signal. For high accuracy it is necessary to have a continuous lead from the measurement tip to the connection on the cold junction. The high temperatures encountered inside engines typically require thermocouples.

RTDs can be used to test from very low to moderately high temperatures. The measurement principle is based on the change in resistance of the device with changing temperature. They are susceptible to mechanical damage, especially if they are subjected to vibrations.

Thermistors are made of semiconducting material that acts as a thermally sensitive variable resistor. Unlike RTDs, the resistance increases with decreasing temperature, so that this device is most useful at low temperatures. Above about 300°F, the signal becomes low and susceptible to error from current-induced noise. Also, the signal from a simple thermistor (i.e., without additional compensation) is very nonlinear and requires quite some calibration effort (Horton, 1990).

Under laboratory conditions, ± 0.2°K, with shielded NiCrNi-thermocouple probes, can be achieved, depending on the flow velocity. The same high accuracy will not be achieved in the field, because the compensation elements have to be kept in an ice water bath of ± 0°C. Also, thermocouples produce the weakest signal of the described devices. The effect of noise in the electronic circuitry is therefore more prominent than with the others. In any case, calibration curves can describe the behavior of the thermocouples precisely. In the field, a total inaccuracy of ± 0.5°K seems achievable. Cleveland (1982) shows instrument accuracies for thermometers to be 0.25°K to 1.0°K, thermocouples to be 0.25°K to 1.0°K, and RTDs to be 0.0025°K to 2.5°K. Schmitt and Thomas (1995) report 0.5°K for RTDs. VDI 2045 (1993) assumes a tolerance of ± 1.0°K for thermocouples and RTDs.

Location, installation, and calibration errors are the dominant contributors to measurement errors, while device and acquisition errors are a smaller contribution to the total temperature error. Also note that the overall temperature errors for the field test are significantly larger than the values quoted by instrument manufacturers, because the manufacturer's numbers normally do not allow for location, installation, and calibration errors.

Critical points for the installation in pipelines are thermowells with sufficient immersion depth of about one-third of the inner pipe diameter, sufficient heat conduction to the device by spring-loading or thermo paste, and a sufficient amount (two to four) of thermowells per measuring point to reduce the influence of flow nonuniformities. For best accuracy, four taps are recommended, which should be positioned 90 degrees apart from one another, and 45 degrees from the pressure taps. Because the flow velocities in the inlet of gas compressors (or other locations of practical interest) are clearly subsonic, inaccuracies due to recovery factors should be small.

Temperature measurements in highly nonuniform flows, such as the exhaust of a gas turbine or the discharge flow of a gas compressor using a volute system, require a large number of probes. Twelve RTDs are used to measure the exhaust temperature of a gas turbine accurately. They are arranged symmetrically at the center of equal areas. Because no information on velocity distributions is available, arithmetic averaging is the only feasible procedure for data reduction, and the resulting uncertainties are considerable. In such situations, it is recommended to conceive a second, independent method (such as the heat balance) to cross check the results.

Flow

A variety of methods exist for flow measurements. Nozzles, orifices, venturi nozzles, and pitot type probes use pressure differentials to measure the volume flow (Figure 6). Ultrasonic flowmeters measure average velocities in the flow field by determining the time that a traveling ultrasound signal needs to cross through the pipe. Turbine flowmeters measure the flow velocity via a turbine wheel where the speed of the turbine wheel is related to the flow velocity. In vortex flowmeters, the frequency of variations of pressure or velocity that occur in a wake of a blunt body and that depend on the flow rate is determined. The gas pressure and temperature have to be measured at the location of the flowmeter to relate the actually measured physical property (pressure differential, time, speed) with the volume flow.

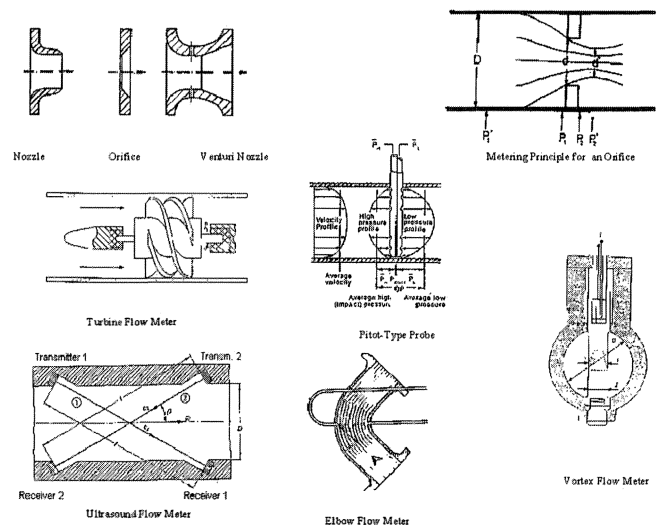

Figure 6. Flow Measuring Devices.

All these devices measure velocities and volumetric flows. Only by simultaneously measuring pressures and temperatures can we also determine the mass flow.

The flow rate uncertainty, ΔQ, depends on the device type employed for the measurements. A detailed discussion of flow measurement uncertainty is provided in ASME PTC 19.1 (1985), while the application of flowmeters is discussed in ASME PTC 19.5 (1971) and ISO 5167 (1980). In Table 3, some typical values of ΔQ from field testing experience are provided. The values that can be obtained with labquality calibrated devices in perfect arrangements are provided in parentheses for comparison. It must be noted that any alteration of the device geometry (example: wear on the sharp edges of orifices, fouling of nozzles, venturis, or turbine flowmeters, etc.) will impact the accuracy of a previously calibrated device.

Table 3. Typical Magnitudes of Volume Flow Rate Measurement Errors (Percent Full Scale).

Component	Measurement Error	Sensitivity to Location	Unrecovered Pressure Loss
Orifice	1.5% (0.5%)	medium to low	high
Venturi	1.5% (0.5%)	medium	moderate
Ultrasound	0.5%	medium to high	nil
Turbine Flow Meter	0.5-2.0% (0.5%)	medium	high
Pitot type probes	1.5% (0.5%)	medium to low	low

None of the values in Table 3 accounts for errors in density. The gas analysis has a certain error margin. Density errors of up to 10 percent due to an inaccurate determination of the gas composition are reported. Even for pipeline applications, error margins are one percent or larger (Meier and Rhea, 1982). Especially cumbersome is the analysis if the gas samples are taken far away from the compressor or upstream of a separator or knockout vessel (McRoberts, 1984). In any case, it is necessary to measure the pressure and temperature of the gas at the location of the flow sensing device in order to achieve a correct calculation of the flow density.

ISO 2314 (1989) assumes 0.5 percent accuracy for flow measurements. Schmitt and Thomas (1995) report an uncertainty for an orifice metering run per ISO 5167 (1980) of 1.4 percent. A properly selected and calibrated fuel flow measuring device can be suitable to achieve measurement accuracies of ± 1.0 percent or better of the measured quantity. However, the additional effort to get from one percent accuracy to the accuracy required by ISO 2314 (0.5 percent) might not always be justified when the driven compressor is used for the power measurement. With a three percent accuracy of the power measurement, the gas turbine heat rate uncertainty will only improve by 1/10 percent. Valenti (1997) reports, in an evaluation of gas flow measuring methods, that the measurement errors for orifices are one percent (mainly depending on piping configuration and diameter ratio of the orifice), and the measurement errors for turbine flowmeters are more than one percent (especially due to flow pulsations). This publication emphasizes the wide flow range and the very low pressure losses of ultrasonic flowmeters. Ongoing research is trying to quantify the sensitivity of these devices to distortions of the flow profile. AGA Report No. 3 (1991) assumes an accuracy of an orifice designed and installed in accordance with this code of ± 0.67 percent including an allowance for the uncertainties in gas compositions. However, in site installations, the uncertainty of the gas composition, together with the fact that the requirements for the length of straight pipe runs are usually not met, will usually yield an accuracy not better than ± one percent.

Errors in the measurement of the flow through a compressor will not influence the accuracy of the compressor efficiency, although they will cause wrong information about the actual operating point of the machine. Nonetheless, they play a significant role in determining the shaft power. Inaccurate flow measurements can be identified by plotting the head/flow test data for an entire speed line on the predicted speed line. If the shape of the curves is the same, but seems to be shifted to higher or lower flows, the flow measurement might be flawed. Flow measurement problems occur frequently when multiple-compressor stations use only one accurate flow device for the whole station, which naturally will operate at the low end of its range when only one machine is operated for test purposes.

The airflow through a gas turbine can be measured by using the pressure differential between inlet flange and bell mouth. Because the flow is accelerated, a venturi effect is created that can be used to measure the flow, if the device were previously calibrated during the factory test. A similar method can be used for some gas compressors. In both cases, the accuracy is clearly lower than for the other methods mentioned above.

For flow measurements, one of the critical requirements is the piping configuration upstream and downstream of the device, because all practical flow measurement devices assume a certain flow profile within the pipe. If the actual flow profile is different, the flow measurement will deviate from the exact values (Figure 7). Grimley and Bowles (1998) demonstrated that for ultrasonic flowmeters, a 90 degree elbow 40 pipe diameters upstream of the measurement device, can increase the error in mass flow from 0.5 percent to a level of one to four percent. Flow conditioners, or the use of multipath ultrasound devices can improve these numbers significantly.

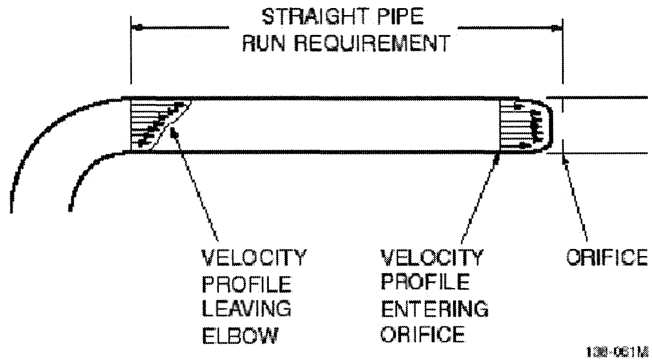

Figure 7. Flow Profile in a Pipeline Downstream of an Elbow.

Another significant error can be caused by entrained liquids. Tests reported by Ting (1998) show indicated wet gas flows that are up to three percent lower than for the same gas after drying it. The effect becomes more pronounced with increased beta ratio (ratio between pipe diameter and bore diameter of an orifice) and increased Reynolds number.

Unlike factory tests where the requirements for straight runs will always be met, in the field, overall installation consideration might lead to shorter straight runs than desirable. ISO 5167 (1980) describes the required straight lengths for orifice plates, nozzles, and venturi nozzles (Table 2). It also shows that considerably shorter straight runs are allowed if additional uncertainties are acceptable. Similar information can also be found in AGA Report No. 3 (1991).

Gas Composition

Gas samples of fuel gas and compressed gas should be taken before and after the test. If changing gas composition is suspected, samples should also be taken during the test. Besides having a serious impact on the measured gas turbine heat rate, the composition of the fuel gas also affects the gas turbine power output. On gas compressor tests, changing gas composition has to be suspected when the head versus flow test points start to deviate from the predicted curve or from the factory test curves. In high-pressure applications, some of the gas may condense at the sample bottle walls, so it is especially important to assure that the gas samples do not form liquid drop outs.

Speed

The power turbine and gas producer speed, as well as the gas compressor speed, must be recorded for all test points. The speed pickups supply the data to be displayed by the programmable logic control (PLC). These numbers are always accurate enough for the test purpose. It is particularly elegant if the electronic data acquisition system is able to communicate with the PLC and provide the data online.

Torque

Various torque measuring systems are available in the industry. All of them measure the torque applied to the coupling shaft by

determining the twist of that shaft. If a torquemeter is used, the total uncertainty for the gas turbine power can be reduced to be about one percent to 1.5 percent. Even lower values are reported (Schmitt and Thomas, 1995). Most manufacturers claim accuracies of about one percent, which has been confirmed in successful performance tests. The crucial point with torquemeters is to maintain the calibration during the test in a harsh environment, and at relatively high speeds. Using an independent measurement is recommended as a means to check the results and to verify the calibration before and after the test.

A torquemeter measurement also gives a good baseline for cross-checking the compressor performance. It must be noted that care has to be taken when selecting a torque measuring coupling, because these couplings can introduce unwanted vibrations into the train.

Data Acquisition

Field testing has been simplified in recent years by the widespread application of portable computers that have introduced powerful analytical tools directly to remote locations where standard digital controls measure and display all necessary parameters.

In particular, it is now possible to monitor all measured parameters, such as pressures, temperatures, and speeds, with a specially programmed laptop computer. A typical test setup is shown in Figure 8. It isalso possible to extract some of the necessary data from the package control system. An analog signal from a pressure transmitter or a thermocouple is transformed into digital data. The advantage of this procedure lies in the fact that all data for one measuring point can be taken at virtually the same instant, thus eliminating measurement inaccuracies due to the inevitable slight fluctuations in operating conditions. This becomes especially important when correlating the gas turbine performance and the gas compressor performance. Modern portable computer data acquisition systems also allow taking a large number of data samples for each test point, thus reducing the overall statistical error. Setup time and, especially, test time are reduced dramatically. These systems also allow online data reduction. It is, therefore, easy to identify faulty data. Reading errors that may occur when gauges are read and data are entered manually can be avoided. The savings in testing time due to faster data acquisition and reduction can be considerable.

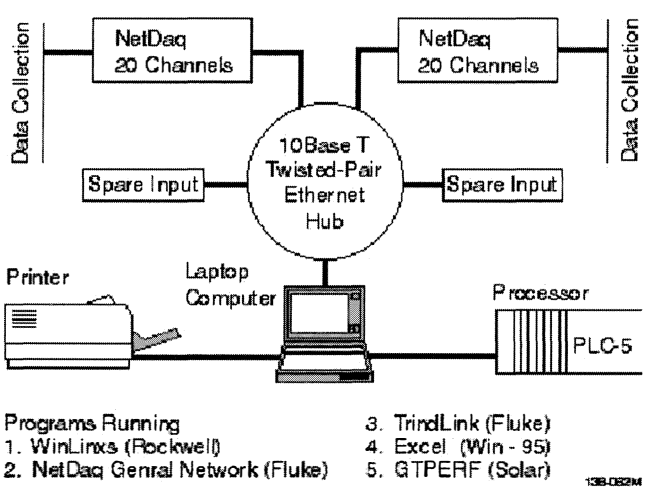

Figure 8. Data Acquisition System.

The data acquisition system in Figure 8 can be adapted to virtually all configurations of gas turbine driven compressor sets and is also easy to transport.

EXECUTING THE TEST

Steady-State Conditions

Stable conditions are critical for a good test. If the process is not in a steady-state, the conservation equations for flow and energy have to consider the storage effects. Component temperatures and clearances between stationary and rotating parts show distinct time lags. These differences cannot be resolved by using the "normal" test instrumentation. The test setup cannot account for time-dependent effects. All test codes, therefore, allow only a certain fluctuation of measured parameters. No matter how small these fluctuations are, they will always increase the error margin for the test data. It is sometimes recommended to average the sampled data to eliminate these errors. Since all the governing equations describe nonlinear relationships, the averaging of data will not resolve this problem.

Stable conditions are especially critical for temperature measurements. Temperature probes are often inserted into thermowells and reach equilibrium by convection. Convective heat transfer is not instantaneous. Consequently, it is necessary to maintain the same operating conditions for a longer period of time until the equilibrium ("heat soak") is reached. In addition, the large heat storing capacity of the compressor casing will need time to reach a new equilibrium after operating conditions change.

If a compressor set is operated in a pipeline, the relationship between flow and pressure ratio—the potential operating points of the compressor—cannot be selected arbitrarily, but is imposed by the pipeline conditions upstream and downstream of the compressor station. Also, the supply into the upstream pipeline and the demand from the downstream pipeline cannot be controlled for test purposes. They are determined by the supply of various sources into the pipeline and the demand from the consumers downstream. Running the gas turbine for test purposes at full load may cause the gas compressor throughput to exceed the upstream supply and the downstream demand. This will cause unsteady conditions, which are deliberately employed in typical pipelines and referred to as "line pack." Under such conditions, a steady drift in measured pressure values will be observed (Figure 9). The temperature measurements will not reach equilibrium and the efficiency of the compressor will be overstated. Because the compressor is used to measure gas turbine power, the gas turbine power and, thus, the gas turbine efficiency will be represented too low. Many stations have recycle loops designed into the station. With an aftercooler designed for full recycle, closed loop testing at steady-state conditions is possible. If, however, the aftercooler is not capable of removing the energy input of the compressor, the loop temperature will rise continuously and, again, make it impossible to measure performance accurately.

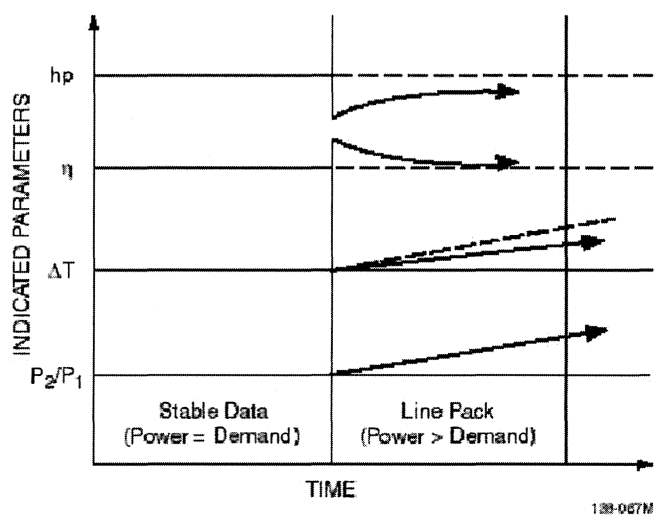

Figure 9. Line Pack Conditions During Field Test.

Steady-state operations not only include stability of pressures, temperatures, and speeds, but also the stability of the gas composition (Table 4 (ASME PTC 10, 1997 and PTC 22, 1997)).

Table 4. Allowable Fluctuation of Test Readings During a Test Run.

Measurement	Fluctuation
Inlet Pressure	2%*
Discharge Pressure	2%*
Orifice Differential Pressure	2%
Orifice Temperature	0.5%
Inlet Temperature	0.5%
Speed	0.5%
Torque	1.0%
Specific Gravity, Test Gas	0.25%
Electrical Power Output (Gen Sets)	2%
Torque	2%
Power Factor	2%
Speed (Gas Producer, Power Turbine)	1%
Site Barometric Pressure	0.5%
Air Temperature at Intake	4°F **
LHV of Gas Fuel	1%
Pressure of Gas Fuel	1%
Exhaust Back Pressure (abs)	0.5%
Air Inlet Pressure	0.5%
Exhaust Temperature	5°F
Fuel Consumption	2%

* = Especially for low pressure ratio compressors, 1% should be used.
** = We recommend 1°F

Furthermore, care must be taken when averaging the data from several readings. To get correct results, the averaging should not be performed on the raw test data, such as pressures and temperatures, but only for the results, including efficiency, power, etc. Because many relationships are nonlinear, averaging the measured parameters and calculating the results with these averages will create different results than calculating the results for each sample point and averaging the results. Only the latter procedure will yield correct averages. Data averaging by itself does not remove the fundamental requirement of steady-state conditions.

DATA REDUCTION

The objective of the test is typically to verify acceptance criteria such as heat rate, specific fuel consumption,

$$HR = \frac{W_f{*}LHV}{P} \qquad HRpackg = \frac{W_f{*}LHV}{P{*}\eta^*} \qquad (1)$$

the shaft power, P, of the engine and the gas power, P_g, of the compressor,

$$P_g = W \cdot H \qquad (2)$$

the compressor efficiency, η, compressor actual head, H, isentropic head, H^*, and mass flow W.

The acceptance criteria need to be clearly defined. In addition to the absolute number of the property in question, the ambient conditions, such as temperature, elevation, fuel gas composition, and pressure for the engine, and isentropic head, suction temperature, actual flow, and gas composition for the compressor, need to be well defined. While the test uncertainties cannot be determined until the test is executed, the method to calculate the uncertainties should be agreed upon in advance.

The following paragraphs describe the path from measuring the physical parameters to the desired end results.

Gas Turbine Power

While testing a gas compressor in the field can yield similar test uncertainties as the factory test, given that the field test is conducted using the same standards as for the factory test, the field testing of the engine will typically yield higher test uncertainties than the test in the factory.

The main reason lies in the methodology of measuring the shaft power. In the factory, the shaft power is measured by running the gas turbine against a dynamometer or other calibrated device. The power turbine applies torque onto the dynamometer. A load cell is used to measure the reaction force on the casing. The shaft power is calculated by multiplying the measured torque, τ, and the measured shaft speed:

$$P = 2\,\pi\,N\,\tau \qquad (3)$$

In the field test, the shaft power is determined in one or more of the following ways:

- Using the calculated power of the driven compressor (heat balance)
- Using a torque measuring coupling between power turbine and driven equipment
- Verifying with a redundant measurement

Field tests with a torque metering coupling can achieve almost a similar accuracy as the factory test method. Using the power input into the driven compressor is subject to significantly higher measuring uncertainties, as described later in this paper. The absorbed power is calculated by:

$$P = \rho\,Q\,H\,\frac{1}{\eta_m} \qquad (4)$$

The third method takes advantage of the conservation of energy in a thermodynamic system (Figure 10), thus requiring that the energy flowing in be balanced by the energy leaving the system:

$$w_1h_1 + w_fE_f\eta_{comb.} + w_fh_f = (w_1 + w_f)h_7 + P + E_r + E_m \qquad (5)$$

The mass flow and enthalpy of the air at the gas turbine inlet, as well as the fuel flow, fuel enthalpy, lower heating value of the fuel, and enthalpy of the exhaust gas, h_7, can be measured. The radiated heat energy, E_r, and the mechanical losses, E_m, leaving the system as heat transferred to the lube oil can be estimated, but will be rather small. The combustion efficiency can be estimated as well and is typically about 99 percent. Therefore, the shaft power of the turbine, P, can be calculated. It is essential for this method to measure the airflow through the gas turbine. This is difficult to do in the field with high accuracy. However, the equation is useful to verify one of the three other methods, because most of the gas turbine characteristics, including airflow versus gas producer speed, are recorded during the factory test. Using the pressure differential that is created when the air is accelerated in a

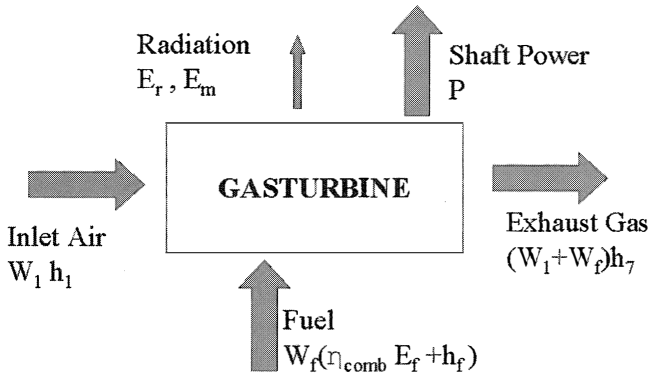

Figure 10. Energy Balance for a Gas Turbine.

converging inlet system, or by using the fact that the first stage gas producer turbine nozzle often operates at a choked condition, allows determination of the airflow into the engine, if the necessary correlations between the pressure differential and the airflow were established during the factory test. Thus, it is possible (at least to some degree) to substitute properties that cannot be measured in the field with information gathered during the factory test.

Heat Balance with Gas Compressor

The gas power or aerodynamic power of the compressor can be calculated as described in the section below. To get from the aerodynamic power to the shaft power, the mechanical losses of the compressor need to be considered to calculate the shaft or brake power of the gas turbine (Equation (4)). The uncertainties related to the measurement of the power required for the gas compressor will be discussed later.

Gas Turbine Heat Rate or Fuel Flow

If the output power of the gas turbine is known, the heat rate is calculated from the fuel flow by:

$$HR = \frac{W_f\, LHV}{P} \tag{6}$$

The fuel gas composition allows the determination of the low heating value (LHV) of the fuel and is also used to calculate the thermodynamic properties of the combustion gas. These properties determine, among others, the power output and the efficiency of the hot section of the gas turbine, and thereforehave to be taken into account. The fuel mass flow has to be measured with a metering run.

Note that for determining the heat rate, the low heating value (LHV) and not the high heating value (HHV) has to be used. When the heating value of a gas is measured, the combustion products have to be cooled down to the temperature of the initial components. The heat that can be extracted during this process is the heating value. When hydrocarbons are burned, one of the reaction products is water. During the process of cooling the combustion products back to feed temperature, water may become liquid. HHV is the total heat released by unit mass of fuel burned, while the LHV is the HHV less the heat absorbed by vaporized water formed during combustion. With the typical exhaust temperature in a gas turbine, the water will not condense, thus it does not contribute to the energy generation in the gas turbine.

Gas Turbine Air Flow

One of the challenges in field performance testing lies in the correct determination of the air flow. The air flow is necessary to calculate the exhaust flow (necessary for combined cycle and cogeneration applications) and, by using Equation (11), to determine the correct setting of the control temperature.

In the absence of a venturi nozzle in the inlet (as in a factory test), there are two potential methods of determining the air flow:

- *Method 1*—Pressure differential between inlet muff and bell housing
- *Method 2*—Using the first stage nozzle flow

Both methods require the analysis of factory test data.

Method 1 uses the fact that the area contraction between the inlet muff and the bell housing (Figure 3) creates a pressure differential that can be correlated with the airflow. The correlation can be derived from factory test data, and typically has the form:

$$W \cdot \frac{\Theta}{\delta} = A \cdot \left(\frac{\Delta P}{P}\right)^B \tag{7}$$

A and *B* are constants derived from the factory test data. However, changes in the inlet configuration can make these correlations relatively inaccurate.

Method 2 uses the fact that the first gas producer turbine nozzle operates in choked or near choked condition, which means that the velocity at the throat of the nozzle is equal to the speed of sound (Figure 11 (Kurz, 1991)), and thus only a function of pressures and temperatures upstream of the nozzle. That means that the nozzle exit area, A_n, determines the actual flow through that nozzle (Cohen, et al., 1996):

$$W \frac{\sqrt{T_3}}{P_3} = A_n \cdot \sqrt{\frac{\gamma}{R}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}} \tag{8}$$

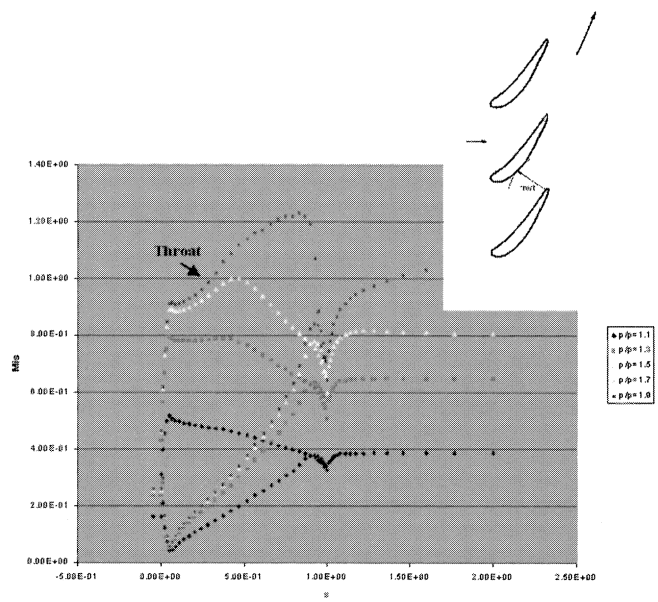

Figure 11. Velocity Distribution in a Turbine Nozzle.

Knowing the pressure and temperature (and gas composition, which determines *R* and γ) at the entrance into the nozzle allows the determination of the mass flow, W, through that nozzle. The cooling and, potentially, the bleed flow can be included such that *An* is replaced by A^*, such that:

$$A^* = A_n + A_{cool} + A_{bleed} \tag{9}$$

A^* is unique for every turbine and can be determined during the factory test.

Equation (5) can be rewritten (using $\Delta h = c_p\, \Delta T$):

$$w_1 c_{pa}(T_1 - T_7) + w_f c_{pf}(T_f - T_7) + w_f E_f \eta_{comb.} - E_r - E_m = P \tag{10}$$

with c_{pa} and c_{pf} appropriate averages of the pertinent c_p.

One can also use another heat balance to estimate the thermodynamic turbine inlet temperature, T_3, from a similar balance for the combustor:

$$w_1\, c_{p,1}(T_2-T_1)\eta_{mech} = (w_1+w_f)c_{p,3}(\tilde{T}_3-T_5)$$

with (11)

$$c_{p,3}\tilde{T}_3 = \frac{c_{p,3}(w_1+w_f-w_{cool})T_3+c_{p,cool}T_{cool}w_{cool}}{w_1+w_f} \approx c_{p,3}T_3$$

Note that the properties of the cooling air and the cooling air flow, w_{cool}, can only be guessed, because no direct way of measuring it is available. The above equation also assumes that no bleeding occurs.

Gas Compressor Power, Efficiency, Head, and Flow

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

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

$$H^* = h(p_d,\Delta s=0) - h(p_s,T_s) \tag{12}$$

The actual head, *H,* is:

$$H = h(p_d, T_d) - h(p_s, T_s) \tag{13}$$

The isentropic efficiency then becomes:

$$\eta_s = \frac{H^*}{H} \tag{14}$$

It should be noted that the polytropic efficiency is defined similarly, using the polytropic process instead of the isentropic process for comparison. The actual head, which determines the absorbed power, is not affected by the selection of the polytropic or isentropic process.

The actual flow, *Q,* will be measured by any flow measuring device, such as an orifice or a nozzle. The aerodynamic or gas power of the compressor, then, is determined to be:

$$P_g = \rho_1\, Q_1\, H = \frac{p_1}{Z_1\, R\, T_1}\, Q_1\, H \tag{15}$$

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 nonsteady 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. The method to use increased vibration levels as an indication of surge, or incipient surge, is even more inaccurate, because the increased vibration levels might be generated by the onset of rotating stall (which is by no means identical with the onset of surge), or other conditions.

Equations of State

The state of any fluid consisting of known components can be described by any given pair of its pressure, specific volume, and temperature. Equations of state (EOS) approximate these relationships. 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, 1981).

The simplest equation of state is the equation for a perfect gas:

$$p\,v = p/\rho = RT$$

$$H = h_2-h_1 = c_p(T_2-T_1) \tag{16}$$

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

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

$$\frac{p}{\rho} = Z \cdot R \cdot T \tag{17}$$

They also allow for the calculation of properties derived from the p-v-T relationships, such as enthalpy, *h,* and entropy, *s.* Because EOS are semiempirical, 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.

Note that not only enthalpy and entropy, but also the compressibility factor is a function of temperature and pressure. The compressibility factor at the discharge of a compressor thus has to be taken at discharge pressure and temperature. Correcting it only for pressure will lead to erroneous results.

Because different EOS will yield different values for density, enthalpies, and entropies, the EOS has to be agreed upon before the test. Table 5 shows the effect of different EOS on the results for a given set of typical test data. Beinecke and Luedtke (1983) have conducted thorough evaluations on the accuracy of the Lee-Kesler-Ploecker (LKP) method, the Benedict-Webb-Rubin-Starling (BWRS) method, and the Soave-Redlich-Kwong (SRK) method. The SRK method is an extension of the Redlich-Kwong (RK) method. Because the extension is limited to gases close to their critical point, there are no differences in the results between these two methods for typical compressor applications. Both the LKP and the BWRS method are modern versions of the original Benedict-Webb-Rubin (BWR) method.

Table 5. Head, Isentropic Head, Efficiency, and Compressibility Factors Calculated with Different EOS.

EOS	H (ft lb_f/lb_m)	H* (ft lb_f/lb_m)	0*=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 (Robinson and Jacobi, 1965)	-	-	-	0.9259	-

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)

All the EOS mentioned can predict the properties of hydrocarbon mixtures quite accurately over a wide range of pressures (Beinecke and Luedtke, 1983). Still, deviations of 0.5 percent to 2.5 percent and more in the values for the

compressibility factor, Z, are common. Even more important than the compressibility factor is the calculation of the enthalpy and entropy using the EOS. Because derivatives of the EOS have to be used to perform these calculations (Reid, et al., 1977), the deviations can be even larger than for the compressibility factor.

As shown in Table 5, four different EOS deliver four different results for the same measured conditions. All compressibility factors Z are relatively close together, except for Peng-Robinson (PR). Even though the isentropic efficiency differs by only one point, the highest and lowest actual head H are almost 2.5 percent apart. Because the actual head is used (together with the mass flow) to calculate the compressor power (Equation (2)), considerable differences can arise from the choice of the EOS. It should be noted that even the results for the same EOS may differ from program to program because sometimes different mixing rules are used or the different interaction parameters between the gases are assumed to calculate the constants in the EOS.

The aerothermodynamic 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 using EOS. Usually, it is not possible to select a "most accurate" EOS to predict enthalpy differences, since there is usually no "calibration normal" against which to test. All the frequently used EOS (RK, BWR, BWRS, LKP, SRK, PR) show reasonably correct enthalpies. It is just not possible to decide which of them is more accurate for a given application (Kumar, et al., 1999). Therefore, 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 VDI 2045 (1993) to avoid additional test uncertainties.

TEST UNCERTAINTIES

Theoretical Background

For the uncertainty analysis, it is assumed that all measurement parameters can be considered to be independent and that parameters have associated statistical bounds, such as a 95 percent confidence interval Δu, rather than absolute limits of errors. All parameters also are assumed to have Gaussian normal distributions around their respective mean values such that the uncertainties can be properly combined using the root-square sum method. However, an uncertainty correction is added for parameters that have sample sizes smaller than 30, which means the uncertainty is widened for individual parameters to account for a student t-type distribution (Brun, 1996). The total uncertainty, ΔF, for a given function, $F = f(u_1, u_2, ..u_n)$, is, thus, determined from:

$$\Delta F = \sqrt{\left(\Delta u_1 \frac{\partial f}{\partial u_1}\right)^2 + \left(\Delta u_2 \frac{\partial f}{\partial u_2}\right)^2 + ... \left(\Delta u_n \frac{\partial f}{\partial u_n}\right)^2} \qquad (18)$$

For this method, the overall uncertainty, ΔF, has the same statistical meaning as the individual uncertainties, Δu. Namely, if Δu represents a 95 percent confidence, then the result for the total uncertainty, ΔF, is also a 95 percent confidence interval. A more detailed analysis of the subject matter can be found in Brun and Kurz (1998).

VDI 2045 (1993) and VDI 2048 (1978) propose to use an uncertainty ellipse to introduce the test uncertainties into the evaluation. In a statistical sense, the test is used to prove the "hypothesis" of the performance prediction. The performance prediction is true if it lies within the uncertainty ellipse around the test result (Figure 12). This has the advantage of using a physically sound way of introducing the unavoidable uncertainty of the test results. Using an uncertainty ellipse acknowledges the fact that both independent variables have an uncertainty margin. Since it is not possible to distinguish between bias errors and data scatter in a field test environment, they will not be treated independently here.

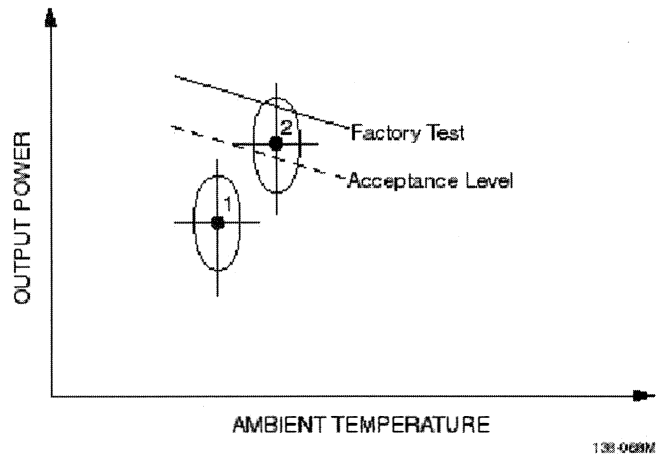

Figure 12. Uncertainty of Two Independent Parameters. (Point 1 shows the tested engine power at site. The uncertainty ellipse is below the level established in the factory test. Point 2 shows the retest after a thorough cleaning of the air compressor. While still slightly below the factory test, the uncertainty ellipse confirms the factory test.)

For complex relationships (e.g., when equations of state have to be considered), Equation 18 is rather difficult to use, because the partial derivatives are not easy to obtain. An easy way out is the following (Moffat, 1988): "If a data reduction program exists—for example a program that calculates compressor shaft power from flow, pressure and temperature measurements—then the same program can be used to estimate the uncertainty in the result. This is accomplished by sequentially perturbing the input values by their respective uncertainties, and recording their effects."

Any term in Equation 18 can be approximated (assuming that the error is relatively small) by:

$$\left(\Delta u_1 \frac{\partial f}{\partial u_1}\right) \cong f(u_1 + \Delta u_1) - f(u_1) \qquad (19)$$

That means that the contribution of the variable, u_1, to the uncertainty in f can be found by calculating f twice. Once with the observed value of u_1 and once for $u_1 + \Delta u_1$, and then subtracting the two values of f. When several variables are involved, the overall uncertainty can be found by sequentially perturbing the individual variables, u_1, and then finding the square root sum of the squares of the individual terms. This can be done on a spreadsheet.

To illustrate the aforementioned, the uncertainty for the gas turbine heat rate is demonstrated. The heat rate can be calculated by:

$$HR = W_{fuel} \frac{LHV}{P_{shaft}} \qquad (20)$$

Assuming the fuel flow, W_{fuel}, was determined to be 1553 scfm (2500 nm^3/h, 0.9 percent uncertainty), the lower heating value (LHV) to be 940 Btu/scf (36,940 kJ/nm^3) with an uncertainty of 0.75 percent, and the shaft power, P_{shaft}, to be 13,410 hp (10,000 kW) with three percent uncertainty (which is an uncertainty that is achievable by using a driven compressor to measure the shaft power), the uncertainty of the heat rate can be calculated by applying Equations (18) and (19), as demonstrated in Table 6. Any effort in improving the test uncertainty in this example would thus focus on improving the measurement of the shaft power.

The beauty of this scheme lies in the fact that:

- It does not matter whether the uncertainty is given as an absolute or relative number.

Table 6. Spreadsheet for Test Uncertainty Calculation.

Parameter	Unit	Uncertainty	Nominal Value	W_{fuel}	LHV	P_{shaft}
W_{fuel}	Nm3/h	0.9%	2500	2522.5	2500	2500
LHV	kJ/Nm3	0.75%	36940	36940	37217.05	36940
P_{shaft}	kW	3.0%	10000	10000	10000	9700
Heat Rate	GJ/h		9235	**9318.115**	**9304.2625**	**9520.6186**
Deviation	GJ/h		0	83.115	69.2625	285.6186
Deviation^2	(GJ/h)^2		0	6908.1	4797.3	81578.0
Sum of Deviations	(GJ/h)^2		93283.4			
HR uncert.	(GJ/h)		305.42			
HR uncert	%	**3.3**				

- The procedure can be implemented using any of the commercial spreadsheet programs.
- Any value in the table can be the result of a complex, even iterative calculation.

When evaluating the relationships that govern test uncertainties, some considerable differences in the uncertainties for the overall package and the individual components can be seen. For example, in a gas turbine driven compressor package, the uncertainties for gas turbine power, gas turbine heat rate, and compressor efficiency may be determined individually, if the acceptance criteria require it. On the other hand, an overall package acceptance criteria might be specified that simply requires the necessary gas turbine fuel flow to achieve a certain compressor operating point.

Economy Versus Accuracy

The cost of a field performance test includes not only the cost for the instrumentation, the test hardware, and the personnel, but also the cost of lost or interrupted production. Thus, the test should be conducted as quickly and with as little effect on production as possible. Repeating the test because test data are questioned after the test should be avoided. Good planning and a speedy, but thorough, test execution using electronic data acquisition cannot be overemphasized.

Part of preparing for the test is to analyze which effects will most likely have the largest contribution to the test uncertainty and which effects will not. For example, an uncertainty analysis (Brun and Kurz, 1998) should show which are the most critical measurements for a given test situation. Figure 13 shows the effect of improving the accuracy of temperature measurements for test situations with different pressure ratios. While an improvement in the accuracy of the temperature measurement from the baseline accuracy of 0.6°C (1°F) will hardly affect the overall accuracy for high pressure ratios, its effect becomes substantial for low pressure ratios. For low pressure ratios, as frequently found in pipeline applications, any extra effort to improve the temperature measurements will be worthwhile. However, if the power measurement is only possible with three percent to four percent uncertainty, it is not worthwhile to enhance the fuel flow measurement from one percent to 0.5 percent uncertainty.

A carefully designed and executed field test can provide results that can closely reproduce prediction and factory tests (Figure 14). For a gas turbine driving a compressor, however, the factory test is normally more accurate than the field test, because the test uncertainties for power are much higher in the field test than in the factory test, where the power can reliably be measured with a calibrated dynamometer. Furthermore, the factory test allows a more accurate determination of the airflow through the engine. The factory test will also provide valuable data that can be used in the field, such as correlations for the airflow at different gas producer speeds.

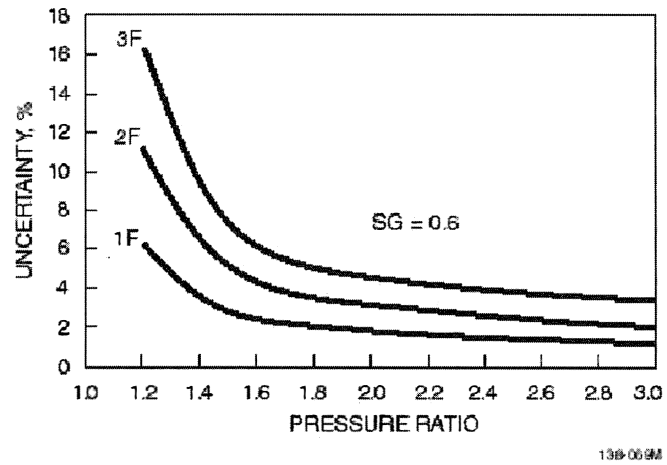

Figure 13. Effect of Temperature Accuracy for Different Pressure Ratios (138-069M).

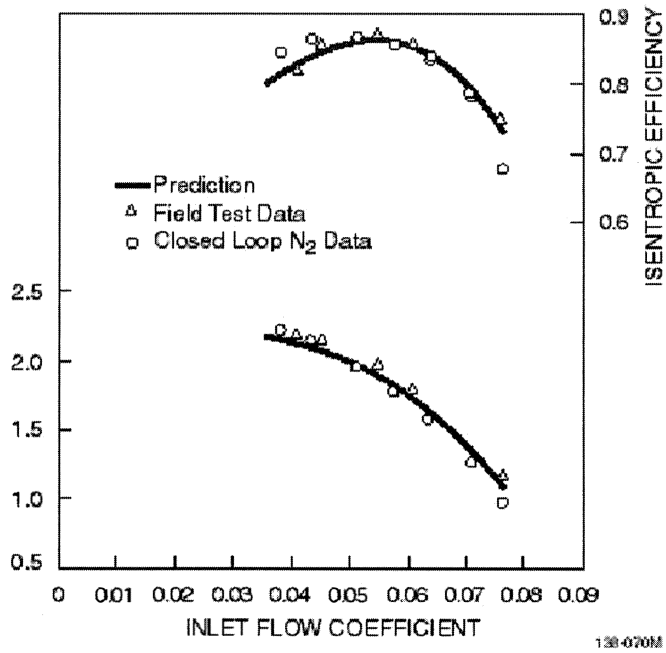

Figure 14. Comparison of Test Results from a Closed Loop Factory Test and a Field Performance Test with the Prediction.

CORRECTION OF TEST CONDITIONS TO DESIGN POINT CONDITIONS

Because the package will invariably be tested under conditions that deviate from the design conditions, adjustments must be made. This is accomplished by conducting the tests following the laws of similarity theory; in other words, the flow characteristics through the machines must be similar for the test and the acceptance condition. Preferably, the test conditions should copy the acceptance conditions as closely as possible.

It must be noted that the governing conditions for the laws of similarity are different for gas turbines and gas compressors. Therefore, for compressor sets, the correction typically has to be performed independently for the compressor and for the gas turbine.

Similarity Conditions for Gas Turbines

The most influential characteristics for the gas turbine operation are inlet temperature, power turbine speed, ambient

pressure, and, to a lesser degree, fuel gas composition and relative humidity. According to ASME PTC 22 (1997), "it is necessary to have the test conditions within limits agreed to by the parties to the test to avoid running the gas turbine at extreme conditions far from its design or rated condition, which would make the determination of accurate corrections impossible....the off-design characteristics of each gas turbine are unique. Hence, the manufacturer's....performance curves for the particular engine must be used to correct the actual test data to rated or standard conditions."

Because a gas turbine consists of three major rotating components (air compressor, gas producer turbine, and power turbine) and the combustor, the application of similarity considerations, as typically used for gas compressors, is virtually impossible. The goal would be to operate the gas turbine in such a way that the component efficiencies are the same as for the acceptance point. For the gas turbine compressor, this is achieved by maintaining identical corrected speeds, thus maintaining the same Mach numbers. For the gas producer turbine, this is similarly achieved by additionally maintaining the same temperature ratio. The power turbine, which is not mechanically connected to the gas producer shaft in two shaft gas turbines, will invariably run at a nonsimilar operating point. Additionally, since the fuel gas composition during the test might differ from the design values, the gas behavior in the hot section can be different. ISO 2314 (1989) mentions a correction for ambient temperatures and ambient pressures:

$$T_{3t} / T_{3a} = T_{1t} / T_{1a}$$

$$N_{GPt} / N_{GPa} = \sqrt{\frac{(kRT_1)_a}{(kRT_1)_t}} \qquad (21)$$

For compressor sets with two-shaft gas turbines, the above correction is only possible if the free gas producer reaches an operating condition that fulfills both conditions at the same time. Additionally, the power turbine must run at the same speed relative to the optimum power turbine speed under given conditions. All this is normally not possible (Cohen, et al., 1996).

Therefore, the only accurate way of correcting the operation of gas turbines to acceptance conditions is by using manufacturer software that models the gas turbine or manufacturer-supplied correction curves. This is especially important if part-load operating points were agreed upon. Since many modern gas turbines are controlled by gas producer speed, firing temperature, and, possibly, variable guide vanes or bleed valves, simple curve matches are not sufficient to describe the gas turbine performance variation due to different operating points.

During field test, the gas turbine is operated at the prevailing ambient conditions. To correct the measured power and heat rate to acceptance conditions, gas turbine specific performance curves (Figures 15, 16, 17, and 18) are used, showing power and heat rate as a function of ambient temperature, with correction factors for barometric pressure, inlet and exhaust losses, and power turbine speed. The observed values are compared with predicted values for the same ambient conditions. Then, the percentage difference between observed and predicted values for the test conditions can be applied to the predicted values for the acceptance conditions. The same procedure, but with higher accuracy, is possible by using computer programs that generate the curves. The gas turbine load should also be similar under test conditions and acceptance conditions. If a gas compressor is used to determine the gas turbine shaft power, this may lead to additional test points where the driven equipment is operated at other than the acceptance operating points. The reason is that gas turbine performance is very sensitive to ambient conditions (pressure and temperature), while the performance of the driven compressor is not.

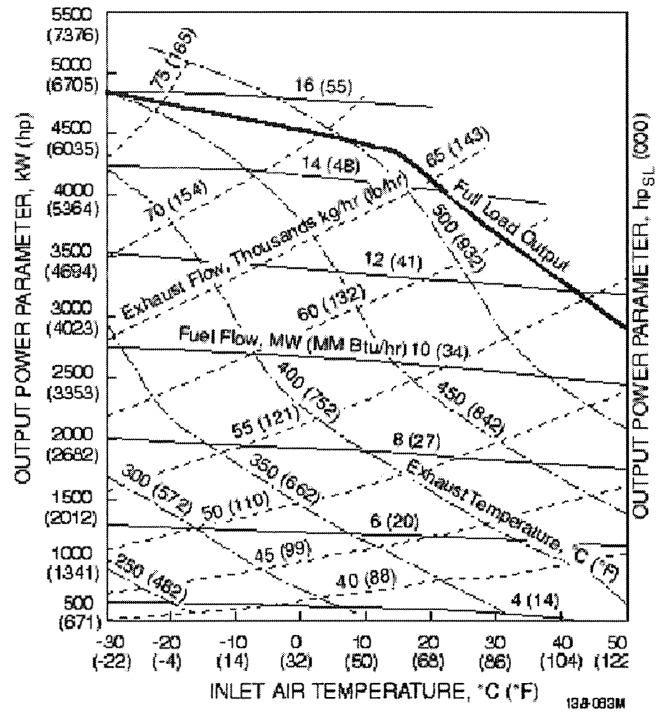

Figure 15. Gas Turbine Correction Curve.

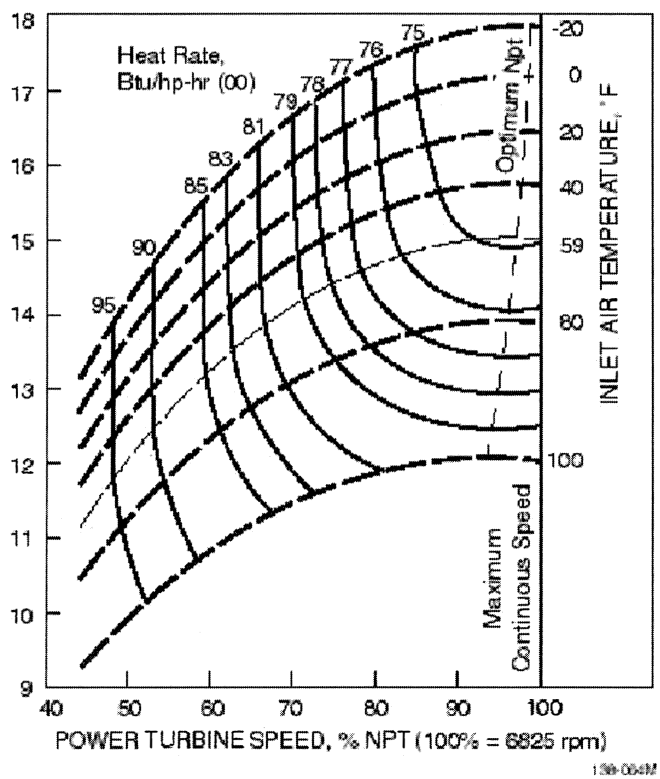

Figure 16. Gas Turbine Correction Curve.

A deviation, for example, of 11°C (20°F) from design ambient temperature has hardly any influence on the operating point and especially the power consumption of the gas compressor. For the gas turbine, it may create the difference of operating at full load or at 95 percent load. Because the heat rate of the gas turbine is sensitive to part load, the results can be quite different for the different loads, particularly for gas turbines that bleed air at part-load operations (such as for emissions control).

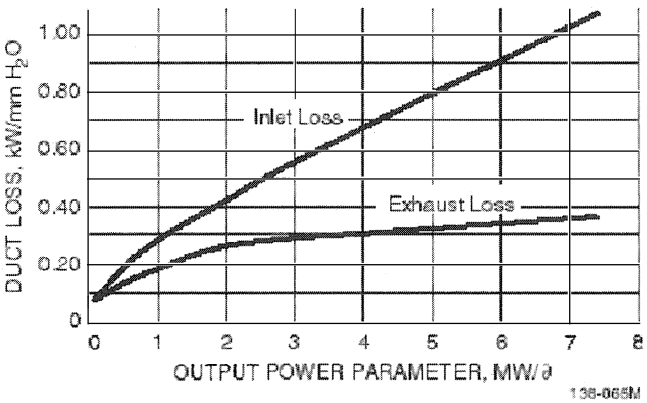

Figure 17. Gas Turbine Correction Curve.

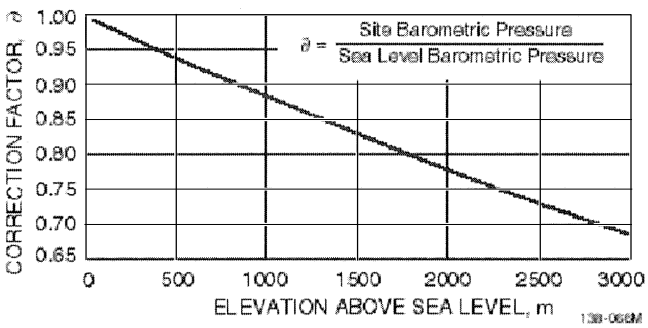

Figure 18. Gas Turbine Correction Curve.

Based on the correction methods mentioned above, it will always be possible to come to test conditions that allow a meaningful demonstration of the acceptance criteria.

The authors want to review briefly why the Mach number is such an important variable to observe. Figure 11 (Kurz, 1991) shows the behavior of the compressible gas flow through a turbine nozzle. In this example, the flow through a gas turbine nozzle shows extremely different behavior for different Mach numbers. It is not just a shift of the velocities, but the entire shape of the velocity distribution along the surface of the nozzle changes. Along with these changes, flow capacity and losses change significantly. The example refers to a turbine nozzle, but similar effects can also be observed both in axial as well as in centrifugal compressors (Figure 19, with flow versus head and flow versus efficiency curves for different Ma_u). It becomes quite clear that the Mach number, therefore, has significant effects on losses and operating range.

On the other hand (Figure 20 (Dejc, 1973)), the influence of Reynolds numbers is usually not as significant, in particular because the Reynolds number levels as well as the turbulence levels in industrial gas turbines and gas compressors are rather high.

Similarity Conditions for Gas Compressors

For the compressor, similarity is accomplished if the following similarity parameters are the same for test and acceptance conditions:

- Flow coefficient:

$$\varphi = \frac{Q_s}{\frac{\pi}{4} D^2_{1,tip}\, u} \tag{22}$$

- Head coefficient (isentropic or polytropic):

$$\psi^* = \frac{H^*}{\frac{u^2}{2}} \qquad \psi^p = \frac{H^p}{\frac{u^2}{2}} \tag{23}$$

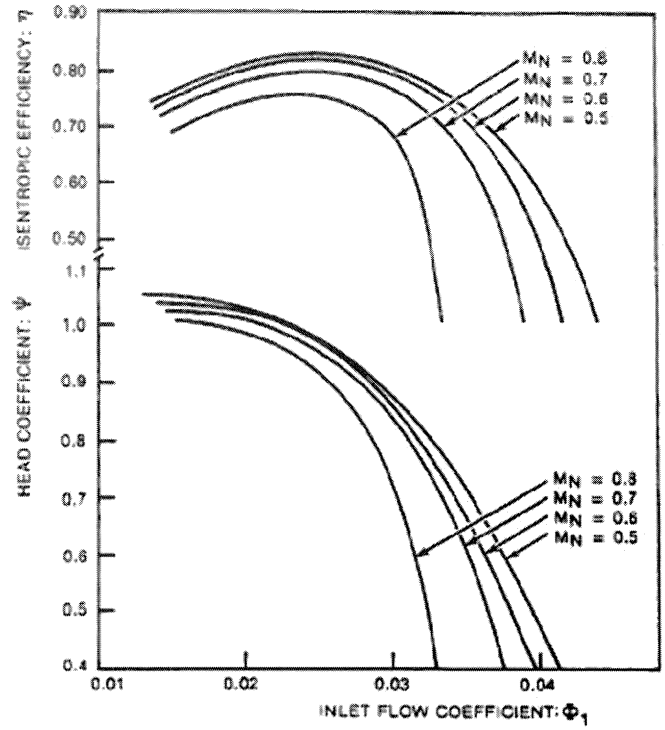

Figure 19. Effect of Machine Mach Number on Head and Efficiency of a Centrifugal Compressor.

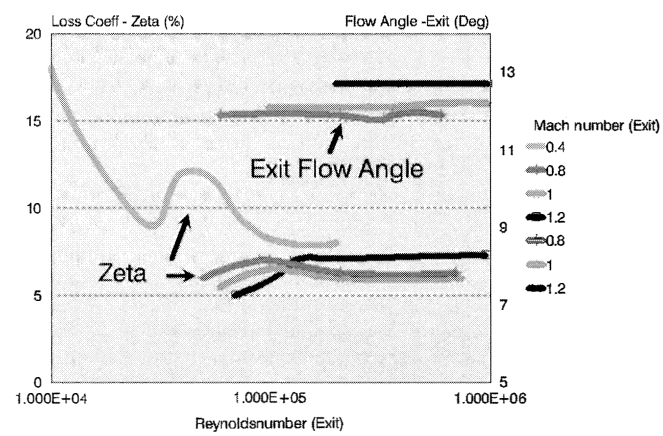

Figure 20. Effect of Reynolds Number and Mach Number on Losses and Turning of a Turbine Blade.

- Machine Mach number:

$$Ma_u = \frac{\pi\, D_{tip}\, N}{\sqrt{k_s Z_s R T_s}} \tag{24}$$

- Machine Reynolds number:

$$Re_u = \frac{\pi\, D_{tip}\, N\, b_{tip}}{v_s} \tag{25}$$

- Isentropic exponent:

$$k = \left(\frac{v\, \delta p}{p\, \delta v}\right) \tag{26}$$

- Ratio of volume flow ratios:

$$(Q_1/Q_2)_t = (Q_1/Q_2)_a \tag{27}$$

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.

Typically, 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 and the machine Mach number. Maintaining the ratio of volume flow ratios is also desirable. 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 same average isentropic exponents over the machine. For most applications, the Reynolds number similarity is of lesser importance because 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. ASME PTC 10 (1997) allows the deviations between design and test case for the parameters as listed in Table 7. In general, as long as the deviations between test and design stay within these limits, 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 (Equations (22) and (23)) as the design point. The limitations of the fan law are covered by Brown (1991).

Table 7. Acceptable Departures of the Test Conditions from Design Conditions.

	Symbol	Departure %
Inlet Pressure	ps	5
Inlet Temperature	Ts	8
Specific Gravity of Gas	SG	2
Speed	N	2
Capacity	Q_i	4
Inlet Gas Density	ρ_1	8

VDI 2045 (1995) provides very specific guidelines about the deviations in volume ratio. If the volume ratio between acceptance criteria and test exceeds ± one percent, additional tolerances have to be applied.

If the test conditions are considerably different from the design conditions, for example outside the limits established in ASME PTC 10 (1997) (Table 7), 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. Colby (1987) states that, especially for compressors in applications where the compressibility factors change rapidly from suction through discharge, the deviations allowed for a PTC 10 Type 1 test might still be too high to simulate field conditions.

PTC 10 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, the compressor performance can be recalculated for the actual test gas, or the provisions for a PTC 10 Type 2 test or the provisions as put forward in VDI 2045 need to be followed.

Deviations also occur if the gas were specified incompletely, for example, by only defining the specific gravity rather than a full gas composition.

INTERPRETATION OF TEST DATA

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.

Another necessary step 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 points to an incorrect flow measurement. 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.

For gas turbine power, it is helpful to use two different, independent measurements. The power can be measured by using the compressor gas power. This result can be checked by a gas turbine heat balance (Equation 5) or by comparing the results with predictions corrected by factory test data. Even if the gas turbine airflow cannot be determined accurately during the field test, this cross check might shed light on the discrepancies. It is also recommended to thoroughly clean the air compressor prior to the test: three percent and more engine power has been recovered after cleaning the air compressor.

Another reason for the discrepancies is the test uncertainty. If the test point does not meet the prediction, but a test uncertainty ellipse (Figure 12) drawn around it still covers the prediction, the test results might be correct. The uncertainty ellipse expresses the fact that not only the measured power is subject to test uncertainties, but also the ambient temperature.

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 can be used for comparison and verification purposes. 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.

CONCLUSIONS

Field performance testing has been identified as an important part of projects involving gas turbine driven compressor sets. 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 requires steady-state conditions and adequate instrumentation. An overview on the necessary instrumentation was given. The concepts of how to calculate efficiency, power, fuel flow, capacity, and head of 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.

NOMENCLATURE

A	= Area
c_p	= Specific heat at constant pressure
γ	= Ratio of specific heats
$\delta = p/p_{SL}$	= Pressure correction to sea level
$\Theta = T/T_{std}$	= Temperature correction to standard temperature (518 R)
h	= Enthalpy
k	= Isentropic exponent
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
W	= Mass flow
Z	= Compressibility factor
η	= Efficiency
H	= Head
HR	= Heat rate
LHV	= Lower heating value
EOS	= Equation of state
BWRS	= Benedict-Webb-Rubin-Starling
LKP	= Lee-Kesler-Ploecker
PR	= Peng-Robinson
RK	= Redlich-Kwong
SRK	= Soave-Redlich-Kwong
τ	= Torque

Subscripts

amb	= Ambient
d	= Discharge
f	= Fuel
M	= Mechanical
packg	= Package
s	= Suction
SL	= Sea level
TH	= Thermal

Superscripts

*	= Isentropic

APPENDIX A
DISCUSSION OF SOME PERFORMANCE TEST CODES

ASME Power Test Code 22 (1997)

ASME PTC 22 is, in its concept, written for factory tests. It defines acceptable instrumentation and instrumentation accuracies for all necessary test data. For every site performance test, it needs to be discussed whether the situation onsite allows for the stipulated test methods and accuracies.

PTC 22 acknowledges the fact that correcting engine data is only possible using manufacturers' curves or equivalent. They also acknowledge the fact that the correct setting of the control temperature (or the inability to do so without accurate airflow measurement) contributes significantly to the test uncertainties for the engine power.

The test uncertainty calculation is per ASME PTC 19, which, from a theoretical standpoint, is a correct implementation of the statistical basis of uncertainty calculation.

It needs to be decided on a case by case basis whether the instrument accuracies are always practical for a field performance test. As mentioned previously, increasing the accuracy of the fuel flow measurements to the PTC 22 requirements only makes sense if other measurements, such as power, can be performed with about the same level of accuracy. Otherwise, the added cost will not significantly improve the test results for thermal efficiency.

The acceptable variations in measured data during a test run (i.e., during a stable test point) may lead to added uncertainties in the heat rate if totalized fuel flow measurements are used.

Note that ASME PTC 22 was revised only recently (1997). The previous code had, for example, more stringent requirements for fuel flow accuracy on gas fuel, which were almost impossible to meet in the field.

ISO 2314 Gas Turbines—Acceptance Tests (1993)

This code applies to both factory and site tests. It states clearly that machines have to be cleaned, if necessary, and also emphasizes requirement for steady-state operating conditions. The allowable variations in test conditions during the test are somewhat different from PTC 22 (ISO 2314 allows one percent variation in barometric pressure, 2°C variation in ambient temperature, and two percent variation in LHV, PTC 22 allows 0.5 percent, 2.2°C, and one percent, respectively). The authors think that given the possible speed of data recording, and the fact that 2°C variation in temperature can cause two percent variation in power, the limits for temperature variation should be set to 1°C. Since the code is written to encompass a wide variety of applications, some of the required measurements (for example, exhaust temperature) can be waived, depending on what the test is supposed to prove. Both ISO 2314 and PTC 22 are very close in the requirements for fuel flow measurement accuracy (one percent and 0.9 percent, respectively) for gas fuel.

The accuracy requirement for exhaust temperature measurement of 3°C is not easily met in the field. Neither are the requirements for measuring the turbine exit pressure. The code neglects the difference between ambient temperature and compressor inlet temperature to some degree. It also does not acknowledge the effect of relative humidity on engine performance. ISO 2314 also neglects the problem of determining the correct control temperature in the field as a source of uncertainty.

The data correction procedures in ISO 2314, using a similarity approach for gas turbines, are not practical (this has been discussed in the section *"Similarity Conditions for Gas Turbines"*). A particular problem is that this specification gives no guidelines regarding test uncertainty calculations. ISO 2314 mandates taking power requirements for separately driven auxiliary equipment (such as electric motor driven lube oil pumps) into account. ISO 2314 also seems to encourage the use of torquemeters for measuring the shaft power, which may not always be practical, especially for smaller turbines. The calibration procedure (against a dynamometer) only seems to be practical in a factory test environment. ISO 2314 also describes the method of using a heat balance over the gas turbine (Equation 5).

Other not performance related tests required in ISO 2314 will not be discussed here.

ASME Power Test Code 10 (1997)

ASME PTC 10 has recently been revised. In its concept, it is more suitable for factory tests than site tests. The code distinguishes two types of tests. Type 1 requires testing with the specified gas and at or near the specified operating conditions. Type 2 tests allow substitutes such as nitrogen, carbon dioxide, and others. The PTC 10 refers to ASME PTC 19 for test uncertainty analysis.

While the general accuracy and the amount of instrumentation as required in PTC 10 can be met in field performance tests, some added thoughts seem to be in order:

• *Data correction*—PTC 10 assumes that the test is conducted close to the specified conditions. This may not be possible at a site test.

• *Instrumentation*—In many cases, the situation at site does not allow meeting PTC 10 requirements. Often, the amount of instrumentation will be less than required by the code, or the instrument locations have to be adjusted to the site requirements. This does not preclude conducting a valid field performance test, but has to be considered in a test uncertainty analysis.

• *Real gas behavior*—ASME PTC 10 does not specify an accurate method for calculating real gas properties. The specified method calculates a polytropic head rise based on an approximation equation that does not provide the same accuracy as modern equations of state.

APPENDIX B
EXAMPLE OF AN ACTUAL FIELD PERFORMANCE TEST

Scope

A pipeline compressor driven by a 15,000 hp class gas turbine was field tested in January 1999. The intent of this test was to establish baseline data from the date of commissioning for future trending and to validate the performance maps.

As the gas compressor piping configuration did not include a compressor discharge gas cooler and the recycle was close to the suction header, running in recycle was avoided. Head and flow conditions for the test were met by manipulating valving, throttling discharge, and adding or removing horsepower at the station. This test was conducted jointly with personnel from the user and the equipment manufacturer.

The requirement was to cover as much of the compressor operating map as possible without disrupting pipeline requirements for supply. To achieve this, valve and horsepower modulation were used as described above to operate on three different speed lines plus maximum turbine power to take data: 8840 rpm max power, 8500 rpm, 7750 rpm, and 7000 rpm. In this paper, we will only discuss the results for one speed line at 7750 rpm. On each speed line, the compressor was operated as far into choke as conditions permitted, then throttled using the station discharge valve up to the surge control line.

Instrumentation

The instrumentation installed at site was standard field test equipment. This equipment is described as a compressor inlet eye pressure transducer, three suction pressure transducers, three discharge pressure transducers, four suction RTDs, and four discharge RTDs.

All package and process transmitters used for normal operation were calibrated before the test as part of the normal commissioning activities. All special test equipment was calibrated in the turbine manufacturer's calibration laboratory before the test. Gas analyses were collected by the customer every 15 minutes throughout the test.

Data Collection

As described above, all data were collected either via the Fluke NetDAQs or from the turbine control panel. Both data from the NetDAQs and the turbine control panel were imported into Microsoft EXCEL® through dynamic data exchange (DDE) drivers. The gas turbine and gas compressor performances were calculated online using "user defined functions" within EXCEL® written in Visual Basic®. These calculations use a gas composition provided by the customer.

Execution of the Test

During the data collection process, each point was plotted on a nondimensional curve at an average of 10 scans, at a rate of one scan per second. Three points were plotted at each of the operating conditions to verify that the operating point had reached equilibrium and was stable. Suction and discharge temperatures were also monitored to confirm steady-state operation and should not have fluctuated in excess of ± 0.5°F prior to plotting a test point.

Testing was started by setting the power turbine speed at the speed of 8500 rpm. The first test point was taken at this speed at high flow/low head conditions (near choke). The operating point was then walked up this speed line by pinching the suction valve. A minimum of five points was taken at this speed, including points near choke, design point, and the surge margin.

After points near surge control were collected at 8500 rpm, the power turbine speed was decreased to the next speed line of 7750 rpm and the same steps were repeated. This process is duplicated for all speed lines to be tested.

After the test was completed, all data collected during the test were copied to a computer disk and presented to the customer.

Data Evaluation

Compressor isentropic head and efficiency were calculated from the field test temperature and pressure measurements using the Redlich-Kwong equation of state. Gas flow measurements were calculated using the suction to the eye of the impeller. Shaft power was calculated by increasing the aerodynamic power by specified mechanical losses.

Test Results

Measured efficiency using pressures and temperature exceeded the current predictions from choke to about 20 percent to the left of the surge control line by about 2.5 percent.

The efficiency determined from the temperature and pressure measurements showed that the compressor exceeded the predicted performance by three percent. Peak efficiency was measured at 90.3 percent and can be viewed in Figure B-1.

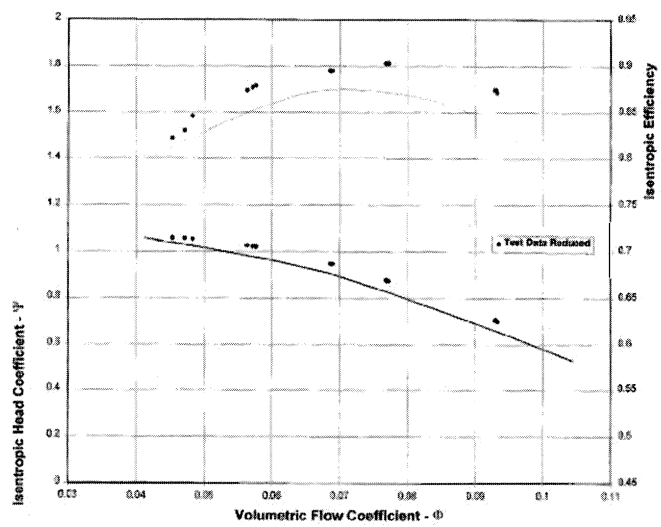

Figure B-1. Test Results from the Site Performance Test.

The test objective includes verification of gas turbine power output, gas turbine heat rate, and compressor efficiency at the site rated conditions of isentropic head and actual volumetric flow. The turbine power output and turbine heat rate were established to be per manufacturer's engine performance program. The prediction for the gas compressor performance is calculated using the manufacturer's program.

REFERENCES

AGA Report No. 3, 1991, "Orifice Metering of Natural Gas," American Gas Association, Washington, D.C.

ASME PTC 10, 1997, "Compressors and Exhausters," American Society of Mechanical Engineers, New York, New York.

ASME PTC 19.1, 1985, "Measurement Uncertainties," American Society of Mechanical Engineers, New York, New York.

ASME PTC 19.5, 1971, "Fluid Meters," American Society of Mechanical Engineers, New York, New York.

ASME PTC 22, 1997, "Gas Turbine Power Plants," American Society of Mechanical Engineers, New York, New York.

Beinecke D. and Luedtke, K., 1983, "Die Auslegung von Turboverdichtern unter Beruecksichtigung des realen Gasverhaltens," VDI Berichte Number 487, pp. 271-279.

Baehr, H. D., 1981, *Thermodynamik,* Berlin, Germany: Springer.

Brown, R. N., 1991, "Fan Laws, the Use and Limits in Predicting Centrifugal Compressor Off Design Performance," *Proceedings of the Twentieth Turbomachinery Symposium,* The Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 91-100.

Brun, K., 1996, "Analysis of the Automotive Torque Converter Internal Flow Field," Ph.D. Dissertation, University of Virginia.

Brun, K. and Kurz, R., 1998, "Measurement Uncertainties Encountered During Gas Turbine Driven Compressor Field Testing," ASME Paper No. 98-GT-1.

Cleveland, A., 1982, "Performance Testing of Gas Turbine Compressor Sets," ASME Paper No. 82-GT-199.

Cohen, H., Rogers, G. F. C., and Saravanamuttoo, H. I. H., 1996, *Gas Turbine Theory,* Harlow, United Kingdom: Addison Wesley Longman.

Colby, C. M., 1987, "Hydraulic Performances Testing of Centrifugal Compressors—Hydrocarbon Versus Inert Gas," Proc. IMechE, Paper C102/87.

Dejc, M. E. and Trojanovski, B. M., 1973, *Untersuchung und Berechnung axialer Turbinenstufen,* Berlin, Germany: VEB Verlag Technik.

Grimley, T. A. and Bowles, E. B., December 1998, "Industry Researchers Evaluate Ultrasonic Meter Performance," *Pipeline and Gas Industry.*

Horton, D., March 1990, "Temperature Sensors Have Little in Common Except That They Measure Temperature," Personal Engineering and Instrumentation News, Brookline, Massachusetts.

ISO 2314, 1989, "Gas Turbines—Acceptance Tests," International Organization for Standardization, Geneva, Switzerland.

ISO 5167, 1980, "Measurement of Fluid Flow by Means of Orifice Plates, Nozzles and Venturi Tubes Inserted in Circular Cross-Section Conduits Running Full," International Organization for Standardization, Geneva, Switzerland.

Kumar, S. K., Kurz, R., and O'Connell, J. P., 1999, "Equations of State for Compressor Design and Testing," ASME Paper No. 99-GT-12.

Kurz, R., 1991, "Transonic Flow through Turbine Cascades with 3 Different Pitch to Chord Ratios," Proc. 10th ISABE, Nottingham, United Kingdom.

McRoberts, I. W., 1984, "Performance Trials of Two Centrifugal Compressor Trains on an Offshore Platform," IMechE C48/84.

Meier, R. H. and Rhea, C. S., 1982, "Centrifugal Compressor Testing Experience and Practice," ASME Paper No. 82-GT-320.

Moffat, R. J., 1988, "Describing the Uncertainties in Experimental Results," Experimental Thermal and Fluid Science, 1:3-17.

Robinson, R. L. and Jacobi, R. H., 1965, "Better Compressibility Factors," *Hydrocarbon Processing,* (4).

Reid, R. C., Prausnitz, J. M., and Sherwood, T. K., 1977, *The Properties of Gases and Liquids,* New York, New York: McGraw Hill.

Schmitt, W. and Thomas, V., 1995, "Comparison of Test Measurements Taken on a Pipeline Compressor/Gas Turbine Unit in the Workshop and at Site," ASME Paper No. 95-GT-125.

Shaw, R. ,1960, "The Influence of Hole Dimensions on Static Pressure Measurements," J. Fluid Mech. 7.

Ting, V. C., December 1998, "Entrained Liquid's Effect on Orifice Meter Accuracy Studied," Pipeline and Gas Industry.

Valenti, M., 1997, "Determining Gas Meter Accuracy," Mechanical Engineering, *3,* (119), pp. 102-106.

VDI 2045, 1993, "Acceptance and Performance Tests on Turbo Compressors and Displacement Compressors," Verein Deutscher Ingenieure, Duesseldorf, Germany.

VDI 2048, 1978, "Inaccuracies in Acceptance Test Measurements," Verein Deutscher Ingenieure, Duesseldorf, Germany.