PRACTICAL METHODS FOR FIELD PERFORMANCE TESTING CENTRIFUGAL COMPRESSORS

by
Ed Wilcox
Senior Staff Engineer
Conoco, Inc.
Westlake, Louisiana

Ed Wilcox is a Senior Staff Engineer with Conoco, Inc., in Westlake, Louisiana, where he is the Reliability Engineer for Excel-Paralubes, a joint venture between Conoco and Pennzoil. He also provides technical support for the adjacent Conoco Refinery. His responsibilities include maintenance, troubleshooting, and specification of rotating equipment.

Mr. Wilcox received his B.S.M.E. degree from the University of Missouri-Rolla and M.S.M.E. degree from Oklahoma State University. He is also doing post graduate work at the Georgia Institute of Technology in the areas of machinery vibration, lubrication, and rotordynamics. He is a Vibration Institute certified Level III Vibration Specialist and a registered Professional Engineer in the State of Oklahoma.

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 invaluable (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

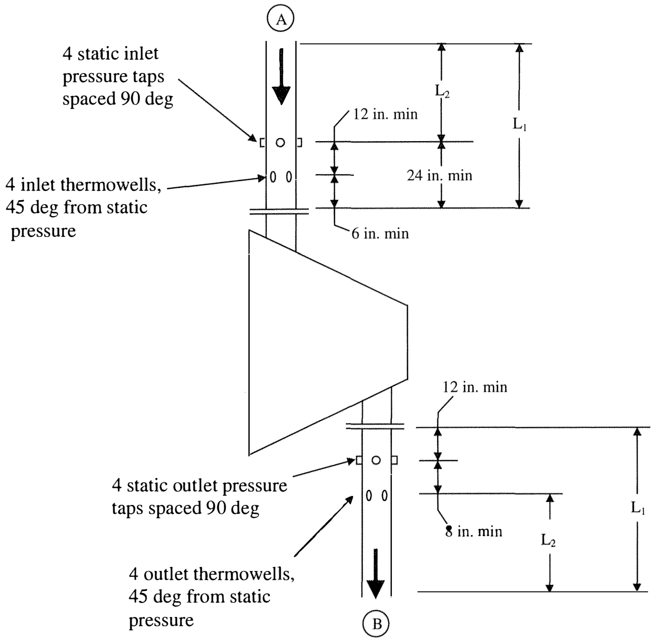
Additionally, other nontraditional data can be helpful in diagnosing the source of a performance problem. These include, but are not limited to:

- 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.

$$P_{TOTAL} = P_{STATIC} + P_{VELOCITY} = P_{STATIC} + \frac{V^2}{2g_c\bar{v}} \quad (1)$$



	Inlet		Outlet	
Piping Config. at A or B	**L_1**	**L_2**	**L_1**	**L_2**
Straight Run/Elbow	3D	2D	3D	2D
Reducer	5D	3D	5D	3D
Valve	10D	8D	5D	3D
Flow Device	5D	3D	10D	8D

Figure 1. Pressure and Temperature Locations.

Table 1. Pressure Instrumentation Accuracy.

Instrument	Accuracy (% error)
Pressure transmitter – Analog mode	0.75% of range
Pressure transmitter – Digital mode	0.1% of value or 0.75% of range
500 psi oil filled pressure gauge	10-25 psi

TEMPERATURE

Contrary to popular belief, temperature is just as important a factor in calculating compressor performance as pressure, because the enthalpy of a gas is a much stronger function of temperature than pressure. Likewise, it is more difficult to obtain accurate temperature measurements due to the slow response nature of temperature and the boundary layer effect in piping. The process temperature is averaged among several temperature readings at different circumferential locations around the pipe (Figure 1) when compressors are tested at the OEM shops according to ASME PTC 10 (1997). This is not feasible in the average petrochemical facility. The most important considerations in the field are the accuracy/calibration of the temperature sensing device, its location, and installation. The two most common devices used to measure temperature are thermocouples and resistance temperature detectors (RTD). RTDs are more accurate than thermocouples (Table 2); however, thermocouples have been more widely used due to ruggedness. If RTDs are used, they *must* be either three or four wire compensated. This eliminates error in the temperature reading due to the resistance in the wires leading to and from the RTD. If thermocouples are used, select a single E or T type for both suction and discharge temperature to eliminate any error caused by thermocouple drift. The sensor should be located in a thermowell that extends at least one-third of the way into the piping. Likewise, conductive grease/paste or spring loading should be used to provide good conduction from the thermowell to the sensor. Similar to pressure, the total temperature is required in the performance calculations. The total temperature is found by adding a portion of the velocity temperature to the static temperature (Equation (2)). Normally the difference between static and total temperature is minimal due to low velocities in the piping.

$$T_{TOTAL} = T_{STATIC} + 0.35{*}T_{VELOCITY} = T_{STATIC} + \frac{0.35{*}V^2}{2g_c c_p} \quad (2)$$

Table 2. Temperature Sensor Accuracy.

Sensor	Error
RTD (100 ohm platinum)	± 0.3%
E Type Thermocouple (0 to 652°F)	± 3.0°F or 0.5%
K Type Thermocouple (0 to 2250°F)	± 4.0°F or 0.75%
J Type Thermocouple (0 to 1350°F)	± 4.0°F or 0.75%
T Type Thermocouple (0 to 660°F)	± 1.8°F or 0.75%

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 ±5°F
- T_2 = 200 ±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 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).

$$Q = K\sqrt{\Delta P} \tag{3}$$

where:

K = f (P, T, Z, Mw)

$$Q_{s,c} = Q_s \sqrt{\left(\frac{P}{P_D}\right)\left(\frac{T_D}{T}\right)\left(\frac{Mw_D}{Mw}\right)\left(\frac{Z_D}{Z}\right)} \tag{4}$$

$$\dot{m}_c = \dot{m} \sqrt{\left(\frac{P}{P_D}\right)\left(\frac{T_D}{T}\right)\left(\frac{Mw}{Mw_D}\right)\left(\frac{Z_D}{Z}\right)} \tag{5}$$

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 a 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.

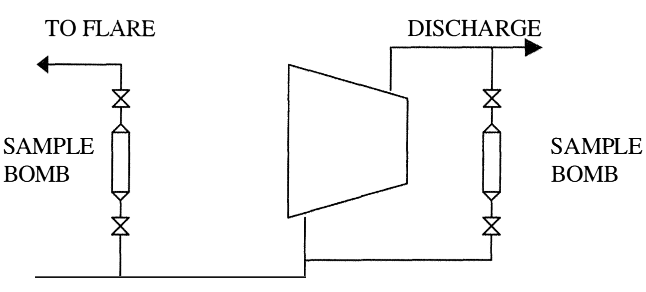


Figure 2. Gas Sampling Examples.

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

	Correct Gas Sample (275°F)	Incorrect Gas Sample (75°F)
Molecular weight	48	34
Flow (MMscfd)	27.4	32.5
Shaft horsepower (hp)	7036	5775

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:

$$\dot{m}_1 = \dot{m}_2 + \dot{m}_3 = \dot{m}_3 + \dot{m}_5 + \dot{m}_6 = \dot{m}_3 + \dot{m}_6 + \dot{m}_8 + \dot{m}_9 \tag{6}$$

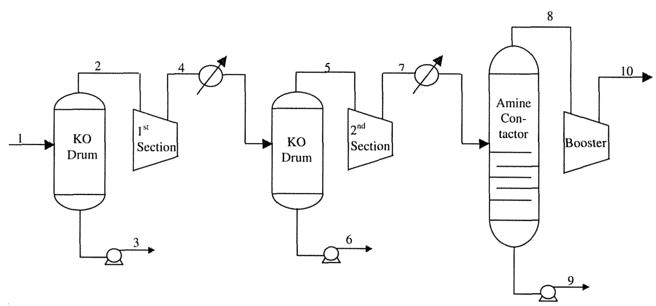


Figure 3. Multisection Wet Gas Compressor.

Likewise, the mass fraction of noncondensables such as H_2 and methane (CH_4) should be constant.

$$\dot{m}_{1,H_2} = \dot{m}_{2,H_2} = \dot{m}_{4,H_2} = \dot{m}_{5,H_2} = \dot{m}_{7,H_2} = \dot{m}_{8,H_2} = \dot{m}_{10,H_2} \tag{7}$$

In addition, the mass fraction of H_2S, 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 H_2S. The result would be that:

$$\dot{m}_{1,H_2S} = \dot{m}_{2,H_2S} = \dot{m}_{5,H_2S} \quad \text{and} \quad \dot{m}_{8,H_2S} = \dot{m}_{10,H_2S} = 0 \tag{8}$$

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 HYSIM™ 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.

$$\mathrm{SHP} = \frac{(\dot{m}_1 - \dot{m}_{S1})(h_2 - h_1) + Q_R}{2545} + \mathrm{HP}_{MECH} \qquad (9)$$

$$\mathrm{SHP} = \mathrm{HP}_{GAS} + \mathrm{HP}_{MECH} \quad , \quad \mathrm{HP}_{MECH} = \mathrm{HP}_{M1} + \mathrm{HP}_{M2} \qquad (10)$$

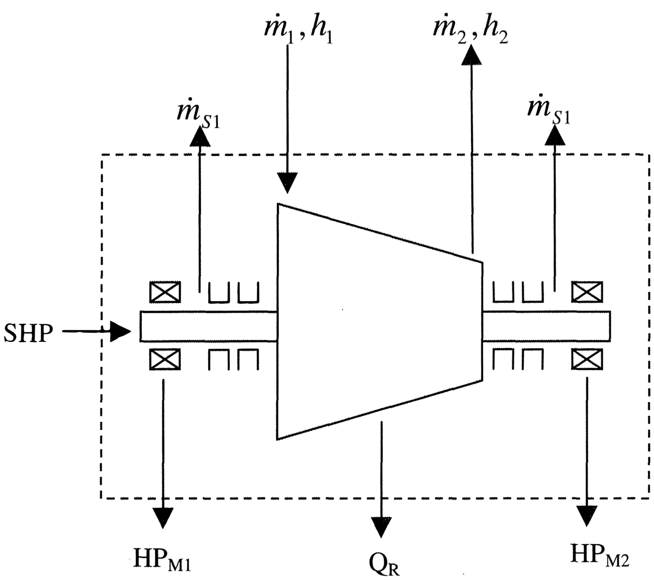


Figure 4. Control Volume Around Compressor.

The radiant heat loss (Q_R) is normally negligible, but it can be approximated by dividing the compressor case into axial sections and approximating the heat transfer from each section. Likewise, the seal leakage on the inlet is normally less than 1 percent, but it can be easily calculated at the orifice in the vent off the seal pots. Note that the internal seal losses (i.e., balance piston and impeller labyrinth seals) are not included in Equation (9). However, they do affect the calculated head and efficiency, which in turn determines the discharge pressure and temperature. The effects of seal losses will be discussed later.

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 (H_2S, CO_2) often require a maximum yield stress of 90 kpsi 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.

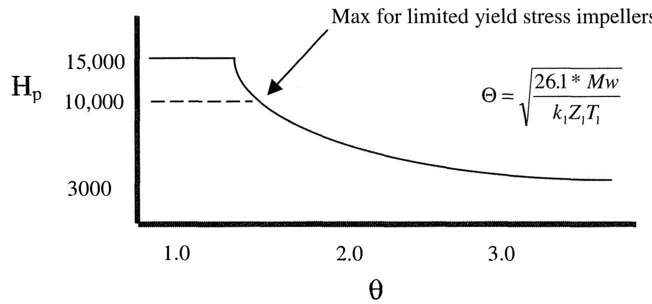


Figure 5. Maximum Polytropic Head Per Impeller.

Polytropic Efficiency

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

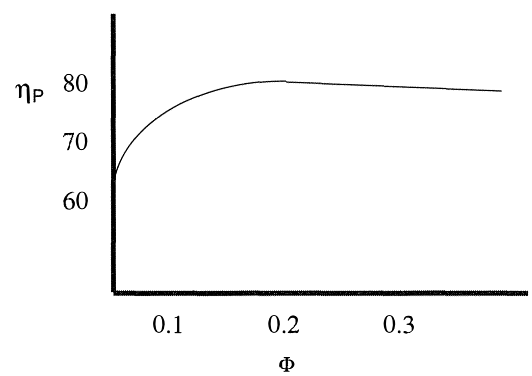


Figure 6. Maximum Polytropic Efficiency Per Impeller.

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.

$$HP_{mech} = \frac{\dot{m} c_p \Delta T_{oil}}{33000} \quad (11)$$

- OEM supplied curves for different bearings and seals (Figures 7 and 8)
- Bearing rotordynamic computer models that calculate horsepower losses as well
- Tables based upon compressor gas horsepower (Table 4) (Lapina, 1982). Table 5 gives approximate gearbox efficiencies

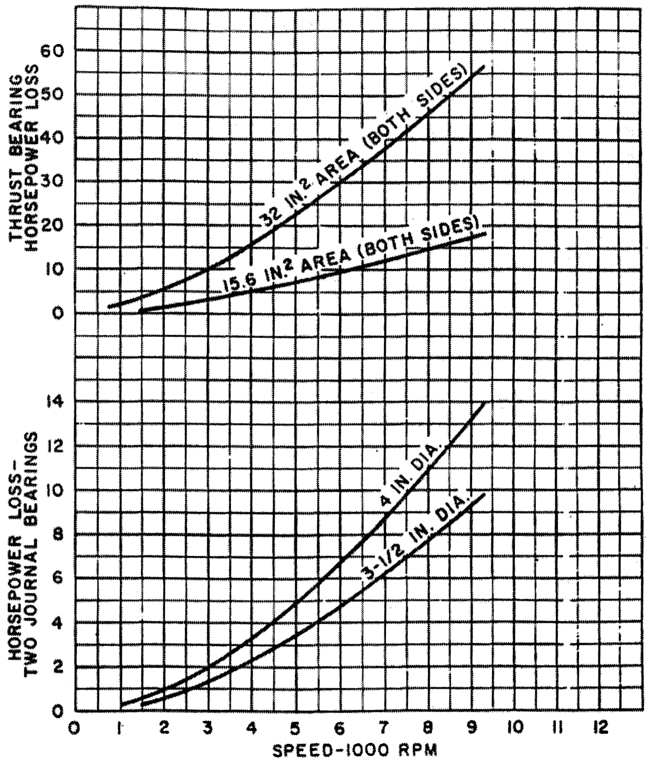


Figure 7. Mechanical Losses for Journal and Tilting Pad Thrust Bearings. (Courtesy of Elliott Co.)

Seal Losses

Balance piston or division wall leakage is the only seal loss evaluated in this discussion since they are usually much larger than impeller labyrinth seal leakage. Seal losses are much more difficult to estimate than mechanical losses because they are not just added to the calculated gas horsepower. Leakage through the balance piston seal to the compressor suction (Figure 9) increases the volume flow through the impellers as well as increases the inlet temperature, both of which decrease the discharge pressure of the compressor (Figure 10). Balance piston leakage causes the calculated head and efficiency to decrease. In reality, because seal losses cause a decrease in head and efficiency, the net result is an increase in compressor horsepower required to maintain the same volume flow rate and discharge pressure (i.e., speed is increased or throttle valve is opened). A feel for the amount of balance piston leakage can be established by monitoring other parameters as well.

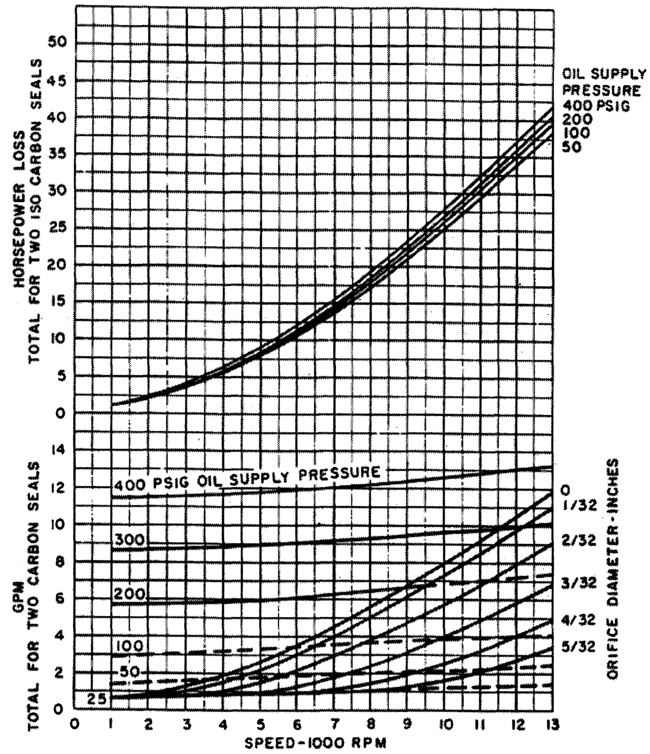


Figure 8. Mechanical Losses for Face Contact Oil Seals. (Courtesy of Elliott Co.)

Table 4. Mechanical Losses.

Gas Power Requirement	Mechanical Losses (%)
0-3000	3.0
3000-6000	2.5
6000-10,000	2.0
10,000 +	1.5

Table 5. Gearbox Efficiencies.

Gear Type	Efficiency (%)
Helical	97-99
Herringbone	96-99
Straight bevel	95-98
Spiral bevel	96-98

- *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).

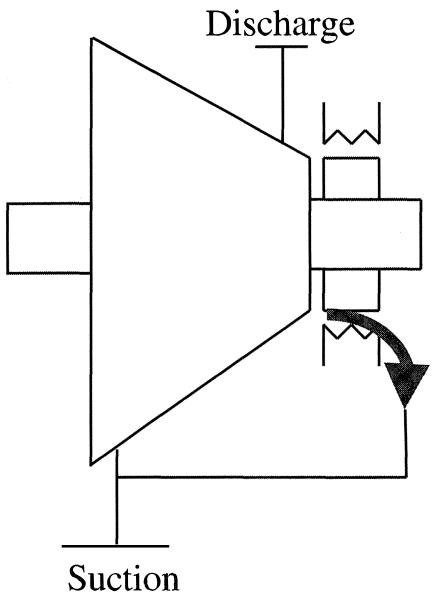


Figure 9. Balance Piston Leakage in Straight-Through Compressor.

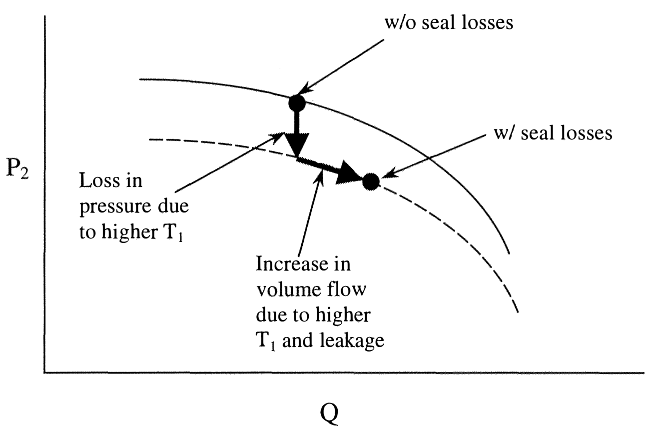


Figure 10. Effect of Balance Piston Leakage on Discharge Pressure.

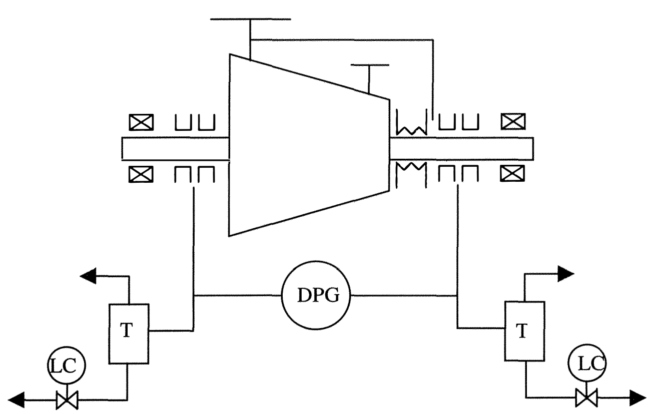


Figure 11. Measuring Balance Piston Differential Pressure Using Seal Oil Traps.

• *Thrust position and thrust bearing temperature*—An increase in thrust position and/or thrust bearing temperature is also an indication of balance piston problems, since the balance piston counters some of the normal thrust load created by the impellers. Of course the thrust position and thrust bearing temperature are 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.

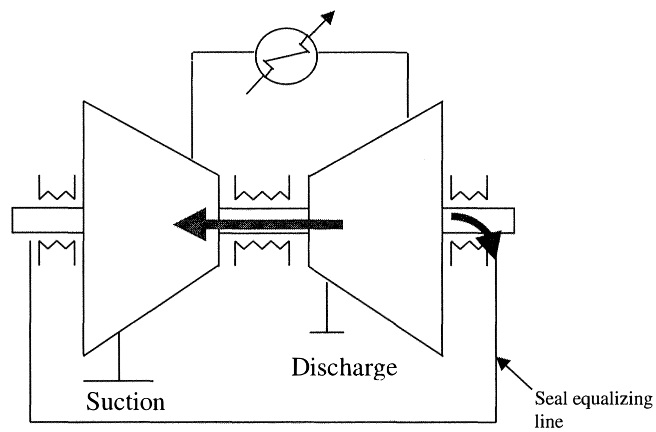


Figure 12. Division Wall Leakage in a Back-to-Back Compressor.

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).

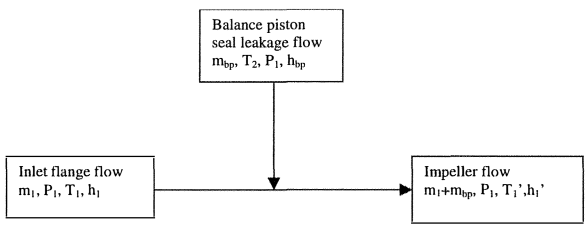
$$\dot{m} = K{*}P_2{*}\frac{\sqrt{1-\left(\frac{P_1}{P_2}\right)^2}}{\sqrt{\frac{Z_2{*}T_2}{Mw}}} \quad \Rightarrow \quad K = \frac{\dot{m}{*}\sqrt{\frac{Z_2{*}T_2}{Mw}}}{P_2{*}\sqrt{1-\left(\frac{P_1}{P_2}\right)^2}} \tag{12}$$

• *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).

$$\dot{m} = \mu_0 \mu_1 H \sqrt{\frac{P_{i-1}^2 - P_i^2}{ZRT}} \tag{13}$$

$$\dot{m}_{NT} = \frac{0.51\mu_0 P_{NC} H}{\sqrt{ZRT}} \tag{14}$$

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 causes 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.



where

$$h_1' = \frac{m_1 h_1 + m_{bp} h_{bp}}{m_1 + m_{bp}}$$

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

Title	Hydrogen Recycle Compressor				
		1st Stage			
Test Data				**Corrected Test Data**	
Suction Conditions:		**Discharge Conditions:**		Inlet	Outlet
Press (psia)	153	256.0		153.0	256.0
Temp(F)	88	190.0		92.7	190.0
Flow	0 MMSCFD	(only one flow is required)			
	0 ICFM				
	2507 lbm/min				
Flowmeter	1 Location (1 - suction, 2 - discharge)				
design data	0 Mole weight				
	0 Temp (F)				
	0 Press(psia)				
Speed	7940 rpm				
Impeller Number	Diameter(in)				
	6 23				
CALCULATED DATA					
Mole weight	7.2				
Properties	**Inlet**	**Outlet**		**Inlet**	**Outlet**
Compressibility	1.0027	1.0060		1.0028	1.0060
Enthalpy (Btu/lbm)	559.0	676.4		564.4	676.4
specific volume (ft^3/lbm)	5.33	3.79		5.37	3.79
Specific heat (Btu/lbm-F)	1.132	1.178		1.134	1.178
cp/cv	1.33	1.31		1.33	1.31
Entropy(Btu/lbm-F)	3.68	3.73		3.69	3.73
	Flow (corrected)				**Flow(correcte**
MMSCFD	189.5				
ICFM	13353.2		Bal Pist flow (lbm/min)		119.0
lbm/min	2507.0		Impeller flow (lbm/min)		2626.0
Q/N	1.682		Impeller q/n		1.777
	Performance				**Performance (corrected)**
Polytropic exponent	1.51				1.47
Polytropic head	65997				66273
Polytropic head coeff	0.557				0.560
Polytropic wf	1.000				1.000
Polytropic effy	0.72				0.76
Gas horsepower	6937				6937
Shaft horsepower	7076				7076
Mach no	0.356				0.358
Volume ratio	1.406				1.418

Figure 14. Calculated Test Data.

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, μ_P

$$\mu_P = \frac{Hp}{(\pi DN)^2} \tag{15}$$

Polytropic efficiency, η_P

$$\eta_P = \frac{H_p}{h_2 - h_1} \tag{16}$$

Flow coefficient, Φ

$$\Phi = \frac{Q}{N^*D^3} \quad \text{or} \quad = \frac{Q}{N} \tag{17}$$

Inlet Mach number, M

$$M = \frac{\pi DN}{\sqrt{k_1 Z_1 RT_1}} \tag{18}$$

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 nondim 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 Φ, which moves the predicted operating point further to the right on the performance curves and always increases the horsepower.

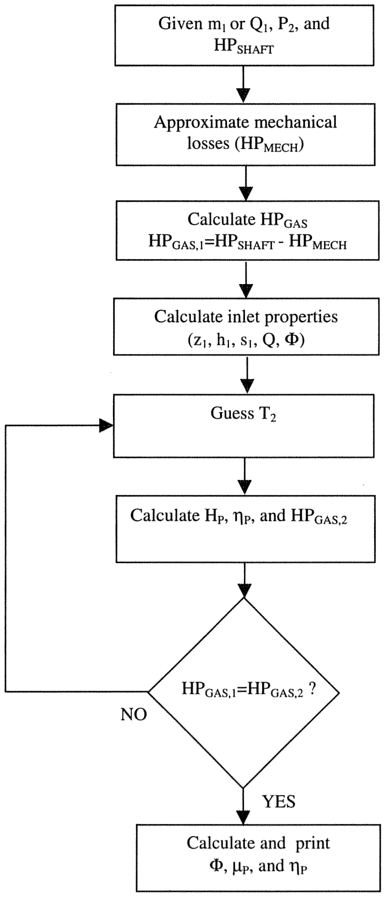


Figure 15. Flowchart for Obtaining Nondim Parameters (μ_p, η_p) from Dimensional Data (P_2, HP_{SHAFT}).

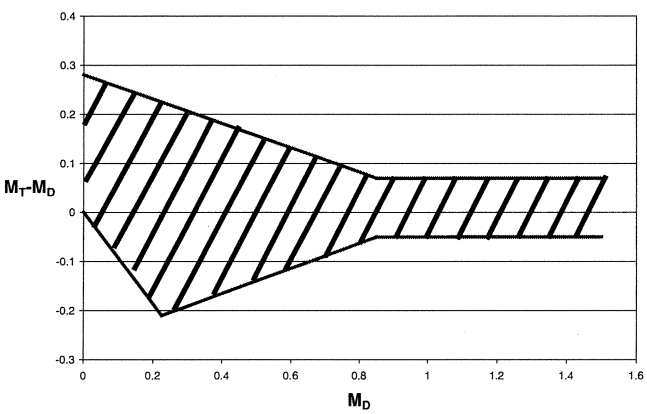


Figure 16. Allowable Shift in Test Mach Number from Design Mach Number.

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

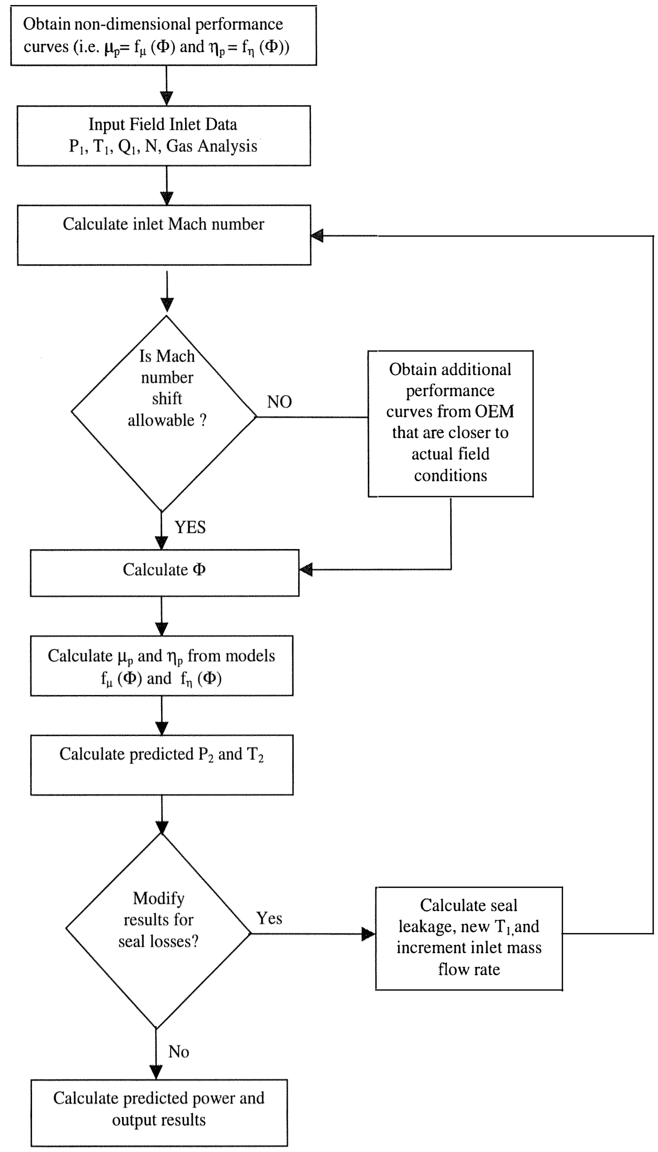


Figure 17. Flowchart for Repredicting Compressor Performance.

- Original design
  - 6370 hp
  - 7940 rpm
  - Inlet flow – 14,600 cfm
  - Molecular weight – 5.18
  - T_1 = 100°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°F
  - P_2 = 248 psia
  - T_2 = 191°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:

$$HP_{MOTOR} = \frac{VIePf}{431} = \frac{4000*978*0.957*0.92}{431} = 7988 \text{ hp} \quad (19)$$

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

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

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 repredicted 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

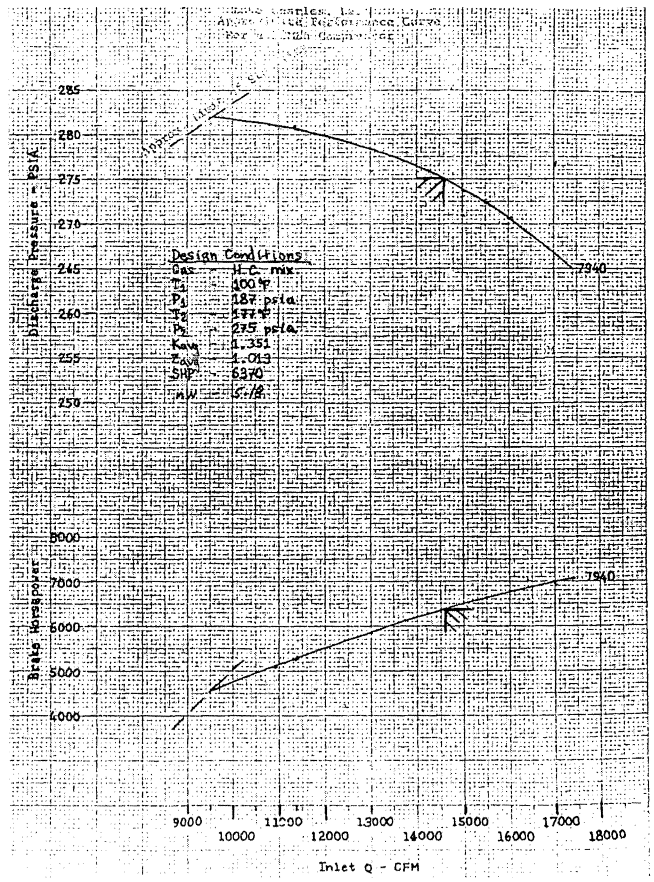


Figure 18. OEM Supplied Predicted Performance Data.

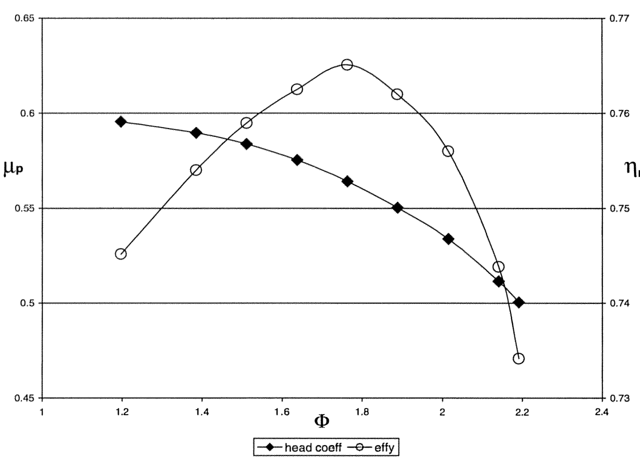


Figure 19. Predicted Nondimensional Polytropic Head Coefficient and Efficiency.

Table 6. Reformer Hydrogen Recycle Compressor Performance Before Cleaning.

Parameter	Measured Field Performance	Predicted Performance (w/o seal losses)	Predicted Performance (w/ seal losses)
P2	248	258	252
T2	191	176	182
Head (lbf-ft/lbm)	64,504	68,898	66,417
Efficiency (%)	65	79	76
Shp	7647	6729	7030

Title	Hydrogen Recycle Compressor					
Test Data			**Corrected Test Data**		**Predicted Data**	
Suction Conditions:		**Discharge Conditions:**	Inlet	Outlet	Inlet	Discharge
Press(psia)	150	248.0	150.0	248.0	150.0	252.1
Temp(F)	80	191.0	84.9	191.0	84.6	182.1

Flow	190 MMSCFD (only one flow is required)	
	0 ICFM	
	0 lbm/min	
Flowmeter design data	1 Location (1 - suction, 2 - discharge)	
	0 Moleweight	
	0 Temp(F)	
	0 Press(psia)	
Speed	7940 rpm	7940
Impeller Number	6	
Diameter(in)	23	

CALCULATED DATA

Mole weight 7.2

Properties	**Inlet**	**Outlet**	**Inlet**	**Outlet**	**Predicted Inlet**	**Predicted Outlet**
Compressibility	1.0026	1.0059	1.0026	1.0059	1.0026	1.0059
Enthalpy (Btu/lbm)	556.1	683.4	561.6	683.4	561.3	673.0
specific volume (ft^3/lbm)	5.39	3.94	5.44	3.94	5.44	3.83
Specific heat (Btu/lbm-F)	1.129	1.175	1.131	1.175	1.130	1.172
cp/cv	1.33	1.32	1.33	1.32	1.33	1.32
Entropy(Btu/lbm-F)	3.76	3.84	3.77	3.84	3.77	3.82

	Flow (corrected)		**Flow(corrected)**	**Flow (predicted)**
MMSCFD	190.0			
ICFM	13456.9	Seal losses (lbm/min)	114.0	120.1
lbm/min	2496.0	Impeller flow (lbm/min)	2610.0	2616.1
Q/N	1.695	Impeller q/n	1.789	1.792

	Performance	**Performance (corrected)**	**Predicted Performance**
Polytropic exponent	1.61	1.56	1.48
Polytropic head	64509	64795	66417
Polytropic head coeff	0.545	0.547	0.561
Polytropic wf	1.000	1.000	1.000
Polytropic effy	0.65	0.68	0.76
Gas horsepower	7497	7497	6892
Shaft horsepower	7647	7647	7030
Mach no	0.357	0.359	0.355
Volume ratio	1.367	1.379	1.421

Figure 20. Example Test and Repredicted Field Performance Data.

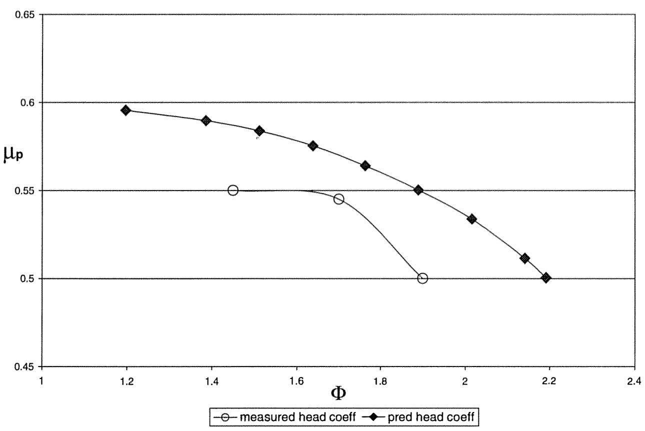


Figure 21. Predicted and Measured Polytropic Head Coefficient.

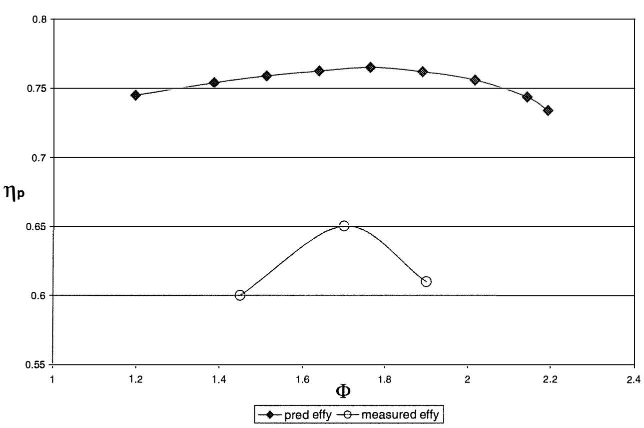


Figure 22. Predicted and Measured Polytropic Efficiency.

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.

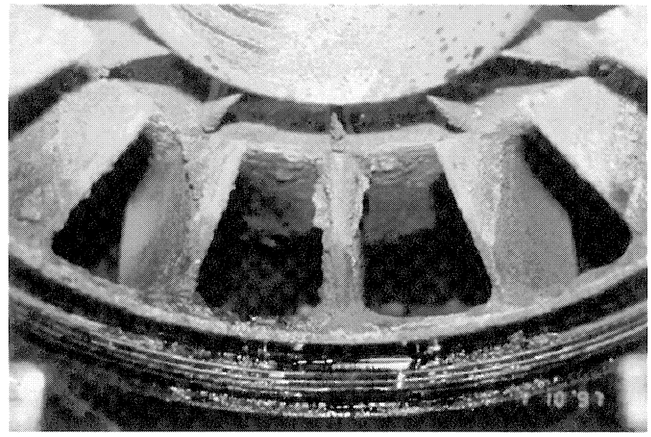


Figure 23. Fouled Inlet Guide Vanes on Hydrogen Recycle Compressor.

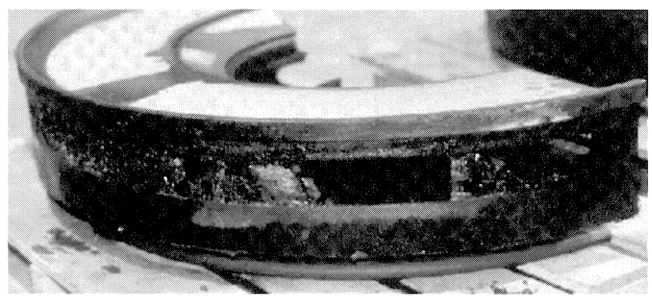


Figure 24. Diaphragm Half from Hydrogen Recycle Compressor Showing Fouled Diffuser.

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.

NOMENCLATURE

C_p	=	Specific heat, constant pressure (Btu/lbm-F)
C_v	=	Specific heat, constant volume (Btu/lbm-F)
D	=	Impeller diameter (in)
h	=	Specific enthalpy (Btu/lbm)
H	=	Head (lbf-ft/lbm)
HP	=	Horsepower
k	=	Specific heat ratio (c_p/c_v)
K	=	Meter factor
m	=	Mass flow (lbm/s)
M	=	Mach number (nondim)
MMscfd	=	Million standard cubic feet per day
Mw	=	Mole weight (lbm/lb-mole)
N	=	Speed (rpm)
N_S	=	Specific speed (nondim)
P	=	Pressure (psia)
Q	=	Volume flow (acfm)
Q_S	=	Volume flow (scfm)
Q_R	=	Radiative heat transfer (Btu)
R	=	Gas constant
T	=	Temperature (F)
V	=	Gas velocity, ft/s
v	=	Specific volume (ft^3/lbm)
Z	=	Compressibility

Subscripts

1	=	Inlet conditions
2	=	Outlet conditions
bp	=	Balance piston
D	=	Design conditions
MECH	=	Mechanical
M	=	Mechanical
P	=	Polytropic state
s	=	Isentropic state
S	=	Corresponds to seal conditions

Symbols

Δ	=	Differential
η	=	Efficiency (nondim)
θ	=	Sonic velocity ratio (nondim)
μ	=	Head coefficient (nondim)
$\mu_{0,1}$	=	Labyrinth seal entrance coefficient (nondim)

APPENDIX A
PERFORMANCE PARAMETERS

Polytropic Head, H_p

$$H_P = \delta * P_1 * \bar{v}_1 * \gamma_P * \left\{ \left(\frac{P_2}{P_1} \right)^{1/\gamma} - 1 \right\} \quad \text{(A-1)}$$

Polytropic Efficiency, η_p

$$\eta_P = \frac{H_p}{h_2 - h_1} \quad \text{(A-2)}$$

Gas Horsepower, ghp

$$HP_{GAS} = \frac{m * H_P}{\eta_P} \quad \text{(A-3)}$$

Specific Speed, NS

$$NS = \frac{N\sqrt{Q}}{H^{3/4}} \quad \text{(A-4)}$$

Polytropic Head Factor, δ

$$\delta = \frac{h_s - h_1}{\left(\frac{\gamma_s}{\gamma_s - 1} \right) \left(p_2 \bar{v}_s - p_1 \bar{v}_1 \right)} \quad \text{(A-5)}$$

Polytropic Exponent, γ_p

$$\gamma_p = \frac{\ln \left(\frac{P_2}{P_1} \right)}{\ln \left(\frac{\bar{v}_2}{\bar{v}_1} \right)} \quad \text{(A-6)}$$

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

$$\gamma_p = \frac{\ln \left(\frac{P_2}{P_1} \right)}{\ln \left(\frac{T_2}{T_1} \right)} \qquad \eta_p = \frac{\gamma_P}{\left(\frac{k}{k-1} \right)} \qquad \text{(ideal gases only)} \quad \text{(A-7)}$$

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 disturbs 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

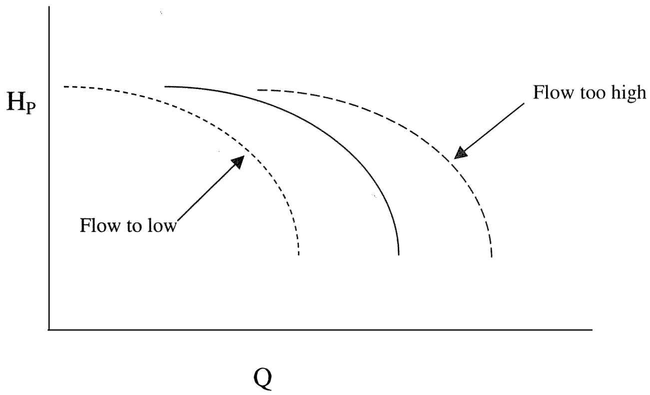


Figure B-1. Effects of Incorrect Flow Measurement.

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 convert 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.

$$Q_{INCORRECT} = Q_{ACTUAL} \sqrt{\frac{Mw_{ACTUAL}}{Mw_{INCORRECT}}} \qquad \text{(B-1)}$$

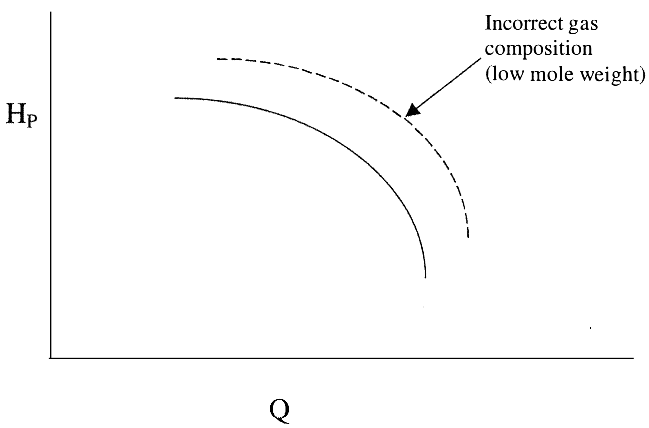


Figure B-2. Effects of Incorrect Gas Composition (Low Molecular Weight).

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.

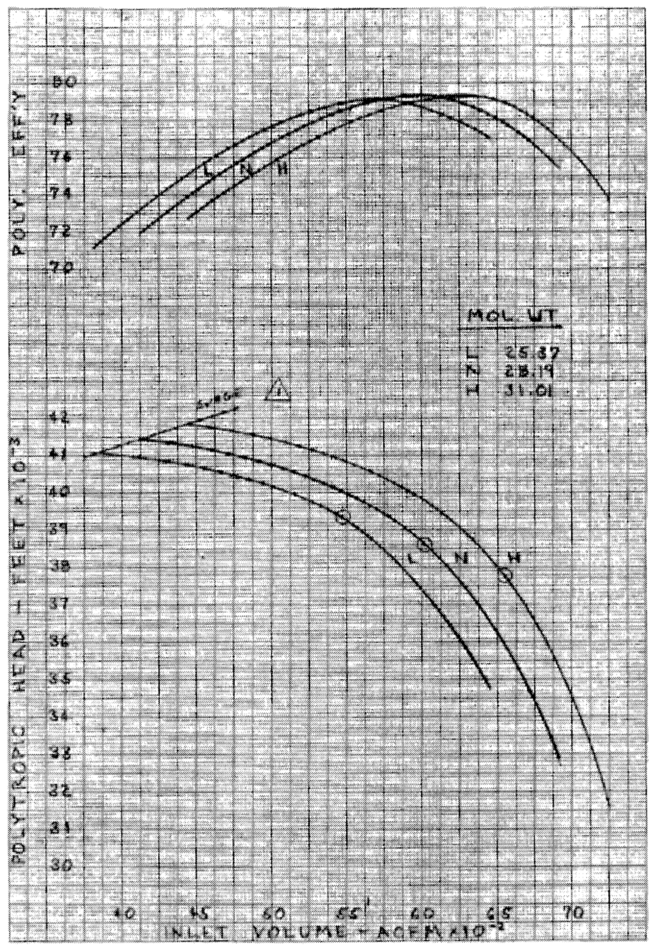


Figure B-3. Effect of Gas Molecular Weight on Head and Efficiency.

REFERENCES

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

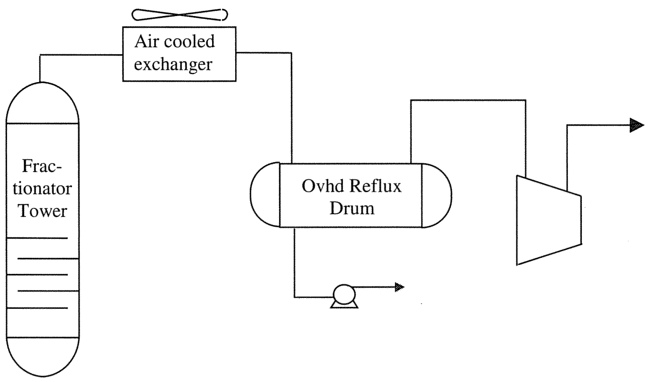


Figure B-4. Typical Petrochemical Fractionator Overhead Gas System.

Childs, D. W., 1993, *Turbomachinery Rotordynamics*, New York, New York: J. C. Wiley.

Lapina, R. P., 1982, *Estimating Centrifugal Compressor Performance*, Houston, Texas: Gulf Publishing.

Lee, B. I. and Kesler, M. G., May 1975, "A Generalized Thermodynamic Correlation Based on Three-Parameter Corresponding States," AIChE Journal, *21*, (3), pp. 510-527.

BIBLIOGRAPHY

Aiche, W., 1993, "Test of Process Turbocompressors without CFC Gases," *Proceedings of the Twenty-Second Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 179-184.

Bartos, J. C., 1990, "Predicting Centrifugal Compressor Performance from Impeller Dimensional Data," Power Machinery and Compression Conference, Houston, Texas.

Boyce, M. P., 1993, "Principals of Operation and Performance Estimation of Centrifugal Compressors," *Proceedings of the Twenty-Second Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 161-175.

Clark, W. A., 1981, "Measure Power by the Simplest Method and Use for Onstream Computer Monitoring," *Proceedings of the Tenth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 151-153.

Davis, H. M., 1973, "Compressor Performance, Field Data Acquisition, and Evaluation," *Proceedings of the Second Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 15-22.

Gresh, M. T., 1991, *Compressor Performance—Selection, Operation, and Testing of Axial and Centrifugal Compressors*, Boston, Massachusetts: Butterworh-Heinemann.

Heftmann, E., Ed., 1975, *Chromatography—A Laboratory Handbook of Chromatographic and Electrophoretic Methods*, New York, New York: Van Nostrand Reinhold Co.

Hill, D. J., 1995, "Centrifugal Compressor Rerates and Redesigns," Power Machinery and Compression Conference, Houston, Texas.

Hunt, J. W., 1981, "Field Testing of Centrifugal Compressors," *Proceedings of the Tenth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 157-164.

Japikse, D., 1982, "Efficiency: The Silent Partner of Machine Performance," *Proceedings of the Eleventh Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 105-111.

Kerlin, T. W. and Katz, E. M., April 1990, "Temperature Measurement in the 1990s," INTECH, pp. 40-43.

Matthews, T., 1981, "Field Performance Testing to Improve Compressor Reliability," *Proceedings of the Tenth Turbomachinery Symposium,* Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 165-167.

Minco Product Catalog, 1997, "Bulletin TS-102," Minco Products, Inc.

Nathoo, N. S. and Gottenberg, W. G., 1981, "Measuring the Thermodynamic Performance of Multistage Compressors Operating on Mixed Hydrocarbon Gases," *Proceedings of the Tenth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 15-22.

Pais, R. B. and Jacobik, G. J., 1974, "Field Performance Testing of Centrifugal Compressors," *Proceedings of the Third Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 84-90.

Schutlz, J. M. "The Polytropic Analysis of Centrifugal Compressors," Journal of Engineering for Power, *1*, pp. 69-83.

VanLaningham, F. L., 1981, "Guidelines for Performance Testing Centrifugal Compressors," *Proceedings of the Tenth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 169-171.