19

COMPRESSOR PERFORMANCE TESTING*

There are many instances when compressor performance must be determined or verified. The fulfillment of contractual obligations may have to be ascertained, or field performance evaluations may be needed, for reasons such as pinpointing the causes of performance degradation. Accordingly, this chapter offers guidelines for performance testing of dynamic compressors. Three case histories, as well as relevant procedures, equations, and sample calculations are included. Also, the first three segments of this chapter convey the (relative) complexity of rigorous performance testing and demonstrate at least one available computer tool. This material is supplemented in Section 19.4 by shortcut methods of predicting compressor performance under "new" conditions those that differ from conditions originally specified and designed.

19.1 PERFORMANCE TESTING OF NEW COMPRESSORS

Completing a performance test on a new purchase is the only way that purchasers can be sure that they getting what they paid for and that the compressors will do the job as specified. For new equipment, a shop test at the factory is most appropriate. Although field testing of new machines is, of course, feasible, the compressor is now far from the factory and out of the hands of the original equipment manufacturer (OEM). Any problems will be the purchaser's to settle. With test equipment not accurately calibrated and the location of test instruments rarely in harmony with the applicable testing code [generally, the widely used ASME Power Test Code 10 (PTC 10)], there can be arguments over the accuracy of field tests. The reliability professional is expected to get the plant running, not resolve compressor problems.

* Sections 19.1 through 19.3 were contributed by M. T. Gresh, Flexware, Inc., Grapeville, PA.

A Practical Guide to Compressor Technology, Second Edition, By Heinz P. Bloch

That said, it might be best to keep the machine at the factory until the owner/purchaser can be certain that he or she is getting the results desired.

19.1.1 Re-rate Options

Re-rating a centrifugal compressor refers to modification of compressor internals so as to achieve a performance different from that for which the machine was originally designed or purchased. For re-rates (see also Section 19.4), the issue of testing is entirely different from that of new machines. Here, a user/owner generally buys "bits and pieces," which differ from case to case. Parts to be changed may include impellers, stationary gas passage components, shaft, seals, bearings, and sometimes even couplings. The machine is opened during an extended turnaround and the new parts are installed. There is rarely an opportunity to ship the machine to the manufacturer for this refurbishing; hence, it will not be possible to test the machine in the factory under controlled PTC 10 conditions. However, a field acceptance test can be done with reasonable accuracy. Compromises will be necessary to accommodate field conditions, but a good test is feasible as long as some special considerations are observed.

The scheduling of maintenance outages and preventive maintenance based on field performance testing is a relatively new practice. Many users are doing continuous on-line monitoring to track compressor performance for various reasons. Not only can field performance analysis be helpful for predicting when maintenance is needed, but certainly, knowing where the compressor is operating on the curve can be beneficial in preventing failures. It stands to reason that knowing compressor performance details is the first step in considering a re-rate. Moreover, online monitoring can help operators optimize throughput or plant production by giving them instant feedback. Having retained or obtained a good compressor performance history can be a powerful aid to resolving compressor problems.

19.1.2 General Guidelines

A number of general guidelines are worth listing.

- All performance testing should be in accordance with ASME PTC 10. Although shop tests must be in strict adherence with this procedure, following it to the letter is generally possible only at the manufacturer's shop; it certainly is not practical for field testing. Even so, it is recommended that PTC 10 be followed as closely as practical to assure accurate results.
- Compressor shaft horsepower is determined by adding the enthalpy rise gas horsepower to standard values of bearing and seal mechanical power losses.
- Mass flow is determined by using the system flowmeter. Mass flow is checked by direct calculation from flowmeter upstream conditions and differential pressure using applicable ASME flow code equations.
- Test points should not be taken until such time as compressor operation is shown to be in equilibrium. *Equilibrium* is defined as the condition in which the discharge temperature does not vary more than 1°F over a 5-minute period at constant inlet conditions.
- Upon achieving equilibrium, three complete scans of data readings per data point are taken over a 20- to 30-minute period and averaged for calculations.
- It is recommended that a minimum of five data points be taken to establish the performance curve shape. Take one point at the design or normal operating point at 95% of this value, 105%, 110%, and 90%. Also, confirm the surge and choke points.

- A gas sample (or purity check) should be completed at the beginning and end of the test points. Take precautions to avoid condensation in the sample bottle. However, it is wise to sample at both suction and discharge. The method of gas analysis recommended is gas chromatography.
- Test accuracy should be checked by comparing test work input to predicted work input. Also, complete a power balance on the equipment string if at all possible. Measured driver-delivered power minus gear (if applicable) power losses can be compared to calculated compressor-absorbed power. The overall accuracy of the test is no better than the power balance.
- The liquid wash injection system, if provided, should be shut down 30 minutes prior to each test data scan. This is required to exclude any quenching effect on the gas and its discharge temperature.

19.1.3 Gas Sampling

Be sure to follow proper precautions when taking a gas sample.

- A stainless steel sampling cylinder should be used. It should be at least 300 mL in size and have straight cylinder valves on both ends. The pressure rating of the cylinder should be high enough so that it can withstand full system pressure. Be sure to check the containers for leaks before using. Helium is a good medium for this leak check.
- Take a gas sample from each compressor nozzle (inlet and discharge for each section) before and after taking compressor data (flow rate, speed, and nozzle pressures and temperatures). Condensate can form on the gas sample container (gas bomb) walls and give erroneous results unless the container is heated. The empty gas bomb should be heated to the temperature of the gas being sampled or higher during the sampling process. Purge the container with the process gas thoroughly before closing the valves and trapping the gas at process pressure.
- The containers are then transported to the laboratory. To minimize the effects of any undetected leaks, it is best not to delay and process the sample as soon as possible. During the transportation the samples cool and therefore drop in pressure. The cooling of the sample may result in condensation of some of the gas. This condensation must be gasified before feeding into the gas chromatograph. The only sure way to do this is to reheat the gas sample to the sampling temperature. At this point it may be wise to then bleed off some of the pressure before feeding the gas chromatograph. This brings you further away from the dew point and provides further assurance of avoiding condensation.
- Be sure that you confirm values by comparing the discharge gas analysis to the inlet gas analysis for the same section. The accuracy of your test results is no better than the agreement between your gas analysis results.
- A cross-check on accuracy can be made by checking the weight of a separate sample. This sample container should be at vacuum and heated before filling with the process gas. Weigh the sample container evacuated and also with the sample. Knowing both weight and volume will give you the specific volume. Check this against the specific volume calculated for the temperature and pressure of the sample point using the composition given by the gas chromatograph.

19.1.4 Instrumentation

General instrumentation requirements for test are detailed in Fig. 19.1. ASME PTC 10-1997 requires four probes in each location. For a shop test, this cannot be compromised. However for a field test, typical instrumentation consists of a single element in each location. If a field acceptance test is being conducted, dual elements at 90° are suggested to offer some redundancy and to check for flow swirl.

- For each section, upstream pressure, upstream temperature, and pressure differential measurements are required at each flowmeter.
- All pressures at the compressor flanges and the primary flowmeter are measured using instruments having a minimum sensitivity of $\frac{1}{4}$% and a maximum error of $\frac{1}{2}$% full scale.
- All pressure-measuring instruments are selected to operate at midscale or greater at the test values expected. Mount the instruments on a vibration-free local panel, connected to the process sensing point using instrument pressure lines at least $\frac{1}{4}$ in. in diameter. The instrument lines should slope continuously down toward the process sensing point to eliminate the possibility of condensation filling the line. Install block and vent valves at each instrument to facilitate in-place calibration. Calibration using a certified and traceable device is preferred.

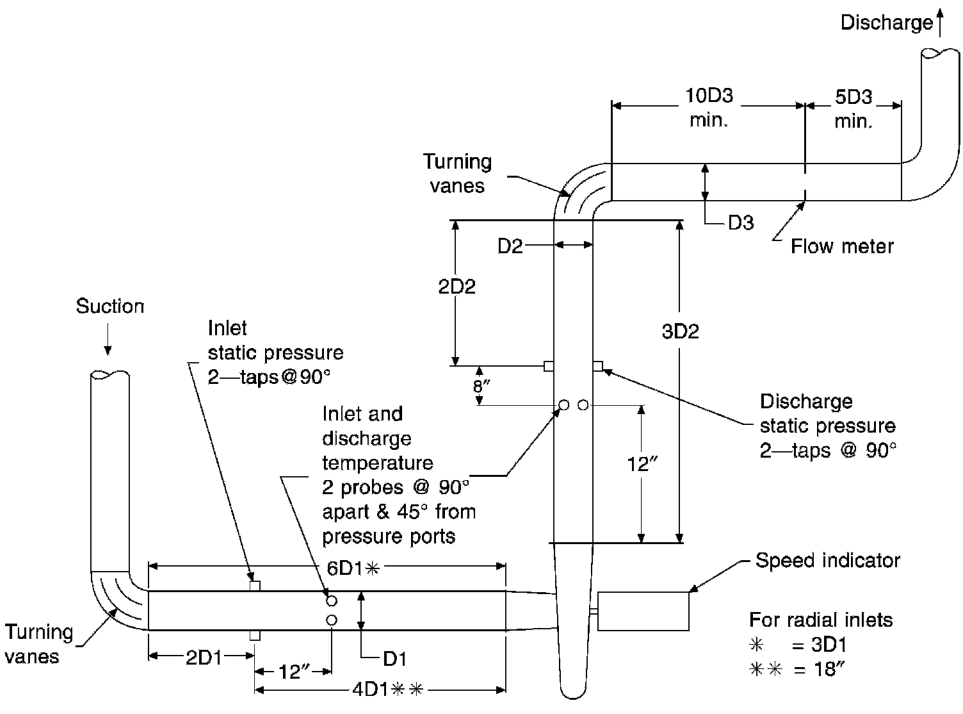

FIGURE 19.1 Typical instrumentation suggested for a field acceptance test. PTC 10-1997 requires four probes in each location, whereas most field installations have only one. Although a single instrument per location is OK for routine monitoring, it is important to confirm the accuracy of the instruments as well as the overall test results. If a field acceptance test is being conducted, dual elements at 90° are suggested to offer some redundancy and to check for flow swirl. (From Ref. 1.)

- The preferred static pressure connection at the process piping must have a pressure tap hole no greater than 0.5 in. in diameter, deburred, and smooth on its inside edge.
- On horizontal runs of pipe, pressure taps should be in the upper half of the pipe only.
- Temperatures should be measured using a thermocouple or RTD system having a minimum sensitivity of $\frac{1}{2}$°F and an accuracy within 1°F. If a thermocouple system is used, care should be taken to avoid intermediate junctions at terminal or switch boxes.
- The temperature-sensing portion of the probe must be immersed into the flow one-third to one-half of the pipe diameter.
- The temperature-sensing element should be in intimate thermal contact if using wells. A suitable heat transfer filling media, such as graphite paste, should be used. Stem conduction errors can be minimized by wrapping the stem and well with fiberglass or rockwool insulation.
- Speed should be determined utilizing two independent systems, one being a Keyphasor on the shaft with a digital readout having ¼% full-scale accuracy.
- Compressor flow for each section at each test point should be determined by direct computation of mass flow rates through the flowmeter using the measured values for upstream pressure, upstream temperature, and meter differential pressure. Thermodynamic properties for the process gas will be based on the gas samples taken for the particular section and test point.
- In general, all instruments used for the measurement of temperature, pressure, and speed should be calibrated prior to test. The calibration of all instruments will be subject to witness and approval by all parties prior to starting the test. Hard copy records of all instrument calibrations must be prepared and will become part of the formal test record. Instruments that malfunction during the test will be replaced with instruments having current calibration. All instruments will be subject to a posttest calibration check at the discretion of either party.
- Pressure instruments are best calibrated by comparison to a certified standard throughout the instrument scale or range. The calibration should be conducted for both increasing and decreasing signals to determine hysteresis. The calibration record should state both actual standard value and indicated instrument value. Instruments that do not demonstrate an accuracy of $\frac{1}{2}$% full scale are not to be used.
- Calibration of temperature-measuring systems should include all components of the system including probe, lead wire, reference junction (if applicable), and readout. Each temperature-measuring system should be calibrated by subjecting it and a certified reference standard concurrently to varying temperatures in a thermostatically controlled oven or oil or sand bath.

Calibrations are to be conducted on both increasing and decreasing signals over the expected operating range for test. The calibration record should state both actual standard value and indicated instrument system value. Temperature systems that do not demonstrate agreement within 1% of the reference standard throughout the calibration range should not be used.

- The orifice meters are pulled. The actual flow-metering element to be used must be checked dimensionally prior to test. Records of the dimensional check are to be made and compared to design standards. The dimensional record is subject to approval by all parties prior to starting the test. Make sure that the sharp edge is still sharp and clean. Clean and/or replace as necessary. Orifice plates must be replaced if there is any evidence of wear.

- Readouts used to measure compressor speed should be checked by input of a signal from a certified frequency-generating device. Readouts that do not demonstrate a minimum accuracy of $\frac{1}{4}$% of full scale are not to be used.
- Blow down all instrument lines to assure that there is no liquid or other blockage in the lines.

19.1.5 Sideload Compressors

As the term implies, sideload compressors are machines where a portion of the gas flow enters somewhere after the first impeller or first stage of compression. The preferred instrument locations for sideload compressors are shown in Fig. 19.2. Sideload and extraction lines, if applicable, are to be treated as inlet and discharge lines, respectively. If existing instrument tapping points must be used, care must be taken that those used are reasonably close to the compressor flanges. There must be no valves, strainers, silencers, or other sources of significant pressure drop between the pressure tap points and the compressor flange.

Evaluation of sideload and extraction compressor performance requires internal pressure and temperature probes at the sideload and/or extraction. Although this is nearly impossible for a field evaluation test, it is typically done for shop testing. For field tests, special data reduction techniques can be used where internal pressures can be estimated

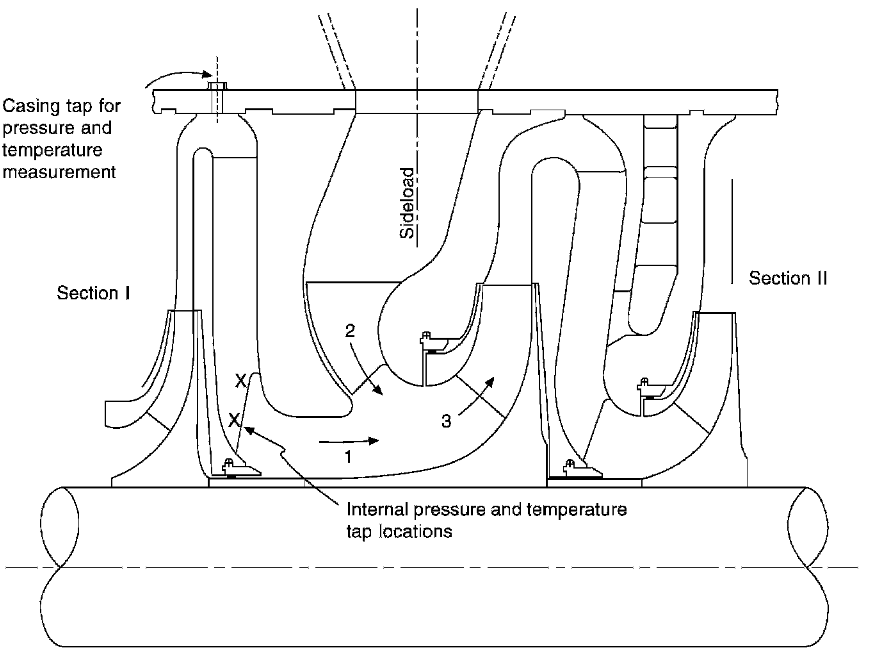

FIGURE 19.2 Pressure and temperature taps for sideload compressors. Although internal measurements are not practical for field installations, the casing tap may be considered. Be sure that the thermal element is installed deep enough that it will not be affected by heat transfer to the casing. (From Ref. 1.)

from flange pressures, gas velocity through the compressor nozzle, and standard pressure drop loss coefficients for a given sideload or extraction nozzle design.

Internal gas temperature at the discharge of each section is also required to determine sectional performance. This can be accomplished through an iterative process which makes use of predicted work input curves for each section. The procedure begins for a given test point by establishing the inlet volume flow for section 1. From the work curves predicted, the estimated work input is obtained. These data, along with the internal pressure determined above, are used to establish the estimated discharge temperature for section 1. A BWR (Benedict–Webb–Rubin) gas properties program such as Gas Flex is best used for this procedure. The sideload flow is then mixed with the discharge flow calculated from section 1 to establish the inlet flow to section 2. This procedure is then repeated for each following compressor section using its respective work input curve. The test on the validity of the work input curves is made by comparing the calculated final discharge temperature to the measured discharge temperature. If these two temperatures agree, the assumption is made that the correct work input has been used. If, however, the two temperatures do not duplicate one another, the work input curves for each section are varied and the process repeated.

Once pressure and temperatures are known at the discharge of section 1, a mixing calculation is required to establish suction conditions for the next section (see Fig. 19.2).

$$P_1 = P_2 = P_3 \tag{19.1}$$

where the subscript 1 represents the discharge of section 1; 2, sideload condition, and, 3, the mixed suction to section 2.

$$\dot{M}_1 h_1 + \dot{M}_2 h_2 = \dot{M}_3 h_3 \tag{19.2}$$

where $\dot{M}_1 + \dot{M}_2 = \dot{M}_3$. T_3 is then found by working back through the gas properties or Mollier diagram knowing h_3 and P_3. T_3 may be approximated very accurately by

$$T_3 = \frac{\dot{M}_1 T_1 + \dot{M}_2 T_2}{\dot{M}_1 + \dot{M}_2} \tag{19.3}$$

Test Evaluation The objective of an acceptance test is to confirm that the compressor will provide the performance predicted for the "design" condition. In addition to proving the originally "as sold" design performance, field testing is useful for determining maintenance schedules and possible modifications as requirements change over time.

For an evaluation test, the compressor is to be operated at a speed to obtain the specified overall head for the compressor string (first main inlet to final discharge), and the process (or test loop) set to obtain the design inlet volume flow at design temperature, pressure, and MW at each compressor section inlet nozzle. All data must be collected via electronic "snapshots" for best accuracy to eliminate the effect of any minor variations in operating conditions and reading errors.

Once data at the design point are collected, the flow is to be increased to choke flow. The flow can then be reduced by a restriction on the discharge of the compressor until surge is detected. Data are collected at a location at the choke point near the surge point in a stable situation at design speed and several points in between (no less than five points total). It should be noted that interaction between sections on a sideload might make this impossible to complete properly for each section in a field test situation.

19.1.6 Calculation Procedures

In general, data reduction will employ the equations shown below. Performance parameters (head, efficiency, and inlet volume flow) must be calculated using Benedict–Webb–Rubin (BWR) equations of state based on the results of the on-site gas analysis. Performance parameters are calculated for each section of compression and/or an overall value based on flange data and internal data.

Polytropic Head

$$H_P = (h_2 - h_1) - \frac{(s_2 - s_1)(T_2 - T_1)}{\ln(T_2/T_1)} \tag{19.4}$$

The polytropic efficiency is given by

$$\eta = \frac{H}{778.16(h_2 - h_1)} \tag{19.5}$$

For sideload compressors,

$$\eta = \frac{H_{1-2}M_1 + H_{\mathrm{SL}1-2}M_{\mathrm{SL}1} + H_{\mathrm{SL}2-2}M_{\mathrm{SL}2}}{778.16[(h_2 - h_1)M_1 + (h_2 - h_{\mathrm{SL}1})M_{\mathrm{SL}1} + (h_2 - h_{\mathrm{SL}2})M_{\mathrm{SL}2}]} \tag{19.6}$$

Flow Measurement Basic flow measurement equations are as follows [2]. For square-edged orifices,

$$M = (5.983)K_4 d^2 (Fa) Y \sqrt{\frac{h_w}{v_1}} \tag{19.7}$$

where v_1 is the specific volume of fluid at inlet to orifice in cubic feet per pound. Gas horsepower is given by

$$ghp = 0.0236(h_2 - h_1)M \tag{19.8}$$

or, for sideload compressors,

$$ghp = 0.0236[M_1(h_2 - h_1) + M_{\mathrm{SL}1}(h_2 - h_{\mathrm{SL}1}) + M_{\mathrm{SL}2}(h_2 - h_{\mathrm{SL}2})] \tag{19.9}$$

Pressure Loss The following equation [3] will be used to determine the pressure drop between the compressor flange and the actual measurement point. This pressure drop will then be added (or subtracted) to the measured value to obtain the value at the flange:

$$\Delta P = 3.62 \frac{K_3 q^2}{v d^4} \tag{19.10}$$

Nomenclature

d = pipe diameter, in. (pressure loss)
d = diameter of orifice, in. (orifice flow)
Fa = thermal expansion factor, obtained from tables in Ref. 2
ghp = gas horsepower
h = enthalpy, Btu/lb
h_w = orifice differential pressure, in. H_2O
H = polytropic head, ft-lb/lb
K_4 = flow coefficient (orifice flow) (if the orifice has not been calibrated individually, obtain K from tables in Ref. 2)
K_3 = resistance coefficient (pressure loss)
M = mass flow rate, lb/min
P = pressure, psia
q = flow rate, ft^3/sec
R = gas constant $= \dfrac{1545}{\text{(molecular weight)}}$
s = entropy, Btu/lb-°F
T = temperature,°R
v = specific volume, ft^3/lb
W = work, ft-lb/lb
Y = net expansion factor for square-edged orifices (ratio of flow coefficient for a gas to that for a liquid at the same value of Reynolds number, obtained from tables or by equation in Ref. 2)
Z_1 = inlet compressibility factor
η = efficiency, polytropic

Subscripts:

1 = main inlet flange
2 = final discharge flange
SL1 = first sideload flange
SL2 = second sideload flange
D = design conditions

19.2 SHOP TESTING AND TYPES OF TESTS

The best way to ensure equipment quality is to specify and carry out compressor performance testing before the machine leaves the OEM factory (Fig. 19.3). Although the performance test can be done in the field once the compressor is installed, the quality of a field test is generally less than that of a shop test, and it is difficult to make corrections if necessary, as timing is always tight during the installation phase. This is true for all compressors, even "off-the-shelf" varieties. Although the design may be proven, parts can be mismachined or installed improperly. Custom-built compressors have the added risk of errors in application or design engineering.

The very well represented and universally accepted ASME Power Test Code (PTC 10-1997) has defined two types of performance tests:

- *Type 1.* This is a test run on the design gas near design conditions. This applies to air compressors, full load shop tests, and field tests.
- *Type 2.* When using the design gas for testing is not practical and a substitute gas is used for the test, a type 2 test is conducted. The gas used for the type 2 test does not have to follow the perfect gas laws.

The type 1 test is relatively simple to complete. Be certain of obtaining good data and complete calculations per PTC 10 (refer to Ref. 1). However, a type 2 test requires some

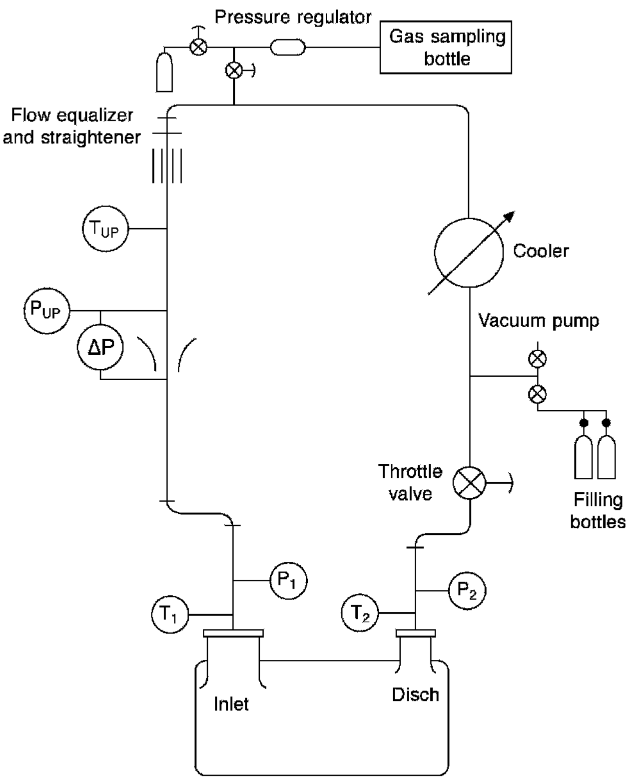

FIGURE 19.3 Typical shop test. (From Ref. 4, by permission.)

special considerations up front. In determining the proper equivalent conditions, the following three items must be reviewed: (1) volume ratio, (2) Mach number, and (3) Reynolds number. Unfortunately it is not always possible to match all three parameters with an equivalent test, so some compromise must be made.

On a multistage compressor, volume ratio is of utmost concern and must be closely matched. According to the API, the variation between test and actual values should not exceed 5%. At other than design volume ratio, downstream stages will not see the design flow and the overall curve will be affected. To assure that the test volume ratio equals the design volume ratio, use the following equation:

$$N_t = N_d \sqrt{\frac{Z_t \mathrm{MW}_d T_t}{Z_d \mathrm{MW}_t T_d}} \sqrt{\frac{[n/(n-1)]_t [(P_2/P_1)^{(n-1)/n} - 1]_t}{[n/(n-1)]_d [(P_2/P_1)^{(n-1)/n} - 1]_d}} \tag{19.11}$$

where N is the speed, N_d the design conditions, and N_t the test conditions. Note that N_t is proportional to $\mathrm{MW}_d/\mathrm{MW}_t$. Assuming a design MW of 70 and a test MW of 28 (other values fixed), the test speed would be much higher than the design speed. This normally puts the speed beyond the mechanical limits, demonstrating why air or nitrogen cannot be used for equivalent tests for a machine designed for compressing a heavy gas such as chlorine or propane.

The Mach number determines the capacity of the compressor. Therefore, it is critical to ensure that the Mach number is within 5% of the design Mach number. The equation for determining test speed to duplicate Mach number is

$$N_t = N_d \sqrt{\frac{(kgZRT)_t}{(kgZRT)_d}} \tag{19.12}$$

It is suggested that the test speed slightly exceeds the Mach number test speed for conservative results [4]. If possible, select a test gas with a k value near the design k value.

With the test speed set by volume ratio and Mach number, we are left with a variation in Reynolds number. According to ASME PTC 10-1997, the performance of a compressor is affected by the machine Reynolds number. Frictional losses in the internal flow passages vary in a manner similar to friction losses in pipes or other flow channels. If the machine Reynolds number at test operating conditions differs from that at specified operating conditions, a correction in the test efficiency is necessary to predict the performance of the compressor correctly. ASME PTC 10-1997 provides correction factors for variations in the Reynolds number.

For a type 1 test, the data are used directly to determine field performance. For a type 2 test, the following is needed to convert test data to actual field conditions.

Capacity

$$Q_d = Q_t \frac{N_d}{N_t} \tag{19.13}$$

where Q is the compressor suction flow and N is the speed.

Efficiency Since frictional losses in the compressor are a function of the machine Reynolds number, it is appropriate to apply the correction to the quantity $1 - \eta$. The magnitude of the correction increases as the machine Reynolds number decreases. The correction to be applied is as follows:

(a) For centrifugal compressors:

$$(1-\eta_p)_d = (1-\eta_p)_t \frac{RA_d}{RA_t}\frac{RB_d}{RB_t} \tag{19.14}$$

$$RA = 0.066 + 0.934\left[\frac{4.8\times 10^6 \times b}{\text{Rem}}\right]^{RC} \tag{19.15}$$

$$RB = \frac{\log(0.000125 + 13.67/\text{Rem})}{\log(\epsilon + 13.67/\text{Rem})} \tag{19.16}$$

$$RC = \frac{0.988}{\text{Rem}^{0.243}} \tag{19.17}$$

where Rem = machine Reynolds number, Ub/v'.

U = tip speed of first-stage impeller or first-stage axial blade, ft/s

b = blade height at the tip of the first-stage centrifugal impeller or the cord at the tip of the first-stage axial rotor blade, ft

v' = kinematic viscosity of the gas at compressor inlet conditions, ft^2/s

ϵ = average surface roughness of the flow passage, in.

(b) For axial compressors, the correction is a function of the machine Reynolds number ratio:

$$(1-\eta_p)_d = (1-\eta_p)_t \left(\frac{\text{Rem}_t}{\text{Rem}_d}\right)^{0.2} \tag{19.18}$$

The limitations of PTC 10-1997 apply.

Head The polytropic head coefficient is corrected for machine Reynolds number in the same ratio as the efficiency:

$$\text{Rem}_{\text{corr}} = \frac{\mu_{p_d}}{\mu_{p_t}} = \frac{\eta_{p_d}}{\eta_{p_t}} \tag{19.19}$$

or for polytropic head,

$$H_{p_d} = H_{P_t}\frac{\eta_d}{\eta_t}\left(\frac{N_d}{N_t}\right)^2 \tag{19.20}$$

19.3 FIELD TESTING

Before attempting a field performance test, review the following checklist and be certain that all the data required are available, preferably in electronic form. As a reference, see your OEM instruction book for design conditions.

- Vane settings
- Pressure and temperature at each flange
- Mass flow rate
- Gas properties
- Equipment speed
- Driver power
- Compressor and driver mechanical losses
- OEM performance curves

For a field acceptance test, it is recommended that dual instrumentation be considered seriously. This will offer some redundancy and make it possible to look for flow swirl. If nothing else, do a thorough job of calibrating instruments and be sure to confirm results by checking work input and completing a power balance.

For online monitoring, the process described above is recommended, but only time will tell how often it will be necessary to recalibrate. Monitoring the work input will go a long way in confirming accurate data.

Case History 1: Hydrogen Recycle Compressor Field Performance Analysis

As part of a debottlenecking procedure, a refinery in Oklahoma was interested in analyzing its hydrogen recycle compressor performance. The compressor string consisted of a small barrel compressor and a condensing steam turbine driver. Of primary concern for data accuracy were the flowmeters and obtaining an accurate gas analysis. Accurate data were a special concern since there is no confirmation of calculation results (the steam turbine is a condensing unit).

Special care was taken to ensure accurate data. Pressure differential data from the compressor gas flowmeter was read and used directly to calculate the flow rate to the compressor. The same was done for the turbine flowmeter. Multiple gas samples were taken to ensure redundancy. It was determined that the compressor was operating at about 71% efficiency (Figs. 19.4 and 19.5; Table 19.1), about 5 percentage points below predicted values. The turbine was operating at about 44% efficiency, about 10 percentage points below its predicted value. Although the data showed that the equipment needed maintenance to get it back to design operating conditions, the compressor was shown to be operating at midrange and there was no need for a re-rate.

Startup Following a major overhaul, all the piping and vessels are filled with air rather than the process gas. So when the compressor starts, it will be on air and eventually nitrogen, once all the air is purged. For operation, the effects will be the same because the MW for air and nitrogen is 28. However, this is very different from the process gas, which has a molecular weight of 3.6.

Power demand is the first consideration because the demand cannot be allowed to exceed the available power of the driver. Also, even with unlimited driver power, the increase in power (torque!) required for operation on nitrogen might well cause a shaft end failure if the machine were operated at the same speed and pressure. To operate the compressor on nitrogen, it is thus necessary to reduce both speed and pressure. Following is a means to estimate this:

$$\text{ghp}_p = \frac{H_p \dot{M}}{\eta_p (33{,}000)} \tag{19.21}$$

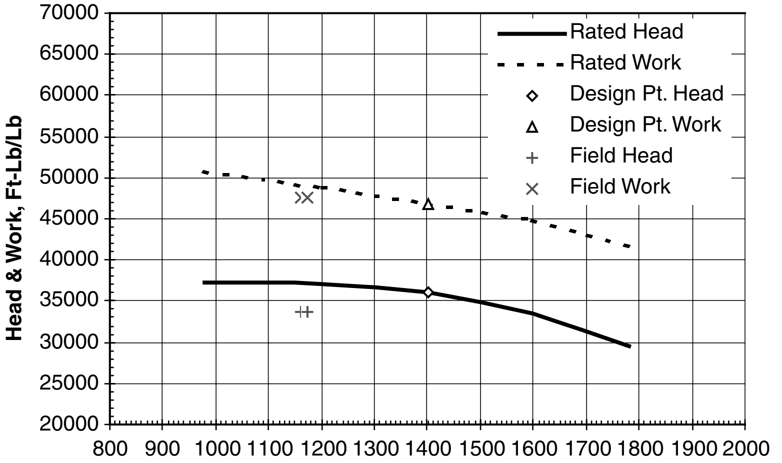

FIGURE 19.4 Hydrogen recycle compressor work and head vs. flow rate (icfm) data obtained during a field test (August 6, 2004). Data have been fan law-corrected to design speed (11,070 rpm) to permit direct comparison with OEM design curve.

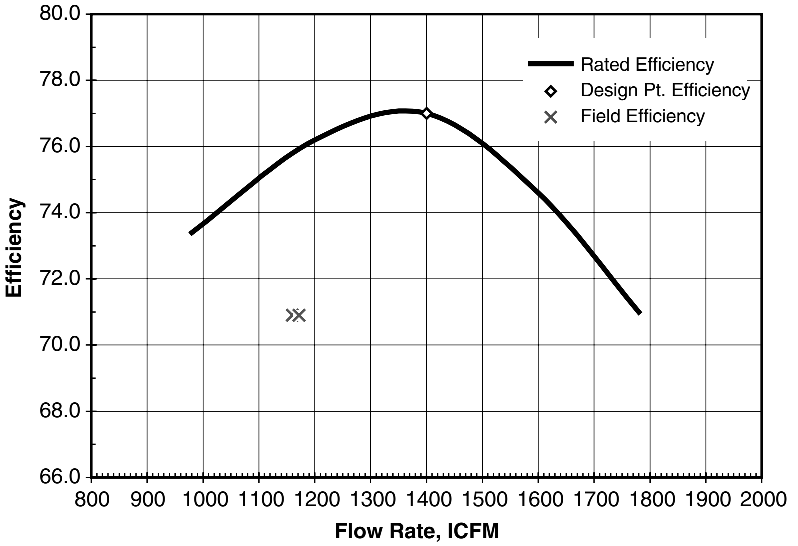

FIGURE 19.5 Hydrogen recycle compressor operating data for August 6, 2004. Data have been fan law–corrected to design speed, 11,070 rpm.

Also note that mass flow is roughly proportional to MW and pressure, so very approximately, we can say that

$$H_1 \mathrm{MW}_1 P_1 = H_2 \mathrm{MW}_2 P_2 \tag{19.22}$$

TABLE 19.1 Summary of Test Results: Hydrogen Recycle Operating Data for August 6, 2004[a]

			Gas	Mole Fraction	Formula
Compressor data					
Time	7:30 A.M.	10:30 A.M.	Hexane	0.0002	C_6H_{14}
Flow (MMscfd)	134	132	Hydrogen	0.92242	H_2
Orifice DP (in. H_2O)	19.65	18.9	Propane	0.00346	C_3H_8
Inlet T (°F)	112.5	114	i-Butane	0.0003	C_4H_{10}
Inlet P (psia)	1721	1724	n-Butane	0.00051	C_4H_{10}
Discharge T (°F)	143	144	Ethane	0.01788	C_2H_6
Discharge P (psia)	1962	1961	Nitrogen	0.0064	N_2
Speed (rpm)	11,289	11,289	Methane	0.04883	CH_4
				MW = 3.5766	
Compressor Results					
Flow (lb/min)	1139	1112			
Flow (icfm)	1195	1183			
Head (ft-lb/lb)	35,118	35,040			
Efficiency	70.9	70.9			
Power (hp)	1710	1665			
Turbine data					
Inlet P (psig)	588	589			
Inlet T (°F)	625	625			
Exhaust P (in. Hg vac.)	28.5	28.5			
Exhaust T (°F)	75	75			
Flow (1000 lbs/hr)	19.7	19.55			
Orifice DP (in. H_2O)	10.9	10.55			
Turbine results					
Flow (lb/hr)	21,568	21,240			
Efficiency	44.3	43.8			
Power (hp)	1710	1665			

[a] Note that the compressor power is identical to the steam turbine power. The compressor power was used as input data for the steam turbine calculation to determine the exhaust conditions of the condensing steam turbine.

If we decide to operate the compressor at 400-psia suction pressure while operating on nitrogen:

$$35{,}000(3.6)(1700) = H_2(28)(400)$$
$$H_2 = 19{,}125 \text{ ft of head}$$

To reduce the head on this compressor to 19,125, it is necessary to reduce the speed. Now consider the fan law equations:

$$H \propto N^2 \tag{19.23}$$

With this we can write

$$\frac{35{,}000}{19{,}125} = \left(\frac{11{,}289}{N_2}\right)^2 \qquad \text{hence, } N_2 = 8344 \text{ rpm}$$

So we will operate the compressor on nitrogen at 8344 rpm at 400-psia suction pressure and 100°F. Now estimate the discharge pressure:

$$\begin{aligned} P_2 &= \left[\frac{H_P}{Z_1 R T_1 [n/(n-1)]} + 1 \right]^{n/(n-1)} P_1 \\ &= \left[\frac{19{,}125}{1.0(55.18)(560)[1.36/(1.36-1)]} + 1 \right]^{1.36/(1.36-1)} (400) \\ &= 710\,\text{psia} \end{aligned} \tag{19.24}$$

Using the Gas Flex estimation calculation shown in Table 19.2, the power demand is calculated to be 1845 hp. Although still high, it is close to where we need to be. Operation at a slightly lower speed will bring the power down to a more conservative level.

Another variable to consider is the discharge temperature. Be sure that the discharge temperature does not exceed guidelines for the compressor. In this case the discharge temperature (~240°F) is relatively low and well within limits (Table 19.3).

To be sure, get back to the OEM for an accurate operating curve for startup conditions on nitrogen. The OEM can confirm if there are other limitations to consider at this off-design operating condition.

Case History 2: Impeller Failure on a Feed Gas Compressor

The importance of using calculation methods in failure analysis is demonstrated in this case history. A customer in northern Europe suffered an impeller failure approximately three years following the re-rate of the compressor used in this case history. The failure was on the first wheel in an intercooled compressor in a feed gas string. As part of the failure analysis, the compressor performance was reviewed to determine where the compressor was operating on the performance curve. Surge, choke, and liquid ingestion are known causes of impeller failures. Liquid can be detected by high work input.

Process and Installation The compressor installation was reviewed. Although there were a lot of elbows upstream of the inlet, this had no apparent effect on the compressor performance. As this was an up-nozzle configuration, there is a straight run of piping under the compressor where liquid could accumulate. Standard procedure is to drain any liquid before startup; however, this is not checked once the machine is running.

This unit has water injection to reduce the operating temperature. The polymer buildup in the compressor is very closely related to the gas temperature; thus, evaporation of the injected water reduces the operating temperature of the gas. The water injection nozzles have flowmeters installed, and operating flow rates were recorded for consideration of performance calculations (see Table 19.4 and 19.5).

Review of Compressor Operational History

Vibration The previous rotor (with the cracked impeller) had nominal vibration levels of about 35 μm. The present rotor is vibrating at approximately 70 μm. At startup the vibration level was about 45 μm.

Performance The present performance for compression section 1 of this machine is shown in Figs. 19.6 and 19.7. It was reported that vibration had been increasing rapidly,

TABLE 19.2 Gas Flex Straight-Through Compressor Test Results

Title:	H2 Recycle			
Database name:	H2 Recycle.gdb			
	Units	Inlet	Gas Composition	
Inlet flange data				
Pressure	psia	1724.	HEXA	0.0002
Temperature	°F	114.0		
Compressibility		1.0660	H_2	0.92242
Enthalpy	Btu/lb	971.6	C_3H_8	0.00346
Entropy	Btu/lb-°R	3.7604		
Specific volume	ft^3/lb	1.0641	IBUT	0.0003
$K(C_p/C_v)$		1.3677	BUTA	0.00051
K(temp. exp.)		1.3867		
K(vol. exp.)		1.5080	C_2H_6	0.01788
Specific heat (C_p)	Btu/mol-°R	2.0642	N_2	0.0064
Dynamic viscosity	lb/ft-sec	6.91E-06		
Sonic velocity	ft/sec	3580.1	CH_4	0.04883
Given flow	lb/min	1112.0	Total mole weight	3.5766
Mass flow	lb/min	1112.0		
Volume flow	ft^3/min	1183.3		
Discharge flange data				
Pressure	psia	1961.		
Temperature	°F	144.0		
Compressibility		1.0738		
Enthalpy	Btu/lb	1035.1		
Entropy	Btu/lb-°R	3.7918		
Specific volume	ft^3/lb	0.9917		
$K(C_p/C_v)$		1.3651		
K(temp. exp.)		1.3838		
K(vol. exp.)		1.5159		
Specific heat (C_p)	Btu/mol-°R	2.0751		
Dynamic viscosity	lb/ft-sec	7.16E-06		
Sonic velocity	ft/sec	3695.7		
Volume flow	ft^3/min	1102.7		
Total polytropic data				
Head	ft-lb/lb	35,040.0		
Efficiency		70.93		
Gas power	hp	1664.6		

Source: www.flexwareinc.com.

and according to plant records, the compressor efficiency had been getting lower as well. From this it was concluded that fouling (see Section 12.12.1) was probably occurring. A possible reason for the rapid fouling rate was a change in process gas composition that occurred at the time of the re-rate.

Liquid Ingestion A water injection system is used to reduce the compressor discharge temperature in an effort to minimize fouling. The liquid is injected into the main inlet and at the crossover between each stage. Prior to the re-rate, both water and hydrocarbon

TABLE 19.3 Gas Flex Straight-Through Compressor Estimation for Startup on Nitrogen

Title:	N2			
Database name:	N2 Compressor.gdb			

	Units	Inlet	Gas Composition	
Inlet flange data				
Pressure	psia	400.000	N_2	1
Temperature	°F	100.0	Total mole weight	28.0130
Given flow	icfm	1200		
Volume flow	ft^3/min	1200.		
Mass flow	lb/min	2255.8		
Compressibility		0.9927		
Min. flange diameter	in.	5.1		
Flange velocity	ft/sec	140.0		
Discharge flange data				
Pressure	psia	697.		
Temperature	°F	239.6		
Volume flow	ft^3/min	870		
Compressibility		1.0039		
Min. flange diameter	in.	4.4		
Flange velocity	ft/sec	140.0		
Total head data				
Head	ft-lb/lb	19,158		
Efficiency		71.00		
Gas power	hp	1845		

Source: www.flexwareinc.com.

liquids were used. However, following the re-rate, only water was used. Since the re-rate, no water has been injected in the main inlet, so the water injection cannot be a contributor to the impeller failure. The spray nozzles had a 20-bar pressure differential for good atomization of the water. Other possible sources of liquid were thought to include carryover of knockout liquids from the cooler.

This particular failure seemed to be unrelated to any existing known failure mode. It seemed unlikely that the cause was related to operation in choke flow or to overload. Operation was near the design point and the failure was on the first impeller. Yet the performance was to be monitored to confirm that the compressor remained within the limits of the performance curve. The customer was advised to monitor the vibration and performance of the compressor closely to find the source of the excitation that initiated the impeller cracking.

Effects of the Liquid Injection The effects of the liquid injection were calculated by considering the amount of water being injected and knowing the latent heat of vaporization. The customer was measuring the water flow using an orifice meter, and it was calculated to be 2600 lb/hr for this section. From Fig. 19.8, the latent heat of vaporization of the water is 898 Btu/lb.

The compressor gas mass flow rate was 421,116.8 lb/hr, and from Table 19-4 the discharge enthalpy is 255.2 Btu/lb.

$$421{,}116.8\ \text{lb/hr} \times 255.2\ \text{Btu/lb} = 107{,}469{,}001\ \text{Btu/hr}$$

TABLE 19.4 Gas Flex Calculation Summary of Field Test Results[a]

Title:	Boreallis F8239 18 Aug04
Database name:	F8239 12AUG04b.GFE

Gas Analysis: Total mol. weight = 23.27

Gas	Mol. Fr.	Gas	Mol. Fr.	Gas	Mol. Fr.	Gas	Mol. Fr.	Gas	Mol. Fr.
H_2	0.23420	CH_4	0.22100	C_2H_4	0.30060	C_2H_6	0.08200	C_2H_2	0.00410
C_3H_6	0.07050	C_3H_8	0.00670	PRPD	0.00270	H_2O	0.00980	IBUT	0.00150
IBTE	0.00710	IBYN	0.00500	12BU	0.01500	BUTA	0.00350	2M1B	0.00290
MEK	0.00190	IPRE	0.00120	IPEN	0.00090	NPEN	0.00260	MCYP	0.00620
CYH	0.00130	C_6H_6	0.01340	C_7H_8	0.00280	STYR	0.00310		

Test Point 1

Inlet data		
Pressure	abs. bar	3.347
Temperature	°C	25.7
Compressibility		0.9866
Enthalpy	Btu/lb	191.6
Entropy	Btu/lb-°R	1.7093
Specific volume	m^3/kg	0.3147
$K(C_p/C_v)$		1.2257
K(temp. exp.)		1.2360
K(vol. exp.)		1.2298
C_p	kJ/kmol · K	0.4631
Dynamic viscosity	mPa · s	1.030E-02
Sonic velocity	m/s	359.9
Given flow	kg/h	191,000.0
Mass flow	lb/hr	421,116.8
Volume flow	m^3/h	60,104
Discharge data		
Pressure	abs. bar	8.337
Temperature	°C	100.0
Compressibility		0.9859
Enthalpy	Btu/lb	255.2
Entropy	Btu/lb-°R	1.7378
Specific volume	m^3/kg	0.1576
$K(C_p/C_v)$		1.1933
K(temp. exp.)		1.2038
K(vol. exp.)		1.1979
C_p	kJ/kmol · K	0.5266
Dynamic viscosity	mPa · s	1258E-02
Sonic velocity	m/s	396.7
Volume flow	m^3/h	30,106
Overall polytropic data		
Head	N · m/kg	107,932
Efficiency	%	73.0
Power	kW	7848.9

Source: www.flexwareinc.com.

[a]Discharge temperature as measured in the field. Since water is being injected into the gas stream to reduce the operating temperature, the calculation must be compensated for the effect of the water. The operating efficiency shown is incorrect due to the effect of the evaporating water, as the calculation assumes dry gas.

TABLE 19.5 Gas Flex Calculation Summary of Field Test Results[a]

Title:	Boreallis F8239 18 Aug04
Database name:	F8239 12AUG04.GFE

Gas Analysis: Total mol. weight = 23.27

Gas	Mol. Fr.	Gas	Mol. Fr.	Gas	Mol. Fr.	Gas	Mol. Fr.	Gas	Mol. Fr.
H_2	0.23420	CH_4	0.22100	C_2H_4	0.30060	C_2H_6	0.08200	C_2H_2	0.00410
C_3H_6	0.07050	C_3H_8	0.00670	PRPD	0.00270	H_2O	0.00980	IBUT	0.00150
IBTE	0.00710	IBYN	0.00500	12BU	0.01500	BUTA	0.00350	2M1B	0.00290
MEK	0.00190	IPRE	0.00120	IPEN	0.00090	NPEN	0.00260	MCYP	0.00620
CYH	0.00130	C_6H_6	0.01340	C_7H_8	0.00280	STYR	0.00310		

Test Point 1

Inlet data		
Pressure	abs. bar	3.347
Temperature	°C	25.7
Compressibility		0.9866
Enthalpy	Btu/lb	191.6
Entropy	Btu/lb-°R	1.7093
Specific volume	m^3/kg	0.3147
$K(C_p/C_v)$		1.2257
K(temp. exp.)		1.2360
K(vol. exp.)		1.2298
C_p	$kJ/kmol \cdot K$	0.4631
Dynamic viscosity	$mPa \cdot s$	1.030E-02
Sonic velocity	m/s	359.9
Given flow	kg/h	191,000.0
Mass flow	lb/hr	421,116.8
Volume flow	m^3/h	60,104
Discharge data		
Pressure	abs. bar	8.337
Temperature	°C	105.7
Compressibility		0.9868
Enthalpy	Btu/lb	260.7
Entropy	Btu/lb-°R	1.7459
Specific volume	m^3/kg	0.1602
$K(C_p/C_v)$		1.1914
K(temp. exp.)		1.2013
K(vol. exp.)		1.1959
C_p	$kJ/kmol \cdot K$	0.5308
Dynamic viscosity	$mPa \cdot s$	1.277E-02
Sonic velocity	m/s	399.6
Volume flow	m^3/h	30,593
Overall polytropic data		
Head	$N \cdot m/kg$	108,915
Efficiency	%	67.8
Power	kW	8518.0

Source: www.flexwareinc.com.

[a] The discharge temperature has been adjusted to compensate for the evaporating water. Note the difference in the compressor efficiency between that in Fig. 19.7 and that shown in this table. The temperature change due to the water evaporation is 5.7°C.

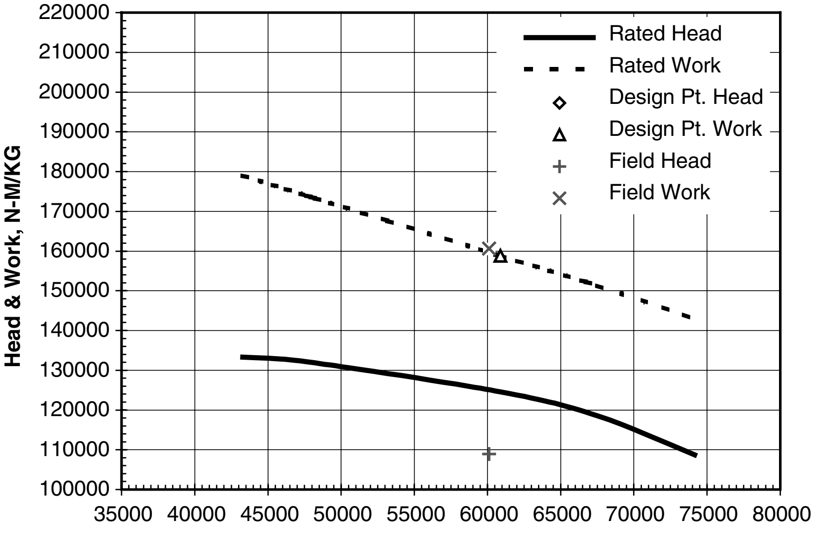

FIGURE 19.6 Performance test data for Case History 2, head and work vs. flow, cubic meters per hour.

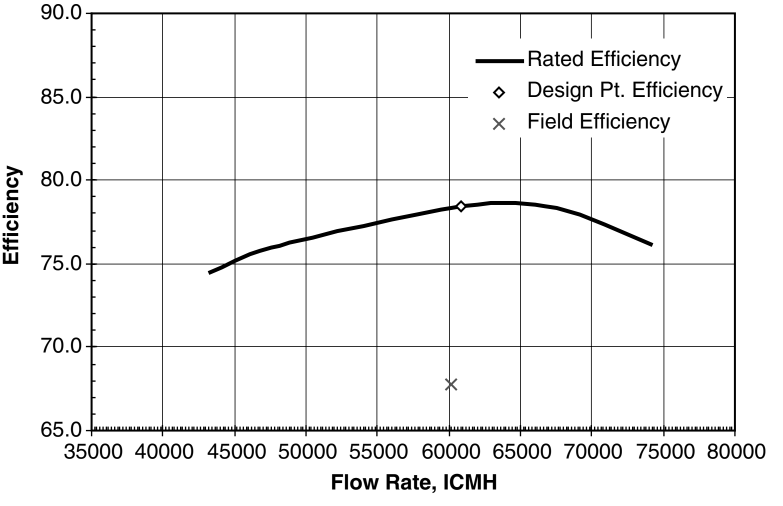

FIGURE 19.7 Compression section 1, performance for August 12, 2004.

The latent heat of vaporization is

$$2600 \text{ lb/hr} \times 898 \text{ Btu/lb} = 2{,}334{,}800 \text{ Btu/hr}$$

Add the latent heat of vaporization:

$$107{,}469{,}001 \text{ Btu/hr} + 2{,}334{,}800 \text{ Btu/hr} = 109{,}803{,}801 \text{ Btu/hr}$$

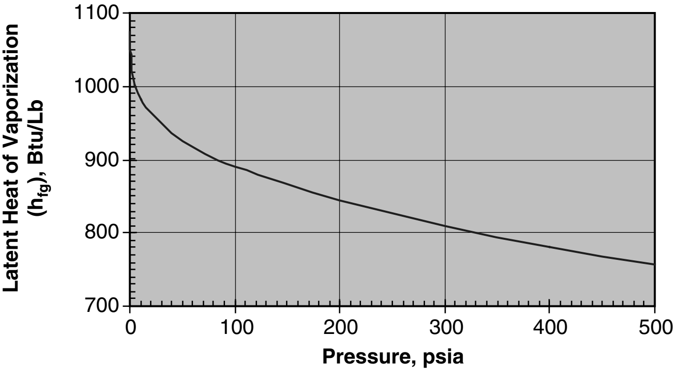

FIGURE 19.8 Latent heat of vaporization of water vs. pressure. (Data from Ref. 5.)

Now divide by the mass flow rate to obtain the discharge enthalpy:

$$109{,}803{,}801\,\text{Btu/hr} \times \frac{1}{421{,}116.8}\,\text{hr/lb} = 260.7\,\text{Btu/lb}$$

Thus, the temperature corresponding to 260.7 Btu/lb is the correct discharge temperature to be used in calculating the actual compressor performance.

Case History 3: Avoiding Damage to Refrigeration Compressors

Refrigeration systems contain liquefied gases, but compressors can be wrecked if allowed to ingest these liquids. When the plant is designed, precautions are taken by the designer to ensure that liquid ingestion is avoided. These precautions include designing components such as evaporators and knockout drums so as to preclude carryover of liquids into the compressor suction. High-speed turbomachinery can withstand only limited amounts of liquid; typical limits are in the range 3 to 5% by weight. However, during periods of high market demand, plants are temped to push the capacity well beyond normal design limits. Although at first causing no apparent distress to the equipment, the increased throughput often results in velocities beyond the design limits of the evaporators and knockout drums. This greatly increases the risk of liquid ingestion and operation in overload.

An event of this type caused a plant in southern Europe to experience multiple impeller failures on the last stage of a propylene refrigeration compressor. It is expected that the high flow operation (Fig. 19.9) is causing the impeller failures. Analysis of the field data shows the compressor to be operating in overload (Fig. 19.10 and Table 19.6). Note the high head, high work input, and high efficiency. This suggests liquid ingestion. The liquid (in mist form) gives the gas more density and results in a greater pressure ratio. The evaporation of the liquid results in a reduced discharge temperature; thus, the calculation, assuming dry gas, shows an abnormally high efficiency and work input.

Work done on other refrigeration compressors where impeller failures have occurred shows similar results (high head, high work input, high efficiency, and operation in overload). Also, work done by Dresser-Rand (Figs. 19.11 and 19.12) shows a high pressure gradient in the last stage wheel due to the volute cutoff. This pressure gradient has the potential to excite impeller frequencies and cause eventual fatigue failures. Liquids in the gas stream generate higher flow rates as the liquid evaporates, pushing the last stage impeller deeper and deeper into choke. The entrained liquid amplifies the pressure gradient of the volute cutoff.

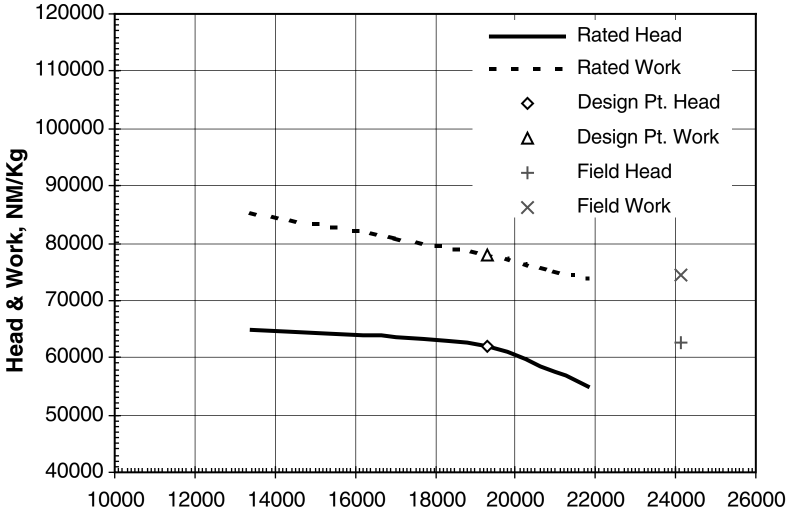

FIGURE 19.9 Comparison of rated head and rated work (vertical scale) vs. flow rate, cubic meters per hour, (horizontal scale). Note the discrepancy between "per design" and "as found in field" data. Note how the field head and work input seem abnormally high and beyond the performance curve limits. The results are fan law-corrected to the OEM design speed of 7060 rpm so the data can be directly compared to the design curve.

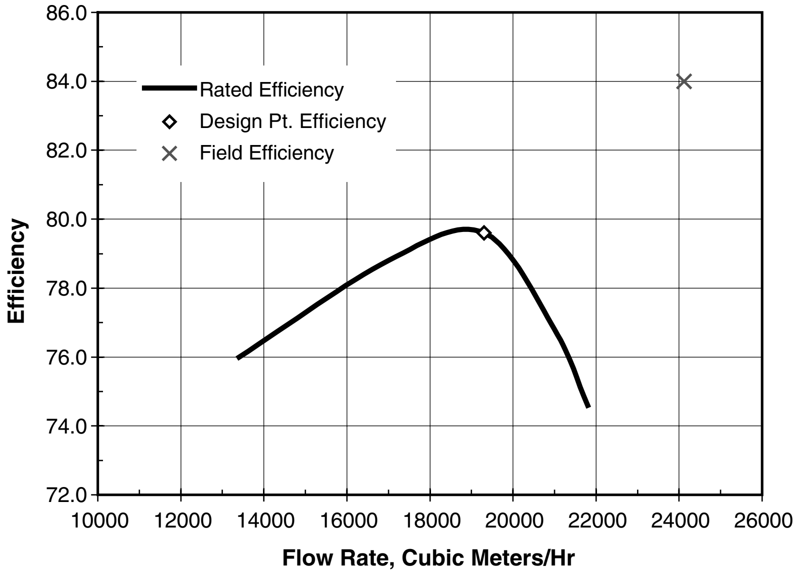

FIGURE 19.10 Performance curve for the third section of a refrigeration compressor. The results are fan law–corrected to a design speed of 7060 rpm. Note how efficiency and flow are beyond the performance curve limits. This suggests that liquids are passing through the compressor. There is a concern related to impeller fatigue failure caused by operation in deep choke and by liquid impingement.

TABLE 19.6 Gas Flex® Sidestream Sectional Analysis Summary[a]

Title: Propylene Refrigeration
Database name: Choke.gdb

	Units	Inlet	SS-1	SS-2	Discharge
Inlet flange data					
Pressure	bar	1.210	2.590	4.580	
Temperature	°C	−30.0	−17.6	2.8	
Given flow	kg/h	73,060.	31,020.	94,605.75	
Volume flow	m^3/h	27,863	5618	11,174	
Mass flow	kg/h	73,060.0	31,020.0	104,045.0	
Compressibility		0.9631	0.9316	0.9046	
Flange diameter	mm	522.8	231.4	326.1	
Flange velocity	m/s	388.2	399.6	400.2	
Discharge flange data					
Pressure	bar				14.18
Temperature	°C				82.8
Volume flow	m^3/h				8880
Mass flow	kg/h				208,125.0
Compressibility					0.8685
Flange diameter	mm				271.3
Flange velocity	m/s				459.3
Inlet section data					
Compressibility		0.9631	0.9515	0.9264	
Temperature	°C	−30.0	8.4	23.2	
Volume flow	m^3/h	27,863	21,215	24,583	
Mass flow	kg/h	73,060.0	104,080.0	208,125.0	
Discharge section data					
Compressibility		0.9575	0.9415	0.8685	
Temperature	°C	19.2	43.2	80.4	
Volume flow	m^3/h	15,557	13,336	8880	
Sectional head data					
Head	N · m/kg	38,506	31,783	65,044	
Efficiency		57	63	84	
Gas power	kW	1382	1460	4470	
Total polytropic data					
Head	N · m/kg	135,333			
Efficiency		69.09			
Gas power	kW	7311			

Source: www.flexwareinc.com.

[a]Results of calculation for propylene refrigeration compressor. Note the exceptionally high efficiency for the last section.

19.4 PREDICTING COMPRESSOR PERFORMANCE AT OTHER THAN AS-DESIGNED CONDITIONS*

As discussed in the preceding sections of this chapter, accurate testing is feasible but could be time consuming. That is where screening studies are often of great value, especially if compressor performance at new conditions is to be predicted.

* Developed by Arvind Godse and originally published in *Hydrocarbon Processing,* June 1989. Adapted by permission of Gulf Publishing Company, Houston, Tex.

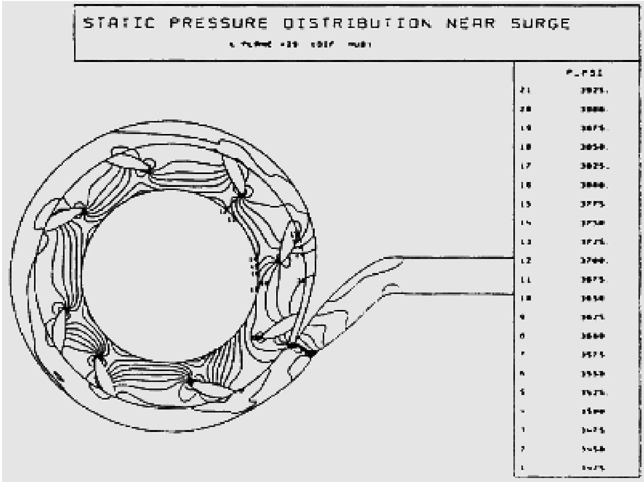

FIGURE 19.11 CFD profile of pressure gradient around the periphery of a discharge volute for a compressor operating near surge. Note how the pressure is relatively uniform. (From Ref. 6; reproduced by permission of the Turbomachinery Laboratory, Texas A&M University, College Station, Tex.)

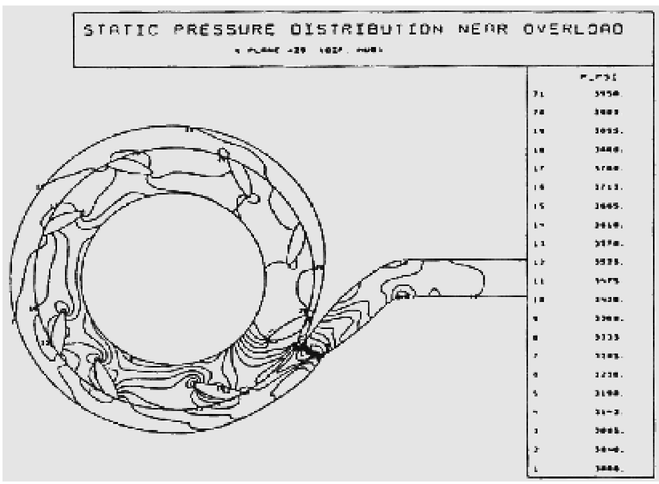

FIGURE 19.12 CFD profile of pressure gradient around the periphery of a discharge volute for a compressor operating in overload conditions. Note the large pressure change at the volute cutoff. This gradient can trigger resonant frequencies in the impeller and eventually cause fatigue failure of the impeller. Testing and CFD evaluation were in general agreement. (From Ref. 6; reproduced with permission of the Turbomachinery Laboratory, Texas A&M University, College Station, Tex.)

To reiterate, the performance of centrifugal compressors is generally documented during shop tests or during field tests at the installation site. Accurate test data will make it possible to predict compressor performance with different gas or operating characteristics at some future date. Moreover, since centrifugal compressor impeller dimensions are often custom-made for a specific operating condition, adjustments may be needed to meet a specific performance.

It is often necessary to arrive at a new speed for variable-speed-driven machines, or to trim the impellers for motor-driven machines, if there is a considerable gap between proposed performance and test stand performance. Similarly, compressor owners may want to determine how a machine would perform with a different gas or with some new operating parameters. Dimensionless numbers provide a convenient way to predict off-design performance.

19.4.1 How Performance Tests Are Documented

Knowing the performance of an existing compressor is a prerequisite for determining performance at new conditions. However, not all compressor manufacturers document performance test results in the same format. Many of them now use computer programs to record or display the results. Still, in some places they resort to manual calculations, which are then displayed in curve shape. Typical examples of performance curve presentation found after surveying a few performance reports show the combinations shown in Table 19.7.

19.4.2 Design Parameters: What Affects Performance

To judge the performance of a machine at off-design conditions, one must know which factors are of prime importance. Examining a compressor is facilitated by understanding the governing laws of similarity as well as through use of dimensionless numbers. This will provide insight as to how a machine can be adapted to varying demands, such as load changes, unforeseen changes in thermodynamic properties, or trying to assess the feasibility of using the existing machine for a different application. When a machine is sold, the basic information on the impeller geometry often remains proprietary. However, using just three rules of thumb and understanding the dimensionless coefficients highlighted earlier in the book allow us to closely approximate machine behavior:

- Head of a compressor varies as the square of tip speed.
- Flow handled by the compressor varies directly with tip speed and impeller diameter.
- The entire design concept of a centrifugal machine can be separated into two areas from a manufacturer's point of view: thermodynamic and mechanical.

Thermodynamic Considerations For the proposed gas, with thermodynamic properties, flow rate, and pressure ratios defined by the original user, the manufacturer selects the correct impeller geometry. Here, the manufacturer will invariably employ dimensionless numbers vs. pressure coefficient ψ flow coefficient ϕ, and their relationship with polytropic

TABLE 19.7 Performance Curve Displays

Sequence	X Axis	Y Axis
1	Inlet volume (icfm)	Discharge pressure
2	Inlet volume (icfm)	Polytropic efficiency
3	Inlet volume (icfm)	bhp
4	Inlet volume (icfm)	Pressure ratio
5	Inlet volume (icfm)	Power ratio
6	Inlet volume (icfm)	Polytropic head
7	Mass flow (lb/min)	Pressure ratio
8	Mass flow (lb/min)	bhp

efficiency η_p. This will help in selecting impeller geometry, number, and configuration. While the manufacturer obviously uses these parameters for his or her selection, we can use them to predict the performance of the machine at other conditions. These dimensionless numbers show the relationship between head, flow, tip speed, and efficiency for a selected blade geometry. They are rarely presented as standard information. However, one can calculate them and plot their graphs, showing their relationship to performance test data. The simplicity of this exercise will be appreciated when investigating engineers have many machines in their plants. Since the theory behind all this has been explained earlier, the rest of this write-up describes the associated formulas and provides an example.

Mechanical Considerations After selecting the impellers, the manufacturer will consider the following:

- Material selection
- Type of split-line orientation and casing configuration
- Rotor layout, depending on the type of casing design and number of impellers. This will involve bearing span, bearing sizing, calculation of critical speeds, and making sure that they are away from the operating speed range. It will also require design decisions to finalize:
  a. Seal design and seal system components
  b. Lube system
  c. Driver rating

From a *future* application point of view, one must have satisfactory answers to a number of questions before the decision can be made to operate an *existing* compressor under *new* conditions:

- Are the existing impellers able to meet flow and head requirements?
- Is the speed selected within an acceptable range for the existing machine?
- Is the casing pressure and type of split orientation satisfactory for the new application?
- Is the material selection of the machine acceptable?
- Is the seal system capable of handling the new application?
- Is the power requirement within the existing driver rating?

When one is trying to adapt the existing machine to changes in operating parameters within a narrow band, the answers will be satisfactory in most cases. The matter can be even simpler for a variable-speed driver.

19.4.3 What to Seek from Vendors' Documents

Many inspection records are submitted by the vendor. It will be beneficial to reduce the data for all centrifugal machines bought through multiple sources to a common format. It is also important to know why reference is initially made to performance data obtained at the manufacturer's facility. Since the machine is new, passages are clean and clearances are per design recommendation. Therefore, these data would obviously help the user to track performance changes occurring with time.

Performance Test Data As pointed out earlier, all manufacturers present the data in different ways. Therefore, arrange to segregate the data for five parameters that normally make up the performance test:

- Polytropic head
- Polytropic efficiency
- Inlet flow
- Mechanical losses
- Critical speed

The last two items are obtained from a mechanical running test. Mechanical run tests are mandatory for centrifugal compressors and are usually conducted prior to the optional shop performance test.

Other inspection records should be reviewed to ascertain:

- Number of impellers
- Individual impeller diameters

In addition, we need to know:

- Casing design pressure and test pressure: hydraulic or pneumatic and its split
- Type of seal and seal system pressure and flow
- Metallurgy of important parts
- Driver rating and speed range

19.4.4 Illustrations and Example

Table 19.8 highlights our nomenclature and associated formulas. Next, an example will show the method employed for arriving at performance at new conditions. The results should always be compared with ratings and capabilities of the compressor and its driver and associated upstream/downstream equipment to verify feasibility.

Figure 19.13 represents a flowchart of the method and procedural sequence we apply to analyze performance of dynamic compressors at new conditions.

Example 19.1

Step 1: Purchase specification data or data known for existing compressor.

• Inlet conditions:	pressure	540 psia (P_1)
	temperature	140°F (T_1)
	compressibility	0.901 (Z)
	K	1.220
	MW	22.23
	flow rate	2654 lb/min (G)
• Discharge conditions:	pressure	1330 psia (P_2)
• Rotor data:	impeller diameter	16.5 in. (D_m and D_s)
	number of impellers	5 (I)
	proposal speed	9600 (N)

TABLE 19.8 Nomenclature and Formulas

Description	Symbol	Formula
Molecular weight	MW	—
Adiabatic exponent	K	C_p/C_v
Polytropic efficiency	η_p	
Polytropic exponent	n	—
Polytropic compression exponent	x	$(n-1)/n = (K-1)/K\eta_p$
Gas constant	R	1544/MW
Compressibility	Z	—
Pressure ratio	r	P_2/P_1
Inlet temperature, °F	T_1	—
Head, ft-lb/lb	H_p	$ZR(T_1 + 460)\,(r^x - 1)/x$
Speed, rpm	N	—
Mean impeller diameter, in.	D_m	—
Suction impeller diameter, in.	D_s	—
Number of impellers	I	—
Weight flow rate, lb/min	G	—
Suction volume, ft^3/min	O_s	—
Flow coefficient	ϕ	$700Q_s/N(D_s)^3$
Pressure coefficient	ψ	$\dfrac{H_p(1300)^2}{IN^2D_m^2}$
Gas hp	W_G	$GH_p/33{,}000\eta_p$
Mechanical loss at speed hp N	W_M	—
Mechanical loss at speed N_1	W_{M1}	$W_M\,(N_1/N)^2$
bhp at speed N	W	$W_G + W_M$

Critical speeds:	first: 6240 rpm
	second: 15,280 rpm
Frictional hp at 9600 rpm:	44

- Driver: variable-speed range, 85 to 105%
rating: 4500 hp
- Performance test data at 9600 rpm

We now proceed to arrange available performance test data in tabular format as follows. Alternatively, we could simply examine the compressor vendor's performance test curve and identify eight points on this curve. For each of these arbitrarily chosen points, we find corresponding head and flow values from the X–Y coordinates associated with this curve plot.

Let us suppose that we obtained the following:

Point	1	2	3	4	5	6	7	8
Head, H_p	38,050	37,875	37,660	37,235	36,500	35,210	32,965	27,458
Flow, Q_s	1046	1120	1190	1296	1410	1556	1726	1950
η_p	0.684	0.697	0.711	0.726	0.739	0.752	0.762	0.731

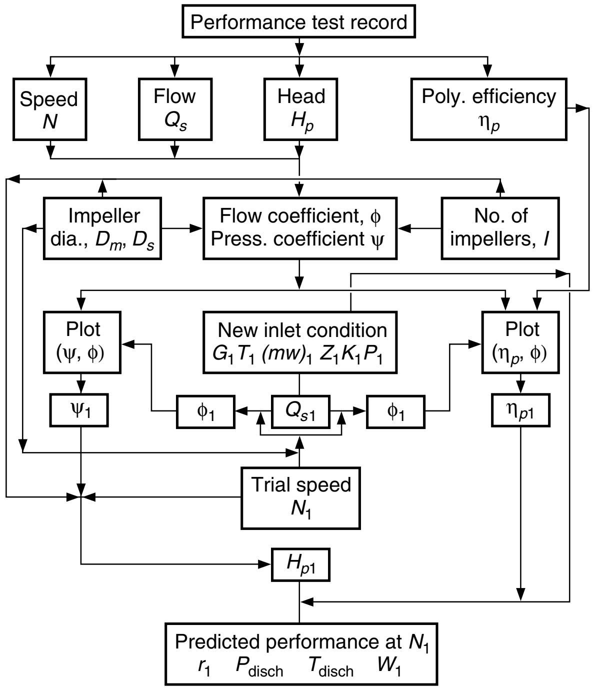

FIGURE 19.13 Flow diagram for reevaluating performance. (From Arvind, Godse, Predicting compressor performance at new conditions, *Hydrocarbon Processing,* June 1989.)

This now enables us to calculate corresponding values of ϕ and ψ for each of the eight points. We use the expressions listed in Table 19.8.*

Step 2: Obtain the flow coefficient and head coefficient.

Point	1	2	3	4	5	6	7	8
ϕ	0.0169	0.0181	0.0194	0.0210	0.0228	0.02524	0.02800	0.03165
ψ	0.5125	0.5102	0.5073	0.5016	0.4916	0.4743	0.4440	0.3698

Step 3: Plot the graph as shown in Fig. 19.14.

* Note again how, for example, data for point 4 were calculated:

$$\begin{aligned}
\phi &= 700Q_s/ND_s^3 \\
&= (700)(1296)/(9600)(16.5)^3 \\
&= 0.0210 \\
\psi &= H_p\ (1300)^2/IN^2D_m^2 \\
&= \frac{(37{,}235)(1{,}690{,}000)}{(5)(9216)(10^4)(272.25)} \\
&= 0.5016
\end{aligned}$$

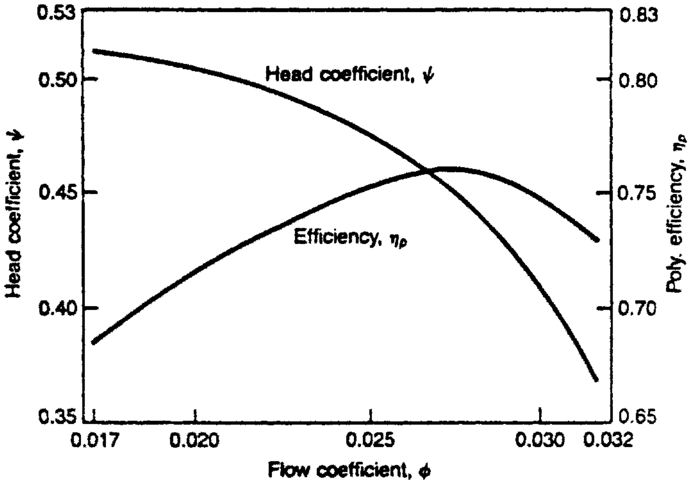

FIGURE 19.14 Relationship of head coefficient, flow coefficient, and polytropic efficiency for a given compressor. (From Arvind, Godse, Predicting compressor performance at new conditions, *Hydrocarbon Processing,* June 1989.)

Step 4: New operating conditions:

- Inlet pressure: 560 psia (P_1)
- Inlet temperature: 130°F (T_1)
- Molecular weight: 24.45 (MW)
- Flow rate: 3000 lb/min (G)
- Other inlet conditions remain as in step 1
- Discharge pressure required, P_2 = 1330 psia

To predict the performance of the compressor at the new operating conditions, take the following steps:

Step 5: Find the inlet volume [from $Q_s = \mathrm{ZGRT_1/MW(144)}P_1$]:

$$Q_s = \frac{(0.901)(3000)(1544)(130+460)}{(24.45)(144)(560)} = 1248.86\ \mathrm{ft^3/min}$$

Step 6: Since the molecular weight is higher than that of the original case, a higher discharge pressure will be generated. Therefore, choose a lower speed.

- Trial 1: Speed selected = 8900 rpm

Step 7: Find the flow coefficient ϕ at the new inlet conditions and a trial speed of 8900 rpm [from $\phi = (700)(1248.86)/(8900)(16.5)^3$].

$$\phi = 0.02186$$

From the graph in Fig. 19.14, find the corresponding:

- Polytropic efficiency, $\eta_p = 0.732$
- Head coefficient, $\psi = 0.497$

Step 8: For the given impeller diameters and number of impellers, the head developed will be

$$0.497 = \frac{H_p(1300)^2}{(5)(8900)^2(16.5)^2}$$

$$H_p = 31{,}709 \text{ ft-lb/lb}$$

Step 9:

$$\frac{n-1}{n} = \frac{k-1}{k\eta_p}$$

$$= \frac{0.22}{(1.22)(0.732)}$$

$$= 0.2463$$

Step 10: Find the pressure ratio r [from $H_p = ZRT(r^x - 1)/(x)(\text{MW})$].

$$31{,}709 = \frac{(0.901)(1544)(590)(r^{0.2463} - 1)}{(0.2463)(24.45)}$$

$$\frac{P_2}{P_1} = 2.3378$$

Step 11: $P_2 = (560)(2.3380) = 1309.17$ psia

From the preceding it is clear that the speed needs to be increased, since the discharge pressure is less than 1330 psia. Let us choose 9000 rpm and find the results.

Step 12: Using the procedure as given in steps 7, 8, 9, and 10, the following information is obtained:

$$\phi = 0.02162 \qquad \text{from} \quad \frac{(700)(1248.86)}{(9000)(16.5)^3} = \phi$$

$$\eta_p = 0.731$$

$$\psi = 0.498$$

$$H_p = 32{,}491 \qquad \text{from} \quad \frac{(0.498)(5)(9000)^2(16.5)^2}{(1300)^2} = H_p$$

$$r = 2.3820$$

$$P_2 = 1334.0 \text{ psia}$$

The speed selected, 9000 rpm, is in order.

$$W_G = \frac{(32{,}491)(3000)}{(33{,}000)(0.731)} = 4040.66 \qquad \text{from } \frac{H_p G}{33{,}000\eta_p} = W_G$$

Step 13: Mechanical losses at 9000 rpm:

$$\begin{aligned} W_M &= (44)\left(\frac{9000}{9600}\right)^2 \\ &= 38.67\ \text{hp} \end{aligned}$$

Step 14:

$$\begin{aligned} \text{bhp} &= 4040.66 + 38.67 \\ W &= 4079.33 \end{aligned}$$

Step 15: Discharge temperature [from $T_{2,\text{actual}} = T_{2,\text{is}} = T_1(P_2/P_1)^{(k-1)/k}$]:

$$T_{dis} = 730.8°\text{F(abs.)}$$

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