PRACTICAL METHODS FOR FIELD
PERFORMANCE TESTING CENTRIFUGAL COMPRESSORS

by

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ABSTRACT

Detailed performance analysis of centrifugal compressors in the
field is essential to evaluate their existing condition. The current
performance of a compressor can also be a valuable tool in
evaluating its reliability. A decrease in compressor performance can
be an excellent indication of internal wear or fouling, which if
allowed to continue may result in unscheduled outages or reduced
throughput. In contrast, perceived performance problems may be a
result of a compressor operating far from its original design.

Obtaining accurate performance data in the field can be very
challenging. The author explains the relative importance that
different process variables have in the performance calculations, as
well as specify the necessary instrumentation to obtain process data
with an acceptable uncertainty. Normal ranges and limitations for
calculated head and efficiency are provided to assist users in
determining if the field data are realistic. Methods to estimate both
mechanical and seal losses are demonstrated. Since original design
conditions almost never match actual operating conditions, the
author demonstrates how to compare actual field data with design
data, using nondimensional head and efficiency. Likewise, the limits
on these comparisons are outlined for users. The author provides
several example field performance evaluations and discusses ways
to avoid some common pitfalls. Examples of the effects of
inaccurate process data are also included in the discussion.

SUMMARY

This paper describes methods to obtain an accurate field
performance analysis that can be used to trend the performance of a
centrifugal compressor and evaluate its reliability. It outlines
the important aspects of field performance testing and provides practical
methods to obtain accurate results. In general, performance testing
to determine if a compressor meets its guaranteed design point
should be done in an original equipment manufacturer’s (OEM) test
facility where the accuracy of the instrumentation is almost always
better and the environment more easily controlled.

OBTAINING ACCURATE FIELD DATA

Accurate performance measurement of a centrifugal compressor is
very dependent upon the quality of the field data. In general,
testing should follow the conditions set forth in ASME PTC 10
(1997). Compressor piping should be designed to accommodate
flowmeter runs and to meet location requirements for pressure and
temperature as well. Small inaccuracies (in certain areas) can make
a large difference between the measured versus actual conditions.

Field data should only be taken during steady-state conditions.
Steady-state is achieved if the suction and discharge temperatures
do not change by more than one degree over a three to five minute
period. Likewise, data obtained at different operating points are
irreversible (i.e., multiple test points will allow construction of an as
tested performance curve, not just a single point to compare against
the OEM performance curve). For this reason, data transmitters are
preferred over local instrumentation because they can provide data
trends, which can be very insightful into the deterioration of the
compressor’s performance. The field data required for an accurate
performance evaluation are:

- Suction and discharge pressure
- Suction and discharge temperature
- Flow rate
- Gas composition
- Rotational speed
- Driver load

Additionally, other nontraditional data can be helpful in
diagnosing the source of a performance problem. These include,
but are not limited to:

- Radial vibration
- Axial position
- Balance line differential pressure
- Thrust bearing temperature

Pressure

Pressure tap locations should follow the guidelines in ASME
PTC 10 (1997) (Figure 1). Generally, pressure transmitters are
more accurate than gauges, but they are usually calibrated with test
gauges. If a digital pressure transmitter is used, the range of the
transmitter should be as narrow as feasible to obtain the greatest
accuracy (Table 1). Oil-filled bourdon tube pressure gauges should
be used for both suction and discharge pressure if transmitters are
not available. Note: Using a single pressure gauge for both suction
and discharge to eliminate calibration error may not be a good
practice, if there is a large pressure differential across the
compressor. For example, do not use a 300 psig gauge to measure
suction and discharge pressure on a compressor that pumps from 5
psig to 250 psig. Wall tap holes need to have sharp edges and be
free of weld slag and/or burrs. Likewise, the tap should be
perpendicular to the process piping. Normally, pressure gauges and
transmitters only measure static pressure; however, the total pressure is required for performance calculations. The total pressure equals the sum of the static and velocity pressures (Equation (1)). Normally, the difference between static and total pressure is minimal because the suction and discharge piping have been adequately designed to keep the gas velocity low.

\[
P_{\text{TOTAL}} = P_{\text{STATIC}} + P_{\text{VELOCITY}} = P_{\text{STATIC}} + \frac{V^2}{2g_e \gamma_p} \tag{1}
\]

![Diagram](image)

\(4\) static inlet pressure taps spaced 90 deg

4 inlet thermowells, 45 deg from static pressure

12 in. min

24 in. min

6 in. min

12 in. min

8 in. min

6 in. min

L_1

L_2

L_3

L_4

L_5

L_6

L_7

L_8

Figure 1. Pressure and Temperature Locations.

![Table](image)

<table>
<thead>
<tr>
<th>Piping Config. at A or B</th>
<th>Inlet</th>
<th>Outlet</th>
</tr>
</thead>
<tbody>
<tr>
<td>Straight Run/Elbow</td>
<td>L_1</td>
<td>L_2</td>
</tr>
<tr>
<td></td>
<td>3D</td>
<td>2D</td>
</tr>
<tr>
<td>Reducer</td>
<td>L_3</td>
<td>L_4</td>
</tr>
<tr>
<td></td>
<td>5D</td>
<td>3D</td>
</tr>
<tr>
<td>Valve</td>
<td>L_5</td>
<td>L_6</td>
</tr>
<tr>
<td></td>
<td>10D</td>
<td>8D</td>
</tr>
<tr>
<td>Flow Device</td>
<td>L_7</td>
<td>L_8</td>
</tr>
<tr>
<td></td>
<td>5D</td>
<td>3D</td>
</tr>
</tbody>
</table>

![Table](image)

<table>
<thead>
<tr>
<th>Instrument</th>
<th>Accuracy (% error)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure transmitter – Analog mode</td>
<td>0.75% of range</td>
</tr>
<tr>
<td>Pressure transmitter – Digital mode</td>
<td>0.1% of value or 0.75% of range</td>
</tr>
<tr>
<td>500 psi oil filled pressure gauge</td>
<td>10-25 psi</td>
</tr>
</tbody>
</table>

Table 1. Pressure Instrumentation Accuracy.

![Table](image)

<table>
<thead>
<tr>
<th>Sensor</th>
<th>Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>RTD (100 ohm platinum)</td>
<td>± 0.3%</td>
</tr>
<tr>
<td>E Type Thermocouple</td>
<td>± 3.0°F or 0.5%</td>
</tr>
<tr>
<td>K Type Thermocouple</td>
<td>± 4.0°F or 0.75%</td>
</tr>
<tr>
<td>J Type Thermocouple</td>
<td>± 4.0°F or 0.75%</td>
</tr>
<tr>
<td>T Type Thermocouple</td>
<td>± 1.8°F or 0.75%</td>
</tr>
</tbody>
</table>

Table 2. Temperature Sensor Accuracy.

As an example to illustrate the importance of temperature accuracy, assume a five-degree error on a reformer hydrogen recycle compressor with the following conditions:

- \(P_1 = 135\) psig
- \(P_2 = 240\) psig
- \(T_1 = 90 \pm 5\) °F
- \(T_2 = 200 \pm 5\) °F

The two extremes of this case would result in a calculated polytropic efficiency of 58 percent and 70 percent, which results in a 17 percent difference in the calculated horsepower (about 1000 hp for this particular machine).

Flow Rate

The flow rate reported by the flowmeter is usually not correct. The meter factor \((K)\), which converts the measured differential pressure into a flow rate, is always a function of the gas pressure, temperature, compressibility, and molecular weight (Equation (3)). Most flowmeters will have a meter factor that is only valid for one set of design conditions. If the actual conditions are different from the meter design, the flow rate calculated from the meter factor must be corrected. Equation (4) gives the correction factor for volumetric flow at standard conditions (14.7 psia and 70°F). In contrast, if the flow rate is displayed in mass units, the correction factor is different (Equation (5)). Occasionally, the meter factor is "compensated" for the actual pressure, temperature, and molecular weight inside the distributed control systems (DCS) or programmable logic controllers (PLC). However, the molecular weight reported by the gravity analyzer should be verified with the gas composition. Likewise, calibrate and range the flowmeter before the performance

\[
T_{\text{TOTAL}} = T_{\text{STATIC}} + 0.35 \times T_{\text{VELOCITY}} = T_{\text{STATIC}} + \frac{0.35 \times V^2}{2g_e \gamma_p} \tag{2}
\]
test. If possible, verify the meter diameter before the test as well. The flowmeter design and location should meet the requirements of the guidelines established by ASME PTC 10 (1997).

\[ Q = K \sqrt{AP} \quad (3) \]

where:

\[ K = f(P, T, Z, MW) \]

\[ Q_{\text{ac}} = Q_k \left( \frac{P}{P_D} \right)^{\frac{T_D}{T}} \left( \frac{MW_D}{MW} \right) \left( \frac{Z_D}{Z} \right) \quad (4) \]

\[ \dot{m}_{\text{ac}} = \dot{m}_k \left( \frac{P}{P_D} \right)^{\frac{T_D}{T}} \left( \frac{MW_D}{MW} \right) \left( \frac{Z_D}{Z} \right) \quad (5) \]

Gas Composition

The gas composition is the most important data required to evaluate a compressor’s performance. Likewise, an accurate gas composition is also the most difficult to obtain. Take a minimum of two gas samples during the performance test. Two samples are required to validate the gas analysis (compare) in case one of the samples is lost or invalid (i.e., without the gas composition, the other process data are worthless). If the performance test lasts for several hours (or days), take multiple gas samples to verify that the gas composition does not change during the test. Use a free flowing arrangement to obtain the sample. Figure 2 shows two examples. If possible, use an insertion probe to obtain the gas sample instead of a wall tap. Samples obtained from wall taps will generally be leaner in the higher molecular weight components due to the boundary layer effect. Heat the sample bomb to the temperature of the process before analyzing it. This will prevent any condensation of liquids that could alter the gas composition. Obviously, this is much more important for higher molecular weight services such as fluid catalytic cracker (FCC) wet gas as compared with reformer recycle hydrogen. As an example, a sample taken from the discharge of a coker wet gas compressor was analyzed at the lab ambient temperature (approximately 75°F) and at 275°F (sample temperature, Table 3). As can be seen, the incorrect gas composition has pronounced effect on the calculated gas horsepower. This effect is magnified because the molecular weight is used to calculate the head as well as correct the flow.

Review the gas composition to determine if it is feasible. One of the most common methods used to determine the composition of a gas sample is the gas chromatograph. A gas chromatograph determines the components by burning them in the presence of a carrier gas. For this reason, a gas chromatograph will not show any water vapor (i.e., water will not burn). However, it is common for process gases to be saturated with water. For this reason, the measured gas composition must be adjusted if it is indeed saturated with water. Likewise, for a multisection compressor, the gas composition will typically become leaner as liquids are condensed out in the intersection knockouts. If the gas samples do not reflect this, there is usually a problem. In addition, certain noncondensable components should remain the same.

For example, the flow diagram of a three section wet gas compressor is shown in Figure 3. Sample liquid knockouts to calculate mass balances around every process split. The measured gas compositions should meet the following constraints:

\[ \dot{m}_1 = \dot{m}_2 + \dot{m}_3 = \dot{m}_4 + \dot{m}_5 + \dot{m}_6 = \dot{m}_7 + \dot{m}_8 + \dot{m}_9 + \dot{m}_0 \quad (6) \]

Figure 3. Multisection Wet Gas Compressor.

Likewise, the mass fraction of noncondensables such as H₂ and methane (CH₄) should be constant.

\[ \dot{m}_{1,\text{H}_2} = \dot{m}_{2,\text{H}_2} = \dot{m}_{4,\text{H}_2} = \dot{m}_{5,\text{H}_2} = \dot{m}_{7,\text{H}_2} = \dot{m}_{8,\text{H}_2} = \dot{m}_{10,\text{H}_2} \quad (7) \]

In addition, the mass fraction of H₂S, which is also a noncondensable, should be constant until the gas stream passes through the amine contactor, which will remove most, if not all, of the H₂S. The result would be that:

\[ \dot{m}_{1,\text{H}_2S} = \dot{m}_{2,\text{H}_2S} = \dot{m}_{4,\text{H}_2S} \quad \text{and} \quad \dot{m}_{8,\text{H}_2S} = \dot{m}_{10,\text{H}_2S} = 0 \quad (8) \]

Table 3. Effects of Incorrect Gas Composition on Corrected Flow and Calculated Horsepower.

<table>
<thead>
<tr>
<th></th>
<th>Correct Gas Sample (275°F)</th>
<th>Incorrect Gas Sample (75°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Molecular weight</td>
<td>48</td>
<td>34</td>
</tr>
<tr>
<td>Flow (MMcfd)</td>
<td>27.4</td>
<td>32.5</td>
</tr>
<tr>
<td>Shaft horsepower (hp)</td>
<td>7036</td>
<td>5775</td>
</tr>
</tbody>
</table>

Note: If normalized gas compositions are used, the mole fraction of noncondensables will actually go up due to the fact that the noncondensables will make up a larger fraction of the total stream.

Calculation and Evaluation of Performance Parameters

The correct performance parameters must be accurately calculated and evaluated to ensure that the field data are realistic. The most critical step in calculating performance parameters is determining the inlet and outlet density, enthalpy, and entropy. For hydrocarbon gas mixtures, performance programs that use equations of state such as Lee–Kesler (1975), Benedict-Webb-Rubin (BWR), or Soave-Redlich-Kwong (SRK) will provide much better results than approximations from Mollier diagrams or ideal gas relationships. Process simulators such as HYSYS™ and ASPEN® can be used as well. Once the gas properties are calculated, the correct parameters must be selected to adequately evaluate the performance of the compressor. Evaluation of the results of these calculations should be made to determine the validity of both the calculations and the field data. Refer to APPENDIX A for a listing of the equations for these parameters.
The compression process can be modeled as either an isentropic process (reversible without heat transfer) or polytropic process (reversible with heat transfer). The author prefers the polytropic process to the isentropic process for the following reasons:

- The sum of the individual impeller polytropic heads is equal to the total compressor head. This is not true for isentropic head.
- The polytropic efficiency is independent of compression ratio, whereas isentropic efficiency is not.

One drawback in using the polytropic process is that the polytropic head is affected by the calculated polytropic efficiency (APPENDIX A). This can make it more difficult in determining the cause of a performance problem.

**Shaft Horsepower**

If a torque meter is available, the shaft horsepower may be calculated directly. However, since this is not commonly the case, the horsepower can be calculated by the heat balance method, as given in ASME PTC 10 (1997) and compared with the calculated driver horsepower. The percent difference between the driver horsepower and the compressor horsepower is a good indication of the accuracy of the performance analysis. Application of the First Law of Thermodynamics to the control volume (Figure 4) around the compressor gives Equation 9.

\[
\text{SHP} = \frac{(\dot{m}_1 - \dot{m}_2)(h_2 - h_1) + Q_R + HP_{\text{MECH}}}{2545} \tag{9}
\]

\[
\text{SHP} = HP_{\text{GAS}} + HP_{\text{MECH}}, \quad HP_{\text{MECH}} = HP_{M1} + HP_{M2} \tag{10}
\]

**Polytropic Head**

This value is limited to between 10,000 ft-lbf/lbm and 15,000 ft-lbf/lbm per impeller, for closed 2D impellers (Lapina, 1982). The sonic velocity of the gas and the yield stress of the impellers set the limit (Figure 5, Lapina, 1982). Impellers designed to operate in the highly corrosive processes (H₂S, CO₂) often require a maximum yield stress of 90 ksi and Rockwell C of less than 22, which limits them to approximately 10,000 ft-lbf/lbm. For example, if a performance test is done on a typical multistage process compressor and the calculated head per impeller is 20,000 ft-lbf/lbm, then either the measured compression ratio is too high or the measured molecular weight is too low. Note: Open 3D impellers (such as for plant air applications or high-speed turboexpander units) can produce heads up to 60,000 ft-lbf/lbm per impeller.

**Polytropic Efficiency**

The maximum value of polytropic efficiency is dependent upon the flow coefficient (\( \Phi \)) of the impeller and its construction, but is limited to approximately 78 percent to 80 percent for shrouded impellers with vanless diffusers (Figure 6). The maximum efficiency occurs at approximately \( \Phi = 0.2 \).

**Figure 5. Maximum Polytropic Head Per Impeller.**

**Figure 6. Maximum Polytropic Efficiency Per Impeller.**

**Losses**

Losses are generally grouped into three distinct areas: mechanical, seal, and aerodynamic. Mechanical losses include power dissipated through bearings, oil or gas seals, shaft driven lube oil pumps, and gearboxes. Seal losses are the decrease in the amount of energy available to convert into pressure head due to internal recirculation inside the compressor. Aerodynamic losses include effects such as friction and pressure losses in the impellers and diffusers.

**Mechanical Losses**

Mechanical losses are mostly a function of size and speed. Larger bearings and seals at higher speeds dissipate more power. Mechanical losses are simply added to the calculated gas.
horsepower. These losses can be approximated by the following methods:

- Measuring the flow rate and temperature increase of the lube/seal oil and using Equation (11). This may be difficult due to many different lube oil return lines and pressure controllers that spill back to the reservoir.

\[
\text{HP}_{\text{mech}} = \frac{m_c \Delta T_{\text{oil}}}{30000}
\]  

(11)

- OEM supplied curves for different bearings and seals (Figures 7 and 8)

- Bearing rotodynamic computer models that calculate horsepower losses as well

- Tables based upon compressor gas horsepower (Table 4) (Lapina, 1982). Table 5 gives approximate gearbox efficiencies

\[\text{Figure 8. Mechanical Losses for Face Contact Oil Seals. (Courtesy of Elliott Co.)}\]

\[\text{Table 4. Mechanical Losses.}\]

<table>
<thead>
<tr>
<th>Gas Power Requirement</th>
<th>Mechanical Losses (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0-3000</td>
<td>3.0</td>
</tr>
<tr>
<td>3000-6000</td>
<td>2.5</td>
</tr>
<tr>
<td>6000-10,000</td>
<td>2.0</td>
</tr>
<tr>
<td>10,000 +</td>
<td>1.5</td>
</tr>
</tbody>
</table>

\[\text{Figure 7. Mechanical Losses for Journal and Tilting Pad Thrust Bearings. (Courtesy of Elliott Co.)}\]

\[\text{Table 5. Gearbox Efficiencies.}\]

<table>
<thead>
<tr>
<th>Gear Type</th>
<th>Efficiency (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Helical</td>
<td>97-99</td>
</tr>
<tr>
<td>Herringbone</td>
<td>96-99</td>
</tr>
<tr>
<td>Straight bevel</td>
<td>95-98</td>
</tr>
<tr>
<td>Spiral bevel</td>
<td>96-98</td>
</tr>
</tbody>
</table>

- Balance piston line differential pressure—The differential pressure between the balance cavity on the outlet of the balance piston seal and the suction of the compressor is a strong indicator of the amount of leakage. Most OEMs design for this differential pressure to be less than 2 psid or 3 psid. Anything above this usually means a balance piston seal that is leaking excessively. A differential pressure gauge is usually required to make this measurement due to the small differential. If pressure taps are not available on the balance line, a differential pressure gauge can be installed on the seal oil traps (Figure 11).
also strong functions of the compressor differential pressure and power as well. For this reason, assumptions about the balance piston seal should not be based on a high thrust position alone. A good method is to plot either thrust position or thrust bearing temperature rise divided by power versus time.

Leakage across a division wall seal in a back-to-back compressor will usually have a lessened effect as compared with balance piston leakage in a straight through compressor, because the increase in temperature is usually removed in an interstage cooler and most of the recirculation occurs only in the second section (Figure 12). The division wall leakage can alter the calculated first section efficiency, if the discharge temperatures of the two sections differ greatly. There is a small amount of recirculation from the suction of the second section to the suction of the first section through a seal equalizing line. However, the fact that this gas is not hot reduces the effects of this leakage. Division wall seals can cause problems due to their location in the center of the machine, which requires added clearance, which in turn increases the leakage rates. Likewise, division wall seals are more likely to rub than balance piston seals as the rotor passes through its first critical speed. Consider all these factors when evaluating the compressor performance.

The seal leakage rate can be estimated from the following methods:

- **OEM test data**—Calculate a seal orifice constant from the measured shop leakage data (Equation 12). This constant can then be used to calculate the leakage in the field. Likewise, the constant can be adjusted to allow for increases in clearance due to wear/corrosion (i.e., the constant is directly proportional to the seal leakage area).

\[
m = K \cdot P_2 \cdot \sqrt{1 - \frac{P_1}{P_2}} \Rightarrow K = \frac{m \cdot \sqrt{Z_2 \cdot T_2}}{P_2 \cdot \sqrt{1 - \left(\frac{P_1}{P_2}\right)^2}} \quad (12)
\]

- **Seal leakage equations**—If measured leakage data are not available, estimate the rate by using a leakage equation. Equation 13 applies for adjacent teeth in a see-through (noninterlocking) labyrinth. This is used to iteratively solve for the labyrinth cavity pressures and seal leakage rate. If the flow is choked, use Equation 14 for the last labyrinth (Childs, 1993).

\[
m = \mu \cdot H \cdot \frac{P_1^2 - P_2^2}{ZRT} \quad (13)
\]
Once the leakage rate is determined, the measured field data must be adjusted to find the actual conditions. In the case of a straight through compressor with a balance piston seal, the hot balance piston seal leakage is mixed with the gas at the inlet. A mass and enthalpy balance is used to determine the actual inlet temperature to the first impeller (Figure 13). This will change the polytropic head and efficiency calculations. An example calculation showing flange and impeller conditions is shown in Figure 14. Note that the balance piston leakage has increased the inlet temperature by five degrees in the corrected test data, which caused the calculated impeller efficiency to change from 72 percent to 76 percent. This does not mean that the higher inlet temperature has increased the efficiency of the compressor. The actual efficiency of the compressor is still 72 percent. However, the efficiency of the impellers is 76 percent, which will put the compressor right on its curve. Adding in the model for the balance piston seal leakage allows us to see why the compressor efficiency is not as designed.

![Balance piston seal leakage flow](image)

\[ h_i' = m_i h_i + m_p h_p \]

**Figure 13. Mass and Heat Balance for Balance Piston Seal Leakage.**

**Aerodynamic Losses**

Aerodynamic losses include various friction, slip, pressure, and shock losses in the rotating and stationary components of the compressor. For the purposes of this discussion, these losses are represented in the polytropic efficiency of the compressor and are not covered.

**COMPARING MEASURED FIELD PERFORMANCE TO SHOP TEST OR PREDICTED PERFORMANCE**

Since field conditions never exactly match the original design, certain nondimensional parameters must be calculated so that the field performance can be compared with the OEM shop test or predicted performance data. While these nondimensional parameters will enable “apple-to-apple” comparisons for different conditions, they have very real limitations based on the aerodynamic characteristics of the impellers. These nondimensional parameters include the following:

**Polytropic head coefficient**, \( \mu_p \)

\[ \mu_p = \frac{H_p}{(\pi DN)^2} \]  \hspace{1cm} (15)

**Polytropic efficiency**, \( \eta_p \)

\[ \eta_p = \frac{H_p}{h_2 - h_1} \]  \hspace{1cm} (16)

**Flow coefficient**, \( \Phi \)

\[ \Phi = \frac{Q}{N^3 D^3} \]  \hspace{1cm} (17)

**Inlet Mach number**, \( M \)

\[ M = \frac{\pi DN}{k_{lZ_lRT}} \]  \hspace{1cm} (18)

The first step in any comparison is to obtain a set of nondimensional curves from the shop test or predicted data. If the OEM did not provide these nondimensional curves, obtain them by iteratively calculating the nondimensional head and efficiency values from the given values of discharge pressure and shaft horsepower. This can be accomplished by guessing a discharge temperature for the given discharge pressure until the correct shaft horsepower has been reached. Once the correct discharge temperature is known, the polytropic head coefficient and efficiency can be calculated to give a set of nondimensional curves (Figure 15). These curves will predict the performance of the compressor for the given Mach number. The curves can be used to compare against the existing operating conditions if the field Mach number is close enough to the Mach number for the curves. The ASME PTC 10 (1997) test code defines the maximum shift in Mach number for a certified shop performance test (Figure 16). These limitations are good to apply in the field as well. If the Mach number shift is too large, the comparison may be inaccurate. If this is the case, obtain a new set of performance curves from the OEM that match the actual inlet conditions.

Plot the polytropic head coefficient and efficiency at the existing operating conditions on the nondimensional graphs to determine if the compressor is operating on its curve. In addition, the nondimensional curves can be used to calculate the field discharge conditions (pressure, temperature, horsepower, etc.) based on the field inlet conditions (Figure 17). Seal losses increase the calculated value of \( \Phi \), which moves the predicted operating point further to the right on the performance curves and always increases the horsepower.

<table>
<thead>
<tr>
<th>Title</th>
<th>Hydrogen Recycle Compressor 1st Stage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test Data</td>
<td>Suction Conditions:</td>
</tr>
<tr>
<td>Press (psig)</td>
<td>150</td>
</tr>
<tr>
<td>Temp (F)</td>
<td>88</td>
</tr>
<tr>
<td>Flow</td>
<td>0 MMSCFD (only one flow is required)</td>
</tr>
<tr>
<td>ICPM</td>
<td>2507 ft/min</td>
</tr>
<tr>
<td>Flowmeter</td>
<td>1 Location (1: suction, 2: discharge)</td>
</tr>
<tr>
<td>design data</td>
<td>0 Mole weight</td>
</tr>
<tr>
<td>0 Temp (F)</td>
<td></td>
</tr>
<tr>
<td>0 Press (psia)</td>
<td></td>
</tr>
<tr>
<td>Speed</td>
<td>7940 rpm</td>
</tr>
<tr>
<td>Impeller Number</td>
<td>Diameter(s)</td>
</tr>
</tbody>
</table>

**CALCULATED DATA**

<table>
<thead>
<tr>
<th>Property</th>
<th>Inlet</th>
<th>Outlet</th>
<th>Inlet</th>
<th>Outlet</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressibility</td>
<td>1.0007</td>
<td>1.0009</td>
<td>1.0028</td>
<td>1.0060</td>
</tr>
<tr>
<td>Entropy (btu/lbm)</td>
<td>0.519</td>
<td>0.564</td>
<td>0.564</td>
<td>0.676</td>
</tr>
<tr>
<td>Specific volume (ft³/lbm)</td>
<td>5.33</td>
<td>5.79</td>
<td>5.37</td>
<td>3.79</td>
</tr>
<tr>
<td>Specific heat (Btu/lbm-°F)</td>
<td>1.132</td>
<td>1.178</td>
<td>1.134</td>
<td>1.178</td>
</tr>
<tr>
<td>cp</td>
<td>1.33</td>
<td>1.31</td>
<td>1.30</td>
<td>1.31</td>
</tr>
<tr>
<td>Enthalpy (Btu/lbm-°F)</td>
<td>3.68</td>
<td>3.93</td>
<td>3.69</td>
<td>3.93</td>
</tr>
</tbody>
</table>

| MMSCFD | 13553.2 | 119.0 |
| ICPM | 2507.0 |
| DN | 1.982 |
| Performance | | |
| Performance (corrected) | | |
| Polytropic exponent | 1.51 | 1.47 |
| Polytropic head | 65997 | 65973 |
| Polytropic head coeff | 0.587 | 0.560 |
| Polytropic ef | 1.00 | 1.00 |
| Polytropic ef | 0.72 | 0.76 |
| Gas horsepower | 6937 | 6937 |
| Shaft horsepower | 7076 | 7076 |
| Mach no | 0.396 | 0.396 |
| Volume ratio | 1.406 | 1.416 |

**Figure 14. Calculated Test Data.**
EXAMPLE PERFORMANCE EVALUATION

Reformer Hydrogen Recycle Compressor

- Configuration
  - Six impellers, straight through, barrel type
  - Vaneless diffusers
  - Interlocking aluminum labyrinth balance piston seal
  - Motor driven through a speed increasing gearbox
  - Contact type oil face seals

- Original design
  - 6370 hp
  - 7940 rpm
  - Inlet flow - 14,600 cfm
  - Molecular weight - 5.18
  - $T_1 = 100^\circ$F
  - $P_1 = 187$ psia
  - $P_2 = 275$ psia

- Balance piston labyrinth dimensional data
  - Stationary labyrinth
    - Nine teeth, 0.375 inch equal spacing, 0.1875 inch tall
    - 14.595 inch internal diameter
  - Rotating labyrinth
    - Eight teeth, 0.375 inch equal spacing, 0.1875 inch tall
    - 14.937 inch outside diameter

- Calculated seal leakage rate - 120 lbm/min
• Measured performance data
  • \( P_1 = 150 \) psia
  • \( T_1 = 80^\circ F \)
  • \( P_2 = 248 \) psia
  • \( T_2 = 191^\circ F \)
• Balance line differential pressure = 6 psid
• Flow = 190 MMScfd
• Speed = 7949 rpm
• Molecular weight = 7.1

• Driver data
  • Amperage = 978
  • Voltage = 4000
  • Speed = 1784
  • \( Pf = 0.92 \)
  • \( E = 0.957 \)

Calculate the horsepower supplied by the motor to the gearbox:

\[
 HP_{\text{MOTOR}} = \frac{V \cdot Pf}{431} = \frac{4000 \cdot 978 \cdot 0.957 \cdot 0.92}{431} = 7988 \text{ hp} \quad (19)
\]

Approximating the gearbox efficiency to be 97 percent, the calculated shaft horsepower delivered to the compressor is:

\[
 HP_{\text{COMP}} = (7988)(0.97) = 7750 \text{ hp} \quad (20)
\]

The OEM supplied a performance curve showing predicted discharge pressure and shaft horsepower (Figure 18). These curves are valid for only the design conditions listed above (i.e., you cannot plot the measured discharge pressure and shaft horsepower on these curves). Likewise, they are only predicted curves (i.e., the compressor was not shop performance tested). These curves must be converted into a nondimensional form so that they can predict the existing field performance (Figure 19).

The discharge conditions are reproduced with and without seal losses to compare against the measured field results (Table 6 and Figure 20). The horsepower for the existing field performance data must be close to the horsepower supplied by the driver for the results to be considered valid; in this case, they are within 2 percent. Three different field data points were measured and compared with the predicted data (Figures 21 and 22). As can be seen, the surge point has moved to the right of the original predicted surge point. The efficiency of the compressor is considerably lower than what is predicted by the OEM performance curve. Adding the seal losses (approximately 4 percent to 5 percent of the total flow) to the predicted curves brings the predicted and actual conditions closer together (Table 6). However, the measured efficiency is just too low to be a balance piston seal problem alone. Likewise, the thrust bearing temperature and axial position were relatively low. Based on the history of fouling in this compressor as well as the high balance line differential pressure, the loss in efficiency was thought to be a result of fouling. The compressor was still meeting the desired discharge pressure, but the low efficiency was causing excessive horsepower consumption, which was limiting unit charge rate. Because the motor was oversized and the loss in efficiency had been gradual, operations was unaware the problem was in the compressor and not the motor (i.e., they thought the motor was dirty).

The compressor was pulled because it could not be washed in place due to a lack of adequate case drains. A large amount of ammonia chloride buildup was found in the stationary components
of the compressor (the diffuser channels had approximately 40 percent blockage, Figures 23 and 24). This large amount of fouling was causing the surge point to be at a higher flow rate. Note: the synchronous vibration amplitudes were relatively low (< 1 mil) because the fouling was mostly on the stationary components, the only fouling on the rotor was on the inside diameter of the impeller eyes.

After the compressor was reinstalled, the measured field performance was within 3 percent of the predicted.

Figure 20. Example Test and Repredicted Field Performance Data.

Figure 21. Predicted and Measured Polytropic Head Coefficient.

Figure 22. Predicted and Measured Polytropic Efficiency.

Figure 23. Fouled Inlet Guide Vanes on Hydrogen Recycle Compressor.

Figure 24. Diaphragm Half from Hydrogen Recycle Compressor Showing Fouled Diffuser.

CONCLUSION

Field performance testing centrifugal compressors is a necessity to monitor the integrity of the machine and to predict losses in performance, which can be used to set turnaround schedules. A single data point of measured performance will not give an accurate indication of the compressor's condition. A history of the performance of the compressor is required to make an accurate estimate of its condition. The accuracy of the field test data is the most important aspect of field performance testing. The gas analysis is the most important piece of the field data and likewise the most difficult to obtain accurately. The calculated performance parameters must be examined to confirm the accuracy of both the test data as well as the calculations. Likewise, the effects of various losses must be considered when looking at the overall compressor performance. Comparisons of field test data to OEM data can only be made using nondimensional parameters and these comparisons are limited by additional nondimensional parameters. For field performance testing to be accurate, the test engineer must follow a set procedure that considers all the above requirements for each individual compressor. Not following all these basic points can lead to incorrect performance predictions and unpredicted drops in performance.
PRACTICAL METHODS FOR FIELD PERFORMANCE TESTING CENTRIFUGAL COMPRESSORS

NOMENCLATURE

\[ \delta = \frac{h_s - h_1}{\left( \frac{\gamma_1}{\gamma_2 - 1} \right) \left( \frac{p_2}{p_1} \frac{v_1}{v_s} \right) } \]  

(A-5)

Polytropic Exponent, \( \gamma_p \)

\[ \gamma_p = \frac{\ln \left( \frac{p_2}{p_1} \right)}{\ln \left( \frac{\gamma_2}{\gamma_1} \right)} \]  

(A-6)

For ideal gases, the polytropic exponent and efficiency can be calculated by the following:

\[ \gamma_p = \frac{\ln \left( \frac{p_2}{p_1} \right)}{\ln \left( \frac{T_2}{T_1} \right)} \]

\[ \eta_p = \frac{\gamma_p}{\left( \frac{k}{k-1} \right)} \]  

(ideal gases only)  

(A-7)

These equations should not be used for multicomponent hydrocarbon mixtures.

APPENDIX B

TROUBLESHOOTING GUIDELINES

Fouling

Fouling is almost always accompanied by a large decrease in efficiency, along with a somewhat lessened decrease in head. The decrease in efficiency is caused by a combination of internal recirculation due to fouled labyrinth seals and changes in the aerodynamic performance of the rotating and stationary components due to obstructed flow passages. Fouling is not always accompanied by an increase in synchronous vibration amplitudes (a common assumption), because the buildup is usually on the stationary components. A very strong indicator of fouling is a decrease in the amount of turndown to surge (i.e., increase in the minimum flow). This is usually caused by buildup in the diffuser that restricts the flow and causes the compressor to surge or build up on the inlet guide vanes, which distorts the flow into the impellers and causes the vanes to stall.

Incorrect Process Data

No matter how carefully data are measured in the field, inaccuracies are many times a reality that must be recognized. Evaluating whether or not the data are correct may be the most important part of field performance testing. Below are some common sources of error and guidelines in detecting them.

Incorrect Flowmeter

An incorrect flow measurement will cause the compressor to appear either low or high in head because the operating point is marked incorrectly on the performance map. The best method to determine if a flow measurement is incorrect is to obtain several data points to compare against the entire curve. As can be seen in Figure B-1, the maximum head should remain the same. The curve is just shifted to the right or left.

Incorrect Gas Composition

If the measured gas composition is lower than the actual gas composition in the compressor, it will have the most pronounced effect on both the corrected flow rate and calculated polytropic head. An incorrect low molecular weight will cause the corrected flow rate (if it is measured in standard cubic feet) to be higher (Equation (B-1)). Likewise, the low molecular weight will cause the calculated polytropic head to be higher than it actually is for the

APPENDIX A

PERFORMANCE PARAMETERS

Polytropic Head, \( H_p \)

\[ H_p = \delta * P_1 * v_1 * r_p * \left( \frac{p_2}{p_1} \frac{v_1}{v_s} \right) - 1 \]  

(A-1)

Polytropic Efficiency, \( \eta_p \)

\[ \eta_p = \frac{H_p}{h_2 - h_1} \]  

(A-2)

Gas Horsepower, ghp

\[ H P_{GAS} = \frac{m * H_p}{\eta_p} \]  

(A-3)

Specific Speed, NS

\[ NS = \frac{N \sqrt{Q}}{H^{3/4}} \]  

(A-4)

Polytropic Head Factor, \( \delta \)
given compression ratio. A molecular weight that is too low will also cause the calculated polytropic efficiency to be to higher, though not as pronounced. The net result is similar to a flowmeter that is reading high, except that the calculated maximum polytropic head and efficiency will be higher than the maximum shown on the performance curve (Figure B-2). In contrast, if the flow rate is measured in mass units, the corrected mass flow rate will be lower because the molecular weights are inverted in Equation (5). However, the corrected volume flow rate will still be higher because the molecular weight is used to correct the mass flow into volumetric flow. Note, these are the effects produced by an incorrect gas composition being used as the input for a field test. These are not the results of a compressor that is operating in a gas that is actually lower in molecular weight than its design.

\[ Q_{\text{INCORRECT}} = Q_{\text{ACTUAL}} \frac{M_{\text{W \, ACTUAL}}}{M_{\text{W \, INCORRECT}}} \]  

(B-1)

If the measured gas composition is too high, the results are just the inverse of above (i.e., the flow, head, and efficiency are all lower).

**Off-Design Operation**

Many times a perceived performance problem is actually just a compressor that is operating far from its design point. The most common cause of off-design operation is a change in gas composition (i.e., mole weight). The inlet Mach number is directly proportional to the molecular weight of the gas (Equation (18)). Centrifugal pumps produce the same amount of head regardless of the fluid specific gravity. In contrast, centrifugal compressors produce more head if the inlet Mach number (i.e., mole weight) is higher, and less if it is lower (Figure B-3). Likewise, as the molecular weight increases, the operating range decreases. Off-design operation can also be a result of variations in rotational speed and inlet temperature. However, the changes in speed are usually obvious and since absolute temperature is used in Equation (5), it takes a large change to significantly affect the inlet Mach number. There are many instances where a change in inlet temperature does cause an off-design operation, but it is usually due to the secondary effect that the molecular weight of the gas has changed. For example, the overhead vapor from a fractionator tower is usually cooled by a fin-fan exchanger before it enters the suction drum of the compressor (Figure B-4). If the exchanger is overloaded (as is commonly the case), the inlet temperature to the suction drum will fluctuate with ambient temperature. This causes the molecular weight to be higher when the ambient temperature is higher because less liquid is knocked out (i.e., the gas contains more high mole weight components). Likewise, the system shown in Figure B-4 is susceptible to surge at lower ambient temperatures if the compressor operates close to its surge point. As the inlet temperature drops, so does the inlet volume flow and molecular weight. If the compressor is already head limited, it can cause the compressor to go into surge.

**REFERENCES**

Figure B-4. Typical Petrochemical Fractionator Overhead Gas System.


BIBLIOGRAPHY


