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GUIDELINE FOR FIELD TESTING OF GAS TURBINE AND CENTRIFUGAL COMPRESSOR PERFORMANCE

RELEASE 2.0

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Guideline for Field Testing of Gas Turbine and Centrifugal Compressor Performance

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Foreword

Field 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 gas turbine package upon delivery. The performance test of the gas turbine and compressor in the field is often necessary to assure that the manufacturer meets performance predictions and guarantee a customer’s return on investment. Economic considerations demand that the performance and efficiency of a gas turbine compressor package be verified at the actual field site. The field environment is not ideal and measurement uncertainties are necessary to characterize the validity of a performance test. As the working field environments shift further from the ideal case, the uncertainties increase. Previous field tests have shown that the compressor efficiency uncertainty can be unacceptably high when some basic rules for proper test procedures and standards are violated.

This guideline applies to a typical gas turbine and centrifugal compressor. The motivation for conducting a field test is based on one of the following objectives:

- The manufacturer is required to verify performance of the gas turbine and compressor to the customer. To the manufacturer, the field test provides a baseline for the gas turbine and compressor at the site of delivery to compare to the factory performance test, although the field test accuracy may be inherently lower. In addition, the field performance test is the final validation from the manufacturer to the customer of the guaranteed performance.

- The user needs to verify performance of the gas turbine and compressor. Baseline performance data is obtained from the initial field performance test. The baseline test can be used for comparing and monitoring the health of the gas turbine-driven compressor package in the future.

- The user or manufacturer needs to assess performance of the gas turbine or compressor because of degradation concerns. Based on the field test results, a performance recovery program may be initiated.

- The user requires calibration of an installed historical trend monitoring system. The field test is used to provide initial calibration of the system based on the first performance of the gas turbine and compressor.

- The user needs to determine the operating range of the installed equipment after an upgrade, restage, or physical system change. In this case, the surge point may also need to be re-assessed.

The following guideline is a suggested best practice for field testing of gas turbines and centrifugal compressors. Specific considerations at a field site may require deviation from this guideline in order to meet safety requirements, improve efficiency, or comply with station operating philosophy.
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Guideline for Field Testing of Gas Turbine and Centrifugal Compressor Performance

RELEASE 2.0

TABLE OF CONTENTS

1. Purpose and Application ......................................................................................................1
2. Performance Parameters .....................................................................................................1
   2.1 Centrifugal Compressor Flow/Flow Coefficient ...........................................................3
   2.2 Centrifugal Compressor Head/Head Coefficient .........................................................4
   2.3 Centrifugal Compressor Efficiency ..............................................................................6
   2.4 Gas Turbine Power .....................................................................................................7
   2.5 Centrifugal Compressor Absorbed Power (Gas Turbine Power Output) .....................7
   2.6 Gas Turbine Heat Rate and Efficiency ........................................................................7
   2.7 Gas Turbine Exhaust Heat Rate .................................................................................8
   2.8 Turbocompressor Package Efficiency .......................................................................8
   2.9 Equations of State .......................................................................................................9
   2.10 Determination of Surge Point and Turndown ............................................................10
   2.11 Similarity Conditions ..............................................................................................12
3. Test Preparation.................................................................................................................15
   3.1 Pre-Test Meeting.......................................................................................................15
   3.2 Pre-Test Operation and Instrumentation Checkout...................................................15
   3.3 Pre-Test Equipment Checkout ..................................................................................16
   3.4 Pre-Test Information..................................................................................................16
   3.5 Test Stability..............................................................................................................17
   3.6 Safety Considerations ..............................................................................................19
4. Measurement and Instrumentation ....................................................................................19
   4.1 Measurement of Pressure .........................................................................................20
   4.2 Measurement of Temperature ...................................................................................21
   4.3 Measurement of Flow ................................................................................................24
   4.4 Measurement of Gas Composition ............................................................................28
   4.5 Measurement of Rotational Speed ............................................................................30
   4.6 Measurement of Torque ............................................................................................30
   4.7 Measurement of Generator Power ............................................................................30
5. Test Uncertainty ................................................................................................................30
   5.1 Ideal Field Test Conditions For Reducing Uncertainties ...........................................32
   5.2 Effects of Non-Ideal Installations on Uncertainty .......................................................36
6. Interpretation of Test Data ................................................................................................38
   6.1 Data Reduction and Checking Uncertainties ..............................................................38
   6.2 Generation of Performance Curve from Recorded Data Points ...............................39
6.3 Standardized Uncertainty Limits ................................................................. 39
6.4 Using Redundancy to Check Test Measurement and Uncertainty ............... 39
6.5 Effects of Fouling on Test Results ............................................................... 40
6.6 Analysis of Measured Results ................................................................. 40
7. Other Field Testing Considerations .............................................................. 40
  7.1 Determination of Influential Test Parameters ............................................ 41
  7.2 Field Testing of Compressor Under Wet Gas Conditions ......................... 41
8. References ..................................................................................................... 41

APPENDICES

APPENDIX A – Determination of Gas Turbine Power ............................................... 45
APPENDIX B – Equation of State Models ............................................................ 49
APPENDIX C – Uncertainty Analysis for Independent Variable Measurements .......... 57
APPENDIX D – Similarity Calculations for Wet Gas Conditions ............................ 67
APPENDIX E – Equation of State Model Comparison of Predicted Performance Data ...... 71
APPENDIX F – Application of Compressor Equations for Side Stream Analysis .......... 77
## LIST OF FIGURES

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Figure 1</td>
<td>Location of Test Instrumentation for Centrifugal Compressor</td>
<td>2</td>
</tr>
<tr>
<td>Figure 2</td>
<td>Location of Test Instrumentation for Gas Turbine</td>
<td>2</td>
</tr>
<tr>
<td>Figure 3</td>
<td>Enthalpy/Pressure Change During Compression and Expansion Process</td>
<td>6</td>
</tr>
<tr>
<td>Figure 4</td>
<td>Typical Compressor Surge Line on Compressor Performance Map</td>
<td>11</td>
</tr>
<tr>
<td>Figure 5</td>
<td>ASME PTC 10 Recommended Installation Configuration for Pressure and Temperature Measurement</td>
<td>22</td>
</tr>
<tr>
<td>Figure 6</td>
<td>Short-Coupled Installation for a Turbine Meter (AGA-7, Rev. 3)</td>
<td>26</td>
</tr>
<tr>
<td>Figure 7</td>
<td>Close-Coupled Installation for a Turbine Meter (AGA-7, Rev. 3)</td>
<td>26</td>
</tr>
<tr>
<td>Figure 8</td>
<td>Sampling Method with Pigtail as Recommended in API MPMS Chapter 14.1</td>
<td>29</td>
</tr>
<tr>
<td>Figure 9</td>
<td>Example of Test Uncertainty Range</td>
<td>40</td>
</tr>
</tbody>
</table>
## LIST OF TABLES

<table>
<thead>
<tr>
<th>Table</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Table 1</td>
<td>Suggested Applications for Equation of Stage Usage</td>
<td>10</td>
</tr>
<tr>
<td>Table 2</td>
<td>ASME PTC 10 Acceptable Deviation in Test Parameters for Similarity Conditions</td>
<td>13</td>
</tr>
<tr>
<td>Table 3</td>
<td>Assessment of Stability of Compressor During Pre-Test – Criteria 1</td>
<td>18</td>
</tr>
<tr>
<td>Table 4</td>
<td>Assessment of Stability of Compressor During Pre-Test – Criteria 2</td>
<td>18</td>
</tr>
<tr>
<td>Table 5</td>
<td>Assessment of Stability of Gas Turbine During Pre-Test</td>
<td>18</td>
</tr>
<tr>
<td>Table 6</td>
<td>Typical Uncertainties in Pressure Measurement (shown as percent of full scale)</td>
<td>21</td>
</tr>
<tr>
<td>Table 7</td>
<td>Recommended Depth of Thermowells</td>
<td>23</td>
</tr>
<tr>
<td>Table 8</td>
<td>Typical Uncertainties in Temperature Measurement (shown as percent of full scale)</td>
<td>24</td>
</tr>
<tr>
<td>Table 9</td>
<td>ISO 5167 Recommended Installation Lengths for Orifice Flow Meter</td>
<td>25</td>
</tr>
<tr>
<td>Table 10</td>
<td>In-Practice Achievable Uncertainty for Measured Test Parameters</td>
<td>32</td>
</tr>
<tr>
<td>Table 11a</td>
<td>Example of Total Uncertainty Calculation for Compressor in “Near Ideal” Case – SI Units</td>
<td>34</td>
</tr>
<tr>
<td>Table 11b</td>
<td>Example of Total Uncertainty Calculation for Compressor in ”Near Ideal” Case – English Units</td>
<td>35</td>
</tr>
<tr>
<td>Table 12a</td>
<td>Ideal Installation for Gas Turbine – Total Uncertainty Calculation – SI Units</td>
<td>35</td>
</tr>
<tr>
<td>Table 12b</td>
<td>Ideal Installation for Gas Turbine – Total Uncertainty Calculation – English Units</td>
<td>36</td>
</tr>
<tr>
<td>Table 13</td>
<td>Effect of Non-Ideal Temperature or Pressure Measurement</td>
<td>37</td>
</tr>
<tr>
<td>Table 14</td>
<td>Non-Ideal Installations Effect on Compressor Uncertainty</td>
<td>37</td>
</tr>
<tr>
<td>Table 15</td>
<td>Non-Ideal Installations Effect on Gas Turbine Uncertainty</td>
<td>38</td>
</tr>
</tbody>
</table>
Definition of Symbols:

- \( A \) = cross-sectional area of pipe
- \( C \) = discharge coefficient for orifice flow meter
- \( D_{tip} \) = tip diameter of compressor
- \( E \) = velocity of approach factor
- \( EHR \) = exhaust heat rate
- \( H \) = head for compressor, either actual or isentropic
- \( HR \) = gas turbine heat rate
- \( LHV \) = fuel gas heating value, as determined through thermodynamic analysis
- \( Ma \) = Machine Mach number
- \( N \) = shaft speed in rpm \([\omega = 2\pi N / 60]\)
- \( P \) = shaft power
- \( P \) = total (stagnation) pressure of gas at suction or discharge side
- \( P_{stat} \) = static pressure at suction or discharge side
- \( Q \) = volumetric flow rate on suction or discharge side of compressor
- \( SM\% \) = surge margin of compressor as % of design flow rate for fixed speed
- \( T \) = temperature of gas at suction or discharge side
- \( TD\% \) = turndown of compressor as % of design flow rate for constant head
- \( U \) = velocity of gas
- \( W \) = mass flow through the compressor
- \( Z \) = compressibility of gas at suction or discharge conditions

- \( c_p \) = specific heat at constant pressure
- \( d \) = bore diameter of orifice meter
- \( f \) = Schultz correction factor for real gas behavior
- \( h \) = enthalpy of gas at suction, discharge or isentropic conditions
- \( k \) = isentropic exponent

- \( \Delta p \) = differential pressure measured across orifice plate
- \( \eta \) = efficiency
- \( \varphi \) = flow coefficient
- \( \gamma \) = ratio of specific heats
- \( \rho \) = density of gas determined at suction or discharge conditions
- \( \tau \) = measured torque from the turbine shaft
- \( \nu \) = specific volume of gas at suction, discharge or isentropic conditions
- \( \psi \) = head coefficient
- \( \omega \) = shaft speed in radians per second \([\text{rad/sec}]\)
Subscripts:

- \( m \) = mechanical efficiency
- \( * \) = isentropic condition
- \( P \) = polytropic condition
- \( C \) = compressor
- \( GT \) = gas turbine
- \( TC \) = turbocompressor package
- \( FG \) = fuel gas properties
- \( s \) = suction gas
- \( d \) = discharge gas
- \( di \) = discharge gas, isentropic conditions
- \( in \) = input to gas turbine
- \( out \) = output to gas turbine/input to compressor
- \( A \) = density of air
- \( E \) = exhaust air temperature
- \( f \) = fuel flow to the gas turbine
- \( GT \) = volume flow of air at gas turbine exhaust
- \( tip \) = tip diameter or tip speed for compressor blade
- \( stat \) = static
Definitions:

1. **Absolute Pressure**: The pressure measured above a perfect vacuum.
2. **Gage Pressure**: The pressure measured with the existing barometric pressure as the zero base reference.
3. **Differential Pressure**: The difference between any two pressures measured with respect to a common reference (i.e., the difference between two gage pressures.)
4. **Total (Stagnation) Pressure**: An absolute or gage pressure that would exist when a moving fluid is brought to rest, and its kinetic energy is converted to an enthalpy rise by an isentropic process from the flow condition to the stagnation condition. In a stationary body of fluid, the static and total pressures are equal.
5. **Absolute Temperature**: The temperature above absolute zero, stated in degrees Rankine or Kelvin. Rankine temperature is the Fahrenheit temperature plus 459.67 degrees; Kelvin is the Celsius temperature plus 273.15 degrees.
6. **Total (Stagnation) Temperature**: The temperature that would exist when a moving fluid is brought to rest, and its kinetic energy is converted to an enthalpy rise by an isentropic process from the flow condition to the stagnation condition. In a stationary body of fluid, the static and total temperatures are equal.
7. **Density**: The mass of gas per unit volume, equal to the reciprocal of the specific volume. The density is a thermodynamic property determined from the absolute total pressure and temperature at a point in the fluid using an equation of state.
8. **Capacity**: The rate of flow, determined by delivered mass flow rate divided by inlet gas density.
9. **Pressure Ratio**: The ratio of absolute total discharge pressure to absolute total suction pressure.
10. **Machine Mach Number**: The ratio of the blade tip velocity at the first impeller diameter to the acoustic velocity of the gas at the suction conditions.
11. **Stage**: A single impeller and its associated stationary flow passages.
12. **Compressor Surge Point**: The capacity below which the compressor operation becomes aerodynamically unstable.
13. **Isentropic Compression**: A reversible, adiabatic compression process.
14. **Polytropic Compression**: A reversible, non-adiabatic compression process between the total suction pressure and temperature and the total discharge pressure and temperature.
15. **Gas Power**: The power transmitted to the gas in a compressor, equal to the product of the mass flow rate compressed and the gas work.
16. **Shaft Power**: The power delivered to the compressor shaft by the gas turbine, also known as brake power. Shaft power is equal to gas power plus mechanical losses.
17. **Mechanical Losses**: The total power consumed by frictional losses in integral gearing, bearings and seals.

**Equation of State**: An equation or series of equations that functionally relate the gas thermodynamic properties, such as pressure, temperature, density, compressibility, and specific heats.

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Definitions for referenced terms in Field Test Guideline are based upon ASME Performance Test Code (PTC) 10-1997, “Performance Test Code on Compressors and Exhausters.”
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Guideline for Field Testing of Gas Turbine and Centrifugal Compressor Performance

RELEASE 2.0

1. PURPOSE AND APPLICATION

The following guideline is intended to serve as a reference for field testing of gas turbine and centrifugal compressor performance. This guideline applies to any party conducting a field test of a gas turbine or centrifugal compressor (manufacturer, user company, or third-party). It is intended to provide the most technically sound, yet practical procedure for all aspects of conducting field performance tests of gas turbines and centrifugal compressors.

The conditions at the field site often cannot be as closely controlled as in a factory environment. The specific site conditions of a particular test may dictate that the test procedure deviates from this guideline or the ideal installation described. This does not preclude a field site test. Nonetheless, when a particular test deviates from the installation requirements or other test procedures, the deviation will affect the test uncertainty and should be accounted for in the uncertainty analysis, as recommended in this guideline.


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 primary measurements required in order to calculate these parameters are discussed in Section 4.0. The uncertainty calculations are discussed in Section 5.0. Accounting for the effect of non-ideal installations on uncertainty is also discussed in Section 5.0.

Performance Parameters:

1. Centrifugal Compressor Flow/Flow Coefficient
2. Centrifugal Compressor Head/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)
7. Gas Turbine Exhaust Heat Rate

The following test data must be measured to determine the above performance parameters. Figures 1 and 2 show the general measurement arrangement for the required test instrumentation on the compressor and gas turbine.
* Flow Rate Measurement and Gas Sample may be on suction or discharge side. Suction side is recommended.

Figure 1. Location of Test Instrumentation for Centrifugal Compressor

Figure 2. Location of Test Instrumentation for Gas Turbine
Centrifugal Compressor Test Measurements:

- Suction Temperature
- Suction Pressure
- Discharge Temperature
- Discharge Pressure
- Flow Through Compressor (*Pressure, temperature also required at the flow measurement point)
- Suction or Discharge Gas Composition
- Barometric Pressure
- Speed of Rotation
- Impeller Diameter
- Upstream and Downstream Piping arrangement
- Pipe Diameter (upstream and downstream)

Gas Turbine Test Measurements:

- Engine Inlet and Ambient Temperature
- Barometric Pressure
- Power Turbine Speed
- Gas Generator Speed
- Fuel Flow (*Pressure, temperature also required at flow measurement point for fuel gas)
- Fuel Gas Composition
- Inlet and Exhaust Pressure Loss
- Relative Humidity of Inlet Air
- Water/Steam Injection Rate

**Important Note:** For the remainder of this document, all pressures and temperatures used for performance and uncertainty calculations are absolute total (stagnation) values unless otherwise noted.

### 2.1 Centrifugal Compressor Flow/Flow Coefficient

The actual flow through the centrifugal compressor \( Q \) should be measured by a flow-measuring device, such as a volumetric flow meter (ultrasonic, turbine, etc.), a differential pressure device (orifice meter, annubar, etc.), or a nozzle. If an orifice meter is used (typical of many installations), the mass flow rate equation is:

\[
W_s = CE \frac{\pi}{4} d^2 \cdot \sqrt{2\Delta p \rho_s} \tag{2.1}
\]

* Note the discharge coefficient, \( C \), is determined from the RG equation (as stated in AGA Report No. 3). The discharge coefficient is dependent on the flow meter Reynolds Number.

The actual suction volumetric flow is given by:

\[
Q_s = \frac{W_s}{\rho_s} \tag{2.2}
\]
The flow coefficient used for similarity comparisons is:

\[
\varphi = \frac{Q_s}{\frac{\pi}{4} D_{tip}^2 U_{tip}} = \frac{Q_s}{\frac{\pi}{4} D_{tip}^2 \omega}
\] (2.3)

* Note the flow coefficient uses the actual volumetric flow rate through the compressor at suction conditions.

* For a 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/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.

The heads are calculated from the enthalpies associated with each state from the EOS as follows:

**Isentropic head:**

\[
H^* = h_d^* - h_s = h(p_d, s_s) - h(p_s, T_s)
\] (2.4)

**Actual head:**

\[
H = h_d - h_s = h(p_d, T_d) - h(p_s, T_s)
\] (2.5)

* Note that \( h_d^* \) is the enthalpy associated with the discharge pressure at the suction entropy, \( s_s \), because the entropy change is zero in an isentropic process. All enthalpies should be directly determined from the EOS.

Isentropic enthalpy can also be for estimation purposes (assuming ideal gas behavior):

\[
h_d^* \approx c_p^* \cdot T_d^* = c_p^* \cdot T_s \cdot \left( \frac{P_d}{P_s} \right)^{\frac{k-1}{k}}
\] (2.6)
Similarly, Polytropic head is determined from:

\[
H^p = \left[ \frac{n^p}{n^p - 1} \cdot \left( \frac{P_d}{P_s} \right)^{\frac{n^p - 1}{n^p}} - 1 \right] \cdot f \cdot P_s V_S
\]  

(2.7)

The polytropic exponent, \(n^p\), is defined as:

\[
n^p = \frac{\ln P_d}{P_s} \frac{n^p}{P_s} \frac{\ln V_S}{V_d}
\]  

(2.8)

The isentropic exponent, \(k\), is defined as:

\[
k = \frac{\ln P_d}{P_s} \frac{k}{\ln V_S} \frac{1}{V_d}
\]  

(2.9)

For equation (2.7), the Schultz Polytropic Head Correction Factor, \(f\), is defined as:

\[
f = \frac{h_d^* - h_s}{\left[ k \frac{k}{k - 1} \left( P_d V_d^* - P_s V_s^* \right) \right]}
\]  

(2.10)

For performance comparisons, it is beneficial to use non-dimensional head and flow coefficients (\(\phi^*\) from equation (2.3) and \(\psi^*, \psi, \psi^p\) from equation 2.11–2.13) rather than actual head and flow.

**Isentropic head coefficient:**

\[
\psi^* = \frac{H^*}{U^2} = \frac{2H^*}{\left( \pi D_{tip} \omega \right)^2}
\]  

(2.11)

**Actual head coefficient:**

\[
\psi = \frac{H}{U^2} = \frac{2H}{\left( \pi D_{tip} \omega \right)^2}
\]  

(2.12)
**Polytropic head coefficient:**

\[
\psi^p = \frac{H^p}{U^2} = \frac{2H^p}{(nD_{tip}N)^2}
\]  

(2.13)

### 2.3 Centrifugal Compressor Efficiency

The isentropic efficiency is calculated from the isentropic and actual head:

\[
\eta^* = \frac{H^*}{H} = \frac{\psi^*}{\psi}
\]  

(2.14)

The polytropic efficiency is calculated based upon the polytropic head and the polytropic exponent, \(n_P\), as defined in equation (2.8):

\[
\eta^p = \frac{H^p}{H} = \frac{n^p}{(n^p - 1)} \cdot \frac{P_d}{P_s} \cdot \frac{n^p - 1}{n^p - 1} \cdot f \cdot P_s \psi_s
\]  

(2.15)

The compression process for a typical centrifugal compressor and the associated enthalpy change are shown on a \(P-h\) diagram in Figure 3 for 100% methane gas mixture. An actual process is compared to the isentropic states.

---

**Figure 3.** Enthalpy/Pressure Change During Compression and Expansion Process  
(Edmister and Lee, 1984)
2.4 Gas Turbine Power

Four methods exist for determining gas turbine power. These are:

1. Direct torque coupling measurements
2. Direct generator power measurements
3. Indirect driven centrifugal compressor shaft power measurements
4. Indirect gas turbine heat balance measurements

The direct measurements methods (1) and (2) either using a torque-measuring coupling (see Section 4.6) or the input power from the generator (see Section 4.7) will normally yield the lowest uncertainty. Using the driven centrifugal compressor shaft power to determine gas turbine shaft output power will usually yield a higher uncertainty but is a widely used and acceptable method, if properly performed. The gas turbine heat balance method yields the highest measurement uncertainty and is generally not recommended. See Appendix A for a complete discussion of the indirect approaches.

If the torque ($\tau$) is measured using a torque coupling, then the shaft power ($P$) developed by the gas turbine is calculated as:

$$P = \tau \times (2\pi N)$$

(2.16)

If gas turbine power is determined for an electric power generation application, the gas turbine shaft output power can be simply determined from the measured electric power at the generator terminals, the generator efficiency, and the gearbox efficiency (see Section 4.7). Indirect methods are more complex and are described in section 2.5 and Appendix A.

2.5 Centrifugal Compressor Absorbed Power (Gas Turbine Power Output)

The absorbed power for the compressor ($P_C$) can be directly used to determine the gas turbine shaft output power, 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 follows:

$$P_C = P_{out} \cdot \eta_m = \rho_s \cdot Q_s \cdot H$$

(2.17)

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.

2.6 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.

$$\eta_{GT} = \frac{P_{out}}{W_f(LHV)}$$

(2.18)

Similarly, the gas turbine heat rate is simply the reciprocal of the efficiency, or:
\[ HR = \frac{W_f \cdot (LHV)}{P_{out}} \]  

(2.19)

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 (36°F) above the ambient temperature, the fuel gas’ sensible heat should be added to the equations above as such:

\[ \eta_{GT} = \frac{P_{out}}{W_f \cdot (LHV + (\rho \cdot c_p \cdot T)_{FG})} \]  

(2.20)

\[ HR = \frac{W_f \cdot (LHV + (\rho \cdot c_p \cdot T)_{FG})}{P_{out}} \]  

(2.21)

Sensible heat represents the energy introduced into the combustor in the form of thermal heat contained in the fuel.

### 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, or:

\[ EHR = W_{GT} \left( h_E - h_R \right) \]  

(2.22)

In equation 2.22 \( h_R \) 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 Appendix A under method 4.

### 2.8 Turbocompressor Package Efficiency

The turbocompressor package efficiency (\( \eta_{TC} \)) may be calculated based upon the previous values. Namely, the total package efficiency is the product of the gas turbine, gearbox, and compressor efficiencies:

\[ \eta_{TC} = \eta_c \cdot \eta_{GB} \cdot \eta_{GT} \cdot \eta_{M, compressor} \]  

(2.23)

Note the compressor efficiency, \( \eta_c \), may be defined as either the isentropic or polytropic efficiency. For compressor drives, the package efficiency may also be calculated as the compressor isentropic gas power divided by the fuel energy rate into the gas turbine:

\[ \eta_{TC} = \frac{\rho \cdot Q_s \cdot H^*}{W_f \cdot LHV} \]  

(2.24)
Note that equation (2.24) does not work for the calculation of efficiency if polytropic head is used instead of isentropic head. If equation (2.24) is used to define the package efficiency, an agreement about the treatment of recirculation and leakage losses must be made to assure that these losses are addressed properly in the turbocompressor package efficiency.

### 2.9 Equations of State

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. Equation of state model accuracy may depend upon the application range and the gas mixture at the site (Sandberg, 2005; Kumar et al., 1999).

The consistent application of the equation of state throughout the planning, testing, and analysis phases of the field test is imperative. The choice of which EOS to use must be agreed upon before the test. 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 ISO 5389 to avoid additional test uncertainties. The selection of a particular EOS can have an important effect on the apparent efficiency and absorbed gas power. An added uncertainty of 1 to 2% can be incurred on the performance results if the EOS is inconsistently applied (Kumar et al., 1999). The formulation of the various EOS is given in Appendix B.

#### 2.9.1 Application of Equation of State

Generally, it is not possible to select a “most accurate” EOS to predict gas properties, since there is generally no “calibration norm” 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 1 provides usage suggestions for the various EOS models based on application. For normal hydrocarbon gas mixtures (such as pipeline quality gas) with diluent content (combined CO$_2$ and N$_2$) below 10%, all equations of state shown in Table 1 provide accurate results. Beyond this range, Table 1 provides some general recommendations on the most applicable EOS.
Table 1. Suggested Applications for Equation of Stage Usage

<table>
<thead>
<tr>
<th>Type of Application</th>
<th>Typically Used EOS Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Typical hydrocarbon gas mixture, standard pressures and temperatures, low CO₂ and N₂ diluents (&lt; 6% total). Air mixtures.</td>
<td>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</td>
</tr>
<tr>
<td>High-pressure applications (&gt;3000 psi).</td>
<td>BWRS, BWR, LKP</td>
</tr>
<tr>
<td>High CO₂ and N₂ diluents (10-30%) and/or high hydrogen content gases.</td>
<td>BWRS, LKP</td>
</tr>
<tr>
<td>High hydrogen content gases (&gt;80% H₂).</td>
<td>PR, LKP, SRK</td>
</tr>
<tr>
<td>Non-hydrocarbon mixtures: ethylenes, glycols, carbon dioxide mixtures, refrigerants, hydrocarbon vapors, etc.</td>
<td>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.</td>
</tr>
</tbody>
</table>

A further comparison of the various EOS models is provided in Appendix E. The calculated enthalpies for various EOS models at different states are used to calculate isentropic efficiency and compressor power for two compressor operating cases.

2.10 Determination of Surge Point and Turndown

The low limit flow on a compressor is the surge point. Oftentimes, rotating diffuser stall can further limit the compressor operating range due to aerodynamically induced vibrations. The surge point is customarily used as the lower flow limit for turndown determination. During a surge event, the flow field within the compressor collapses, and the compressor head and flow rate drop suddenly. Surge is usually a sudden and sometimes catastrophic event and full surge should be avoided.

The turndown of a centrifugal compressor is the allowable operating range between the design point and the surge line at any given speed for a fixed compressor head. It is determined from the difference between design flow rate and the minimum flow rate at which the compressor is aerodynamically stable as a percentage of the design flow rate at the same head, as follows:

\[
TD\% = 100 \cdot \left( \frac{Q_{\text{design}} - Q_{\text{surge}}}{Q_{\text{design}}} \right)_{\text{Head} = \text{constant}} \tag{2.25}
\]

The surge margin is defined as the difference between design flow rate and minimum flow rate as a percentage of the design flow rate, for a fixed speed \(N\), as follows:

\[
SM\% = 100 \cdot \left( \frac{Q_{\text{design}} - Q_{\text{surge}}}{Q_{\text{design}}} \right)_{\text{Speed} = \text{constant}} \tag{2.26}
\]

The determination of the compressor surge point should be conducted with extreme caution at relatively low pressure differential and operating pressure (i.e., “low energy”) conditions. Thus, surge point testing should be at operating conditions that correspond to low energy conditions. If incipient surge tests must
be performed at high head conditions, these tests should be preceded by tests at low energy conditions in order to characterize the compressor behavior and instrument outputs of incipient surge. Prior review of the compressor’s dynamic test reports to identify limiting vibration levels may also be valuable to avoid damaging compressor internals during the test.

Surge point testing conducted on a factory test stand will often produce different results than testing at a field installation because the piping configuration and other installation details do affect the minimum stable flow. Significant gas composition changes will alter the performance map of the compressor and lead to erroneous predictions of head and flow. Thus, surge testing can be important to establish the correct surge line in the field. An example of a typical compressor surge line at various compressor speeds is shown in Figure 4 as the lower operational boundary of the compressor.

![Compressor Curve - Surge Line and Operational Boundaries](image)

**Figure 4. Typical Compressor Surge Line on Compressor Performance Map**

There are a number of instrument readings that can provide an indication that the compressor is approaching its surge point; however, for each compressor geometry and operating conditions, these indications may be of varying magnitude or sometimes may not occur at all. Thus, it is difficult to identify a singular instrument reading that should be utilized for the identification of incipient surge. In general practice, there are five indications that should be monitored:

1. A marked increase in flow fluctuations in the suction and discharge piping
2. An increase in suction or discharge pressure pulsation
3. An increase in shaft vibrations (both axial and radial directions)
4. A decrease in head as flow is decreased
5. An audible indication from the compressor of a significant operating change

As surge is a sudden and significant event, identifying the surge from the compressor’s operating noise can be dangerous, as the compressor may already be in full surge. Using increased vibrations as the criterion to determine the surge point will be considerably more inaccurate than other methods (Kurz and
due to variations in the mechanical responsiveness of different compressor systems. However, the increased vibration can be due to the onset of stalled flow rather than full surge. Also, not all compressors have their surge point corresponding to the maximum head for a fixed speed line.

A good method to determine the surge line is by measuring flow and pressure fluctuations on the compressor suction and discharge piping. As these fluctuations occur at a higher frequency range, pressure transducers and flow meters with a good dynamic response must be utilized (linear to at least 100 Hz). If the dynamic transducers are installed at a distance upstream or downstream from the compressor, the signal may be delayed. Thus, the surge line should always be approached slowly, by carefully throttling suction or discharge flow while maintaining compressor speed. It is recommended that the surge line is determined for a minimum of three different compressor speed lines.

### 2.11 Similarity Conditions

Generally, available field test conditions will deviate from specified test conditions used in the factory test or previous testing of the gas turbine/compressor package. Thus, similarity conditions are used to adjust for differences in the test conditions in order to match the flow characteristics for the machine under different test conditions. The similarity variables that must be calculated are provided in Sections 2.11.1 and 2.11.2 to follow.

#### 2.11.1 For the Centrifugal Compressor

To compare performance data for a centrifugal compressor between predicted performance and actual test data, the non-dimensional parameters for head and flow must be used. Namely, predicted performance, factory test data, and field test results should be normalized using the head coefficient ($\Psi$) and the flow coefficient ($\phi$) that were given previously in Sections 2.1 and 2.2. By matching these coefficients, the data can then be directly compared (as long as the Machine Mach number, isentropic exponent, and volume flow ratio are similar as discussed below).

This comparison of non-dimensional head and flow eliminates the requirement to test the compressor at the identical speed as the factory test (or other baseline tests of the unit) during the field test. To vary $\phi$ and $\psi$ during the test, the compressor speed and actual head must be adjusted.

In addition to the head and flow coefficients for thermodynamic similarity, the machine Mach number, isentropic exponent, and volume/flow ratio should be maintained as closely as possible to the comparison test values to maintain aerodynamic similarity.

#### Machine Mach number:

\[
Ma = \frac{U}{\sqrt{k_s \cdot Z_s \cdot RT_s}} = \frac{\pi \cdot D_{tip} \cdot N}{\sqrt{k_s \cdot Z_s \cdot RT_s}}
\]

#### Isentropic exponent:

\[
k = \left(\frac{\nu \phi}{\rho \partial \Phi \partial v}\right)
\]

#### Volume Flow Ratio:
In general, single- and two-stage compressors may allow deviations up to 10% from the Machine Mach number and isentropic exponent. For machines with multiple stages or high Machine Mach number (> 0.8), a 10% deviation in the Machine Mach number or isentropic exponent may lead to unacceptable deviations in the data sets. Thus, a direct comparison between data sets should not be made unless the Machine Mach number and isentropic exponent are the same, in such cases. If the test data shows unexplainable deviation from the predicted performance, a deviation in the Machine Mach number may be the cause.

An alternative approach to matching compressor performance data is provided by ASME PTC 10 but is generally written for factory testing rather than field testing. ASME PTC 10 allows deviations in the test and design cases for inlet pressure, inlet temperature, specific gravity, speed, capacity, and inlet gas density (see Table 2). Namely, if the flow coefficient and head coefficient remain the same as in the comparative (factory) test, the velocity triangles at the inlet and outlet of each stage of the compressor will remain the same. This is also known as the Fan Law. Deviations in the test within the limits given in Table 2 will require only a correction using the Fan Law. The volume ratio shown in equation (2.29) should also remain the same. The volume ratios may be kept constant by maintaining the Mach number and isentropic exponent (within 10%) over the machine.

<table>
<thead>
<tr>
<th>Condition</th>
<th>Acceptable Deviation (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet Pressure</td>
<td>5</td>
</tr>
<tr>
<td>Inlet Temperature</td>
<td>8</td>
</tr>
<tr>
<td>Specific Gravity</td>
<td>2</td>
</tr>
<tr>
<td>Speed</td>
<td>2</td>
</tr>
<tr>
<td>Capacity</td>
<td>4</td>
</tr>
<tr>
<td>Inlet Gas Density</td>
<td>8</td>
</tr>
</tbody>
</table>

If the intent is to compare identical compressors, $\psi$ and $\phi$ can be simplified to $Q/N$ and $H/N^2$ as shown by the Fan Law Proportionality below.

**Fan Law Proportionality:**

$$Q \propto N; \ H \propto N^2; \ P \propto N^3,$$

where:  
- $Q =$ actual volumetric flow rate, acfm  
- $N =$ rotational speed, rpm  
- $H =$ head, ft-lb/lb  
- $P =$ power, ft-lb/min

However, if the test conditions are considerably different and outside the limits of ASME PTC 10 (Table 2) or the Mach number difference or isentropic exponent difference is outside the acceptable range (as discussed above), the Fan Law Proportionality is no longer valid. It is recommended in this case that the compressor manufacturer design software be used to recalculate compressor performance as well as the head and flow coefficient curves at the changed conditions. Both the actual head and flow and/or the
head coefficient and flow coefficients can be utilized to verify compressor performance. The compressor manufacturer should inform the user of the EOS model used in recalculating the compressor performance.

### 2.11.2 For the Gas Turbine

For the gas turbine, the most important parameters affecting the performance are engine inlet temperature, gas turbine speed and ambient pressure. To a lesser extent, fuel gas composition and relative humidity may also influence the performance characteristics. The achievement of similarity conditions for the gas turbine is more difficult than for the gas compressor because of the gas turbine sensitivity to ambient conditions. ASME PTC 22 (1997) recommends that actual gas turbine performance curves should be used to correct the actual test data. However, these curves are often not available, particularly for older machines.

#### 2.11.2.1 Full Load Operation

The recommended method of correcting the measured performance of the turbine is to use ISO standard or actual performance curves or the manufacturer’s software. In addition, the testing parties should agree on the acceptable departure limits for the gas turbine parameters under test prior to the actual field testing. The procedure for correcting the gas turbine to similarity conditions is as follows:

1. Determine full load power and heat rate for actual ambient conditions and turbine speed.
2. Use the maps or the performance program to calculate the performance of a “nominal” engine at the same conditions as in (1).
3. Calculate the percent difference between the test results in (1) and (2) above, for power and heat rate.
4. Use the maps or the performance program to calculate the performance of the “nominal” engine under desired new conditions.
5. Apply the percent difference for power and heat rate calculated under (3) to reference values in (4) to yield engine performance under desired new conditions.

#### 2.11.2.2 Part Load Operation

The following procedure for correcting the gas turbine to similarity conditions at part load operation applies:

1. Determine part load heat rate for defined ambient conditions and turbine speed.
2. Use the maps (if available for part load operation) or the performance program to calculate the performance of a “nominal” engine at the same conditions as in (1).
3. Calculate the percent difference between the test results in (1) and (2) above for heat rate.
4. Use the maps (if available for part load operation) or the performance program to calculate the performance of the “nominal” engine under desired new conditions.
5. Apply the percent difference for heat rate calculated under (3) to reference values in (4) to yield engine performance under desired new conditions.
3. TEST PREPARATION

A field test agenda or plan should be prepared prior to the test as this is an essential part of test preparation. The plan should include field conditions and equipment layout, instruments to be used and their location, method of operation, test safety considerations, and the pressure, temperature and flow limits of the facility. Piping and station layouts should be made available. Any deviations from normal operation that may be necessary to conduct the test should also be provided.

The field test agenda should include a discussion of the following:

1. The method of data reduction.
2. The selected approach for determining the test uncertainty.
3. The acceptance criteria (specified in terms of maximal uncertainty allowable).
4. The equation of state to be used for all calculations in the test.
5. The use of isentropic or polytropic calculations (either may be used for accurate thermodynamic performance characterization).

Test preparations should also include a discussion on possible 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 upon the tested package) or at fixed ambient conditions (if the necessary gas turbine power is only available on cold days). Because instrumentation is part of the overall station design, the requirements for installation of test instrumentation need to be communicated early.

The selection and calibration of the test instrumentation is important. Generally, the instruments supplied for monitoring and protection of the packages are not accurate enough to meet the stringent requirements necessary for a field test (redundant measurement requirements, small uncertainty margins, detailed sensor location placement, and the effects of improper flow measurement). Whenever possible, calibrated laboratory quality instrumentation should be installed for the tests. (Refer to Section 4.0.) The accuracy of the instruments and the calibration procedure should be such that the measurement uncertainty is reduced to the best attainable uncertainty under ideal conditions (see Section 5.1).

3.1 Pre-Test Meeting

A meeting between the test engineer, the parties involved (supplier, operator, etc.) and the customer to discuss test procedures and the situation on site should be conducted in advance of the performance test. The site P&ID, Site Layout and Mechanical Installation Drawing diagrams should be obtained (if available) and used in preparation for the performance test.

During the pre-test meeting, the parties should reach an agreement on the test purposes, test procedures, safety requirements, responsibilities during the test, availability of necessary operating conditions, and acceptance conditions.

3.2 Pre-Test Operation and Instrumentation Checkout

The following items should be checked during the pre-test checkout:

A. The test engineer should verify that the unit has been proven suitable for continuous operation.
B. The test engineer should note if a gas compressor start-up strainer is installed in the inlet pipe. If so, the strainer should be checked for cleanliness, either by use of a differential pressure gauge, direct inspection, or by borescope inspection.

C. Sufficient gas should be available for proper operation of the gas compressor.

D. All instrumentation should be calibrated in the range in which it will be operated during the test. Check all instrument readings for temperature, pressure, flow, torque, and speed to assure that the sensors are functioning properly. Verify data acquisition system operation prior to starting the field performance test.

E. All RTD’s or thermocouples used in the test should use spring load type fittings, or when necessary, the thermowells will be serviced with oil or other approved heat transfer material.

F. If thermowells are used during the test and a large portion of the thermowell is exposed to the atmosphere, the area around the exposed portion should be insulated to preclude the ambient air from affecting the temperature reading.

G. Check insertion depth of thermowells.

H. Where pressure taps involve tubing runs, the tubing should be checked for leaks.

I. The proper number of capable personnel should be on site to ensure that all the data can be recorded in a reasonable amount of time.

3.3 Pre-Test Equipment Checkout

Prior to running the field performance test, the following should be performed:

- Wash the gas turbine compressor thoroughly.
- Clean air inlet filter panels (if necessary).
- Verify fuel and site load to assure continuous operation of the unit and full-load conditions as required at the time of the test.
- Perform a visual walk-through of the turbine compressor package to eliminate any sources of hot air ingestion or recirculation.
- Consult with gas control on station operation. Check if gas cooling is available and if recirculation of gas is an option during field test. Notify all parties of time frame for test.

3.4 Pre-Test Information

The following information should be obtained as a result of the test preparation and pre-test meeting:

- Impeller diameter.
- Predicted performance curves for compressor (or existing test curves).
• Flow meter information: Pipe ID, orifice bore or beta ratio (for orifice meter), K-factor (for turbine or vortex shedding meter), flow coefficient (for annubar or nozzle) scaling frequency, configuration log (for ultrasonic meter or to adjust turbine or mass flow meters).

• Engine performance data, such as factory test data and predicted performance.

• Manufacturer Engine Performance Maps or their electronic representations.

• Piping geometry between compressor and test instrumentation.

3.5 Test Stability

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 (heating value change of more than 1.0%) in between test points.

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. Note that an increase in pressure ratio due to drift during the test will cause an increase in the temperature as well, though the temperature change will lag behind the pressure change. Refer to Section 5.0 on uncertainty for more discussion of unsteady conditions and drifting conditions during a test. These added uncertainties due to drift during the test interval are in addition to non-ideal effects discussed in Section 5.2.

3.5.1 Compressor Steady State

The compressor should be operated for at least 30 minutes prior to the test or until stable readings are reached. Steady state is achieved if all of the compressor measurements listed in Table 3 apply during a 10-minute interval.
Table 3. Assessment of Stability of Compressor During Pre-Test – Criteria 1

<table>
<thead>
<tr>
<th>Test Reading</th>
<th>Maximum Allowable Variation During 10-min Interval</th>
</tr>
</thead>
<tbody>
<tr>
<td>Suction Temperature</td>
<td>± 1°C (± 1.8°F)</td>
</tr>
<tr>
<td>Discharge Temperature</td>
<td>± 1°C (± 1.8°F)</td>
</tr>
<tr>
<td>Suction Pressure</td>
<td>± 1% of Average Value</td>
</tr>
<tr>
<td>Discharge Pressure</td>
<td>± 1% of Average Value</td>
</tr>
<tr>
<td>Compressor Speed</td>
<td>± 10 rpm</td>
</tr>
<tr>
<td>Compressor Flow</td>
<td>± 1.0% of Average Value</td>
</tr>
</tbody>
</table>

Alternatively, the following performance conditions shown in Table 4 should be satisfied:

Table 4. Assessment of Stability of Compressor During Pre-Test – Criteria 2

<table>
<thead>
<tr>
<th>Test Reading</th>
<th>Maximum Allowable Variation During 10-min Interval</th>
</tr>
</thead>
<tbody>
<tr>
<td>Efficiency</td>
<td>Fluctuations &lt; ± 0.5% of Average</td>
</tr>
<tr>
<td>Head</td>
<td>± 0.5% of Average Value</td>
</tr>
<tr>
<td>Shaft Power</td>
<td>± 1% of Average Value</td>
</tr>
</tbody>
</table>

3.5.2 Gas Turbine Steady State

Before readings are taken for any individual test point, gas turbine steady state operating conditions must be achieved. The gas turbine must be heat soaked according to manufacturer specifications. If manufacturer specifications are not available, gas turbines should be heat soaked for at least 1 hour for aeroderivative gas turbines and small gas turbines (<10,000 hp), and 2 hours for large gas turbines (>10,000 hp), or until stable operations (per Table 5) have been reached. To verify stability of the gas turbine, the parameters given in Table 4 should be checked. Three to ten data points of each parameter over a 10-minute period should be recorded to verify stability. If the gas turbine has reached equilibrium, each of the parameters in Table 5 will fall within the stability criteria provided.

Table 5. Assessment of Stability of Gas Turbine During Pre-Test

<table>
<thead>
<tr>
<th>Test Reading</th>
<th>Maximum Allowable Variation During 10-min Interval</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbine Inlet Temperature Or Ambient Temperature</td>
<td>± 1 °C (± 1.8 °F)</td>
</tr>
<tr>
<td>Gas Producer Speed</td>
<td>± 1% Average Speed</td>
</tr>
<tr>
<td>Shaft Load</td>
<td>± 1% Average Load</td>
</tr>
<tr>
<td>Firing Temperature</td>
<td>± 5 °C (± 9 °F)</td>
</tr>
<tr>
<td>Power Turbine Speed</td>
<td>± 10 rpm</td>
</tr>
<tr>
<td>Fuel Flow</td>
<td>± 2% Average Flow</td>
</tr>
<tr>
<td>Generator</td>
<td>Line Voltage + 1%</td>
</tr>
</tbody>
</table>
3.5.3 Unsteady Operations

If unsteady operations cannot be avoided during the test interval, measurements may still be valid, but the fluctuations have to be accounted for in the uncertainty calculations of the results. If fluctuations during the test exceed quasi-steady conditions as given in Tables 3, 4, and 5, the test may need to be performed again. For measurement cases where there is a simple drift in the average operating condition, the criteria listed above should be employed to determine whether a data point is steady. If the drift in any of the instrument readings exceeds the steady state conditions (as defined in Tables 3, 4, and 5), it is difficult to determine any valid performance result from this measured data because of the high degree of interdependence of all measured parameters and the system as a whole. Namely, as the validity of the data depends on the rate of drift, heat storage capacity of the pipe and measurement system, and the frequency response of the transducers, a total uncertainty cannot be determined.

On the other hand, if the fluctuations in the data can be determined to be varying around a mean value, without the average drifting significantly, the resultant measurement error is primarily due to a time lag of the temperature transducers. Namely, while the pressure and flow transducers generally measure at a high frequency and, thus, capture rapid operating changes accurately, the temperature transducers lag due to the requirement of complete heat soaking of the piping and measurement system. Thus, if the fluctuations produce a mean performance value, the criteria for acceptance of unsteady operation can be extended to allow up to twice (i.e., factor-of-two range) the fluctuations listed in Tables 3, 4, and 5 for compressor and gas turbine steady state testing. In these cases, the fluctuations must be accounted for in the uncertainty calculation. As this can generally result in very high total uncertainties for efficiency and power, one should carefully evaluate whether to accept this test data. Also, once this factor-of-two range is exceeded, the non-linear behavior of the system as a whole makes it unrealistic to determine accurate performance results from experimental data.

3.6 Safety Considerations

Safety considerations should remain a priority during the pre-test phase, as well as the actual testing of the compressor and turbine. Abnormal operating conditions should be discussed with station personnel prior to running the test. If possible, a schematic of the yard piping should be given to all test personnel. Unit vibration equipment operation should be verified. When cables are run to test instrumentation, the cables should be covered with mats or correctly taped down (if possible) to reduce trip hazards. Cable connections should be secured. Finally, the requirements of the field test should not be given priority over station safety precautions in order to reduce measurement uncertainty or meet test schedules.

4. Measurement and Instrumentation

The gas turbine and centrifugal compressor should be equipped to measure the test variables shown in Figures 1 and 2. For testing purposes, a dedicated set of laboratory quality instrumentation should be utilized. This dedicated set of test instrumentation should be maintained and frequently calibrated using acceptable reference standards. A valid calibration certificate for all measurement instrumentation is recommended. An end-to-end calibration of the data acquisition system, wiring, and instrumentation is also recommended prior to the field test but may not always be practical.

If possible, all measurement instrumentation should be installed inside the branch piping to the compressor’s recirculation flow loop such that the measured values represent the true flow through the compressor. If the test instrumentation is located outside the recycle loop for the compressor, the recycle valve must be fully closed during the tests for the results to be valid.
A piping configuration using a closed loop through the compressor or station recycle line may also be utilized for the performance testing of the compressor. In this case, a process gas cooler on the discharge of the compressor will generally be required to maintain the gas temperature stable in the closed piping loop. Also, for this test scenario, the effects of gas lean out must be considered as the heavier components in the test gas may liquefy and drop out due to sequential compression and cooling. Thus, prior to the performance test, the compressor should be run in the closed loop configuration while monitoring the gas using an online gas chromatograph until there is no significant change in the gas composition.

4.1 Measurement of Pressure

4.1.1 Recommended Best Practice

Total (stagnation) pressure must always be used for performance calculations. However, it is often more convenient to measure static pressures \( P_{\text{stat}} \) and then convert static to total pressure \( P \) using:

\[
P = P_{\text{stat}} + 0.5 \rho U^2
\]

In equation (4.1), the flow velocities can be calculated using the measured flow rate and the pipe diameter \( U = Q/A \).

Whenever feasible, it is recommended to use four pressure taps and four temperature taps at the pressure and temperature locations indicated in Figures 1 and 2, consistent with ASME PTC 10 recommendations. The accuracy of the static pressure or temperature measurement is dependent upon the selected location. Four pressure and temperature sensors assure that the average measurement of pressure or temperature will be accurate, even in a non-uniform flow field. Additional pressure and temperature measurements can be employed, if four sensors are not sufficient.

Two different approaches are appropriate for locating the suction and discharge pressure and temperature taps. The first approach is to place measurement taps at locations relatively far upstream and downstream from the compressor in the longest available straight pipe segment to assure a uniform flow field at the transducer taps. These locations may be relatively far away from the compressor, so the pressure measurement values must be corrected using empirical loss factors (i.e., head losses) for the straight pipe, elbows, tees, and reducer that lie in between the measurement location and the compressor inlet/discharge.

The second approach is recommended for field testing, if possible. The approach is to measure the pressure and temperature as close as possible to the compressor, using multiple temperature and pressure-taps at suction and discharge. Generally, the flow field near the compressor will be highly non-uniform and, thus, at least four pressure and temperature taps should be used on both suction and discharge. Non-uniformity of the flow field affects the uncertainty of the measurement data. If less than four transmitters or test taps (for pressure or temperature) are available, the first measurement approach is recommended. Using less than four pressure or temperature sensors will result in an increase in the total uncertainty for pressure or temperature, as discussed in Section 5.2.

4.1.2 Installation

The installation of the pressure measurement device, pressure tap size, and symmetry is critical to the measurement accuracy. ASME PTC 10 provides specific guidelines for correct installation and location of pressure probes. The pressure tapping should be inspected prior to installation of the pressure measurement device. The tube and static tapping used to make the dynamic pressure measurement should have a constant length to diameter ratio and must be greater than 2. The ratio between the pressure tubing...
and the pipe diameter should be as small as possible to prevent the pressure measurement from altering the flow pattern. In addition, the wall taps should be exactly perpendicular and flush to the surface. Burrs or slag in the taps are not acceptable and will influence measurement accuracy.

4.1.3 Calibration

Prior to performing the field test, the transmitters or transducers should be calibrated, such that the maximum device error is less than or equal to 0.1% of the actual value. The calibration procedure should contain at least three points. Recalibration of the pressure transmitters should be performed frequently. The calibration process will not eliminate all measurement errors, since the calibration process itself is subject to non-linearities, hysteresis, and reference condition error.

4.1.4 Accuracy Achieved in Practice

The precision uncertainty in pressure measurement will depend upon the uniformity of the flow field. If piping vibration or flow-induced pulsations are high at the location of the static pressure measurement, the measurement of pressure will show a significantly higher random uncertainty. Non-uniformities, location, installation, and calibration errors will affect the pressure measurement. The signal from the transmitter should be transformed into a digital signal by means of a portable data acquisition system (DAS). The data acquisition system should have an instrumentation accuracy of better than 0.01 to 0.05% of reading.

The main source of pressure measurement error is incorrect installation and location of pressure probes. Table 6 provides typical values for sources of pressure measurement errors encountered during field tests. All values are percent full scale. For cases of multiple transmitters, it is assumed that the transmitters are installed at equal angular intervals in the pipe and the flow field is uniform. Table 6 assumes that the installation meets the upstream and downstream requirements of ASME PTC 10. Installation configurations, which do not meet ASME PTC 10, will have significantly higher location uncertainties in pressure measurement. See Figure 5 for ASME PTC 10 recommended installation.

<table>
<thead>
<tr>
<th>Sensor Type</th>
<th>Location</th>
<th>Installation</th>
<th>Calibration</th>
<th>Device</th>
<th>Acquisition</th>
</tr>
</thead>
<tbody>
<tr>
<td>One Static Transmitter</td>
<td>0.15</td>
<td>0.02</td>
<td>0.10</td>
<td>0.10</td>
<td>0.005</td>
</tr>
<tr>
<td>Two Static Transmitters</td>
<td>0.10</td>
<td>0.02</td>
<td>0.10</td>
<td>0.10</td>
<td>0.005</td>
</tr>
<tr>
<td>Four Static Transmitters</td>
<td>0.10</td>
<td>0.02</td>
<td>0.10</td>
<td>0.10</td>
<td>0.005</td>
</tr>
</tbody>
</table>

Errors in location will largely be dependent on uniformity of flow field at measuring location and the number of pressure measurement devices used at a single location. Wall static error will cause high uncertainty if wall taps are not correct. Wall taps should be exactly perpendicular to the surface and flush (with no burrs or slag).

4.2 Measurement of Temperature

4.2.1 Recommended Best Practice

The approach to measurement of temperature is similar to pressure, in that temperature may be measured very close to the compressor to assure that the measured temperature is representative of the compressor temperature. However, if the temperature measurement is taken at this location, four temperature sensors should be used to identify any inconsistent measurements and assure that bad readings are discarded. Temperature should always be measured downstream of pressure, if possible. The pressure and temperature sensors should not be installed in the same line of sight.
Thermocouples, thermistors, and resistance temperature devices (RTDs) are typically used to measure temperature. RTD’s are recommended for measurement of temperature in the flow stream over a broad temperature range. Thermocouples can be used for high temperature measurements, but below 200°F (93°C) the resolution will be reduced. Also, thermocouples tend to drift more than RTDs and, thus, require more frequent recalibration. At low temperatures, thermistors are useful but should be carefully calibrated because of inherent nonlinearities.

### 4.2.2 Installation

These devices should be inserted into a thermowell, though RTD sensors may be used as direct insert devices. Direct insert RTD’s will provide a faster response time. The temperature sensor should be instrumented to a temperature transmitter that is connected to the field test data acquisition system. The measurement location should assure that the temperature sensor will be relatively insensitive to radiation, convection, and conduction between the temperature sensor and all external bodies. The insertion depth can produce a large error in the temperature measurement, if the sensor is placed too deep or too shallow in the flow stream (see Table 7). The manufacturer safety guideline should be consulted for insertion depth of RTD’s without thermowells and extra long thermowells to ensure the pipe velocity meets acceptable safety levels.
Table 7. Recommended Depth of Thermowells

<table>
<thead>
<tr>
<th>Pipe Diameter (inches)</th>
<th>Thermowell Depth (inches)</th>
</tr>
</thead>
<tbody>
<tr>
<td>6</td>
<td>2.0</td>
</tr>
<tr>
<td>8</td>
<td>2.5-3.0</td>
</tr>
<tr>
<td>10</td>
<td>3.0-3.5</td>
</tr>
<tr>
<td>12</td>
<td>4.0</td>
</tr>
<tr>
<td>14</td>
<td>4.5-5.0</td>
</tr>
<tr>
<td>16</td>
<td>5.0-5.5</td>
</tr>
<tr>
<td>18</td>
<td>6.0</td>
</tr>
<tr>
<td>&gt;18</td>
<td>7.5 minimum</td>
</tr>
</tbody>
</table>

Note: Above 18-inch diameter, a minimum depth of 7.5 inches from the inner wall is enough to avoid pipe influence and breakage.

The ASME PTC 10 standard provides specific guidelines for proper installation and location of temperature sensors. Though the field test constraints may make ideal measurement locations impossible, it is important to be aware of the required specifications to assess measurement error and the propagation of additional measurement uncertainty (see Section 5.2 on Non-Ideal Installations).

4.2.3 Calibration

The temperature sensor shall be calibrated, such that the maximum measurement uncertainty for each sensor is ≤0.15 °C. Transmitters used in acquiring data from the temperature sensor should be calibrated in tandem (i.e., the transmitter used to read the signal from the RTD should be calibrated with the RTD as a single measurement chain). The calibration procedure should involve at least three points. The calibration process can introduce errors into the temperature measurement, primarily through non-linear response, instrument drift, cold junction, and reference temperature error.

4.2.4 Accuracy Achieved in Practice

Table 8 provides typical uncertainty values for the five main sources of temperature measurement errors encountered during field tests. Uncertainties in the temperature originate from the following five major sources of error: (i) location: incorrect position of the thermal sensor in the gas stream; (ii) installation: wall conduction heat transfer to and from the sensor due to inadequate insulation; (iii) calibration: instrument drift, nonlinearities, cold junction, and reference temperature errors; (iv) device: inherent accuracy limitations of the sensor device; and (v) acquisition: amplifier, transmission, noise, read, and analog-digital conversion errors.

Location, installation, and calibration errors may be minimized easily in production or laboratory test facilities. However, for field testing this is more difficult because time and cost constraints can force the test engineer to accept field test arrangements with improperly located, installed, and calibrated instruments. While it is often impossible to correct these problems during the short field test duration, it is imperative to recognize them and account for them in the uncertainty analysis.

Table 8 shows that the location, installation, and calibration errors are the dominant factors, while the device and acquisition errors are a smaller contribution to the total temperature error. Also note that field test device and acquisition errors are significantly larger than values quoted by instrument manufacturers (>0.005 percent full scale). Again, the circumstances and limitations encountered in the field test may not always allow for ideal handling of the sensitive measurement instruments.
Table 8 assumes that the installation meets the upstream and downstream requirements of ASME PTC 10 for temperature sensor installation. Installation configurations, which do not meet ASME PTC 10, will have significantly higher location uncertainties in temperature measurement.

### Table 8. Typical Uncertainties in Temperature Measurement
(Shown as percent of full scale)

<table>
<thead>
<tr>
<th>Sensor Type</th>
<th>Location</th>
<th>Installation</th>
<th>Calibration</th>
<th>Device</th>
<th>Acquisition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hg Thermometer</td>
<td>0.10</td>
<td>0.20</td>
<td>0.03</td>
<td>0.03</td>
<td>0.10</td>
</tr>
<tr>
<td>Thermistor</td>
<td>0.10</td>
<td>0.20</td>
<td>0.10</td>
<td>0.05</td>
<td>0.05</td>
</tr>
<tr>
<td>Thermocouple</td>
<td>0.10</td>
<td>0.20</td>
<td>0.10</td>
<td>0.10</td>
<td>0.05</td>
</tr>
<tr>
<td>RTD</td>
<td>0.10</td>
<td>0.20</td>
<td>0.05</td>
<td>0.05</td>
<td>0.05</td>
</tr>
<tr>
<td>Infrared Sensor</td>
<td>0.40</td>
<td>0.20</td>
<td>0.10</td>
<td>0.25</td>
<td>0.05</td>
</tr>
</tbody>
</table>

1 Location error is based on having four (4) equally spaced sensors in the pipe and assumes uniform flow in the pipe. For a single sensor installation, the location uncertainty should be multiplied by four (4). For two (2) sensors, the location uncertainty should be multiplied by two (2). For highly non-uniform flow fields these values may be larger.

4.2.5 Temperature and Pressure Measurement – Other Considerations

To obtain the total temperature or pressure uncertainty, the individual uncertainties must be added using the root-square sum method. Typically, the stagnation and static values for pressure and temperature are assumed to be the same, and any differences between these two values are assumed to not affect the performance calculation. This assumption is valid for inlet and outlet compressor Mach numbers less than 0.1 (Brun and Kurz, 2003.) However, at higher compressor Mach numbers (≥0.1), an additional uncertainty term should be accounted for because of the difference between stagnation and static pressure and temperatures. At a compressor Mach number of 0.10, the pressure error is approximately 0.6%, while the temperature error is 0.15%. At a compressor Mach number of 0.30, the pressure error is 7.0% and the temperature error 1.6%.

4.3 Measurement of Flow

An accurate measurement of the gas flow through a compressor is essential for proper determination of the performance and is necessary to identify degradation in the performance of a compressor. The most common meter type installed at gas compressor field sites is orifice meters. Other meters that are available at some times are full bore turbine meters, ultrasonic meters, flow nozzles, and a range of insertion type meters, such as vortex shedding meters, insertion turbine meter, and multi-port pitot probes. Fuel gas flow rates are also measured with these common meters, including orifice, turbine, insertion, and Coriolis mass flow meters. The proper sizing, installation, maintenance, adjustments, and calibrations are necessary for any of these meters to achieve the desired level of precision and repeatability in flow measurement.

4.3.1 Recommended Best Practice

Orifice, ultrasonic, and turbine flow meters are typically employed to measure pipeline flow at a high level of accuracy. However, proper installation, maintenance, and calibration are critical to achieve a desired level of precision and repeatability. All three meter types have upstream length requirements, which may be mitigated through the use of a flow conditioner.
When installed correctly with a properly calibrated differential pressure transducer, an orifice flow meter may be used to measure flow over a 3:1 range with an accuracy of 1.0%. Turbine meters have a greater flow range than orifice meters. Turbine meters are very repeatable in both high and low flow situations and can provide accuracies better than 1.0% dependent upon the quality of calibration. A calibrated ultrasonic meter will provide flow measurement accuracy better than 1.0%.

4.3.2 Installation

The upstream piping configurations at field compressor installation are normally not ideal and result in distorted velocity profiles at the meter. Non-ideal meter piping will result in errors in the flow measurement unless something is done to correct the metering configuration. The best solution is the installation of the flow conditioner either of the traditional tube bundle type with relatively long lengths of straight upstream piping per AGA Report No. 3 or the use of the recently developed perforated plate type flow conditions.

Table 9 gives the recommended distance between various upstream disturbances and the orifice flow meter based on ISO 5167. The flow conditioner required by ISO 5167 is any of the various plate type flow conditioners (instead of a 19-tube bundle).

| Required upstream length of primary device (orifice meter) – given in nominal pipe diameters |
|-----------------------------------------------|-----------------------------------------------|-----------------------------------------------|-----------------------------------------------|-----------------------------------------------|-----------------------------------------------|
| Beta Ratio | Single 90° bend or tee | Two or more 90° bends in 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 D |
| 0.20 | 10 (6) | 14 (7) | 34 (17) | 5 | 16 (8) |
| 0.30 | 10 (6) | 16 (8) | 34 (17) | 5 | 16 (8) |
| 0.40 | 14 (7) | 18 (9) | 36 (18) | 5 | 16 (8) |
| 0.50 | 14 (7) | 20 (10) | 40 (20) | 6 (5) | 18 (9) |
| 0.60 | 18 (9) | 26 (13) | 48 (24) | 9 (5) | 22 (11) |
| 0.70 | 28 (14) | 36 (18) | 62 (31) | 14 (7) | 30 (15) |

| Beta Ratio | Full bore ball or gate valve fully open | Abrupt symmetrical reduction with diameter ratio >0.5 | Thermometer pocket or well of diameter <0.03D | Thermometer pocket or well of diameter between 0.03D and 0.13D | Downstream distance - Fittings |
| 0.20 | 12 (6) | 30 (15) | 5 (3) | 20 (10) | 4 (2) |
| 0.30 | 12 (6) |  | 5 (3) | 20 (10) | 5 (2.5) |
| 0.40 | 12 (6) |  | 5 (3) | 20 (10) | 6 (3) |
| 0.50 | 12 (6) |  | 5 (3) | 20 (10) | 6 (3) |
| 0.60 | 14 (7) |  | 5 (3) | 20 (10) | 7 (3.5) |
| 0.70 | 20 (10) |  | 5 (3) | 20 (10) | 7 (3.5) |

1. Values without parenthesis are zero additional uncertainty, while values in parenthesis require an additional 0.5% uncertainty due to installation effect.
If an orifice flow meter is used, it may be beneficial to use a smaller beta ratio plate for the performance test purposes (though this will provide higher DP), because lower beta ratios are less susceptible to installation configurations and will typically provide a more accurate flow measurement.

The measurement calculations, installation requirements and calibration of a turbine meter should follow the specifications of AGA Report No. 7. The recommended installation (AGA Report No. 7, Third Revision) for a turbine meter requires at least ten pipe diameters of straight pipe upstream of the meter inlet, with a flow conditioner outlet located at five pipe diameters upstream of the meter inlet. Downstream, an additional five pipe diameters of straight pipe should be provided. Pipe connections or protrusions upstream of the meter (<10D) are not permitted. In the recommended installation, the additional uncertainty due to location or installation error is negligible.

AGA Report No. 7 allows for optional installation configurations, with a relatively higher measurement uncertainty. The short-coupled installation configuration shown in Figure 6 may be used when space is limited. The close-coupled installation configuration shown in Figure 7 may be used where space is severely limited.

![Figure 6. Short-Coupled Installation for a Turbine Meter (AGA-7, Rev. 3)](image)

![Figure 7. Close-Coupled Installation for a Turbine Meter (AGA-7, Rev. 3)](image)
Ultrasonic meters should be installed with proper flow conditioning and upstream length as stated in AGA Report No. 9. This standard ensures that swirling or distorted velocity profiles will not occur at the meter. Either extremely long, 80 to 100 pipe diameter, straight length of upstream pipe, or flow conditioners are required to prevent distorted velocity profiles from reaching an ultrasonic meter. If a flow conditioner is not available or upstream straight pipe is not sufficient, an additional bias uncertainty should be added to the flow measurement uncertainty term.

Modern perforated plate flow conditioners of the proven types are preferred and allow shorter straight upstream length than tube bundle flow conditioners. Straight upstream lengths of 20 to 30 pipe diameters are usually required with a tube bundle flow conditioner. Total upstream lengths with a proper perforated plate flow conditioner for an ultrasonic flow meter can be as little as 10 to 15 pipe diameters. Meter runs that are shorter than the guideline provided here are subject to significant installation errors.

4.3.3 Calibration

The primary advantages of orifice meters are that they do not require calibration of the meter. The differential pressure transducer used to measure DP must be calibrated, however. Orifice meters for compressor process gas flow measurement and for fuel gas metering should be inspected prior to a performance test to ensure they are clean and in good condition.

Turbine meters require calibration at near the operating condition of the meter. The characteristic non-linearity of the turbine meter calibration curve can cause measurement errors of 1 to 3% at low flows, if the non-linearity is not accounted for in the calibration. Ultrasonic meters must be individually calibrated. Every ultrasonic meter should be calibrated in a gas flow, at pressure conditions similar to the intended operating conditions. A calibration factor or linearization routine may be used to correct the as-found meter to achieving an accuracy of at least 1.0% over the meter range (typically 10:1).

4.3.4 Accuracy Achieved in Practice

Pulsation adversely affects most types of meters and, therefore, must be avoided during centrifugal compressor and gas turbine performance testing. However, as most flow measurement instruments provide a low frequency output response, it is often difficult to determine pulsation magnitudes and frequencies. RMS output variation on the flow meter can be used to estimate pulsation amplitudes, but flow turbulence also contributes to RMS flow velocity and pressure variations. AGA Report No. 3 defines that when the pressure differential or the velocity fluctuations across the measurement device exceed 10% RMS value, the flow meter results cannot be considered valid.

In order to calculate flow accurately for an orifice meter, the temperature, pressure, gas composition and differential pressure must be measured accurately. Properly sized orifice meters are suitable for testing centrifugal compressors over a normal operating range from surge to stonewall. Orifice meters are highly susceptible to installation-effects resulting from improperly conditioned flow, insufficient upstream length, upstream bends, elbows or valves, or extreme beta ratios (>0.65). If installed correctly with a beta ratio less than 0.65, orifice meters will provide a flow measurement accuracy of less than 1.5%.

Turbine meters can be calibrated to obtain a measurement uncertainty of less than 1.0%. If the gas pressure or flow rate is outside the calibration curve, the measurement will contain a bias error, which can be as large as 1.5 to 2.0% additional measurement uncertainty. If a turbine meter is over-ranged by exceeding its maximum velocity, permanent damage to the rotor may cause the measurement uncertainty of the meter to exceed 2.0%. 

Guideline for Field Testing of Gas Turbine and Centrifugal Compressor Performance
The accuracy of an ultrasonic meter decreases at flow velocities of less than 5 to 7 feet per second and at high flow velocities, above 70 to 90 feet per second. Therefore, ultrasonic meters should only be used for compressor testing, if the flow range in the pipe is between 5 to 70 ft/sec. Ultrasonic meters should not be oversized such that low flow velocities routinely occur, or undersized, such that high velocities are experienced. The field accuracy of ultrasonic meters is normally in the range of 0.5 to 1.0 % and is based on the meter’s calibration and having a suitable piping configuration.

Other differential pressure devices (annubar, v-cone, etc.) and venturi meters (sonic nozzles) may also be used to measure gas flow through the compressor. These meter types typically have a lower pressure drop than an orifice meter, but the flow measurement error is highly sensitive to the measurement of differential pressure across the device. Typical meter accuracy is 0.5 to 1.5% if the differential pressure sensor is calibrated and operating well within its range. Venturi meters and differential pressure (DP) type meters are similar to an orifice meter, in that large installation errors can occur (1 to 5%) if installed incorrectly. Installation guidelines for venturi meters are provided in ISO 5167, Measurement of Fluid Flow by Means of Orifice Plates, Nozzles and Venturi Tubes, and ASME MFC-3M-1989, Measurement of Fluid Flow in Pipes Using Orifice, Nozzle and Venturi Tubes. All differential pressure flow devices must meet a test protocol standard specified in American Petroleum Institute Manual of Petroleum Measurement Standards (API MPMS) Chapter 5.7, Testing Protocol for Differential Pressure Flow Measurement Devices. Specific installation requirements for DP type meters should be provided by the meter manufacturer and should assure that the meter conforms to API MPMS Chapter 5.7.

4.4 Measurement of Gas Composition

4.4.1 Recommended Best Practice

Both the gas turbine fuel gas and compressor process gas composition should be evaluated at regular intervals throughout the field test, either through automatic gas chromatograph sampling or by taking regular gas samples. At a minimum, a gas sample should be taken before and after the test. Multiple sampling methods are defined in GPA Standard 2166 and the American Petroleum Institute Manual of Petroleum Measurement Standards (API MPMS) Chapter 14.1. These include both spot methods and composite sampling (on-line methods).

If an automatic sampling probe is used (such as a gas chromatograph probe regulator), the probe should be properly sized to draw samples from the center one-third of the pipe, so that liquids that may appear in the flow cannot be easily ingested into the probes and sample lines. Velocities in the location where samples are taken should not exceed 150 fts/sec in order to avoid possible probe vibration and failure.

To properly determine the gas properties from the gas samples, the gas composition should be evaluated up to C₆+. Namely, as a minimum, the gas samples should be analyzed for all hydrogen, oxygen, water, sulfur compounds, carbon dioxide, nitrogen, nitrous oxides, other inert gasses, and all hydrocarbon gasses or vapors between C₁ and C₆.

If the hydrocarbon dewpoint of the gas mixture is within 0 to 11ºC of the temperature of the sampling equipment, or above the temperature of the sampling equipment (ambient temperature), additional precautions must be taken. The hydrocarbon dewpoint will vary with different gas mixtures and primarily be influenced by the presence of heavier hydrocarbons, above C₆. The best practice in this case is to preheat the sample line and sampling containers prior to taking the gas sample. If pre-heating is not practical, “dead-end” spot sampling methods may be used in combination with re-heating the sample above the mixture dew point during the analysis process.
“Dead-end” spot sampling methods include the water or glycol displacement method, the piston displacement method, the evacuated cylinder methods (evacuated, reduced pressure, or helium pop), and the fill and empty method. These methods work best because the depleted gas is not convected out of the cylinder in the sampling process. If condensate is formed on the wall of the sampling cylinder, re-heating the sample will cause the condensate to re-vaporize as part of the sampled gas mixture.

4.4.2 Installation

All sampling lines and equipment that come in contact with the sample streams should be made of stainless steel or other materials that are inert, compatible with the gas and minimize adsorption of heavy hydrocarbons from the gas stream. Polyethylene, Nylon, and Teflon will cause sample distortion because of these materials preferential absorption of specific hydrocarbon components. The probes and lines should be insulated to avoid condensation of the heavier hydrocarbon constituents or water vapor in the sample. The probes and sample lines should also be arranged above the pipeline. As stated in API MPMS Chapter 14.1, the sampling bottle should have a pigtail line connected to its outlet to assure that the process gas is kept above the hydrocarbon dewpoint – see Figure 8 below. Prior to sampling the gas, the gas sampling equipment should be cleaned, preferably by steam cleaning or using acetone or liquid propane.

Filters in the sample lines are required in most cases. All fittings, tubing, and pressure regulators should be rated for the appropriate operating pressure of the station.

![Sampling Method with Pigtail as Recommended in API MPMS Chapter 14.1](image)

**Figure 8.** Sampling Method with Pigtail as Recommended in API MPMS Chapter 14.1

4.4.3 Calibration

Gas chromatographs are almost exclusively used to determine the gas chemical composition, in order to determine gas density, compressibility, and energy content. A gas chromatograph should not be regarded as an infallible device. A calibration gas standard should be used to calibrate the gas chromatograph regularly.

4.4.4 Accuracy Achieved in Practice

The gas composition will almost always differ from the gas composition used in the baseline or factory test. Deviations will occur in the calculated performance values if only the specific gravity of the new gas is used, instead of the entire chemical gas composition. (Note: ASME PTC 10 assumes that for a Type 1
test the gas is almost identical to the gas specified at acceptance conditions.) Errors in the determination of the gas composition will affect the density, compressibility, and the energy content determination. Density errors will propagate in the flow measurement when converting between mass and volume flow.

Online gas chromatographs must be calibrated regularly (at least once per week) to ensure accurate gas composition with a calibration gas that is similar in composition to the process gas being measured.

### 4.5 Measurement of Rotational Speed

The speed of rotation should be measured using magnetic speed pickups on the power turbine or the key phasor probes of the gas compressor. Either method is acceptable and can be used with sufficient accuracy. The signal from the magnetic speed pickup or key phasor should be transformed into a digital signal using the field package data acquisition system.

### 4.6 Measurement of Torque

Determination of torque with a torque meter is recommended to reduce test uncertainty in the measured shaft power (gas turbine output power). Various torque measuring systems are available in the industry. If the torque meter is used, the total uncertainty in the gas turbine power calculation will be considerably reduced to possibly less than 1.0%. The torque meter calibration must be maintained during the entirety of the test. Practice has shown that harsh test environments and high speeds can easily affect torque meter calibration. Thus, calibration should be verified before and after the test. A torque meter can also provide a good baseline for verifying the compressor performance determined from heat balance methods.

### 4.7 Measurement of Generator Power

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 equation (4.2). The generator output can be calculated by equation (4.3).

\[
PF = \frac{P_{el,active}}{P_{el,apparent}}
\]  

\[
P_{el} = \sqrt{3}E \cdot I \cdot PF
\]

For increased accuracy, specially calibrated current transformers and potential transformers can be used.

### 5. TEST UNCERTAINTY

Test uncertainty must be calculated to determine the accuracy or quality of the test and the bounds of any measured quantity. There are two primary components to uncertainty of any physical measurement: random (precision) uncertainty and bias (fixed) uncertainty.

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.

An increased test uncertainty increases the risk of failing the test if the turbomachinery is actually performing better than the acceptance level, but reduces the risk of failing if the turbomachinery is performing below 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. The following definitions should be applied to the discussion of uncertainty:

**Precision (Random) Error:** The error due to random fluctuations of the measured quantity. The true value of the measurement should lie within the scatter of the data points, if no bias error exists. This error is reduced by taking more measurements of the test quantity.

**Bias (Fixed) Error:** A systematic deviation of an instrument’s output from a fixed input. Bias can be a complex functional form over the instrument's operational range, but in most cases, it is just the consistent over- or under-reading of input data. It is often due to installation effects or calibration errors. Bias errors must be estimated in the uncertainty analysis.

Note: In a field site test, the bias error and precision error may not be distinguishable. These two components of the uncertainty are often treated as a single combined uncertainty.

**Linearity:** Compares the deviation of a system’s output to a straight-line assumption. Clearly, few physical systems behave linearly over a wide range and, thus, linearity must always be stated with an upper and lower limit.

**Hysteresis:** Refers to the system or instrument output dependency on directionality of the input. Hysteresis has nothing to do with an instrument’s accuracy degradation over time. In most cases, it is defined as the maximum difference in instrument reading for a given input value when the value is approached first with increasing, and then with decreasing, input signals. Hysteresis is often caused by energy absorption in the elements of the measuring instrument or system.

All of the above are factors that contribute to, but are fundamentally different than the definition of measurement uncertainty. Uncertainty does not refer to a single instrument’s accuracy, but evaluates the complete range of possible test results given a singular test condition. The field test cannot be performed with all variables fixed. Consequently, the measured performance calculations and test results must also be a range rather than a point and must account for all possible input combinations of all input variables.

It is important to understand that if the input ranges to the system are defined as statistical bounds, such as 95% confidence intervals, then the output from the uncertainty analysis will also present the same 95% confidence interval statistical bounds. Similarly, if the inputs are absolute errors of measurements, then the uncertainty analysis will also yield absolute errors (i.e., whatever is the type of uncertainty range for the input variables will be the type of uncertainty range for the result). Consistent application and definitions of the input variable’s uncertainty ranges is, thus, critically important in any uncertainty analysis.
Furthermore, prior to determining a test uncertainty, it is important to know whether the measured variables in the test are independent or dependent, as this determines the method of uncertainty calculation that must be employed. For almost all real measurement scenarios, there is some physical dependency between the input variables and, thus, unless one is absolutely certain that all measured and given system inputs are independent, it is safer to opt for the more conservative assumption of measurement dependence. Thus, as the determination whether an experiment’s measured variables are interdependent directly establishes the uncertainty analysis method that must be employed, a thorough physical understanding of the measured system is imperative.

The test uncertainty calculation should be performed using one of the three methods described in Appendix C. To evaluate the test data and the gas turbine/compressor package itself, the uncertainty of the test must be calculated correctly, and the required uncertainty limits must be understood prior to the test. For example, data with an uncertainty of 3.0% cannot yield conclusions requiring an accuracy of 1.0% and, thus, if 1% accuracy is required, the test preparation, instrumentation, and planning must reflect this requirement. Also, in comparing the field test data to any other set of data of the same machine (such as historical data or the factory test data), the uncertainties in both tests must be considered in the comparison.

The test uncertainty tolerances are recommended in Table 10 for the primary measurement parameters in the performance test.

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Achievable Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure</td>
<td>0.3 - 2.0% Full Scale</td>
</tr>
<tr>
<td>Temperature</td>
<td>0.3 - 4.0 ºC (0.5-7.5 ºF)</td>
</tr>
<tr>
<td>Flow</td>
<td>0.5 - 2.0% of value</td>
</tr>
<tr>
<td>Torque</td>
<td>0.5 - 1.5% of value</td>
</tr>
<tr>
<td>Gas composition</td>
<td>0.2 – 3.0% of value</td>
</tr>
<tr>
<td>(Density, compressibility,</td>
<td></td>
</tr>
<tr>
<td>gas constant, specific</td>
<td></td>
</tr>
<tr>
<td>heat, energy content)</td>
<td></td>
</tr>
</tbody>
</table>

For all parameters derived from an equation of state (such as compressibility, heating value, isentropic coefficient, density, specific heat, and gas constants), there is an added inherent uncertainty, since the equation of state is an empirical model. Unless direct experimental data for comparison is available for the gas composition used in the performance test, it is difficult to quantify the added EOS model uncertainty. However, a consistent application of the selected EOS between the factory test, field test, and predictions will minimize any potential performance analysis differences and, thus, reduce the contribution of the EOS model uncertainty to a negligible contribution.

The effect of typical “near ideal” measurement uncertainties on the total compressor and gas turbine uncertainty is provided in Section 5.1. Non-ideal installation effects on uncertainty are provided in Section 5.2 below.

5.1 Ideal Field Test Conditions For Reducing Uncertainties

In an ideal field installation, the uncertainty in measured power and efficiency for the centrifugal compressor and gas turbine is at a minimum. Departures from the ideal installation will increase these uncertainties. Uncertainties should be calculated using the methods described in Appendix C and instrument uncertainty values listed in the preceding sections.
This section describes an example of a near ideal field test installation and provides a typical baseline uncertainty in power and efficiency for this case. The effects of non-ideal measurement conditions on the total performance uncertainties are discussed and compared in Section 5.2. In the example, uncertainty calculations given in Sections 5.1.1 and 5.1.2 and in the non-ideal installations shown in Section 5.2, the perturbation method was used to determine the total performance uncertainty, as described in Appendix C.

5.1.1 Compressor Uncertainty Example

The validity of a compressor performance field test depends on the level of uncertainty of measured efficiency and power. Power and efficiency uncertainties should be calculated from the individual measurement uncertainties (temperature, pressure, flow rate, and gas properties).

An example uncertainty calculation for a centrifugal compressor test is given in Table 11(a-b). The values of the measured variables shown in Table 11(a-b) represent a typical centrifugal compressor application in pipeline (low compression ratio) service. Operating conditions are also shown on these tables. A representative gas composition was used to compute the gas properties, consisting of the following components:

- 90.0% methane
- 5.37% ethane
- 1.7% propane
- 0.274% isobutene
- 0.331% n-butane
- 0.055% isopentane
- 0.09% n-pentane
- 0.07% n-hexane
- 1.06% carbon dioxide
- 1.05% nitrogen

The Benedict-Webb-Rubin (BWR) equation of state model was used to compute the compressibility, specific heat, and molecular weight.

The measurement uncertainties calculated in Table 11(a-b) assume near-ideal test conditions, procedures, and efficiencies. These uncertainties are based on proper installation, application, and acquisition of the test instrumentation, as recommended previously. The calculated property uncertainties \((Z, cp, k, R)\) are based on typical variations in a sampled gas composition due to sample variation and uncertainty introduced by the gas sampling process \((\Delta = +0.3\% \text{ for methane and ethane}, \Delta = +0.1\% \text{ for propane}, \Delta = -0.3\% \text{ for carbon dioxide}, \Delta = -0.4\% \text{ for nitrogen})\). The calculated property uncertainties include the uncertainty due to gas chromatograph analysis for a calibrated gas chromatograph.

Based on all the input uncertainties, the resulting uncertainty in compressor power is 1.43%. The resulting uncertainty in compressor efficiency is 2.39%. These values of measurement uncertainty for the compressor are close to the minimum attainable test uncertainty for this case.

5.1.2 Gas Turbine Uncertainty Example

The level of uncertainty in measured gas turbine efficiency and shaft output power determines the validity of the field performance test. The efficiency uncertainty of the gas turbine is typically calculated from the individual measurement uncertainties in fuel flow, fuel heating value, and gas turbine shaft power output.
The shaft output power and its associated uncertainty is usually determined from the driven equipment via torque coupling, compressor heat balance, or generator terminal power as discussed in the preceding sections.

Table 12(a-b) provides an example of a representative gas turbine in a pipeline compression application, which matches the typical centrifugal compressor example described in Table 11(a-b). The fuel flow is given as a mass flow, although this would typically be measured with a volumetric meter and converted to mass flow through density (which adds another uncertainty term). However, these terms can be combined since the density uncertainty term of the fuel is usually small. The heating value uncertainty is based on the uncertainty due to sampling variations and gas chromatograph analysis. The measurement uncertainties in fuel flow and heating value represent ideal test conditions, procedures, and efficiencies. The measured uncertainty in gas turbine shaft power output is based on the compressor power uncertainty calculated for the near-ideal case in Table 11(a-b). Based on the input uncertainties, the resulting uncertainty in gas turbine efficiency is 1.77%, which represents a minimum attainable test uncertainty for this example.

### Table 11a. Example of Total Uncertainty Calculation for Compressor in “Near Ideal” Case – SI Units

<p>| Case 1: Ideal Case |</p>
<table>
<thead>
<tr>
<th>Input Parameter</th>
<th>Value</th>
<th>Input Δ[%]</th>
<th>Δ Power</th>
<th>Δη</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measured properties:</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ps [MPa]</td>
<td>1.10</td>
<td>0.3</td>
<td>19.47</td>
<td>0.006</td>
</tr>
<tr>
<td>Ts [K]</td>
<td>300</td>
<td>0.1</td>
<td>55.17</td>
<td>0.005</td>
</tr>
<tr>
<td>Pd [MPa]</td>
<td>1.55</td>
<td>0.3</td>
<td>19.52</td>
<td>0.006</td>
</tr>
<tr>
<td>Td [K]</td>
<td>340</td>
<td>0.1</td>
<td>48.80</td>
<td>0.005</td>
</tr>
<tr>
<td>Qa [m3/s]</td>
<td>9.13</td>
<td>0.5</td>
<td>32.45</td>
<td>0.000</td>
</tr>
<tr>
<td>Calculated properties based on gas composition at suction conditions:</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Z</td>
<td>0.9763</td>
<td>0.05</td>
<td>3.25</td>
<td>0.000</td>
</tr>
<tr>
<td>cp [J/kgK]</td>
<td>2177.14</td>
<td>0.3</td>
<td>19.47</td>
<td>0.000</td>
</tr>
<tr>
<td>k</td>
<td>1.31</td>
<td>0.08</td>
<td>0.00</td>
<td>0.003</td>
</tr>
<tr>
<td>R [J/kgK]</td>
<td>460.1</td>
<td>0.3</td>
<td>19.47</td>
<td>0.000</td>
</tr>
<tr>
<td>Performance Parameter</td>
<td>Value</td>
<td>Total ΔPower [%]</td>
<td>Total Δη [%]</td>
<td></td>
</tr>
<tr>
<td>Pactual [kW]</td>
<td>6490</td>
<td>1.35</td>
<td>1.81</td>
<td></td>
</tr>
<tr>
<td>Efficiency, η [%]</td>
<td>0.636</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

1. Qa is actual flow rate at suction conditions, v=20 m/s in a 30” diameter pipe.
2. Z, cp, k and R represent a typical hydrocarbon transmission grade gas.
3. Typical values of uncertainty represent ideal pressure and temperature measurement using four sensors on suction and discharge, recommended flow meter installation and gas property calculation made with consistent EOS model and accurate gas sample.
Table 11b. Example of Total Uncertainty Calculation for Compressor in “Near Ideal” Case – English Units

<table>
<thead>
<tr>
<th>Case 1: Ideal Case</th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Input Parameter</td>
<td>Value</td>
<td>Input $\Delta$[%]</td>
<td>$\Delta$ Power</td>
</tr>
<tr>
<td>Measured properties:</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ps [psia]</td>
<td>159.54</td>
<td>0.3</td>
<td>26.11</td>
</tr>
<tr>
<td>Ts [R]</td>
<td>540</td>
<td>0.1</td>
<td>73.97</td>
</tr>
<tr>
<td>Pd [psia]</td>
<td>224.80</td>
<td>0.3</td>
<td>26.18</td>
</tr>
<tr>
<td>Td [R]</td>
<td>612</td>
<td>0.1</td>
<td>65.44</td>
</tr>
<tr>
<td>Qa [ft³/s]¹</td>
<td>322.42</td>
<td>0.5</td>
<td>43.51</td>
</tr>
<tr>
<td>Calculated properties based on gas composition at suction conditions:</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Z</td>
<td>0.9763</td>
<td>0.05</td>
<td>4.35</td>
</tr>
<tr>
<td>cp [Btu/lbmR]²</td>
<td>1.69</td>
<td>0.3</td>
<td>26.11</td>
</tr>
<tr>
<td>k</td>
<td>1.31</td>
<td>0.08</td>
<td>0.00</td>
</tr>
<tr>
<td>R [Btu/lbmR]</td>
<td>0.3563</td>
<td>0.3</td>
<td>26.11</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Performance Parameter</th>
<th>Value</th>
<th>$\Delta$ Power [%]</th>
<th>Total $\Delta$ $\eta$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pactual [hp]</td>
<td>8704</td>
<td>1.35</td>
<td>1.81</td>
</tr>
<tr>
<td>Efficiency, $\eta$ [%]</td>
<td>0.636</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

¹ Qa is actual flow rate at suction conditions, $v$=65.6 ft/s in a 30" diameter pipe.
² Z, cp, k and R represent a typical hydrocarbon transmission grade gas.
³ Typical values of uncertainty represent ideal pressure and temperature measurement using four sensors on suction and discharge, recommended flow meter installation and gas property calculation made with consistent EOS model and accurate gas sample.

Table 12a. Ideal Installation for Gas Turbine – Total Uncertainty Calculation – SI Units

<table>
<thead>
<tr>
<th>Case 1: Ideal Case</th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Input Parameter</td>
<td>Value</td>
<td>Input $\Delta$[%]</td>
<td>$\Delta$ Power-in</td>
</tr>
<tr>
<td>Mf, Fuel Flow [kg/s]¹</td>
<td>0.60</td>
<td>1</td>
<td>300</td>
</tr>
<tr>
<td>HV, Heating Value [kJ/kg]</td>
<td>50000</td>
<td>0.3</td>
<td>90</td>
</tr>
<tr>
<td>Power-out [kW]²</td>
<td>6223</td>
<td>1.35</td>
<td>0</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Performance Parameter</th>
<th>Value</th>
<th>Total $\Delta$ Power [%]</th>
<th>Total $\Delta$ $\eta$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power-in [kW]</td>
<td>30000</td>
<td>1.04</td>
<td>1.71</td>
</tr>
<tr>
<td>Turbine Efficiency</td>
<td>0.207</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

¹ Fuel flow is given as a mass flow for typical natural gas.
² Power out is delivered power of turbine = absorbed power for centrifugal compressor.
³ Input fuel flow is typical for volumetric meter with density conversion or orifice meter.
Input heating value is typical due to gas sampling / gas chromatograph.
Output power uncertainty based on calculated power uncertainty of compressor for ideal case.
Table 12b. Ideal Installation for Gas Turbine – Total Uncertainty Calculation – English Units

<table>
<thead>
<tr>
<th>Case 1: Ideal Case</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Input Parameter</strong></td>
</tr>
<tr>
<td>Mf, Fuel Flow [lbm/s] ¹</td>
</tr>
<tr>
<td>HV, Heating Value [Btu/lbm]</td>
</tr>
<tr>
<td>Power-out [hp] ²</td>
</tr>
<tr>
<td><strong>Performance Parameter</strong></td>
</tr>
<tr>
<td>Power-in [hp]</td>
</tr>
<tr>
<td>Turbine Efficiency</td>
</tr>
</tbody>
</table>

¹ Fuel flow is given as a mass flow for typical natural gas.
² Power out is delivered power of turbine = absorbed power for centrifugal compressor.
³ Input fuel flow is typical for volumetric meter with density conversion or orifice meter.

Output power uncertainty based on calculated power uncertainty of compressor for ideal case.

5.2 Effects of Non-Ideal Installations on Uncertainty

Deviations in the ideal test conditions or procedures (as recommended previously) will increase the individual measurement uncertainties and result in a higher total performance measurement uncertainty for the compressor and gas turbine. Depending upon the effect of the non-ideal installation on the measurement, the resulting increase in uncertainty can range from a small increase of 0.20% in some cases to above 5.0%, in either efficiency or power. Some typical non-ideal effects are listed in Tables 13 through 15. The resulting increase in total uncertainty is shown in these tables, based upon the example test case described in Section 5.1.

Table 13 illustrates the effect of non-ideal temperature and pressure measurements for the previously described pipeline compression example. The increase in uncertainty in the temperature or pressure measurement is primarily due to non-uniformities in the compressor upstream or downstream flow field when less than four sensors are used. For a uniform flow field where the upstream and downstream straight pipe length meets the installation requirements of ASME PTC 10, the increase in temperature or pressure measurement uncertainty is usually not as severe as shown in Table 13. However, conservative “worst case” assumptions should be used for uncertainty calculations.

Other frequently encountered non-ideal effects in compressor testing are listed in Table 14. Table 15 provides some examples of non-ideal effects that could occur in the gas turbine performance test for the subject pipeline compression example. The common departures from an ideal test are principally due to field test installations with insufficient upstream length involving a single elbow or double elbows directly upstream of the flow measurement point or temperature/pressure sensor. In the non-ideal cases where upstream disturbances affected the flow measurement, an additional uncertainty was assumed for the flow meter based on published literature. These tables give the increase in uncertainty for each measured property and the additional compressor uncertainty, for the typical baseline case presented in Table 11(a-b).
Table 13. Effect of Non-Ideal Temperature or Pressure Measurement

<table>
<thead>
<tr>
<th>No. of RTD's / Pressure Sensors</th>
<th>4</th>
<th>3</th>
<th>2</th>
<th>1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Suction Temperature (Ts)</td>
<td>0</td>
<td>0.64</td>
<td>1.41</td>
<td>2.21</td>
</tr>
<tr>
<td>Discharge Temperature (Td)</td>
<td>0</td>
<td>0.52</td>
<td>1.17</td>
<td>1.86</td>
</tr>
<tr>
<td>Suction Pressure (Ps)</td>
<td>0</td>
<td>0.02</td>
<td>0.04</td>
<td>0.07</td>
</tr>
<tr>
<td>Discharge Pressure (Pd)</td>
<td>0</td>
<td>0.02</td>
<td>0.04</td>
<td>0.07</td>
</tr>
</tbody>
</table>

Increase in Compressor Efficiency Uncertainty
(Additional uncertainty above baseline Δη = 1.81%)

<table>
<thead>
<tr>
<th>No. of RTD's / Pressure Sensors</th>
<th>4</th>
<th>3</th>
<th>2</th>
<th>1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Suction Temperature (Ts)</td>
<td>0</td>
<td>0.53</td>
<td>1.2</td>
<td>1.95</td>
</tr>
<tr>
<td>Discharge Temperature (Td)</td>
<td>0</td>
<td>0.53</td>
<td>1.2</td>
<td>1.95</td>
</tr>
<tr>
<td>Suction Pressure (Ps)</td>
<td>0</td>
<td>0.14</td>
<td>0.29</td>
<td>0.46</td>
</tr>
<tr>
<td>Discharge Pressure (Pd)</td>
<td>0</td>
<td>0.14</td>
<td>0.29</td>
<td>0.46</td>
</tr>
</tbody>
</table>

Table 14. Non-Ideal Installations Effect on Compressor Uncertainty.

<table>
<thead>
<tr>
<th>Non-Ideal Installation</th>
<th>Total ΔPower (%)</th>
<th>Deviation from Baseline ΔPower</th>
<th>Total Δη (%)</th>
<th>Deviation from Baseline Δη</th>
</tr>
</thead>
<tbody>
<tr>
<td>Single elbow 1-3 pipe diameters upstream of flow measurement point (no flow conditioner)</td>
<td>1.73</td>
<td>0.38</td>
<td>1.81</td>
<td>0</td>
</tr>
<tr>
<td>Double elbows in-plane, second elbow 1-3 pipe diameters upstream of flow measurement point (no flow conditioner)</td>
<td>1.95</td>
<td>0.60</td>
<td>1.81</td>
<td>0</td>
</tr>
<tr>
<td>Double elbows out of plane, second elbow 1-3 pipe diameters upstream of flow measurement point (no flow conditioner)</td>
<td>2.79</td>
<td>1.44</td>
<td>1.81</td>
<td>0</td>
</tr>
<tr>
<td>Partially closed gate or ball valve within 1-3 pipe diameters upstream of flow measurement point (no flow conditioner)</td>
<td>3.72</td>
<td>2.37</td>
<td>1.81</td>
<td>0</td>
</tr>
<tr>
<td>Single elbow within 1-3 pipe diameters of temperature and pressure sensors</td>
<td>2.39</td>
<td>1.04</td>
<td>2.85</td>
<td>1.04</td>
</tr>
<tr>
<td>Double elbows in-plane, second elbow within 1-3 pipe diameters of temperature and pressure sensors</td>
<td>3.19</td>
<td>1.84</td>
<td>3.66</td>
<td>1.85</td>
</tr>
<tr>
<td>Double elbows out of plane, second elbow within 1-3 pipe diameters of temperature and pressure sensors</td>
<td>4.84</td>
<td>3.49</td>
<td>5.37</td>
<td>3.56</td>
</tr>
<tr>
<td>RTD not fully inserted into flow stream</td>
<td>2.37</td>
<td>1.02</td>
<td>2.66</td>
<td>0.85</td>
</tr>
<tr>
<td>Use of thermocouple instead of RTD</td>
<td>3.15</td>
<td>1.8</td>
<td>3.38</td>
<td>1.57</td>
</tr>
<tr>
<td>Error in gas sampling (no pigtail used) with heavy hydrocarbon gas</td>
<td>1.38</td>
<td>0.03</td>
<td>2.01</td>
<td>0.2</td>
</tr>
</tbody>
</table>
Table 15. Non-Ideal Installations Effect on Gas Turbine Uncertainty.

<table>
<thead>
<tr>
<th>Non-Ideal Installation</th>
<th>Total ΔPower-in (%)</th>
<th>Deviation from Baseline ΔPower-in (%)</th>
<th>Total Δη (%)</th>
<th>Deviation from Baseline Δη (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Single elbow 1-3 pipe diameters upstream of fuel gas flow meter (no flow conditioner)</td>
<td>1.4</td>
<td>0.4</td>
<td>2.0</td>
<td>0.3</td>
</tr>
<tr>
<td>Double elbows in-plane, second elbow 1-3 pipe diameters upstream of fuel gas flow meter (no flow conditioner)</td>
<td>1.7</td>
<td>0.7</td>
<td>2.2</td>
<td>0.5</td>
</tr>
<tr>
<td>Double elbows out of plane, second elbow 1-3 pipe diameters upstream of fuel gas flow meter (no flow conditioner)</td>
<td>2.7</td>
<td>1.7</td>
<td>3.0</td>
<td>1.3</td>
</tr>
<tr>
<td>Partially closed gate or ball valve within 1-3 pipe diameters upstream of fuel gas flow meter (no flow conditioner)</td>
<td>3.7</td>
<td>2.7</td>
<td>4.0</td>
<td>2.2</td>
</tr>
<tr>
<td>Error in fuel gas sampling procedure or gas composition analysis of 1.0%-3.0% in density</td>
<td>1.5-3.4</td>
<td>0.4-2.3</td>
<td>2.0-3.6</td>
<td>0.3-1.9</td>
</tr>
<tr>
<td>Error in Power-out of 2.0-5.0%</td>
<td>1.0</td>
<td>0.0</td>
<td>3.5-6.4</td>
<td>1.8-4.7</td>
</tr>
</tbody>
</table>

Other non-ideal effects typically seen at a field site include non-recommended gas sampling procedures, flow meter errors, or temperature sensor installation errors. Note that an error in the gas sample or gas composition will affect the gas properties uncertainties used in calculating compressor power (as shown in Table 11(a-b)) but may also affect the flow measurement uncertainty, if the density is used to convert mass flow to volumetric flow. As Tables 13 through 15 illustrate, non-ideal installations will affect the overall performance measurement uncertainty significantly.

6. INTERPRETATION OF TEST DATA

6.1 Data Reduction and Checking Uncertainties

Averaging of temperatures, pressures, and flows from different test points taken at different points in time should be avoided as the non-linear effects will not be averaged correctly. Instead, an average of the resulting performance parameter (flow coefficient, head coefficient, efficiency, etc.) should be calculated once the performance results have been computed for each data point. Data reduction procedures should be aimed at minimizing time and cost in the determination of performance.

If the test data from the field test deviate more than the level of test uncertainty, the source of the deviation should be explored further. Repeating the performance test is not recommended by this guideline unless the data is clearly un-usable. Data taken during a field test should be corrected to common datum conditions.

For the gas compressor, Mach number and volume/flow ratio differences can cause considerable deviation in the overall performance of the compressor. The prediction procedure should be repeated for actual conditions. If the head versus flow curve has shifted horizontally, the flow may have been measured incorrectly or contain a bias error. If some of the points on the curve match predictions and others do not, the gas composition (or another influential parameter) may not have been stable during the test. A shift towards lower efficiency or head is often caused by worn compressor seals.

For the gas turbine, separate calculations of power can be used to check the data. The power calculated using the gas compressor power should be checked against the predicted power from the initial factory test data.
6.2 Generation of Performance Curve from Recorded Data Points

Each data point recorded during the field test should be evaluated individually. The average of all data points at a particular condition should be used to compute the average head, flow, and efficiency. Two methods may be applied to the data reduction procedure in order to determine the centrifugal compressor performance curve.

The first method uses linear interpolation between the individually measured isentropic head data points and measured efficiencies. These two interpolations provide a curve for the as-tested efficiency and isentropic head. The absorbed power is calculated based on the as-tested isentropic head and efficiency curves. At the flow rate conditions of interest, the predicted isentropic head, efficiency, and power may be determined based on the field test data points.

The second method relies on the non-dimensional head and flow coefficients. The test points are plotted for the head coefficient versus flow coefficient map and the flow coefficient versus efficiency map. A data fit curve is determined (linear or polynomial). At the flow coefficient values of interest, the corresponding point on the curve fits for efficiency and head coefficient is found. The speed needed to meet the head coefficient and flow coefficient values is obtained. The tested absorbed power should be calculated according to the mechanical efficiency, the mass flow at the flow point of interest, the head at the flow point of interest, and the determined test efficiency:

\[ P_{\text{test}} = \eta_{\text{test}} W_i \frac{H_i}{\eta_{\text{test}}} \]  

(6.1)

6.3 Standardized Uncertainty Limits

ASME PTC 10 provides uncertainties as a guideline for acceptable uncertainty limits in factory testing. However, these uncertainty limits are generally not realistic in a field setting and should, therefore, not be utilized. The uncertainty calculation method described in this guideline should be employed to calculate the actual field testing uncertainty, with the goal of minimizing these uncertainties whenever possible.

6.4 Using Redundancy to Check Test Measurement and Uncertainty

Redundant calculations should be performed, if possible, to check test measurements. The gas turbine driver full load performance should be known based on factory testing. The compressor shaft power measurement during the field test should match this value, if the engine can be run at full load output during the field test. Similarly, the gas turbine fuel flow measurement may be checked against the heat rate. Use of the indirect measurements for gas turbine power input to the compressor may be used to check the direct measurement of shaft power or generator power, whichever is used in the field test.

The calculated values or factory test (manufacturer supplied) values should match the measured values within the associated uncertainty for both values. The uncertainty on any measured value obtained in the field test should be calculated (or estimated as accurately as possible). This uncertainty will be a plus or minus value. It should overlap with the uncertainty band (also a plus or minus) on the calculated/factory test value. This analysis will determine if the redundant measurement is statistically equal to the measured value. This method of comparison should always be used in order to practically determine if measured values are correct.
6.5  Effects of Fouling on Test Results

The gas turbine should be thoroughly cleaned prior to a field test. Fouled compressor blades can cause deviations in predicted gas turbine power of more than 3%. Like any prime mover, a gas turbine is susceptible to the effects of wear and tear. The problem is in predicting the effects of degradation on the engine after many hours of operation. Power and efficiency (and related speed and firing temperature) are the evident indicators of fouling or degradation of the gas turbine.

On driven equipment, such as the centrifugal compressor, the increased clearance in labyrinth seals, balance piston seals and shroud seals will cause the absorbed power to increase. On high-pressure ratio machines with low flow, worn balance piston seals will cause a significant increase in power because of the increased balance piston recirculation caused by the seals (Kurz and Brun, 2001).

6.6  Analysis of Measured Results

The true value of the compressor performance parameters (head, efficiency, and power) and the gas turbine performance lies within the measured data points, assuming the data has been recorded correctly without a significant bias error. The measured data points should be viewed as a representation of the bracket surrounding the true value. If the measured head and measured flow data points are plotted on an x- and y-axis, an uncertainty band on each measured head and measured flow exists. An ellipse surrounds the measured data point (see Figure 9). The predicted performance test point or manufacturer factory test curve may lie within the uncertainty band produced from the field test, though the exact values from the field test do not exactly match the manufacturer suggested curve. This example shown in Figure 9 shows good performance of the compressor during the field test. The compressor performance in the field test is statistically equal to the factory test in this example (Kurz and Brun, 2005).

![Figure 9. Example of Test Uncertainty Range](image-url)

7.  OTHER FIELD TESTING CONSIDERATIONS

The additional recommendations included in this section highlight the importance of understanding influential test parameters and provide considerations for testing under wet gas conditions.
7.1 Determination of Influential Test Parameters

The determination of influential test parameters should be used to assess the measurement parameters in order to improve the accuracy of the field test. The uncertainty analysis can be used to determine the influential test parameters. The terms in the total uncertainty equation can be compared at different operating conditions. If the comparison reveals that a certain term becomes more significant to the overall uncertainty, then extra efforts to improve this measurement will be worthwhile.

7.2 Field Testing of Compressor Under Wet Gas Conditions

Wet gas conditions will negatively influence the accuracy of the test, as well as the correct comparison of the test data with data gathered with dry gas. Test results may significantly deviate from predictions, if the test gas is not completely dry.

Relative to a dry compressor, the field test with wet gas conditions will result in a higher-pressure ratio because of the increased gas-volume fraction. The increased pressure ratio is a result of heavier gas. A decreased temperature ratio is to be expected as well because of the transfer of energy from the gas to the liquid phase and the limited condensate phase transition (Brenne et al., 2005.)

The use of the polytropic head coefficient and the polytropic efficiency is recommended in dealing with wet gas conditions. The specific volume of gas-liquid mixture may be treated as a combined homogenous fluid or a two fluid model. Either approach is valid, though the phase transition contribution is assumed to be negligible. For multi-stage compressors with a high-pressure ratio, this assumption may not be valid. Similarity variables for the compressor will need to be compensated by the correct two-phase terms. The modified thermodynamic equations for the two-phase head and flow coefficients are given in Appendix D.

8. REFERENCES


Brown, R. N. “Fan Laws, the Use and Limits in Predicting Centrifugal Compressor Off Design Performance,” Proceedings of the Twentieth Turbomachinery Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, 1991.


APPENDIX A

METHODS OF CALCULATING GAS TURBINE POWER
APPENDIX A

If a field test is conducted using the same standards as for a factory test, the field testing of the gas turbine will typically yield higher test uncertainties than the factory test. The main reason for this lies in the methodology of measuring the input turbine power. In a factory, shaft power is measured by running the gas turbine against a generator, a dynamometer, or a water brake. The power turbine applies torque directly to the generator, dynamometer, or water brake. The absorbed power is accurately measured with a load cell or other method.

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

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

Field tests with a torque metering coupling (method 1) or a driven generator can achieve accuracies similar to factory test methods. The generator electrical power output (method 2) can be measured directly at the generator terminals with the proper instrumentation. These two methods are the most accurate as they are direct measurements. Using the power input into the driven compressor to determine gas turbine power (method 3) is subject to much higher measuring uncertainties, because of the indirect method of measurement.

Calculation of gas turbine shaft power using method 3 requires the aerodynamic power of the compressor, the mechanical efficiency, and any losses in the gearbox (if present). The gas turbine power output is then determined from the following:

\[ P = \frac{P_g}{\eta_m} + P_{GB} \]  

(A.1)

Where:  
- \( P_g \) = aerodynamic or gas power delivered by the process compressor  
- \( \eta_m \) = mechanical efficiency of the compressor, typically given by the manufacturer as 98 to 99% of the absorbed power of the compressor (\( P_C \))  
- \( P_{GB} \) = losses in the gearbox

The fourth method takes advantage of the conservation of energy in a thermodynamic system, requiring the energy flowing into the system to be balanced by the energy leaving the system:

\[ w_i h_i + w_f \times LNV \times \eta_{comb} + w_j h_j = (w_i + w_f) \times h_1 + P + E_r + E_m \]  

(A.2)

The mass flow and enthalpy of the air at the gas turbine inlet (\( w_i h_i \)), as well as the fuel flow and fuel enthalpy (\( w_f h_f \)), 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 (\( \eta_{comb} \)) can be
estimated as well and is typically about 99% or better. Therefore, the shaft power of the turbine ($P$) 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.
APPENDIX B

EQUATIONS OF STATE
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APPENDIX B

The following explanation of the equation of state models is taken from *Equations of State for Gas Compressor Design and Testing* by Kumar, Kurz and O’Connell, 2003.

While the operating conditions for gas compressors are typically defined in terms of pressures, temperatures, and mass or standard flows, the relevant data that describe the behavior of a compressor are the head \( H \), which is related to the work input, the volumetric flow \( Q \) and efficiency \( \eta \), which compares the real process to an isentropic process between the same inlet state and outlet pressure.

The head, or specific enthalpy difference between two states (e.g., inlet and discharge side of the compressor), is defined by:

\[
H = h(p_2, T_2, \{y\}) - h(p_1, T_1, \{y\}) \tag{B.1}
\]

The enthalpy \( h \) is a function of pressure, temperature, and gas composition defined through a set of mole fractions \( \{y\} \). The actual absorbed power \( P_{\text{gas}} \) involves the mass flow rate \( W \):

\[
P_{\text{gas}} = WH \tag{B.2}
\]

The mass flow rate is obtained from the actual or volumetric flow rate \( Q \) and the gas density \( \rho \):

\[
W = \rho Q \tag{B.3}
\]

\( \rho \) with the compressibility factor \( Z \).

When \( Z \) differs from unity, the gas is not ideal and its value is a:

\[
\rho = p/ZRT \tag{B.4}
\]

function of \( T, p, \) and gas composition.

The result of these definitions is that \( P_{\text{gas}} \) is found from:

\[
P_{\text{gas}} = \rho QH = \frac{p}{ZRT} QH \tag{B.5}
\]

In order to define the quality of the compression process, \( H \) is usually compared to the head for an ideal compression process, which is defined as compression between the same inlet \( T_1 \) and \( p_1 \) and outlet \( p_2 \), with the outlet temperature being fictitious \( T_{2s} \):

\[
\Delta s = s(p_2, T_{2s}, \{y\}) - s(p_1, T_1, \{y\}) = 0 \tag{B.6}
\]

This isentropic change of state defines an isentropic head, \( H_s \), such that:

\[
H_s = h(p_2, T_{2s}, \{y\}) - h(p_1, T_1, \{y\}) \tag{B.7}
\]

The quality or efficiency of the compression is defined by:
Compressor characteristics, in terms of head versus flow and efficiency versus flow, are found by comparing test data, taken with test gases such as Nitrogen, with results obtained from the thermodynamic calculations above. The characteristics can later be used to calculate the performance of the compressor under arbitrary conditions of pressures, temperatures, and gas compositions. As long as the same EOS is used for obtaining compressor performance predictions and data reduction, errors are minimized.

An EOS is a relation among variables of a fully specified system: $T$, $p$, $\rho$ and the N-1 component mole fractions $y_i$ (Alberty and Silbey, 1997). This is usually expressed in the form:

$$Z = Z(\rho, T, \{y_i\})$$  \hspace{0.5cm} (B.9)

since in a multiphase region, multiple values of $\rho$ give the same value of $p$.

Thermodynamics gives rigorous relations for enthalpy and entropy differences from derivatives and integrals of $Z$ from any EOS and ideal gas specific heat, $C_p^0$. A gas is said to be in a specified state if it has zero degrees of freedom. The degrees of freedom are the number of properties that can be arbitrarily set before all other properties become specified. The formula for the degrees of freedom of $N$ non-reacting gases is:

$$DF = N - \#phases + 2$$  \hspace{0.5cm} (B.10)

In gas compressor design calculations, only one phase exists and the gas composition is usually specified, so two more degrees of freedom must be chosen. Generally, $p$ and $T$ are specified and the number of phases is always one. Then, all other thermodynamic properties are fixed and calculated via an EOS. Since real gas behavior commonly plays a role in gas compressors, knowledge of the relationships between pressures and temperatures, on one hand, and enthalpies, entropies and densities, on the other hand, is of great importance in compressor design, their performance under arbitrary operating conditions, and test data reduction. Especially during gas compressor performance tests, the selection of a particular EOS can have an important effect on the apparent efficiency and absorbed gas power.

**Thermodynamic Approach**

In order to decide on the most appropriate (EOS) to be used for designing and testing gas compressors for natural gas applications, five frequently applied EOS were studied: original Redlich-Kwong, Redlich-Kwong-Soave, Peng-Robinson (Reid et al., 1986), Lee-Kesler-Ploecker (Ploecker et al., 1978) and Starling version of the Benedict-Webb-Rubin model (Starling, 1973).

The variation in entropy or enthalpy between two states of a gas or mixture, each defined by a temperature and pressure, is independent from the path chosen from one state to the other (Reid et al., 1986). A convenient path involving three steps of changing the real gas to an ideal gas at $T_1$, changing the ideal gas from $T_1$ to $T_2$ and changing the ideal gas back to the real gas at $T_2$ (Figure B-1).
Figure B-1. Calculation Path for Equations of State

\[ h = f(p,T) \]
\[ dh = \left( \frac{\partial h}{\partial p} \right)_T dp + \left( \frac{\partial h}{\partial T} \right)_p dT \]
\[ h_2 - h_1 = \int_{p_1}^{p_2} \left( \frac{\partial h}{\partial p} \right)_T dp + \int_{T_1}^{T_2} \left( \frac{\partial h}{\partial T} \right)_p dT \]
\[ h_2 - h_1 = \left( h_0 - h_{p_1} \right)_{T_1} + \int_{T_1}^{T_2} C_p^0 dT - \left( h_0 - h_{p_2} \right)_{T_2} \]

The terms in the parentheses of equation (B.11) are called departure functions, real gas contributions, or residual properties, which relate the enthalpy at some \( p \) and \( T \) to that at an ideal gas reference state at \( T, H^0 \). These departure functions can be calculated solely from the EOS. The same approach can be used for the entropy.

The ideal gas law is based on the assumption that the molecules of the gas do not interact with each other or that there is no attractive or repulsive forces between two molecules. The heat capacity of a gas is the amount of energy, which the gas needs to absorb before its temperature increases one unit. For an ideal gas, the heat capacity \( C_p^0 \) is a function only of \( T \). An empirical equation for the ideal gas heat capacity can be stated as a polynomial, e.g., third order polynomial:

\[ C_p^0 = A + BT + CT^2 + DT^3 \]

\( A, B, C, \) and \( D \) are empirical parameters or constants based on the type of gas being analyzed. Once an equation for \( C_p^0 \) is found, the ideal gas enthalpy change, which is the change in total energy in the gas as it goes from state one to state two, can be found by:
Even for an ideal gas, the entropy change depends upon the initial and final temperatures and pressures. The entropy change is expressed by:

\[
\Delta S^0 = \int_{T_1}^{T_2} \frac{C_p}{T} dT - R \ln \left( \frac{P_2}{P_1} \right)
\]

(B.14)

When calculating the enthalpy or entropy of a given state, an arbitrary reference state must be selected whose enthalpy and entropy are set to zero. The enthalpy and entropy for a given state is calculated relative to this reference. Therefore, any absolute value of the enthalpy or entropy of a gas at a given state has no real meaning, given its dependence on the reference state. However, when the enthalpy difference between two states is calculated, the reference state cancels out, so an enthalpy or entropy difference is an actual value that does not depend on the reference state.

**Functionality of Equations of State**

The departure functions for enthalpy and entropy for each of the five EOS can be found in the literature (Reid et al., 1986; Peng and Robinson, 1976; Ploecker et al., 1978; and Starling, 1973). Herein, the RK, RKS, and PR EOS are referred to as cubics.

In equation (B.15), \( Z \) represents the compressibility factor of the gas, defined as:

\[
Z = \frac{pv}{RT}
\]

(B.15)

The quantities \( X \) and \( Y \) are two other types of compressibility factors used in compressor design. The formulas for each:

\[
Y = 1 - \frac{p}{Z} \left( \frac{\partial Z}{\partial p} \right)_T
\]

(B.16)

\[
X = \frac{T}{Z} \times \frac{\partial Z}{\partial T}
\]

(B.17)

The calculation of the molecular weight and the heat capacity at given temperatures of the gas mixture is completed by using the following mixing rules:

\[
\tilde{MW} = \sum y_i MW_i
\]

\[
\tilde{C}_p = \sum y_i C_{P_i}
\]

(B.18)

\( MW_i \) and \( y \) are the molecular weight and mole fraction of each component in the mixture. The heat capacities are divided by \( R \) to make them dimensionless, so when the linear function is found at a given temperature, the result should be multiplied by \( R \) in the desired units to get the heat capacity in those units.
The linear function for ideal \( \frac{C_p^0}{R} \) is calculated using the \( \frac{C_p^0}{R} \) values at 10 and 149°C (50 and 300°F). These two points are used to find the slope of a straight line on a \( \frac{C_p^0}{R} \) versus temperature plot. This slope is used to solve for the \( y \) intercept of the following simple linear equation:

\[
\frac{C_p^0}{R} = CT + B \tag{B.19}
\]

Finally, the specific gravity (\( SG \)) and the real gas parameter (\( RG \)) are calculated. \( SG \) is calculated relative to the molecular weight of air:

\[
SG = \frac{\tilde{MW}}{28.964} \tag{B.20}
\]

The \( RG \) parameter is given by:

\[
RG = \frac{0.287kJ/kgK}{SG} \tag{B.21}
\]

The Redlich-Kwong and Peng-Robinson models are cubic equations of state. The LKP equation is like the BWRS, a modification of the original BWR EOS. The LKP EOS has mixing rules that are very different from the cubics. The Starling version (BWRS) of the original BWR EOS added three extra parameters for improving the temperature dependence of the eight parameter forms. These parameters must be found for each pure gas. There also are mixing rules for the 11 parameters (Starling, 1973).

For the cubic EOS, an analytical method can be used to solve for the three roots of \( \rho \), thus, yielding \( Z \). There are three roots to any cubic equation; however, when \( Tr >1 \) only the largest real root has any physical significance. After \( Z \) is calculated, the \( X \) and \( Y \) compressibility factors, along with specific heat, are calculated. For the LKP and BWRS models, \( Z \) is found by an iterative method.
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APPENDIX C

UNCERTAINTY ANALYSIS OF INDEPENDENT VARIABLE MEASUREMENTS
Prior to determining a test uncertainty, it is important to know whether the measured variables in the test are independent or dependent as this determines the method of uncertainty calculation that must be employed. For almost all real measurement scenarios, there is some physical dependency between the input variables and, thus, unless one is absolutely certain that all measured and given system inputs are independent, it is safer to opt for the more conservative assumption of measurement dependence. If the measured variables in an experiment are truly found to be independent, then the method to determine total uncertainty is simply an addition of all individual measurement uncertainties. This is mathematically expressed as:

\[ \Delta F = \left| \Delta x_1 \right| + \left| \Delta x_2 \right| + \left| \Delta x_3 \right| + ... + \left| \Delta x_n \right| \]

where \( \Delta F \) is the total result uncertainty and \( \Delta x \) are the individual measurement uncertainty ranges.

This is the absolute value method rather than square root of the sum of squares method, which is more commonly utilized (as shown in C.2). The absolute value addition presents a true superposition of individual uncertainties rather than a blended sum. This method yields more conservative uncertainty results than the square-root-sum method, but both approaches are generally acceptable for uncertainty analyses.

\[ \Delta V = \sqrt{\Delta x_1^2 + \Delta x_2^2 + \Delta x_3^2 + ... + \Delta x_n^2} \]

**Uncertainty Analysis for Independent Variable Measurements**

If the measured variables in an experiment are dependent, which is usually the case, then the analysis becomes more complex. Specifically, the individual measurement uncertainties, \( \Delta x \), are now functionally related and this must be accounted for in the analysis. There are three methods that are commonly used by modern engineers for this type of uncertainty analysis:

- Partial Derivative Method
- Coefficient Method
- Perturbation Method

All three methods are based on a functional transfer from input to output variable but employ a different approach to the determination of the proper transfer function.

**The Partial Derivative Method**

The most traditional method for uncertainty calculations is based on determining the transfer function using a partial derivative and adding the individual uncertainty transformations. Namely,

\[ \Delta F = \Delta F(\Delta x_1) + \Delta F(\Delta x_2) + \Delta F(\Delta x_3) + ... + \Delta F(\Delta x_n) \]

\[ \begin{align*} 
\Delta F &= \left| \Delta x_1 \cdot \frac{\partial F}{\partial x_1} \right| + \left| \Delta x_2 \cdot \frac{\partial F}{\partial x_2} \right| + \left| \Delta x_3 \cdot \frac{\partial F}{\partial x_3} \right| + ... + \left| \Delta x_n \cdot \frac{\partial F}{\partial x_n} \right| \\
&= \sum_{i=1}^{n} \left| \Delta x_i \cdot \frac{\partial F}{\partial x_i} \right| 
\end{align*} \]
To understand this method, one must analyze the principal term $\Delta x_i \cdot \frac{\partial F}{\partial x_i}$ and derive some basic physical understanding. Figure C-1 shows a graphical interpretation of the functional transformation from range $\Delta x$ to $\Delta F$ using this term. Effectively, the input range $\Delta x$ is multiplied by the slope of the function $F$ at the measurement point $x_1, x_2, x_3, \text{etc.}$, to determine the $\Delta F$ output range. This assumes that the function $F$ is linear over the interval $\Delta x$ from the specified measurement point, which is a reasonable assumption for small $\Delta x$ and any linear functional form. However, few physical laws are linear over a wide range and, thus, this method will be inaccurate for steeply sloped functions combined with large individual measurement uncertainties. Also, this method assumes that the function $F$ is in an algebraic form that can be readily differentiated. This is obviously not always the case as many physical governing equations include ordinary and partial differential terms.

![Figure C-1. Determination of Uncertainty using Differential Methods](image)

**Coefficient Method**

Clearly, the above partial differential $\frac{\partial F}{\partial x_i}$ can be determined numerically using a simple forward difference approach. This is commonly called the coefficient method and is shown below.

$$c_i = \frac{\partial F}{\partial x_i} = \frac{F(x_j) - F(x_j + \Delta x_j)}{\Delta x_i}$$ (C.4)

and

$$\Delta F = |c_1 \cdot \Delta x_1| + |c_2 \cdot \Delta x_2| + |c_3 \cdot \Delta x_3| + \ldots |c_n \cdot \Delta x_n| = \sum_{i=1}^{n} |c_i \cdot \Delta x_i|$$ (C.5)

Both the partial derivative and coefficient method should yield identical answers when properly applied. Again, as long as $\Delta x$ is small and the slope of the function $F$ is moderate, this approach will yield
reasonably accurate uncertainties. A slight variation of this approach centers the numerical derivative around $x$, specifically:

$$c_i = \frac{\delta F}{\delta x} = \frac{F(x - 0.5\Delta x) - F(x + 0.5\Delta x)}{\Delta x}$$

(C.6)

This modified method often provides an improved determination of the uncertainty for measurements centered distributions.

Unfortunately, for convenience, the coefficient method is often misapplied by assuming fixed coefficients for a standard analysis. A number of well established engineering codes and specifications publish fixed numbers for uncertainty coefficients of standard engineering analysis problems. This approach can only be valid if the actual physical equation is strictly linear, which is seldom the case. Also, unless all units of measurement are identical to those of the published coefficients, largely incorrect uncertainty results will be obtained.

**Perturbation Method**

The most accurate analysis to determine total uncertainty of dependent variable measurement systems is the perturbation method, as it is based on the actual function $F$ and does not require any linearity assumptions. It is simply expressed as:

$$\Delta F = \Delta F(\Delta x_1) + \Delta F(\Delta x_2) + \Delta F(\Delta x_3) + \ldots + \Delta F(\Delta x_n)$$

$$= \left| F(x_1) - F(x_1 + \Delta x_1) \right| + \left| F(x_2) - F(x_2 + \Delta x_2) \right| + \left| F(x_3) - F(x_3 + \Delta x_3) \right| + \ldots + \left| F(x_n) - F(x_n + \Delta x_n) \right|$$

(C.7)

$$= \sum_{i=1}^{n} \left| F(x_i) - F(x_i + \Delta x_i) \right|$$

The term $\left| F(x_i) - F(x_i + \Delta x_i) \right|$ is graphically reviewed in Figure C-2 and demonstrates that the $\Delta F$ uncertainty obtained using this method is the actual transformation of $\Delta x$.

![Figure C-2. Determination of Uncertainty Using Perturbation Methods](image)

*Figure C-2. Determination of Uncertainty Using Perturbation Methods*
The variation of parameters method is implemented by sequentially perturbing the input values (temperature, pressure, etc.) by their respective uncertainties and recording the effects on the calculated output quantity (i.e., efficiency, power, etc.). Assuming the uncertainty perturbation is fairly small, any term in equation (C.8) can be determined in this manner:

\[
\Delta u_i \times \frac{\partial f}{\partial u_i} \approx f(u_i + \Delta u_i) - f(u_i)
\]  

(C.8)

The contribution of the variable \( u_i \) towards the overall uncertainty can be determined by calculating \( f \) twice—at the observed value at \( u_i \) and at the perturbed value of \( u_i + \Delta u_i \). For several variables, the results for each term should be summed using the square-root-sum or absolute value of the individual terms. The benefits of this approach are that it does not matter if the uncertainty is an absolute or relative number, the procedure can be implemented using any spreadsheet program, and the values in the spreadsheet can be the results of complex, iterative relationships.

**Implementation of the Partial Derivative Method for Compressors**

The partial derivative method was described in detail by Brun and Kurz [ASME Journal of Engineering for Gas Turbines and Power, 2001]. The implementation for both compressor and gas turbine is briefly described herein:

For the specific heat uncertainty, \( \Delta c_p \) is obtained:

\[
\Delta Z = \sqrt{\left( \frac{\Delta T \cdot \gamma}{\gamma - 1} \frac{R_{Univeral}}{L \cdot MW} \right)^2 + \left( \frac{\Delta L \cdot \gamma \cdot Z}{L \cdot MW} \right)^2 + \left( \frac{\Delta MW \cdot R_{Univeral} \cdot Z}{L \cdot MW^2} \right)^2}
\]  

(C.9)

The above equation (C.9) is valid if the physical gas properties, specific heat ratio, compressibility factor, and molecular weight are directly determined from testing. A physical property uncertainty, due to the effect of applying uncertainties in \( T \) and \( p \) to the non-ideal gas state equation has to be included; i.e., since there is a measurement error in \( T \) and \( p \), there will be an added error in determining \( c_p \) from the gas equation. This uncertainty is most conveniently obtained numerically by varying temperatures and pressures parametrically in the gas equation and, thus, determining the gradients \( d\gamma/dT \), \( d\gamma/dp \), \( dZ/dT \), and \( dZ/dp \) indirectly. Recognizing that \( dL/dT = dL/dT \) and \( d\gamma/dp = dL/dp \), one can easily determine corrections for \( \Delta Z \) and \( \Delta L \):

\[
\Delta Z = \sqrt{\left( \Delta T \cdot \frac{\partial \gamma}{\partial T} \right)^2 + \left( \Delta p \cdot \frac{\partial \gamma}{\partial p} \right)^2}
\]  

(C.10)

\[
\Delta Z = \sqrt{\left( \Delta T \cdot \frac{\partial Z}{\partial T} \right)^2 + \left( \Delta p \cdot \frac{\partial Z}{\partial p} \right)^2}
\]  

(C.11)

The uncertainty in \( c_p \), is also affected by the variation of the gas properties during the duration of the test. This effect is again mathematically difficult to describe but can be easily handled numerically using a procedure similar to the one shown above for the variations in \( T \) and \( p \). It is beyond the scope of this paper to list all possible gas composition variations; however, it is important to realize that they can strongly affect \( Z \), \( L \), and \( MW \).
The uncertainty of the compressor head is determined using equation (C.11). Since the head uncertainty is not dependent on the absolute temperatures, but rather on the temperature difference \( (T_d - T_s) \), and since the specific heats \( (c_p) \) for the discharge and suction are functionally related, the temperature difference \( (T_d - T_s) \) should be employed for the derivation rather than the absolute temperature values \( (T_d, T_s) \).

\[
\Delta H = \sqrt{\left(\Delta c_p \cdot (T_d - T_s)\right)^2 + \left(\Delta T_d \cdot c_p T_d\right)^2 + \left(\Delta T_s \cdot c_p T_s\right)^2}
\]  \(\text{(C.12)}\)

The uncertainty for the isentropic (ideal) compressor outlet temperature is obtained using equation (C.12).

**Isentropic Temperature**

\[
\Delta T_d^* = \sqrt{\left(\Delta T_s \cdot \left(\frac{P_d}{P_s}\right)^L\right)^2 + \left(\Delta P_d \cdot LT_s p_{d}^{L-1}\right)^2 + \left(\Delta P_s \cdot LT_s p_{d}^{L+1}\right)^2}
\]  \(\text{(C.13)}\)

The uncertainty of the compressor efficiency is given in equation (C.15). The temperature difference should be used rather than the absolute temperature values for the derivation of the isentropic enthalpy given in equation (C.14).

**Isentropic Enthalpy**

\[
\Delta h^* = \sqrt{\left(\Delta c_p \cdot (T_d^* - T_s)\right)^2 + \left(\Delta T_d^* \cdot c_p T_d\right)^2 + \left(\Delta T_s \cdot c_p T_s\right)^2}
\]  \(\text{(C.14)}\)

**Isentropic Efficiency**

\[
\Delta \eta^* = \sqrt{\left(\frac{\Delta h^*}{H}\right)^2 + \left(\frac{\Delta H}{H^2}\right)^2}
\]  \(\text{(C.15)}\)

**Mass Flow**

\[
\Delta W^2 = \left(\Delta P_s \cdot \frac{MW \cdot Q}{R_{\text{Universal}} Z T_s}\right)^2 + \left(\Delta MW \cdot \frac{P_s Q}{R_{\text{Universal}} Z T_s}\right)^2
\]  
\[
+ \left(\Delta Q \cdot \frac{P_s \cdot MW}{R_{\text{Universal}} Z T_s}\right)^2 + \left(\Delta Z \cdot \frac{P_s \cdot MW \cdot Q}{R_{\text{Universal}} Z^2 T_s}\right)^2
\]  
\[
+ \left(\Delta T_s \cdot \frac{P_s \cdot MW \cdot Q}{R_{\text{Universal}} Z^2 T_s}\right)^2}
\]  \(\text{(C.16)}\)
Power

\[ \Delta P = \sqrt{\left( \Delta H \cdot \frac{W}{\eta_M} \right)^2 + \left( \Delta W \cdot \frac{H}{\eta_M} \right)^2 + \left( \Delta \eta_M \cdot \frac{H \cdot W}{\eta_M^2} \right)^2} \]  

(C.17)

The flow rate uncertainty, \( \Delta Q \), depends strongly on the device type employed for the measurements. A detailed discussion of flow measurement uncertainty is provided in ASME PTC 19.1 [3] and is, thus, not further discussed herein.

By evaluating equations (C.14) through (C.17), estimates of the total measurement uncertainties for the compressor efficiency, head, and required driver power can be obtained. However, one source of measurement uncertainty that is often overlooked is the uncertainty due to a finite sample size. The above uncertainty statistics are valid only for mean parameters with an assumed Gaussian normal distribution. This is a good assumption for measurements where sample sizes are larger than 30. But for field tests, it is sometimes difficult to maintain a steady state system operating condition for a time period adequate to collect 30 or more samples.

**Implementation of the Partial Derivative Method for Gas Turbines**

To complete the above field test measurement uncertainty evaluation, one also needs to look at the complete turbocompressor train (gas turbine and compressor efficiency) performance. The gas turbine shaft output power has to equal the compressor required power \( (P_{GT} = P) \). Thus, the following two equations can be used to define the gas turbine thermal efficiency, \( \eta_{TH} \), and the total package efficiency, \( \eta_p \):

**Thermal Efficiency**

\[ \eta_{TH} = \frac{P}{W_{fuel} \cdot q} \]  

(C.18)

**Package Efficiency**

\[ \eta_p = \eta^* \cdot \eta_M \cdot \eta_{TH} \]  

(C.19)

Here \( W_{fuel} \) is the fuel flow into the engine and \( q \) is the fuel heating value. The fuel flow is typically measured using an orifice plate in a metering run and the heating value is determined from the chemical composition of the fuel (often the centrifugal compressor discharge gas is used as the fuel gas). Based on the above equations, the corresponding gas turbine uncertainty, \( \Delta \eta_{TH} \), and package uncertainty, \( \Delta \eta_p \), are given by:

**Thermal Efficiency**

\[ \Delta \eta_{TH} = \sqrt{\left( \frac{\Delta P}{W_{fuel} q} \right)^2 + \left( \frac{\Delta W_{fuel} P}{W_{fuel}^2 q} \right)^2 + \left( \frac{\Delta q P}{W_{fuel}^2 q^2} \right)^2} \]  

(C.20)
Package Efficiency

\[
\Delta \eta_p = \sqrt{\left(\Delta \eta \eta_M \eta_{TH}^*\right)^2 + \left(\Delta \eta_M \eta \eta_{TH}^*\right)^2 + \left(\Delta \eta_{TH} \eta_M \eta^*\right)^2}
\]  

(C.21)

To complete the above equations (C.20) and (C.21), the only additional information needed is the fuel flow uncertainty and the fuel heating value uncertainty. Since the fuel flow is measured in the same way as the flow through the gas compressor, uncertainty values in Table C-1 can be used. Also, since the heating value is obtained directly from gas composition, the same percent uncertainty as was obtained for the specific heat equation (C.9) can be used, namely:

\[
\frac{\Delta q}{q} = \frac{\Delta c_p}{c_p}
\]

(C.22)

By introducing the uncertainty experience values from those suggested in this guideline, the measurement uncertainty for a field test can be predicted prior to the test. Consequently, the above method allows the gas turbine/compressor manufacturer and the end-user to determine reasonable test uncertainties, as well as necessary requirements for the test instrumentation prior to the test. This method can also be employed to resolve observed variations of field test performance results from theoretically predicted and/or factory test results.

The different equation of state models will provide different values of head, isentropic head, and compressibility for the compressor based on the differences in calculated enthalpy and compressibility.
APPENDIX D

SIMILARITY CALCULATIONS FOR WET GAS CONDITIONS
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APPENDIX D

The following equations should be used to calculate the head and flow coefficients when wet gas conditions exist in the process gas through the centrifugal compressor. Additional information on performance evaluation in wet gas conditions is provided in Performance Evaluation of a Centrifugal Compressor Under Wet Gas Conditions by Brenne et al., 2005.

**Polytropic Exponent – Two Phase**

\[ n_{TP} = \frac{\ln \left( \frac{P_2}{P_1} \right)}{\ln \left( \frac{v_{TP1}}{v_{TP2}} \right)} \]  \hspace{1cm} (D.1)

**Mass Flow Rate – Two Phase, Polytropic Process**

\[ W_{p-TP} = \frac{n_{TP}}{n_{TP} - 1} \times \left( P_2 \times v_{TP2} - P_1 \times v_{TP1} \right) \]  \hspace{1cm} (D.2)

where the two-phase specific volume is defined as:

\[ v_{TP} = \frac{1}{GVF \times \rho_g + (1 - GVF) \times \rho_i} \]  \hspace{1cm} (D.3)

and the Gas Volume Fraction (GVF) is calculated as:

\[ GVF = \frac{Qdot_g}{Qdot_g + Qdot_i} \]  \hspace{1cm} (D.4)

Alternatively, the Mass Flow Rate for the Two Phase mixture may be calculated as:

\[ W_{p-TP} = x_1 \times \frac{n}{n-1} \times \frac{R_g}{MW_g} \times Z_i \times T_i \times \left( P_2 \frac{n-1}{P_1} \right)^{-1} \times (1 - x_1) \times v_{i1} \times (P_2 - P_1) \]  \hspace{1cm} (D.5)

where \( x \), the fluid quality, is defined as:

\[ x_1 = \frac{mdot_g}{mdot_g + mdot_i} \]  \hspace{1cm} (D.6)

**Head Coefficient – Two Phase, Polytropic Process**

\[ \varphi_{p-TP} = GVF_i \times \frac{v_{i1}}{v_{TP1}} \times \frac{W_{p-TP}}{U^2} \]  \hspace{1cm} (D.7)
**Flow Coefficient – Two Phase**

\[ \phi_{TP} = \frac{Q_{dot, TOT}^{-1}}{GVF_1 \times 2 \times \pi \times \frac{N}{60} \times D^3} \]  

(D.8)

**Efficiency – Two Phase, Polytropic Process**

\[ \eta_{p-TP} = \frac{W_{p-TP}}{P_{cs}} \]  

(D.9)

where \( P_{cs} \) is the specific compressor shaft power defined as the power consumed by the compressor per unit mass of wet gas.
APPENDIX E

EQUATION OF STATE MODEL COMPARISON OF PREDICTED PERFORMANCE DATA
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APPENDIX E

The following comparison of the equation of state models relies on results presented in *Enthalpy Determination Methods for Compressor Performance Calculations* by David Ransom, Rainer Kurz, and Klaus Brun.

**Approach for EOS Model Comparison**

For this comparison, a matrix of three gas compositions and two pressure ratios are considered. Enthalpy values are calculated using various EOS models and used to calculate compression power and isentropic efficiency. The three gas compositions (Table E-1) are intended to represent a variety of typical compression products including natural gas, high hydrogen, and high diluent compositions. (Note that gas mixture 1 is the same composition used in the uncertainty analysis in Section 5.0.) The two pressure ratios included in this comparison (PR = 1.3 and 2.2) are consistent with typical two- and six-stage machines respectively, although neither value represents any specific application. In all cases, inlet conditions of 1000 psia and 80°F are assumed. For each analysis configuration (gas mix and pressure ratio), both the isentropic and actual gas horsepower are determined, followed then by the isentropic efficiency. Gas power is a function of mass flow (assumed to be measured) and the change in enthalpy of the working fluid.

**Table E-1. Gas Mixtures Used in EOS Model Comparison**

<table>
<thead>
<tr>
<th>Component</th>
<th>Mix 1</th>
<th>Mix 2</th>
<th>Mix 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Methane</td>
<td>90.00</td>
<td>80.00</td>
<td>7.65</td>
</tr>
<tr>
<td>Ethane</td>
<td>5.37</td>
<td>5.37</td>
<td>1.06</td>
</tr>
<tr>
<td>Propane</td>
<td>1.70</td>
<td>1.70</td>
<td>0.20</td>
</tr>
<tr>
<td>Iso-Butane</td>
<td>0.27</td>
<td>0.27</td>
<td>0.00</td>
</tr>
<tr>
<td>N-Butane</td>
<td>0.33</td>
<td>0.33</td>
<td>0.00</td>
</tr>
<tr>
<td>Iso-Pentane</td>
<td>0.06</td>
<td>0.06</td>
<td>0.00</td>
</tr>
<tr>
<td>N-Pentane</td>
<td>0.09</td>
<td>0.09</td>
<td>0.00</td>
</tr>
<tr>
<td>Hexane</td>
<td>0.07</td>
<td>0.07</td>
<td>0.00</td>
</tr>
<tr>
<td>Carbon Dioxide</td>
<td>1.06</td>
<td>6.06</td>
<td>0.85</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>1.05</td>
<td>6.05</td>
<td>3.85</td>
</tr>
<tr>
<td>Hydrogen</td>
<td>0.00</td>
<td>0.00</td>
<td>86.34</td>
</tr>
<tr>
<td>Hydrogen Sulfide</td>
<td>0.00</td>
<td>0.00</td>
<td>0.05</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td>100.00</td>
<td>100.00</td>
<td>100.00</td>
</tr>
</tbody>
</table>

**Table E-2. Assumed Measured Conditions; PR = 1.3**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Mix 1</th>
<th>Mix 2</th>
<th>Mix 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_1$ (psia)</td>
<td>1000</td>
<td>1000</td>
<td>1000</td>
</tr>
<tr>
<td>$T_1$ (°F)</td>
<td>80</td>
<td>80</td>
<td>80</td>
</tr>
<tr>
<td>$P_2$ (psia)</td>
<td>1295</td>
<td>1295</td>
<td>1295</td>
</tr>
<tr>
<td>$T_2$ (°F)</td>
<td>119</td>
<td>117</td>
<td>127</td>
</tr>
</tbody>
</table>
Four EOS models are included in this comparison: Redlich-Kwong-Soave (Soave, 1972), Peng-Robinson (1976), Lee-Kesler-Ploecker (Ploeker et al., 1978), and Benedict-Webb-Rubin-Starling (Starling, 1973). Using the appropriate EOS, three enthalpy values are determined as follows: Determine the inlet enthalpy (h1) and entropy (s1) as a function of the inlet conditions (P1, T1); determine the isentropic discharge enthalpy (h2s) as a function of isentropic discharge conditions (P2, s1); and determine the actual discharge enthalpy (h2) as a function of the actual discharge conditions (P2, T2). A graphic representation of this process is provided below on a generic T-s diagram (Figure E-1).

![Figure E-1. Compression T-S Diagram](image)

Once these values are determined, it is a very simple calculation to determine the isentropic and actual gas horsepower values (equations E.1 and E.2).

\[
P_{Actual} = \dot{m}(h_2 - h_1) \quad \text{(E.1)}
\]

\[
P_{is} = \dot{m}(h_{2s} - h_1) \quad \text{(E.2)}
\]

Isentropic efficiency is calculated using equation (2.14).
Enthalpy Determination Results

The results for gas horsepower and isentropic efficiency for each of the EOS models are shown in Tables E-4 and E-5. For the sake of comparison, the mass flow for Mix 3 (high hydrogen gas) is adjusted to provide a similar horsepower as the natural gas compositions. Note that results are not shown for BWRS for the High H₂ gas composition (Mix 3) since BWRS does not contain hydrogen data.

These results demonstrate the relative agreement between the four EOS methods applied in this study. At the lower pressure ratio, for the first two gas mixtures, the standard deviation is about 30 Hp, or 1.2% of the average value. In the case of the high hydrogen gas (Mix 3), the standard deviation is approximately 140 HP, or 1.7% of the average value at the same pressure ratio. For the higher pressure ratio, the deviation in the first two mixtures between the EOS models is approximately the same as the lower pressure ratio. The deviation increases to 1.8% at the higher pressure ratio for the high hydrogen gas (Mix 3). It should also be noted that the Peng-Robinson model consistently predicts slightly lower horsepower for all three gas mixtures, while the SRK model typically predicts higher horsepower within the deviations stated above.

The isentropic efficiencies calculated using the various EOS models show relatively close agreement as well. However, the standard deviation among the four methods used in this study is as high as 2%, which can be significant when evaluating compressor performance against the promised performance, usually specified within 1%. In the extreme case, the isentropic efficiency between one particular EOS model and another can be as high as 3.8%. These results underscore the importance of applying the same EOS model throughout the performance analysis.

Table E-4. Horsepower and Efficiency Calculations for EOS Models at Pressure Ratio of 1.3

<table>
<thead>
<tr>
<th>EOS Model</th>
<th>Mix1</th>
<th>Mix2</th>
<th>Mix3</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Hp-Act</td>
<td>Eff-is</td>
<td>Hp-Act</td>
</tr>
<tr>
<td>LKP</td>
<td>2530</td>
<td>68.2%</td>
<td>2249</td>
</tr>
<tr>
<td>BWRS</td>
<td>2522</td>
<td>64.6%</td>
<td>2262</td>
</tr>
<tr>
<td>SRK</td>
<td>2528</td>
<td>65.6%</td>
<td>2261</td>
</tr>
<tr>
<td>PR</td>
<td>2460</td>
<td>65.0%</td>
<td>2200</td>
</tr>
<tr>
<td>Stdev</td>
<td>33.30</td>
<td>0.659</td>
<td>29.07</td>
</tr>
<tr>
<td>Avg</td>
<td>2510</td>
<td>0.659</td>
<td>2243</td>
</tr>
<tr>
<td>Stdev - %avg</td>
<td>1.33%</td>
<td>2.44%</td>
<td>1.30%</td>
</tr>
</tbody>
</table>

Table E-5. Horsepower and Efficiency Calculations for EOS Models at Pressure Ratio of 2.2

<table>
<thead>
<tr>
<th>EOS Model</th>
<th>Mix1</th>
<th>Mix2</th>
<th>Mix3</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Hp-Act</td>
<td>Eff-is</td>
<td>Hp-Act</td>
</tr>
<tr>
<td>LKP</td>
<td>8840</td>
<td>61.3%</td>
<td>7879</td>
</tr>
<tr>
<td>BWRS</td>
<td>8843</td>
<td>60.9%</td>
<td>7922</td>
</tr>
<tr>
<td>SRK</td>
<td>8940</td>
<td>61.7%</td>
<td>7986</td>
</tr>
<tr>
<td>PR</td>
<td>8695</td>
<td>60.8%</td>
<td>7767</td>
</tr>
<tr>
<td>Stdev</td>
<td>101.07</td>
<td>0.376%</td>
<td>92.13</td>
</tr>
<tr>
<td>Avg</td>
<td>8830</td>
<td>0.612</td>
<td>7889</td>
</tr>
<tr>
<td>Stdev - %avg</td>
<td>1.14%</td>
<td>0.61%</td>
<td>1.17%</td>
</tr>
</tbody>
</table>
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APPENDIX F

APPLICATION OF COMPRESSOR EQUATIONS FOR SIDE STREAM ANALYSIS
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APPENDIX F

The following equations should be used to calculate the power and isentropic efficiency of a centrifugal compressor when a side stream is used in the compression process (as shown in Figure F-1).

![Diagram of Compressor Stage Points with Side Stream](image)

**Figure F-1. Diagram of Compressor Stage Points with Side Stream**

**Compressor Power (with side stream)**

\[
P_C = m_3 h_3 - m_1 h_1 - m_2 h_2 = (m_1 + m_2) h_3 - m_1 h_1 - m_2 h_2 \quad (F.1)
\]

**Isentropic Efficiency (with side stream)**

If power from driver \((P_{out})\) is available:

\[
\eta^* = \frac{P_C}{P_{out}} \quad (F.2)
\]

Alternatively, the isentropic efficiency may be calculated by comparing the ideal and actual power:

\[
\eta^* = \frac{P_{ideal}}{P_{actual}} = \frac{\dot{m}_1 (h_3^* - h_1) + \dot{m}_2 (h_3^* - h_2)}{\dot{m}_1 (h_3 - h_1) + \dot{m}_2 (h_3 - h_2)} \quad (F.3)
\]