PERFORMANCE EVALUATION OF TURBOMACHINERY IN THE PETROLEUM EXPLORATION AND PRODUCTION AREA

Alisson Cardoso Gomes da Silva, alissoncardoso@petrobras.com.br
Petrobras Santos Basin Exploration & Production Operation Unit, Av. Conselheiro Nébias, n° 175, Santos, São Paulo, Brasil.

Silvio de Oliveira Júnior, soj@usp.br
State University of São Paulo, Mechanical Engineering Department, Av. Prof.: Mello Morais, n° 2231, Cidade Universitária, São Paulo, Brasil.

Abstract. *In light of the recent discoveries of new oil and natural gas reserves in the coast of São Paulo and Rio de Janeiro states, it will be required a large amount of investments for the development of hydrocarbons exploration and production. In this scenario, the turbomachinery play a crucial role because of its strong influence in the areas of power generation and ensuring the flow of current oil and natural gas produced in offshore platforms to the continent. Among the turbomachinery, it is possible to highlight gas turbines and centrifugal compressors, since they have high reliability, high thermodynamic efficiency and great operation and maintenance flexibility. However, gas turbines and centrifugal compressors show performance caractheristics that distinctly depend on ambient and operating conditions. They are not only influenced by site elevation, ambient temperature and relative humidity, but also by the speed of the driven equipment, the fuel and the load conditions. Because of that, performance testing of gas turbines and compressors has become increasingly common due to the need to verify efficiency, power, fuel flow, capacity and head of the packages uppon delivery. This paper suggests the use of seven performance parameters that generally describe the performance of a gas turbine and centrifugal compressor. Special consideration will be given to show a comprehensive view of the physical models and mathematical formulations required for evaluating this seven performance parameters based on the basic relationships of pressure, temperature, flow and head. Since the test conditions are rarely fully controlled, it will be discussed an approach to perform measurement uncertainties analysis, with the interest of assuring results validity. Finally, the paper present some considerations for conduct a proper performance evaluation test.*

Keywords: Turbomachinery, Performance, Factory Acceptance Test, Field Testing

1. INTRODUCTION

Gas turbines have been used for many aerospace and industrial applications for many years. They are used successfully to power aircraft as well as in industrial applications. The primary focus of this paper is on gas turbines for industrial applications, which consist either of an air compressor driven by a gas generator turbine with a separate power turbine or of an air compressor and a turbine on one shaft, where the turbine provides both power for the air compressor and the load. The power and efficiency characteristics of a gas turbine are therefore the result of a complex interaction of different turbomachines and combustion system.

Centrifugal gas compressors are used in many applications in the oil and gas industry, such as in pipelines, for gas gathering, gas reinjection, gas lift, gas storage, in onshore as well as offshore environments. The predominant driver for these compressors is two shaft gas turbines, and, to some extent, variable speed electric motors. Both types of drivers have in common that the speed of the compressor can be varied easily over a large range. To some extent, variable speed electric motors are used in similar tasks.

Gas turbines and centrifugal compressors are high technological complexity machines, where it is common to identify state of art applications in disciplines like structural mechanics, thermodynamics, automation and control. Thus, making an assessment of performance and reliability of this equipment is not a trivial task. Over the past years, the international technical community has concentrated its efforts to spread the use of standards and codes produced by ASME (American Society of Mechanical Engineers), API (American Petroleum Institute) and ISO (International Organization for Standardization) as a reference in planning, implementation and evaluation of performance tests turbomachinery. The main objective being to allow that results from different tests can be compared on equal basis. However, there are some differences in implementing the recommendations found in the standards, mainly because of conflict of interest between machine manufacturers and end users responsible for equipment operation.

Therefore, the factory acceptance tests of turbomachinery are critical to ensuring that all contract criteria will be respected. They also enable that machines intrinsic problems are detected even in the test bay of manufacturers, so any necessary interventions should be realized in the manufacturer facility, reducing delays in delivery and do not represent additional costs for final customers, ensuring operational quality equipment and extending its life cycle. Problems found after the equipment final installation are much more expensive and complicated to be solved, causing start operation delays of the projects, or even interrupting the operation, thereby causing significant losses due to production interruption.

As a conclusion, there is great need to systematize the use of technical literature available through careful review, seeking a better understanding of international standards and their relevance for implementing the Brazilian context. Therefore, this implies in facilitate the implementation of all recommendations and best engineering practices already adopted internationally. However, the definition of best practices should be built through an extensive discussion promoted by the national technical and scientific community.

Finally, this paper has the main objective to provide a comprehensive view of the physical models and mathematical formulations required for evaluating the performance of gas turbines and centrifugal compressors. Another task is to present special considerations for conduct the evaluation of performance of gas turbines and centrifugal compressors.

2. PERFORMANCE PARAMETERS

The following seven performance parameters generally describe the performance of a gas turbine and centrifugal compressor. These parameters are commonly used in acceptance testing, testing to determine degradation of the machine, and operational range testing. The performance parameters are listed below: 1) Centrifugal Compressor Flow vs. Flow Coefficient, 2) Centrifugal Compressor Head vs. Head Coefficient, 3) Centrifugal Compressor Efficiency, 4) Centrifugal Compressor Power Absorbed, 5) Gas Turbine Full Load Output Power, 6) Gas Turbine Heat Rate (thermal efficiency) and 7) Gas Turbine Exhaust Heat Rate. Such parameters are determined by mathematical formulae that correlate the measurement results of various physical quantities (e.g. pressure, temperature, power, speed, etc.). Since the test conditions are rarely fully controlled, it is necessary to perform analysis of measurement uncertainties, with the interest of assuring results validity (Brun and Nored, 2006). Figure 1 shows the general measurement arrangement for the required test instrumentation on the gas turbine.

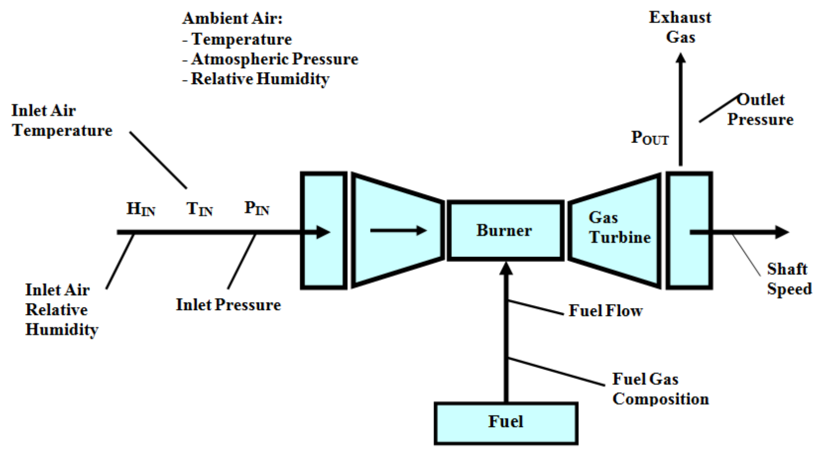

Figure 1. Location of Test Instrumentation for Gas Turbine

Figure 2 presents test data and general measurement arrangement for the required test instrumentation on the centrifugal compressor.

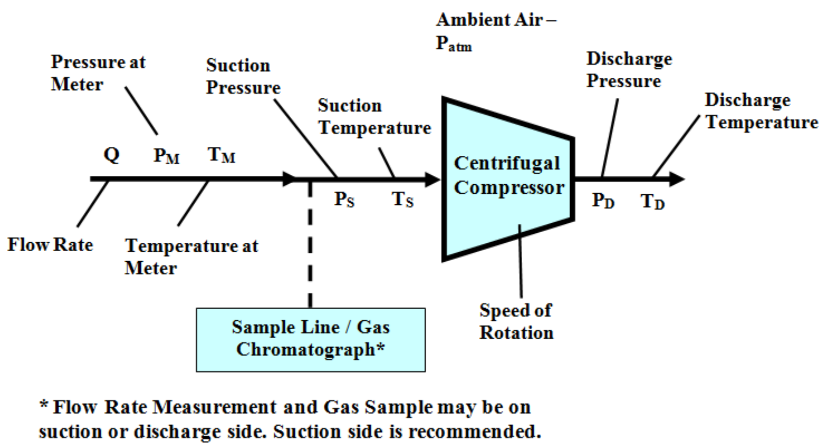

Figure 2. Location of Test Instrumentation for Centrifugal Compressor

2.1. Centrifugal Compressor Flow vs. Flow Coefficient

The actual flow through the centrifugal compressor (Q_m) should be measured by a flow-measuring device, such as a volumetric flow meter (ultrasonic, turbine, etc.), a differential pressure device (orifice meter, Venturi, nozzle, etc.). If an orifice meter is used, which is typical of many installations, we can calculate the mass flow rate from the following equation (AGA n° 3, 2003).

$$Q_m = \frac{C}{\sqrt{1-\beta^4}}\varepsilon\frac{\pi}{4}d^2\sqrt{2\Delta p\rho} \quad (1)$$

Where: C – discharge coefficient, ε – expansion factor, β = d/D: d – orifice plate bore diameter / D – meter tube internal diameter, Δp – orifice differential pressure and ρ – fluid density. The discharge coefficient is determined from the equation developed by Reader–Harris/Gallagher (AGA n° 3, 2003), as showed below:

$$\begin{aligned} C = {} & 0.5961 + 0.0291\beta^2 - 0.2290\beta^8 + 0.003(1-\beta).M_1 + [0.0433 + 0.0712e^{-8,5L_1} - 0.1145e^{-6L_1}] \\ & \cdot\left[1 - 0.23\left(\frac{1900\beta}{Re_D}\right)^{0,8}\right]\frac{\beta^4}{1-\beta^4} - 0.0116\left[\frac{2L_2{}'}{1-\beta} - 0.52\left(\frac{2L_2{}'}{1-\beta}\right)^{1,3}\right]\beta^{1,1} \\ & \cdot\left[1 - 0.14\left(\frac{19000\beta}{Re_D}\right)^{0,8}\right] + 0.000511\left(\frac{10^6\beta}{Re_D}\right)^{0,7} \\ & + \left[0.021 + 0.0049\left(\frac{19000\beta}{Re_D}\right)^{0,8}\right]\beta^4\left(\frac{10^6}{Re_D}\right)^{0,35} \end{aligned} \quad (2)$$

Where:

$$M_1 = max\left(2.8 - \frac{D}{N_4}, 0.0\right) \quad (3)$$
$L_1 = L_2 = dimensionless\ correction\ for\ the\ tap\ location$
$N_4 = 1.0\ when\ D\ is\ in\ inches\ or\ 25.4\ when\ D\ is\ in\ milimeters$
$Re_D = pipe\ Reynolds\ number$

The actual suction volumetric flow is given by:

$$Q_v = \frac{Q_m}{\rho} \quad (4)$$

The flow coefficient proposed to be used for similarity comparison is given by Eq. (5).

$$\varphi = \frac{Q_v}{\frac{\pi}{4}D_{tip}{}^2U_{tip}} = \frac{Q_v}{\frac{\pi^2}{4}D_{tip}{}^3\omega} \quad (5)$$

Where: D_{tip} – tip diameter of compressor, U_{tip} – impeller tip speed (tangencial) and ω – impeller angular speed.

In case of multi-stage compressor, the tip diameter may be defined as the first impeller diameter or a geometric average for all stage diameters. When comparing flow coefficients for multi-stage machines, the definition of tip diameter should be verified.

2.2. Centrifugal Compressor Head vs. Head Coefficient

Compressor head and efficiency are commonly defined based on either isentropic or polytropic ideal processes. Both definitions are appropriate for performance comparison as they provide a ratio of the actual enthalpy difference (head) to the ideal (isentropic or polytropic) enthalpy difference across the compressor.

The isentropic process assumes a reversible adiabatic process without losses (i.e., no change in entropy). The polytropic process is also a reversible process, but it is not adiabatic. It is defined by an infinite number of small isentropic steps followed by heat exchange. Both processes are ideal, reference processes.

The compressor actual head (H), isentropic head (H*) and polytropic head (H^P) are determined from the measurement of pressure and temperature on the suction and discharge sides and the calculation of enthalpy and specific volume using an equation of state (EOS) model. This subject will be presented later.

The heads are calculated from the enthalpies associated with each state from the EOS. Equation (6) presents the isentropic head calculation.

$$H^* = h_d^* - h_s = h(p_d, s_s) - h(p_s, T_s) \tag{6}$$

Equation (7) presents the actual head calculation.

$$H = h_d - h_s = h(p_d, T_d) - h(p_s, T_s) \tag{7}$$

The value of h_d^* is the enthalpy associated with the discharge pressure at the suction entropy (s_s), because the entropy change is zero in an isentropic process. Isentropic enthalpy can also be calculated for estimation purposes (assuming ideal gas behavior) by Eq. (8).

$$h_d^* \approx c_p^* \cdot T_d^* = c_{ps} \cdot T_s \cdot \left(\frac{p_d}{p_s}\right)^{\frac{k-1}{k}} \tag{8}$$

Similarly, polytropic head is determined from Eq. (9).

$$H^P = \left[\frac{n^P}{n^P - 1}\right] \cdot \left[\left(\frac{p_d}{p_s}\right)^{\frac{n^P-1}{n^P}} - 1\right] \cdot f \cdot p_s v_s \tag{9}$$

The polytropic exponent (n^P) is defined using Eq. (10).

$$n^P = \frac{\ln \frac{p_d}{p_s}}{\ln \frac{v_s}{v_d}} \tag{10}$$

The isentropic exponent (k) is defined using Eq. (11).

$$k = \frac{\ln \frac{p_d}{p_s}}{\ln \frac{v_s}{v_d^*}} \tag{11}$$

Considering Eq. (9), the Schultz Polytropic Head Correction Factor (f) is defined using Eq. (12).

$$f = \frac{h_d^* - h_s}{\left(\frac{k}{k-1}\right) \cdot (p_d v_d^* - p_s v_s)} \tag{12}$$

For performance comparisons, it is beneficial to use non-dimensional head coefficient (ψ) and flow coefficient (φ), rather than actual head and flow. Equation (13) shows the isentropic head coefficient calculation.

$$\psi^* = \frac{H^*}{\frac{U^2}{2}} = \frac{2H^*}{\left(\pi D_{tip} \omega\right)^2} \tag{13}$$

Equation (14) presents the actual head coefficient calculation.

$$\psi = \frac{H}{\frac{U^2}{2}} = \frac{2H}{\left(\pi D_{tip} \omega\right)^2} \tag{14}$$

Equation (15) presents the polytropic head coefficient calculation.

$$\psi^P = \frac{H^P}{\frac{U^2}{2}} = \frac{2H^P}{\left(\pi D_{tip} N\right)^2} \tag{15}$$

Where: N – compressor shaft speed.

2.3. Centrifugal Compressor Efficiency

The isentropic efficiency is calculated from the isentropic and actual head from Eq. (16).

$$\eta^* = \frac{H^*}{H} = \frac{\psi^*}{\psi} \tag{16}$$

The polytropic efficiency is calculated based upon the polytropic head and the polytropic exponent, as shown in Eq. (17).

$$\eta^P = \frac{H^P}{H} = \frac{\left(\frac{n^P}{n^P - 1}\right) \cdot \left[\left(\frac{p_d}{p_s}\right)^{\frac{n^P - 1}{n^P}} - 1\right] \cdot f \cdot p_s v_s}{h_d - h_s} \tag{17}$$

2.4. Centrifugal Compressor Absorbed Power

The absorbed power for the compressor (P_{comp}) can be directly used to determine the gas turbine shaft output power (P_{out}), considering a mechanical efficiency of the compressor (η_m), typically given by the manufacturer as 98% to 99%, if no gearbox is present. Otherwise, the gearbox power losses must be included to determine gas turbine shaft output power. Compressor absorbed power is calculated using the compressor suction gas conditions and the actual head (enthalpy change) as defined in Eq. (18).

$$P_{comp} = P_{out} \cdot \eta_m = \rho_s \cdot Q_v \cdot H \tag{18}$$

If the driven compressor is rated for less power than the gas turbine output power, the full load power of the gas turbine cannot be determined using this approach.

2.5. Gas Turbine Full Load Output Power

According with Kurz et al. (1999), while testing a gas compressor in the field can yield test uncertainties similar to those found in the factory test, if the field test is conducted using the same standards as for the factory test, the field testing of gas turbine 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 generator, dynamometer or other calibrated device. The power turbine applies torque directly to the generator, dynamometer or other calibrated device. The absorbed power is accurately measured with a load cell or another method.

In a field test, shaft power is determined in one or more of the following ways, as shown in Tab. 1.

Table 1. Shaft power measurements: direct and indirect ways

Direct Measurement	Indirect Measurement
Using a torque measuring coupling between power turbine and driven equipment (high accuracy, small uncertainty)	Using the calculated power of the driven compressor (high uncertainty due to uncertainty in aerodynamic power of compressor)
Using the measured output of the generator (high accuracy, small uncertainty)	Verifying with a redundant measurement, such as a heat balance (high uncertainty due to measurement of air flow)

If the torque (τ) is measured using a torque coupling, then the shaft power (P_{out}) developed by the gas turbine is calculated in accordance with Eq. (19).

$$P_{out} = \tau \cdot (2\pi N) \tag{19}$$

For electric generator gas turbine applications, the generator output is measured in order to determine the gas turbine shaft power. If a gearbox is present between gas turbine and generator, the power losses also need to be considered. The generator electrical power output can be measured directly at the generator terminals. Typically, three current transformers (CTs) and two potential transformers (PTs) are used to measure the line voltages (E) and currents (I). The power factor (PF) can be determined from the phase angle between the voltage and the current as shown in Eq. (20). The generator output can be calculated using Eq. (21). For increased accuracy, specially calibrated current transformers and potential transformers can be used.

$$PF = \frac{P_{el,active}}{P_{el,apparent}} \quad (20)$$

$$P_{el} = \sqrt{3} \cdot E \cdot I \cdot PF \quad (21)$$

Calculation of gas turbine shaft power using the conservation of energy in a thermodynamic system, requires to know the energy flowing into the system be balanced by the energy leaving the system, as given by Eq. (22).

$$q_{m,1}h_1 + q_{m,f} \times LHV \times \eta_{comb} + q_{m,f}h_f = \left(q_{m,1} + q_{m,f}\right) \times h_7 + P_{out} + E_r + E_m \quad (22)$$

The mass flow and enthalpy of the air at the gas turbine inlet ($q_{m,1}h_1$), as well as the fuel flow and fuel enthalpy ($q_{m,f}$ h_f), lower heating value (LHV) of the fuel, and the 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 small. The combustion efficiency (η_{comb}) can be estimated as well and is typically about 99% or better. Therefore, the shaft power of the turbine (P_{out}) can be calculated from the above equation.

For this field method, it is essential to measure the airflow through the gas turbine, which usually is not possible in the field with sufficient accuracy for precise test results. 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 factory testing. Thus, it is possible to substitute parameters that cannot be measured in the field with information gathered during a factory test.

2.6. Gas Turbine Heat Rate and Efficiency

If the gas turbine shaft output power is known (or determined from the driven equipment or heat balance methods), then the gas turbine efficiency is determined by dividing the gas turbine shaft output power by the fuel energy flow rate, as given by Eq. (23).

$$\eta_{GT} = \frac{P_{out}}{Q_{m,f} \cdot (LHV)} \quad (23)$$

Similarly, the gas turbine heat rate is simply the reciprocal of the efficiency. Equation (24) presents the heat rate calculation.

$$HR = \frac{Q_{m,f} \cdot (LHV)}{P_{out}} \quad (24)$$

As heat rate is often expressed in mixed units, the appropriate unit conversion may need to be applied. The actual fuel gas composition should be used to determine the lower heating value (LHV) of the fuel. If the fuel gas temperature is greater than 20 °C above the ambient temperature, the fuel gas sensible heat should be added to the equations above as given in Eq. (25) and Eq. (26). Sensible heat represents the energy introduced into the combustor in the form of thermal heat contained in the fuel.

$$\eta_{GT} = \frac{P_{out}}{Q_{m,f} \cdot \left[LHV + \left(\rho \cdot c_p \cdot T\right)_{FG}\right]} \quad (25)$$

$$HR = \frac{Q_{m,f} \cdot \left[LHV + \left(\rho \cdot c_p \cdot T\right)_{FG}\right]}{P_{out}} \quad (26)$$

2.7. Gas Turbine Exhaust Heat Rate

The gas turbine's exhaust heat rate is often important for combined cycle or cogeneration applications. Exhaust heat rate is the remaining energy in the exhaust flow of the gas turbine. Equation (27) shows the exhaust heat rate calculation.

$$EHR = Q_{m,GT} \cdot (h_E - h_R) \quad (27)$$

In Eq. (27) h_R is a mutually agreed reference enthalpy. Direct measurement of the mass flow is not recommended to determine the gas turbine exhaust heat rate because of the difficulty of accurately performing this measurement without a large pressure differential. In addition, test uncertainties will be high due to the flow measurement and temperature measurement uncertainties. An energy balance of the system may be used to estimate the gas turbine exhaust heat rate, as described in Eq. (22).

2.8. Equations of State

The state of any fluid of given 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 (Kurz et al, 1999). The simplest equation of state is the equation for a perfect gas, as given in Eq. (28).

$$pv = \frac{p}{\rho} = RT \qquad (28)$$

Real gases and in particular gas mixtures, however, display complex relationship between pressure, volume and temperature (p-v-T). Equations of state use semi-empirical equations to describe these relationships, in particular the deviations from perfect gas behavior, as showed in Eq. (29).

$$\frac{p}{\rho} = Z(p,t) \cdot R \cdot T \qquad (29)$$

They also allow for the calculation of properties that are derived from the p-v-T relationships, such as enthalpy (h) and entropy (s). Because equations of state are semi-empirical, they might be optimized for certain facets of the gas behavior, such as liquid-vapor equilibriums and not necessarily for the typical range of temperatures and pressures in pipeline applications. According with Brun and Nored (2006), in the field performance test of the compressor and turbine, the correct determination of the thermodynamic properties of the gas (such as enthalpy, entropy, and density) plays a critical role. The measured quantities (such as pressure, temperature, and composition) are used as inputs to an equation of state (EOS) to determine thermodynamic properties. The enthalpy change is used to determine the head and the isentropic or polytropic efficiency of a compressor. The choice of the EOS used in calculating enthalpy and density affects the accuracy of the results and needs to be considered in the uncertainty calculation.

The possible equations of state commonly used in the gas industry are: Redlich-Kwong (RK), Soave-Redlich-Kwong (SRK), Peng-Robinson (PR), Benedict-Webb-Rubin (BWR), Benedict-Webb-Rubin-Starling (BWRS), and Lee-Kesler-Plocker (LKP), and AGA-10. The final selection of the equation of state to be used in the field test should depend on the applicability of the particular equation of state model to the gas and temperatures encountered along with the process of interest (Sandberg, 2005; Kumar et al., 1999).

Generally, it is not possible to select a most accurate EOS to predict gas properties, since there are no calibration standards to test against for typical hydrocarbon mixtures. All the frequently used EOS models (RK, BWR, BWRS, LKP, SRK, PR) can predict the properties of hydrocarbon mixtures accurately below 20 MPa for common natural gas mixtures. Outside this pressure range, deviations between the EOS models of 0.5 to 2.5% in compressibility factor Z are common, especially if the natural gas contains significant amounts of diluents. Because derivatives of the compressibility factor (Z) must be used to calculate the enthalpy differences (i.e., head), the head deviations can be larger than the compressibility factor for different EOS. Table 2 provides usage suggestions for the various EOS models based on application (Brun and Nored, 2006).

Table 2. Suggested applications for Equation of State usage

Type of application	Suggested used EOS Model
Typical hydrocarbon gas mixture, standard pressures and temperatures, low CO_2 and N_2 diluents (< 6% total). Air mixtures.	All EOS Models may be used for this application: Redlich-Kwong (RK), Soave-Redlich-Kwong (SRK), Peng Robinson (PR), Benedict-Webb-Rubin-Starling (BWRS), Benedict-Webb-Rubin (BWR), Lee-Kesler-Plocker (LKP), AGA-10
High-pressure applications (> 3000 psi)	BWRS, BWR, LKP
High CO_2 and N_2 diluents (10-30%) and/or high hydrogen content gases	BWRS, LKP
High hydrogen content gases (> 80% H_2)	PR, LKP, SRK
Non-hydrocarbon mixtures: ethylenes, glycols, carbon dioxide mixtures, refrigerants, hydrocarbon vapors, etc.	Specific EOS model designed for particular application or chemical mixture will result in greater accuracy. The literature should be consulted for the particular gas and application

3. TEST UNCERTAINTIES CONSIDERATIONS

Traditionally the uncertainty of the result of a measurement is viewed as having two error components, the random error, which presumably arises from unpredictable, stochastic temporal and spatial variations of the measurand, and the systematic error or bias, considered as constant in magnitude and direction for repeated observations.

This approach of categorizing errors can be ambiguous because, depending on how an error quantity appears in a mathematical model that describes the measurement process, a random component may become a systematic component and vice versa. According with ISO/IEC Guide 98-3 (2008), to avoid such ambiguity, the concept presented is to categorize the uncertainty components by the methods of their evaluation rather than the components themselves.

3.1. Type A evaluation of uncertainty

For the uncertainty calculation of a test result, the Type A uncertainty component is calculated based on a series of *n* sample readings of the random variable quantity (q) with the experimental standard deviation of the mean $s(\bar{q})$, as given in Eq. (30) and Eq. (31).

$$s(\bar{q}) = \frac{s(q_k)}{\sqrt{n}} \tag{30}$$

$$s(q_k)^2 = \frac{1}{n-1}\sum_{k=1}^{n}[q_k - \bar{q}]^2 \tag{31}$$

Where q_k is the individual observation of a variable input quantity, and $\bar{q}$ is the arithmetic mean or average of *n* observations.

In this case the random variable q with $\upsilon = n - 1$ degrees of freedom is considered normally distributed according to the probability density function of Laplace-Gauss, also termed t-distribution or Student's distribution. Then the standard uncertainty $u(x_i)$ of the estimate $x_i = \bar{X}_i$ of the input quantity X_i is $u(x_i) = s(\bar{X}_i)$ with $s(\bar{X}_i)^2$ calculated according to Eq. (31).

3.2. Type B evaluation of uncertainty

For an estimate x_i of an input quantity X_i that has not been obtained from repeated observations, the associated estimated variance $u^2(x_i)$ or the standard uncertainty $u(x_i)$ is evaluated by scientific judgement based on all of the available information on the possible variability of X_i. The Type B component of the uncertainty calculation is based on the data provided in calibration or other certificates, such as manufacturer specifications, of the test instruments and devices. The proper use of the pool of available information for a Type B evaluation of standard uncertainty calls for insight based on experience and general knowledge, and is a skill that can be learned with practice. It should be recognized that a Type B evaluation of standard uncertainty can be as reliable as a Type A evaluation, especially in a measurement situation where a Type A evaluation is based on a comparatively small number of statistically independent observations.

3.3. Combined standard uncertainty

The combined standard uncertainty $u_c(y)$ of a measurement result is an estimated standard deviation and characterizes the dispersion of the values that could reasonably be attributed to the measurand *Y*. For uncorrelated input quantities it is the positive square root of the combined variance $u_c^2(y)$ obtained by the *Law of Propagation of Uncertainty*, in common parlance the RSS (root-sum-of-squares) method, calculated as given by Eq. (32).

$$u_c^2(y) = \sum_{i=1}^{n}\left[\frac{\partial f}{\partial x_i}\right]^2 u^2(x_i) \tag{32}$$

Equation (33) presents the combined standard uncertainty calculation for correlated input quantities.

$$u_c^2(y) = \sum_{i=1}^{n}\sum_{j=1}^{n}\frac{\partial f}{\partial x_i}\frac{\partial f}{\partial x_j}u(x_i, x_j) = \sum_{i=1}^{n}\left[\frac{\partial f}{\partial x_i}\right]^2 u^2(x_i) + 2\sum_{i=1}^{n-1}\sum_{j=i+1}^{n}\frac{\partial f}{\partial x_i}\frac{\partial f}{\partial x_j}r(x_i, x_j)u(x_i)u(x_j) \tag{33}$$

The component $r(x_i, x_j)$ is the correlation coefficient which characterizes the degree of correlation between x_i and x_j with $-1 \leq r(x_i, x_j) \leq +1$. For independent estimates x_i and x_j, $r(x_i, x_j) = 0$.

The combined standard uncertainty allows combining the individual standard uncertainties $u(x_i)$, whether arising from the Type A evaluation of uncertainty or Type B evaluation of uncertainty.

The partial derivates $\partial f / \partial x_i$ of the function $y = f(x_i)$, often referred to as sensitivity coefficients, are either mathematically determined by derivating y for each input estimate x_i or numerically by calculating the change $(\Delta y)_i$ of the output estimate y with varying the input estimates at the expected x_i, considering Eq. (34).

$$\frac{\partial f}{\partial x_i} = \frac{(\Delta y)_i}{\Delta x_i} \tag{34}$$

3.4. Determining expanded uncertainty

The expanded uncertainty U, which is a measure that defines the interval about the measurement result encompassing a large fraction of the measured values with a defined level of confidence, is obtained by multiplying the standard uncertainty $u(x_i)$ with a coverage factor k. Equation (35) shows the expanded uncertainty calculation. In practice, a coverage factor k = 2 is frequently applied for a level of confidence of 95% and for degrees of freedom as given by $\nu \geq 30$.

$$U = k\, u(x_i) \tag{35}$$

Test uncertainties need to be clearly distinguished from machine 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 machine building tolerances. Building tolerances are entirely the responsibility of the manufacturer and must be excluded in any uncertainty calculation. In addition, the test uncertainty is not equivalent to the contractual test tolerance. The contractually agreed upon test tolerance might be influenced by consideration of how accurate a test can be performed or by more commercial considerations, such as the amount of risk the parties are willing to accept. This optional section must be placed before the list of references.

4. FIELD PERFORMANCE EVALUATION

Performance testing requires complex and detailed preparations. Since the purpose of such tests may vary, it is important to establish upfront the test objectives, identify the participating parties and their role in the process. A clear determination of the equipment boundaries and associated instrumentation shall avoid any potential disagreements after the test. A detailed test procedure specific to the test site/supplier's test facility and conditions shall be agreed by all parties involved. Because of that, the standards and codes produced by ASME, API and ISO have been increased as a reference in planning, implementation and evaluation of performance tests turbomachinery. Compliance with such specifications as listed above, it is a relatively easy matter in a factory environment where the facilities are designed specifically for testing, there are available qualified support personnel, instrumentation and calibration laboratories. At least, real time on-line computers routinely monitor the test progress. This usually is not the case at actual installation sites designed for commercial operation of turbomachinery.

During field performance tests an accurate determination of the performance of the package or its components are often difficult because of working environments that are not optimized for testing. 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. Field test also provide the operator and the equipment manufacturers with information complementary to the data collected during factory testing. Thus, for the end user and manufacturer, an accurate determination of the package field performance is critical.

Site performance tests generally require concerted planning and execution, including development of a unique test agenda prepared jointly by manufacturer and the equipment end user. Ideally, such an agenda should include field conditions and equipment layout, list of instruments to be used and their location. It is also necessary to 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. The agenda also should describe the methods of data reduction and determining the test uncertainties, as well as the acceptance criteria.

The selection and calibration of the test instrumentation is extremely important. Generally, the instruments supplied for protection and monitoring the packages are not accurate enough to achieve the small uncertainty margins necessary for a field test. This is mainly due to more stringent calibration requirements. 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 unit performance.

The requirement for special instrumentation is especially important for field test of compressors sets with low pressures ratios.

In order to obtain steady state conditions, the gas turbine and compressor should be started prior to the initiation of the test (compressors require at least 30 minutes of heat soak time, gas turbines require between 1 to 2 hours of heat soak time). The field test should be performed when the gas turbine and compressor operating conditions have reached steady state and the operating conditions should stay constant during each test point.

Power fluctuations should not occur during the performance testing. As it is very difficult to determine fuel gas composition variations during the short test intervals, it is important to ensure that the fuel and process gas compositions will remain unchanged for the duration of the testing period for each test point. Multiple gas samples of the process gas and fuel gas must be taken for each test point if the gas composition significantly changes.

Temperature measurements will especially be affected by any instability during the test. Temperature probes reach equilibrium through relatively slow heat transfer and heat soaking, while the system operating conditions vary at much faster rates. The heat storing capacity of the compressor and system piping will need adequate time to reach equilibrium after any operating conditions have changed. It is, thus, critical to maintain extended stable operating conditions prior to beginning the test in order to reach thermal equilibrium and measure accurate gas temperatures.

Regardless of the assumption of steady state test operation, any variation in measured parameters during the test interval should be accounted for in the uncertainty calculation.

5. CONCLUSIONS

Based on the information provided in this paper, it was presented a comprehensive view of the physical models and mathematical formulations to determine the suggested seven performance parameters that generally describe the performance of a gas turbine and centrifugal compressor, using basic relationships of pressure, temperature, flow and head. Since the test conditions are rarely fully controlled, it was discussed an approach to perform measurement uncertainties analysis, with the interest of assuring results validity.

Once there is an enormous difference between factory environment and site environment, a successful field test needs an elaborated agenda or plan that should be prepared prior to the test. This preparation should include field conditions, equipment layout and instruments to be used and their location. It is also necessary to describe method of operation, test safety considerations and the facility's limits (pressure, temperature and flow). As a conclusion, an appropriate field test should include a discussion of the method of data reduction, the selected approach for determining the test uncertainty, the acceptance criteria (specified in terms of maximum uncertainty allowable), the equation of state to be used for all calculations in the test and the use of isentropic or polytropic calculations.

6. ACKNOWLEDGEMENTS

The authors thank PETROBRAS for supporting the development of this work.

7. REFERENCES

AGA Report n° 3, Part 1, 2003, "Orifice Metering of Natural Gas and Other Related Hydrocarbon Fluids – General Equations and Uncertainty Guidelines", American Gas Association, Washington, USA.

Brenne, L., Bjorge, T., Gilarranz, J. L., Koch, J.M., and Miller, H., 2005, "Performance Evaluation of a Centrifugal Compressor Operating Under Wet Gas Conditions", Proceedings of 34th Turbomachinery Symposium, Houston, Texas, USA.

Brun, K. and Nored, M.G., 2006, "Guideline for Field Testing of Gas Turbine and Centrifugal Compressor Performance", Release 2.0, Gas Machinery Research Council, SouthWest Research Institute, San Antonio, USA.

ISO/IEC Guide 98-3, 2008, "Uncertainty of measurement – Part 3: Guide to the expression of uncertainty in measurement (GUM 1995)", 1st edition, Geneva, Switzerland.

ISO 2314, 2009, "Gas Turbines – Acceptance tests", 3th edition, Geneva, Switzerland.

Kurz, R., Brun, K., Legrand, D. D., 1999, "Field Performance Testing of Gas Turbine Driven Compressor Sets", Proceedings of 28th Turbomachinery Symposium, Texas A&M University, College Station, Texas, USA.

Kurz, R. and Brun, K., 2005, "Site Performance Test Evaluation for Gas Turbine and Electric Motor Driven Compressors", Proceedings of 34th Turbomachinery Symposium, Houston, Texas, USA.

Sandberg, M.R., 2005, "Equation of State Influences on Compressor Performance Determination", Proceedings of 34th Turbomachinery Symposium, Houston, Texas, USA.

8. RESPONSIBILITY NOTICE