FIELD PERFORMANCE TESTING OF CENTRIFUGAL COMPRESSORS

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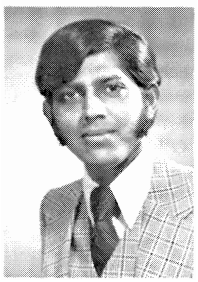

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Symbols listed at the end of the paper.

ABSTRACT

The paper describes the type of data that must be recorded in order to performance test a centrifugal compressor in the field. The instrumentation required to measure different test parameters is also discussed. The need for accurate measurements, as well as the effect errors in the data have on the calculated performance, is explained.

In order to obtain the characteristics of the compressor it is necessary to record data for different values of $\frac{Q}{N}$ (flow to speed ratio). Techniques for varying the $\frac{Q}{N}$ ratio for both fixed and variable speed drives are discussed.

A method of calculating horsepower consumed and the capacity of the compressor is shown. Limitations of a field performance test are also commented upon.

Field Performance testing of Centrifugal compressors with meaningful results is an area of considerable interest to the process and gas industry. Although much time and money are spent on a field test, one must remember that the accuracy of the results is not always in proportion to the amount of effort put into the test. The preferred method, of course, is to perform the test in the Manufacturer's Shop where facilities are designed for accurate data acquisition, and subsequent performance computations.

Proper planning is essential for conducting a meaningful test. A test agenda must be drawn up to cover the scope of the test. It should discuss the kind of test to be performed and also define the basic objectives. The agenda should describe the instrumentation to be used and also list the data required. When the compressor manufacturer gets involved in a field test, it is essential that the test conditions be discussed and an agreement on the acceptance of the test agenda by both the manufacturer and user be made prior to the test.

DATA REQUIRED TO DETERMINE COMPRESSOR PERFORMANCE

Evaluation of compressor performance involves the determination of the inlet capacity, pressure ratio and horsepower consumed for the specific conditions of the test, which include inlet pressure and temperature, discharge pressure, compressor speed and gas properties. The following observations and measurements are required:

1) Inlet Temperature
2) Discharge Temperature
3) Inlet Pressure
4) Discharge Pressure
5) Barometric Pressure
6) Compressor Speed
7) Differential Pressure across flowmeter. Temperature and Pressure at the flow measuring device are also required.

8) Specific Gravity, thermodynamic properties and analysis of the test gas

TEMPERATURE MEASUREMENT

Accurate temperature measurements are very necessary in evaluating performance. Unfortunately, accurate measurement is not easily achieved. Different kinds of instrumentation are available to measure temperature and the choice of any one particular kind depends on the accuracy desired and accessibility at the point of measurement. Thermocouple and glass thermometers are commonly used devices. Thermocouples are more advantageous to use because of the following:

a) They are simple in basic design and operation.

b) They can attain a high level of accuracy.

c) They are suitable for remote reading.

d) They are relatively inexpensive.

e) They can be installed in relatively inaccessible areas and can also be inserted into the gas flow.

Since the output signals of thermocouples are small, sensitive measuring devices must be used to measure them.

Though glass thermometers are available in wide ranges of sensitivity and accuracy, they are not recommended for field testing. They are fragile and also have the disadvantage of not being adaptable for remote reading. Also, thermometers are placed in a fluid bath, and since heat transfer losses are difficult to account for, temperatures measured may not be very accurate.

PRESSURE MEASUREMENT

Pressures are generally measured with high quality Bourdon tube test gauges which must be calibrated against a deadweight tester in their normal operating range. In view of the considerable amount of time involved in reading different pressures, deadweight gauges are rarely used for field testing. When selecting a pressure gauge it is important to note that the measured value is above the midpoint of the scale. If one reads a 2 PSIG pressure with a 0-100 PSIG gauge the reading will be very inaccurate. Sometimes a set of different pressures are measured with one gauge by manifolding and valving. This approach eliminates gauge errors. However, reading too many pressures with one gauge has a disadvantage in that flow conditions may vary during the time interval between recording the first and last pressure measurements.

Differential pressures and subatmospheric pressures are generally measured with manometers with suitable gravity fluids to permit easy reading. A fluid that is chemically stable when in contact with the test gas, as well as one which does not attack the materials of the compressor or process piping, should be selected. Precautions should be taken to prevent foreign matter from blocking the sensing lines and mercury traps should be used to prevent the manometer fluid from entering the process piping.

SPEED MEASUREMENT

In the case of a synchronous motor drive, the synchronous speed can be calculated from the number of poles and line frequency. For other drives, speed can be measured by an electric counter actuated by a magnetic pulse generator or a 60 tooth gear. The latter method is preferred and most driver manufacturers install a gear on the driver shaft for this purpose.

FLOW MEASUREMENT

The following are the various ways by which flow can be measured:

a) Flow Nozzles

b) Concentric Orifice Plates

c) Venturi Tubes

d) Elbow Flow Meters

e) Calibrated pressure drops from the inlet flange to the eye of the first stage impeller. A calibration curve is used for this purpose. When using the curve, it is important to ensure that the Reynold's number during the field test is identical to the one during the calibration test.

Usually one of the above flow measuring techniques is incorporated as a part of plant piping. The choice of any one system depends on the cost, allowable pressure drops and accuracy required.

LOCATION OF INSTRUMENTATION

Figure 1 shows the location of the pressure and temperature probes. Instrumentation at the compressor inlet must be placed downstream of any valves, elbows or screens. Similarly, instrumentation at the discharge should be placed upstream of any valves or elbows.

Temperature and Pressure measurements for the purpose of calculating flow must be measured right at the flow measuring device and not at some place hundreds of feet away.

It is recommended that provisions be made right at the design stage for location of instrumentation in the process piping.

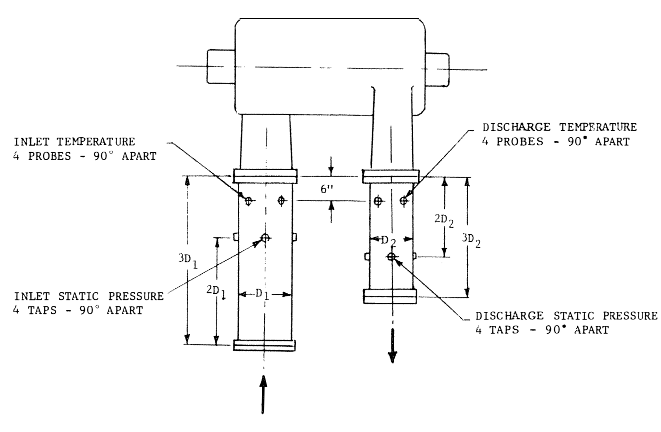

Figure 1. Location of Instrumentation

ACTUAL TEST

After selection, calibration and installation of instrumentation, the test procedure can get underway. In selecting a particular procedure, consideration is given to plant operation and the kind of variables available.

In order to obtain the characteristics of the compressor, it is necessary to record data for different values of Q/N (flow to speed ratio). The different techniques used for varying the Q/N ratio are dependent on the flexibility of the process and whether or not the compressor has a fixed or variable speed driver.

For fixed speed drives the compressor is run near its overload condition. By throttling either the suction or discharge valve the flow is decreased in increments and measurements are made at each value of flow until a minimum flow operating point is reached.

For variable speed drives, the flow can be varied by changing the speed of the compressor. However, it is common procedure to run at a particular speed and vary flow by throttling. By running through this procedure at different speeds, characteristics of the compressor at different speeds can be obtained.

After setting the speed and flow for the first point, the compressor should be "levelled out" by maintaining a constant speed, inlet temperature and inlet pressure. When monitoring reveals that different parameters have stabilized and a condition of equilibrium has been established, only then are the different measurements recorded. Once a complete set of data has been taken, a second set must be recorded for the same conditions. A minimum of two sets of readings for each point is essential. The flow is then varied again and the compressor again levelled and the above procedure repeated for the second point. Flow is varied from the maximum flow operating point(close to overload) to the minimum flow operating point (close to surge). Care must be taken to ensure that the compressor is not surged when setting conditions for the minimum flow operating point.

While throttling it is important to ensure that the throttled gas entering the compressor does not contain any liquid. Liquid in the gas will adversely affect test results.

Many times good data cannot be taken while the compressor is on process. In such cases, a preferred method is to put the unit in recycle and obtain flow variation by varying recycle.

During the test, gas samples of the gas being compressed should be obtained. The samples must be analyzed in a laboratory, to determine the volumetric percentage of the constituents in the gas.

It is extremely important that the molecular weight does not change during a test. Changes due to knock out must be noted and taken into account. Sometimes this is very hard to achieve, particularly if molecular weight changes occur in the compressor itself. As a result, it is impossible to obtain a meaningful performance test.

CARE REQUIRED IN TAKING TEST DATA

Before recording the data, the first step is to ensure that the observed data is consistent within limits given in Figure 2 and Figure 3. These are based on recommendations given in the ASME Power Test Code PTC-10 and hold true in most cases. However, there are many times where caution and good judgment must be exercised. In high molecular weight applications, where there is a very small range of operation as a result of a very steep characteristic, a 4% departure in capacity may be excessive. Likewise, on a low temperature ratio machine of 0.5% fluctuation in inlet temperature may be too high. Also, a tolerance tighter than 2% may be required on pressures for a low pressure ratio compressor. In such cases, it is essential that there be good understanding and judgment on the part of the user and manufacturer in deciding what further refinements in tolerance are required.

FIG. 2. ALLOWABLE DEPARTURE FROM SPECIFIED OPERATING CONDITIONS

VARIABLE	DEPARTURE %
1. Inlet Pressure, psia	5
2. Inlet Temperature, °R	8
3. Specific Gravity	2
4. Speed, rpm	2
5. Capacity, cfm	4

NOTE:
1. The combined effect of 1, 2, and 3 shall not produce more than 8% departure in inlet gas density.
2. See paragraph on Care Required in Taking Test Data.

FIG. 3. ALLOWABLE FLUCTUATION OF TEST READINGS DURING A TEST RUN

MEASUREMENT	FLUCTUATION %
1. Inlet Pressure, psia	2%
2. Inlet Temperature, °R	0.5%
3. Discharge Pressure, psia	2%
4. Flow Nozzle Differential Pressure, psi	2%
5. Flow Nozzle Temperature, °R	0.5%
6. Speed, rpm	0.5%
7. Specific Gravity of Gas	0.25%

CALCULATION PROCEDURE

Pressure gauge readings must be corrected for any liquid leg present in the connecting line. Since the performance is evaluated on the basis of total pressures, the velocity pressure must be calculated and added to the static pressure to obtain the total pressure. Velocity pressure is determined by the following equation:

$$P_v = \frac{(V_{av})^2 \rho}{2g \times 144} \qquad (1)$$

If the velocity pressure is more than 5 percent of the pressure rise, it is determined by a pitot tube traverse. For details of tube design and location of points, the reader is referred to the ASME PTC-10.

From P_1, P_2, T_1, and T_2 the polytropic exponent is evaluated according to the formula:

$$\frac{n-1}{n} = \frac{\log\left(\frac{T_2}{T_1}\right)}{\log\left(\frac{P_2}{P_1}\right)} \qquad (2)$$

The polytropic efficiency is calculated from:

$$\eta_p = \frac{\frac{k-1}{k}}{\frac{n-1}{n}} \qquad \text{(for perfect gases)} \qquad (3)$$

$$\eta_p = \frac{\frac{n'-1}{n'}}{\frac{n-1}{n}} \qquad \text{(for real gases)} \qquad (4)$$

Perfect Gases: This is a gas that follows the equation of state $pv = RT$.

Real Gases: Most gases pumped in industry do not follow the equation of state perfectly and are called real gases.

For a perfect gas k is the ratio of C_p to C_v.

For a real gas $\frac{n'-1}{n'}$ is evaluated according to equation 5.

$$\frac{n'-1}{n'} = \frac{R'}{M_w C_p}\left[Z + T_r \left(\frac{\partial Z}{\partial T_r}\right)_{P_r}\right] \qquad (5)$$

Charts given in figures 4 and 5 are used to calculate $\frac{n'-1}{n'}$. In equation 5, P_r and T_r are the mean reduced

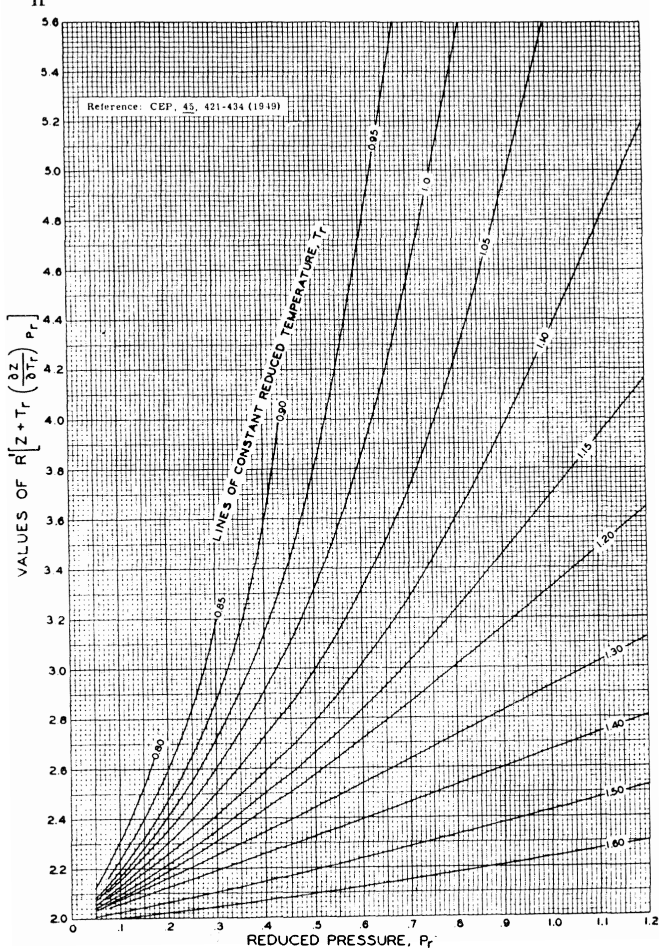

FIG. 4

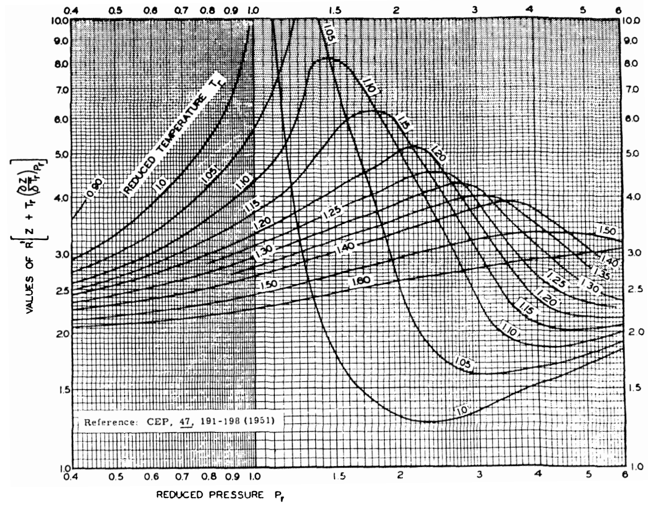

FIG. 5

pressure and temperature for a section of compression.

The head developed by the machine is calculated from the following:

$$H_p = \frac{Z_{av} R T_1 \;(r^M - 1)}{M} \qquad (6)$$

$$\text{Where } M = \frac{n-1}{n} \qquad (7)$$

Knowing P_1, T_1, inlet capacity to the machine is:

$$Q = \frac{Z_{av} W R T_1}{144 P_1} = \frac{1545\, W Z_{av} T_1}{144 M_w P_1} \qquad (8)$$

The horsepower required is:

$$HP = \frac{W H_p}{33000 \eta_p} \qquad (9)$$

Evaluation of Z

Z is evaluated using Pitzer's acentric factor method. The acentric factor is defined by the reduced vapor pressure at a reduced temperature equal to $T_r = 0.7$.

$$\omega = [\, \log P_r + 1 \,] \qquad (10)$$

For a simple fluid $\omega = 0$. Values of acentric factors for some gases are given in figure 6. Knowing ω, Z is calculated according to equation 11.

$$Z = Z^{\circ} + \omega Z' \qquad (11)$$

FIG. 6 ACENTRIC FACTORS FOR SOME GASES (Ref. 1)

GAS	ACENTRIC FACTOR
Nitrogen	0.04
Ethane	0.105
Propane	0.152
Ammonia	0.250
Oxygen	0.0213
Hydrogen sulphide	0.10

Where Z° is the compressibility factor for a simple fluid and Z' is the compressibility factor correction for deviation from a simple fluid.

A plot of Z^{o} vs P_r for different T_r is shown in figure 7 while figure 8 gives a plot of Z' vs P_r for different T_r.

In the above relation Z is calculated at the inlet and discharge temperatures and pressures for the particular section of compression. The arithmetic mean of the Z_{inlet} and $Z_{discharge}$ gives an average Z_{av} which is used in equations 6 and 8.

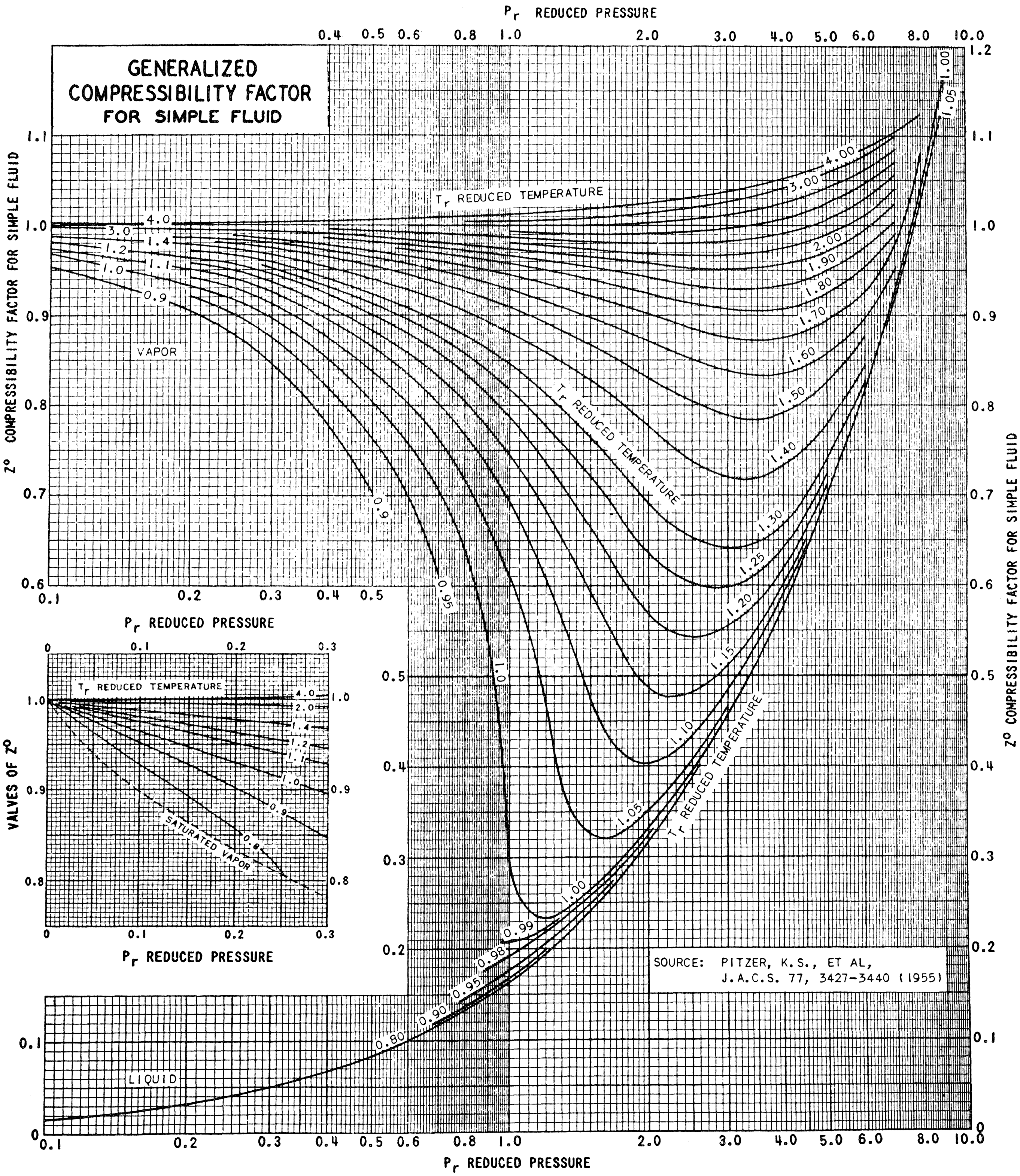

FIG. 7

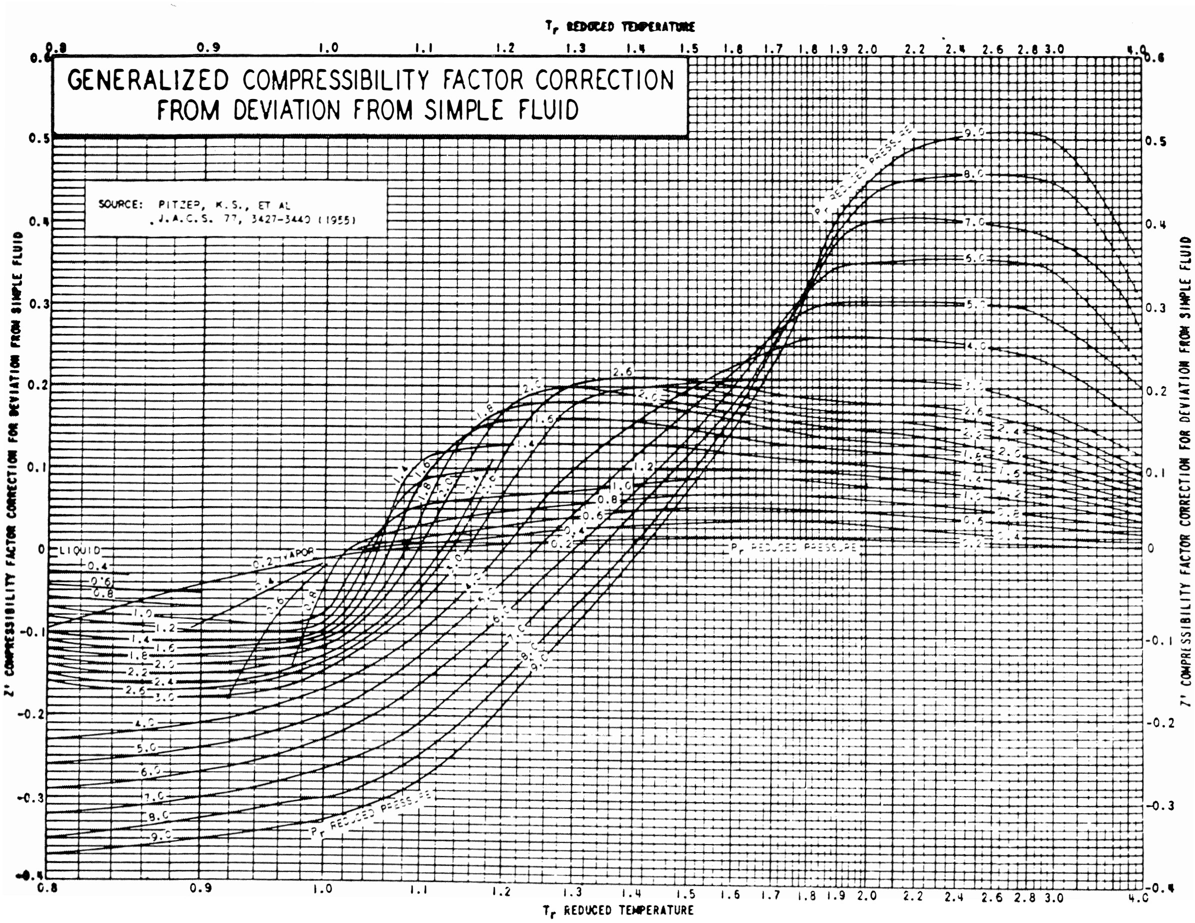

FIG. 8

For Gas Mixtures

From the volumetric gas analysis, the molecular weight of the mixture is obtained from:

$$M_w = \Sigma x_i M_{w_i} \text{ (i = 1, to n for n constituents)} \quad (12)$$

For critical pressures and temperatures the following relationships are used:

$$\text{Mixture } P_{crit} = \Sigma x_i \, P_{crit_i} \quad (13)$$

$$\text{Mixture } T_{crit} = \Sigma x_i \, T_{crit_i} \quad (14)$$

where P_{crit} is the critical pressure and T_{crit} the critical temperature of the i^{th} constituent.

For acentric factors, ω for each constituent is obtained from a chart and the mixture acentric factor is calculated according to:

Mixture $\omega = \Sigma x_i \omega_i$ where ω_i is the acentric figure of the i^{th} constituent. (15)

CONCLUSION

It would seem to appear from the foregoing equations, that calculating the performance of a compressor is a relatively simple task. Undoubtedly, it is easy to plug numbers into the equations and generate results, but the important question is: How accurate are the results? Accuracy depends on the reliability and accuracy of the input data. A seemingly small error of 1% in n′ can throw efficiency η_p off by as much as 10% for gases with n′ values around 1.1.

Also, a field test is done using a process gas, properties of which are not accurately known. Correct knowledge of gas properties is essential if one is to conduct a meaningful field test. The use of a test gas for which the thermodynamic properties are well defined will undoubtedly give better results. Since it is not possible to use a test gas in the field, it is recommended that all compressors be tested in the manufacturer's test facility where a test gas can easily be used.

Finally, since the accuracy of results is always a question mark, a field test should be done only where all parties involved recognize the limits of accuracy.

NOMENCLATURE

C_p = Specific heat at constant pressure, Btu/lb °R
C_v = Specific heat at constant volume, Btu/lb °R
g = Acceleration due to gravity, ft/sec^2
H_p = Polytropic head developed, ft
k = Ratio of $\frac{C_p}{C_v}$
M = $\frac{n-1}{n}$
M_w = Molecular weight of gas mixture

M_{wi} = Molecular weight of i^{th} constituent of the gas mixture
n = Polytropic exponent
n' = Temperature change exponent
p = Pressure, psia
P_v = Velocity pressure, psi
P_1 = Total inlet pressure, psia
P_2 = Total discharge pressure, psia
P_c = Critical pressure, psia
P_r = Reduced pressure
Q = Inlet capacity of machine, cfm
r = Pressure ratio $\frac{P_2}{P_1}$
R = Gas constant, ft-lbs/lb °R
R' = Gas constant, Btu/lb mole °R
T = Temperature, °R
T_1 = Total inlet temperature, °R
T_2 = Total discharge temperature, °R
T_c = Critical temperature, °R
T_r = Reduced temperature
v = Specific volume, ft^3/lb
V_{av} = Average gas velocity, ft/sec
W = Weight flow to machine, lbs/min
x_i = Mole percent of i^{th} constitutent of a gas mixture
Z = Compressibility factor
Z_{av} = Average compressibility factor
ρ = Density of gas, lbs/ft^3
η_p = Polytropic efficiency
ω = Acentric factor

REFERENCES

1. Applied Hydrocarbon Thermodynamics by Wayne C. Edmister, Copyright 1961 by Gulf Publishing Co., Houston, Texas.

2. ASME Power Test Code PTC-10-1965.