FIELD TESTING OF COMPRESSORS

by

Dr. Meherwan P. Boyce

Director, Gas Turbine Laboratories

Texas A&M University

College Station, Texas

Richard D. Bayley

Technical Services Superintendent

Columbia Gulf Transmission Company

Houston, Texas

Vankayalapati Sudhakar

and

Vijayvardhan Elchuri

Graduate Research Assistants

Gas Turbine Laboratories

Texas A&M University

College Station, Texas

INTRODUCTION

An accurate evaluation of the performance of a dynamic rotary compressor is of paramount interest to the builder and user of this type of machinery because it defines the ability of a machine to perform a specific job and the related requirement for energy expenditure. Since the compression process in this type of machine is continous rather than batch, as in reciprocating compressors, physical dimensions of the compressor will not permit accurate performance evaluation. Instead, the characteristics of the machine must be determined by calculation from the job that has been done on the gas as indicated by measurement of observable gas conditions. While a compressor builder might be quite satisfied with relative values for gauging development work, absolute determinations must be made for effective comparison of a machine with its peers. To facilitate such comparisons, a code of practice has been established by ASME,: Power Test Code 10 (PTC 10). It sets guidelines for conducting and reporting tests on a compressor under such conditions and in such a way that its thermodynamic performance on a specified gas of known properties, under specified conditions, may be determined where, during the compression process, no condensation or evaporation occurs and there is no injection of liquids.

The code defines conditions and provides methods by which a compressor may be tested on a suitable test gas, and the results can be converted into anticipated performance of the same compressor when pumping the gas for which it was designed at the design conditions. Also, guidelines are provided for testing compressors with interstage side stream inlets or outlets, internally cooled compressors, and uncooled tandem driven compressors with externally piped intercoolers.

PLANNING OF TEST

The desire to field test centrifugal compressors in accordance with PTC 10 usually occurs for one of two reasons. Most commonly, the user had indication that an equipment train is not performing to expectations. This means that the user knows the compressor facility is not doing the job that operations wants done, but not necessarily the reason why. The facility's energy consumption may be exceeding its design allowance, or it may be an apparent source of plant steam balance upset, overloading the plant cooling water system, or an apparent bottleneck to achieving desired production rates. These and many other factors can dictate that a test be run in an effort to completely define the operating characteristics of a compressor.

The other reason for field testing in accordance with the code is to satisfy contract requirements that spell out a code test prior to acceptance of a machine from the builder.

Regardless of the reason for a test, planning is an essential element for testing that should be considered early in the conceptual stages of a compressor facilities installation.

In addition to the above, PTC 10 can be a helpful guide to monitoring equipment performance. However, in this type of testing, the results are rarely reported in detail and are only of interest to the user as a relative indication of a machine's condition. The real requirement in this type of test is consistency in test procedure, so that successive test results are readily comparable with previous results.

Historically, most field test situations preclude precise application of the test code. Once it is recognized that strict adherence to the code is not possible, it then becomes mandatory that all interested parties; i.e., user, purchaser, contractor, and all equipment suppliers; have agreement in advance as to the intent of a test, how it is to be accomplished, and what machine performance will constitute satisfactory accomplishment of a test.

There is always the possibility that the results of the test planning meeting will reveal that there is no practical means of determining the real performance capability of an installed compressor. It may be that, in this type of situation, agreement can be reached to utilize the code as a guide for a test that merely determines if a machine will do the job for which it was installed.

If there is intent to ever field test a compressor in accordance with PTC 10, the initial test planning needs to be done while the compressor installation is in its planning stage. While the incremental cost of providing a proper number of adequate instrumentation facilities is relatively small at the time of construction, the lost production and incorporation costs for an operating machine generally prohibit their addition.

As will be evident from review of ASME Test Codes and supplements, accuracy of measurement is of prime importance to the successful testing of machinery. However, too many times the installation is complete and the machine is running before consideration is given to providing for the required accuracy. The descriptions of required piping and instrumentation are provided in the latter sections. Some of the more common handicaps that frequently cast doubt on the validity of a field test are also discussed later.

CLASSIFICATION OF TESTS

Three distinct classes of tests are prescribed by the code and are as follows:

Class I: This class includes all tests made on the specified gas (whether treated as perfect or real) at the speed, inlet pressure, inlet temperature, and conditions of cooling for which the compressor is designed and intended to operate. The specified operating conditions and the fluctuations of the test readings should be closely controlled within the limits shown in Tables 1 and 2. By this, the adjustments that have to be made to the results would be minimum, and the accuracy of the results would be maximum. Class I tests should be made wherever feasible, since they usually yield the most accurate results.

Class II and Class III: These tests are intended for use where the compressor cannot be reliably tested on the specified gas at specified operating conditions. Methods for predicting the performance at specified conditions from a test made at different operating conditions and/or with other gases are discussed later.

The reliability of these methods is controlled by the accuracy of the knowledge of the gas properties, the choice of methods used for computing and converting test results, and the extent of deviation from the fundamental design param-

Table 1, [1]
Allowable Departure From Specified Operating Conditions For Class I Test

Variable	Symbol	Unit	Departure (%)	(1)
(a) Inlet pressure	p_i	psia	5	(2)
(b) Inlet temperature	T_i	R	8	(2)
(c) Specific gravity of gas	G	ratio	2	(2)
(d) Speed	N	rpm	2	—
(e) Capacity	q_i	cfm	4	—
(f) Cooling temperature difference	—	°F	5	(3)
(g) Cooling water flow rate	—	gpm	3	—

(1) Departures are based on the specified value where pressures and temperatures are absolute.
(2) The combined effect of items (a), (b) and (c) shall not produce more than 8 per cent departure in inlet gas density.
(3) Difference is defined as inlet gas temperature minus inlet cooling water temperature.

Table 2, [1]
Allowable Fluctuation Of Test Readings During A Test Run — For All Tests Class I, II, And III

Measurement	Symbol	Unit	Fluctuation	(1)
Inlet pressure	p_i	psia	2%	
Inlet temperature	T_i	R	0.5%	(2)
Discharge pressure	p_d	psia	2%	
Nozzle differential pressure	$p_1 - p_2$	psi	2%	
Nozzle temperature	T_1	R	0.5%	
Speed	N	rpm	0.5%	
Torque	τ	lb-ft	1.0%	
Electric motor input	—	KW	1.0%	
Specific gravity test gas	G	ratio	0.25%	
Cooling water inlet temperature	—	F	3F	(2)
Cooling water flow rate	—	gpm	2%	
Line voltage	—	volts	2%	

(1) Pressure and temperature fluctuation for the gas expressed as per cent of average absolute values. Temperature fluctuation for the water is deviation from average in degrees F.
(2) Values do not apply for power measurements by heat balance or heat exchanger methods.

Table 3, [1]
Allowable Departure From Specified Design Parameters For Class II and Class III Tests

Variable	Symbol	Range of Test Values Limits — % of Design Value	
		Min	Max
Volume ratio	q_i/q_d	95	105
Capacity-speed ratio	q_i/N	96	104
Machine Mach Number	M_m		
0 to 0.8		50	105
Above 0.8		95	105
Machine Reynolds Number where the design value is	R_e		
Below 200,000 Centrifugal		90	105
Above 200,000 Centrifugal		10*	200
Below 100,000 Axial Compressor		90	105
Above 100,000 Axial Compressor		10**	200

Mechanical losses shall not exceed 10 per cent of the total shaft power input at test conditions.

*Minimum allowable test Machine Reynolds number is 180,000
**Minimum allowable test Machine Reynolds number is 90,000

Table 4, [1]
Departure of Gas Properties From Perfect Gas Laws Of Test and Specified Gas Permissible For Class II Tests

Pressure Ratio	Maximum Ratio $\frac{k\,max^*}{k\,min}$	Maximum Compressibility Functions X	Maximum Compressibility Functions Y	Minimum Compressibility Functions X	Minimum Compressibility Functions Y
1.4	1.12	0.279	1.071	−0.344	0.925
2	1.10	0.167	1.034	−0.175	0.964
4	1.09	0.071	1.017	−0.073	0.982
8	1.08	0.050	1.011	−0.041	0.988
16	1.07	0.033	1.008	−0.031	0.991
32	1.06	0.028	1.006	−0.025	0.993

When these limits are exceeded by either the test gas or the specified gas at any state point along the compression path, the methods described for computing Class III tests shall be used.

*Maximum and Minimum values of k over either test of specified range of conditions.

$$X = \frac{T}{V}\left(\frac{\delta v}{\delta T}\right)_P - 1 \qquad Y = -\frac{p}{v}\left(\frac{\delta v}{\delta p}\right)_T$$

eters of volume ratio, volume-speed ratio, Mach number, and Reynolds number. Limits for these deviations are shown in Table 3 and are MANDATORY. Class II tests differ from Class III only in the methods of computation. If the thermodynamic properties of either test gas or the specified gas depart from the perfect gas laws beyond the limits shown in Table 4, computation methods specified for Class III tests should be used. Otherwise, the computation methods for Class II can be used.

One of the first deviations from code, that will frequently require consideration, is test gas. While the compressor builder can usually make his shop test with nearly any desired gas, in the field the gas will be whatever is in the process or pipeline. In some instances, it will be difficult to supply the same gas for the entire test or even long enough to obtain an accurate data point. This sort of problem can frequently be accommodated to some degree by obtaining frequent gas samples in the course of the test, but in some instances the gas will change slightly by reaction with the sample bomb, thus destroying the relevance of the laboratory analysis.

PIPING ARRANGEMENTS

The location of pressure, temperature, and flow measuring devices should have a specific relation to the compressor inlet and outlet openings, as described and illustrated in this section. Minimum lengths of straight pipe are required for flow measuring devices and for certain pressure measurements. Flow straighteners and/or equalizers should be used in the vicinity of throttle valves and elbows, as shown in Figure 1.

Piping configuration is most often not subject to change in the field test situation. The torturous path often taken by piping as it approaches or leaves the compressor can have impact on how the compressor will perform, since the piping associated with the compressor will also contain the flow measurement device. This may cause the flow meter to be located in such a manner that the meter primary element is not in

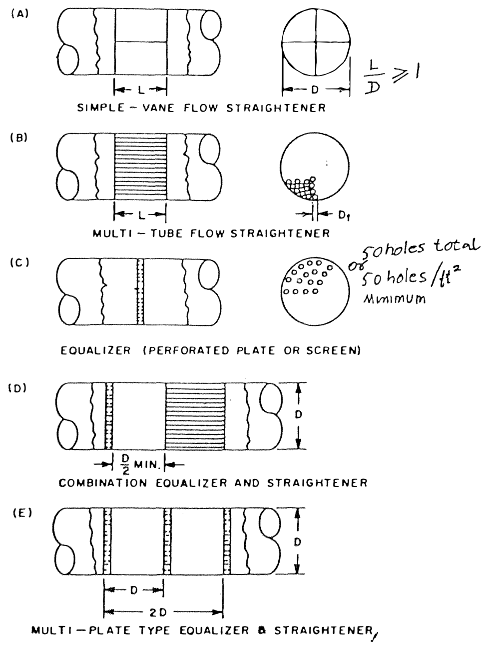

Figure 1. Flow Equalizers and Straighteners [1].

accordance with ASME codes. Frequently, the only flow meter associated with an installed compressor is there for a surge control system that is calibrated empirically and not intended to have absolute accuracy. Even a process flow meter will ultimately have greater utility as a relative indicator than it will have as an absolute meter. In any event, inspection of the primary element of a flow meter prior to test is mandatory.

Another item in the piping system that requires consideration is valves. Care must be exercised to ensure that any branch connections in the system are properly positioned to account totally for the gas flow through the compressor. Thought must be given to proving that a closed valve is not providing significant flow as a result of leakage. This item can be particularly important if side streams are involved as some systems will not have provision for side stream flow measurement. Any test will be judged a complete failure if after the fact it is determined that there was unaccounted for gas involved in the test that did or did not pass through compressor.

Inlet Piping

In shop tests, compressors may be operated without an inlet pipe if the air at the prevailing atmospheric pressures and temperatures will satisfy the conditions required for the test. However, the inlet should be protected with a screen and a suitably designed bellmouth should be used to minimize the entrance losses. In this case, the inlet stagnation pressure is measured by the barometer, and the temperature should be measured at the screen.

Compressors with axial inlets, which have an elbow or a throttle valve preceding the inlet, should use a "long inlet pipe," as shown in Figure 2, not less than ten pipe diameters in length. A flow straightening device should be installed upstream of the pressure measuring station because, under some conditions in an axial inlet, the impeller produces a vortex which would cause substantial error in the measurement of inlet pressure.

Compressors which do not have an axial inlet may be tested with a "short inlet pipe" not less than three pipe diameters in length, as shown in Figure 4. Location of the pressure, temperature, and flow measuring devices are shown

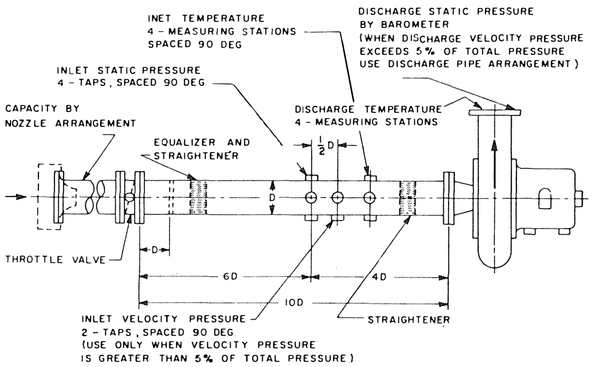

Figure 2. Long Inlet Pipe [1].

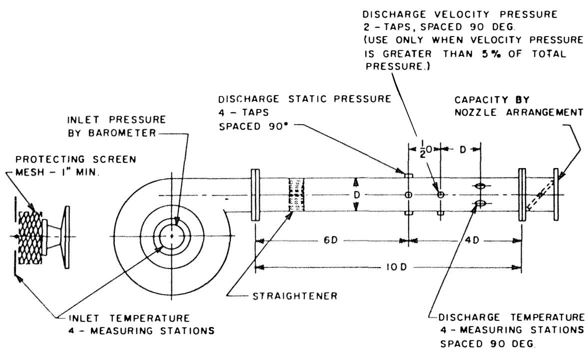

Figure 3. Long Outlet Pipe [1].

in the figures. The above conditions described may be impossible to achieve in field testing. Thus, a pitot traverse may have to be used to determine the velocity profile and calculate the flow rate.

Discharge Piping

For compressors operating as exhausters, discharge piping is not necessary when the velocity pressure is not more than five percent of the pressure rise. The static discharge pressure is measured by the barometer, and the discharge gas temperature should be measured at the outlet of the compressor.

Compressors with a volute type diffuser should be tested with a "long outlet pipe" not less than ten pipe diameters in length, as shown in Figure 3. Flow straighteners and vanes are necessary, because volute type diffusers tend to produce unsymmetrical flow in the discharge pipe, which will cause appreciable error in the measurement of discharge pressure.

Compressors which do not use volute type diffusers may be tested with the "short pipe" arrangement, as shown in Figure 4.

Closed Loop Piping

This type of testing is limited usually to a manufacturer's shop facility. The closed arrangement provides an economical method for testing with many gases other than air and under precisely controlled conditions of pressure and temperature. The essential elements of this arrangement, as shown in Figure 5, are the heat exchanger, the throttle valve, the primary element for flow measurement, the inlet pipe, and the discharge pipe.

A modification of the closed loop system for testing compressors with side streams is shown in Figure 6. For testing with a condensible gas, the piping arrangement is shown in Figure 7.

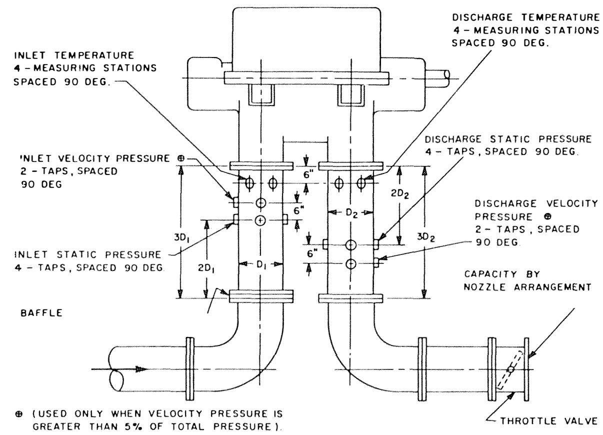

Figure 4. Short Inlet Pipe [1].

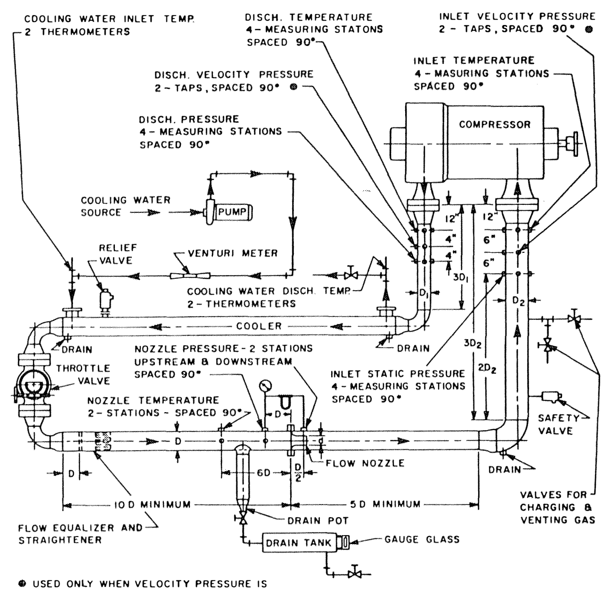

Figure 5. Closed Loop Test Arrangement-1 [1].

A combination of these arrangements may be used as needed by the circumstances, provided the piping includes stations for measuring flow, temperature, and pressure for each side stream independently and in accordance with the requirements in this section.

Provisions should be made to remove liquids from the test loop. The test loop should not be charged with air or any oxidizing gases when the compressor shaft seals are lubricated with combustible fluids. Precautions should be taken against over-pressures, over-temperatures, loss of cooling water, and other unsafe malfunctions.

DATA ACQUISITION

It should be evident that a proper test set up can be an expensive time-consuming job. This being the case, it is apparent that one major objective of any test is to obtain good data. The best check is rough field calculation that permits plotting the results. With the advent of programmable hand calculators, this task is greatly simplified. Bad data will invariably result in an irregular curve or reveal calculated values that are impossible. Embarrassment and excessive cost become plenti-

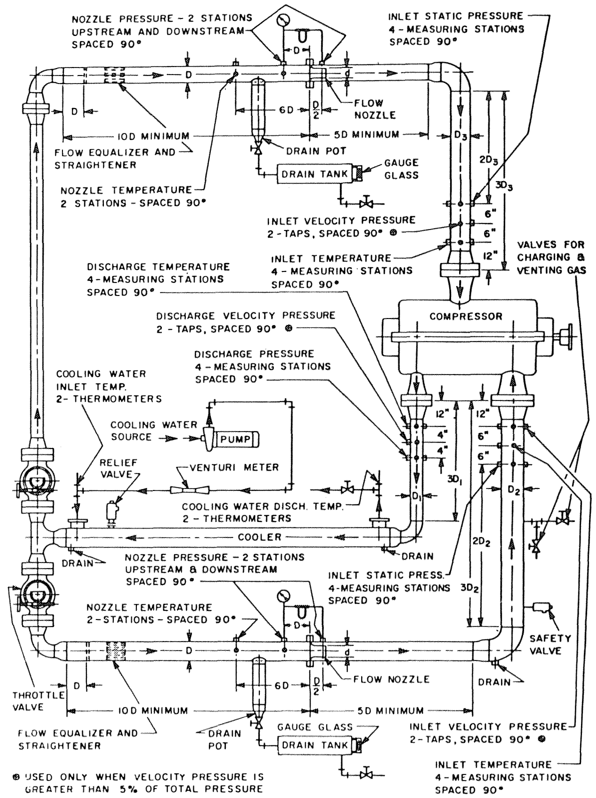

Figure 6. Closed Loop Test Arrangement-2 [1].

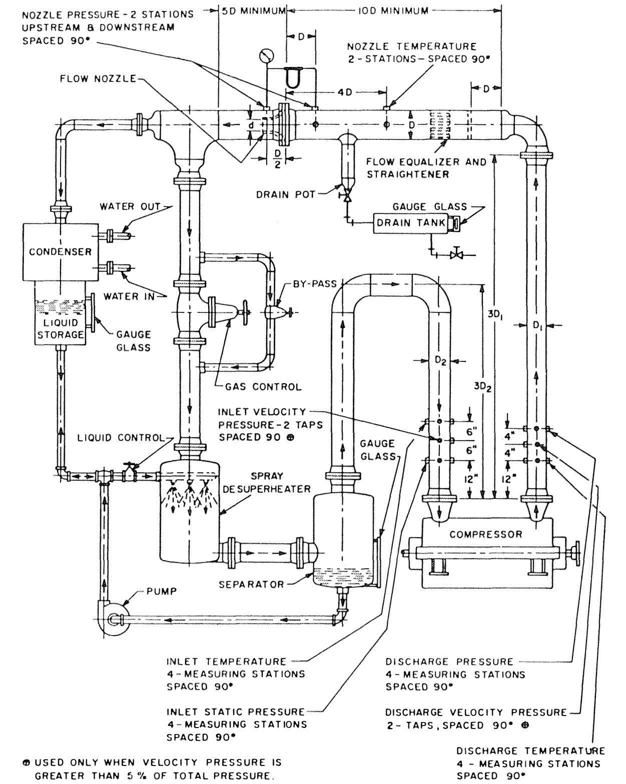

Figure 7. Closed Loop for Condensible Gases [1].

ful when, long after the test set up has been dismantled, the reduced data reveals a machine to have an efficiency greater than 100%.

Field testing requires one final admonition. There is no way that a single head-flow point will verify the performance characteristics of a machine. The motor driven compressor in process service that will not allow for a 20% variation in flow is essentially an untestable machine. Confronted with a single point test situation even greater than normal precautions must be taken to provide the best possible data, if any meaningful information is to be obtained.

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

1. Inlet temperature
2. Inlet pressure
3. Discharge temperature
4. Discharge pressure
5. Barometric pressure
6. Compressor speed
7. Differential pressure across flow meter (or pitot traverse)
8. Temperature and pressure at flow meter
9. Properties of the gas

Measurement of Pressure

The following types of instruments are used to make pressure measurements.

1. Bourdon tube gauges
2. Dead weight gauges (used for calibration purposes only)
3. Liquid manometers
4. Impact tubes
5. Pitot static tubes
6. Pressure transmitters
7. Pressure transducers
8. Barometers

Good quality Bourdon tube test gauges are highly suitable for pressure measurements over 20 psi. They should be calibrated against a dead weight tester in their normal operating range. When selecting a pressure gauge, it is important to see that the measured value is above the midpoint on the scale.

Differential pressures and subatmospheric pressures should be measured with manometers with a fluid that is chemically stable when in contact with the test gas. Mercury traps should be used where necessary to prevent the manometer fluid from entering the process piping. Errors in these instruments should not exceed ¼ percent.

An outstanding failure in pressure measurement is the uncertainty of the configuration of static pressure taps penetration through the pipe wall. This is another early planning concern, as proper taps are easy to provide prior to placing the machine in service, but inspection of the taps after operation has commenced is a luxury rarely afforded the test team.

Another pitfall in pressure measurement, particularly important in flow measurement, is the potential for liquids in gauge lines. All too often gauge lines coming from overhead pipes have no provision for maintaining a liquid-free status, even though the flowing fluid may be condensable at gauge line temperatures.

Calibration of the pressure measuring device presents another pitfall for test crews. All too often a test is conducted through the field calculation step before bad data reveals that gauges, possibly with too large a minimum increment, were removed from the shipping carton and installed, relying on the vendor's calibration. On site calibration of all instruments is always good insurance against a bad test.

Frequently, new machines are put into service with a "start up screen" in the compressor inlet piping to guard against the inevitable weld slag and construction debris that will remain in a new or rebuilt piping system after construction. Regardless of the age of the installation, care must be exercised to ensure that measurements defining suction or discharge conditions are not influenced by such devices.

Inlet and discharge pressures are defined as the stagnation pressures at the inlet and discharge, which are the sum of static and velocity pressures at the corresponding points. Static pressures should be measured at four stations in the same plane of the pipe as illustrated in the piping arrangements. Velocity pressure, when less than five percent of the pressure rise, can be computed by the formula:

$$p_V = \frac{(V_{av})^2\ \rho}{2g_c \times 144} = \frac{(V_{av})^2\ \rho}{9266.1} \tag{1}$$

where V_{av} is the ratio of measured volume flow rate to the area of cross section of the pipe.

When the velocity pressure is more than 5% of the pressure rise, it should be determined by a pitot tube traverse of two stations. For each station, the traverse consists of 10 readings at positions representing equal areas of the pipe cross section, as shown in Figure 8. The average velocity pressure p_v is given by:

$$p_V = \frac{\rho\ \Sigma V_p{}^3}{288\ g_c\ n_t\ V_{av}} \tag{2}$$

where, at each traverse point $V_p = \sqrt{\dfrac{9266.1\ p_V}{\rho}}$

and n_t = number of traverse points.

Barometric pressure should be measured at the test site at 30-minute intervals during the test.

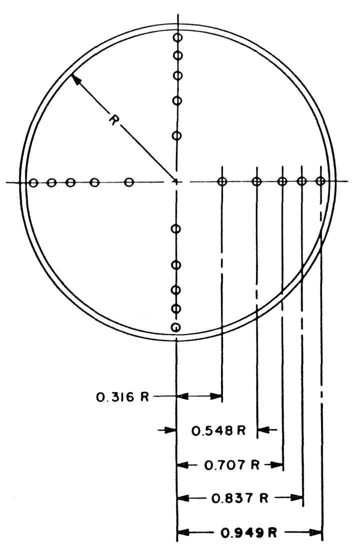

Figure 8. Traverse Points in Pipe [1].

Temperature Measurement

Temperature may be measured by any of the following instruments:

1. Mercury-in-glass thermometers
2. Thermocouples
3. Resistance thermometers
4. Thermometer wells

Thermocouples are the preferred type of instruments because of the simplicity in basic design and operation. They can attain a high level of accuracy, are suitable for remote reading, and are robust and relatively inexpensive.

Regardless of the temperature measuring device to be used, on-site calibration of the entire measurement system is desirable. Usually a two-point check can be made by employing frozen and boiling water. At the very least, all devices can be checked at a common temperature, preferably in the mid-range of expected temperatures, so that any deviant devices can be discarded. This check is particularly desirable for low head machines where the temperature rise will be slight.

Test plans frequently are prepared on the assumption that a laboratory thermometer can replace an operating instrument in an existing thermometer well. While this may be satisfactory, the prudent tester needs to be aware that, because of the propensity of thermo-wells to break off and perhaps enter the machine or cause a hazardous leak, their design is compromised such that true gas temperature determination is impossible. The compromise may be to make the well short and/or to make it thick walled. In either event, the mass of metal exposed to ambient temperature may exceed that exposed to the gas, resulting in significant error if the gas temperature is much different from the ambient. High pressure systems requiring thick wall pipe are particularly susceptible to this fault. However, the use of a good heat transfer fluid can minimize the error. Recognize that the best gas temperature is generated by a calibrated fine wire thermocouple whose junction is directly exposed to the gas near the center flow. As deviations from this ideal are made, the potential for error is increased.

Inlet and discharge temperatures are the stagnation temperatures at the respective points, and should be measured within an accuracy of 1°F. When the velocity of the gas stream is more than 125 fps., the velocity effect should be included in the temperature measurement by use of a total temperature probe. This probe is a thermocouple with its hot junction provided with a shielded cup. The cup opening points upstream. A trade-off has to be made in a field test situation where the gas is not clean.

Flow Measurement

Gas flow through the compressor is measured by flow nozzles or other devices installed in the piping. Among the various devices used are:

1. *Orifice plates* — Orifice plates are of the concentric orifice, eccentric orifice or segmented orifice type and their choice depends upon the quality of the fluid handled.

2. *Venturi tubes* — Consists of a well-rounded convergent section at the entrance, a throat of constant diameter and a divergent section. Their accuracy is high; however, installation, unless planned for in advance, is very difficult in the field.

3. *ASME Flow Nozzle* — These nozzles provide for accurate measurements. Their use is limited in that they are not easily placed in a process plant; however, they are excellent for shop tests. Venturi meters and nozzles can handle about 60% more flow than orifice plates, with varied pressure losses.

4. *Elbow flow meters* — Uses the principle of centrifugal force at the bend to obtain the difference in pressure at the inside and outside of the elbow, which is then related to the discharge pressure.

5. Calibrated pressure drops from the inlet flange to the eye of the first stage impeller in centrifugal compressors, when such data is available from the manufacturer.

6. A flow tracer technique in which Freon is injected into the stream, and flight time between two detection points is measured.

7. Velocity traverse techniques must be used when, due to the configuration in piping, nozzles or orifice plates, etc..., cannot be used. These techniques have been described previously in the pressure measurement section.

Usually, one of the above flow measuring devices and the required instrumentation is incorporated as a part of the plant piping. Choice of the technique depends upon the allowable pressure drop, type of flow, accuracy required and cost.

Nozzle arrangements for various applications vary considerably. For sub-critical flow measurement at the outlet end where nozzle differential pressure, p, is less than the barometric pressure, flow should be measured with impact tubes and manometers, as shown in Figure 9.

For critical flow measurement, where the drop, p, is more than the barometric pressure, flow should be measured with static pressure taps upstream from the nozzle, as illustrated in Figure 10. For compressors operating as exhausters, differential pressure is measured at two static taps located down stream from the nozzle at the inlet, as shown in Figure 11. Nozzle arrangement to measure flow within a closed loop system is shown in Figure 5.

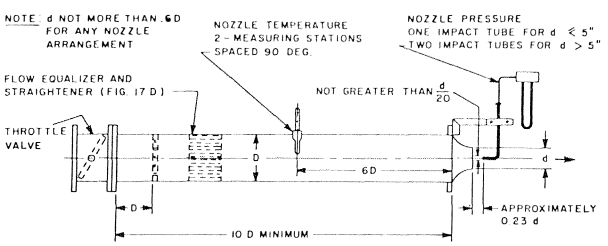

Figure 9. Flow Nozzle for Subcritical Flow [1].

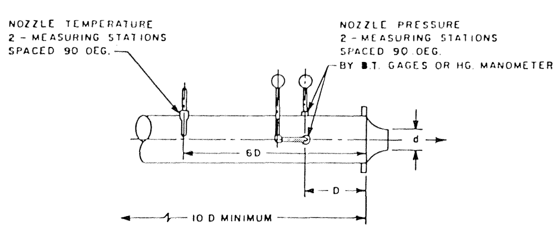

Figure 10. Flow Nozzle for Critical Flow [1].

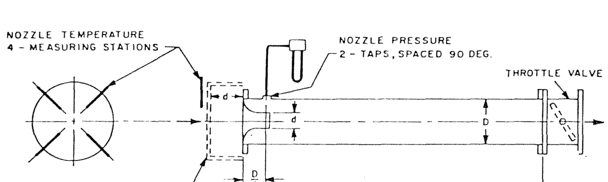

Figure 11. Nozzle for Exhausters [1].

Power Measurement

Power consumed by a compressor can be measured directly at the drive shaft or at the coupling. It can also be determined indirectly from measurement of electrical input to the driving motor, by a heat balance method, or by the heat absorbed at the heat exchanger in a closed loop arrangement. Typical impediments to good field tests are steam turbines without flow meters, engines without fuel meters, motors with unreliable potential, and current measuring devices.

The torque at the drive shaft can be measured by strain gauges or optical devices. Torque together with compressor speed would yield shaft power, HP_{sh}, consumed.

$$HP_{sh} = \frac{\tau N}{5252.1} \tag{3}$$

When an electric motor is used to drive the compressor,

$$HP_{sh} = \frac{\text{Net kW input} \times \text{motor efficiency}}{0.7457} - \text{gear losses} \tag{4}$$

By heat balance method,

$$HP_{sh} = \frac{w(h_d - h_i) + Q_r + Q_m + Q_{sl}}{42.408} \tag{5}$$

Where

w =	mass rate of flow	lbm per min.
h_d =	enthalpy at discharge	Btu per lbm
h_i =	enthalpy at inlet	Btu per lbm
Q_r =	external heat loss from casing	Btu per lbm
Q_m =	total mechanical losses	Btu per lbm
Q_{sl} =	external seal loss equivalent	Btu per lbm

Speed Measurement

In most cases, speed can be measured by an electrical counter actuated by a magnetic pulse generator or a 60-tooth gear. The latter method is preferred and the driver manufacturer usually installs a gear on the driver shaft for this purpose. Optical probes can be used by placing reflective tape on a shaft; however, care should be taken not to place it in direct light. In case of synchronous motor drive, the speed can be calculated from the number of poles and line frequency.

TEST PROCEDURES

The goal of any type of compressor testing is to obtain a compressor map, as shown in Figure 12. This map plots the corrected flow versus the pressure rise at various aerodynamic speeds. The aerodynamic speed shows us the effect that ambient temperature has on the operating characteristic of the compressor.

In selecting a particular test procedure, consideration should be given to plant operation and the kind of variables available. In order to obtain the characteristics of a compressor, data should be recorded for different values of Q/N, flow to speed ratio. The choice of techniques used for varying Q/N ratio depends upon the flexibility of the compressor or the variable speed prime mover.

For fixed speed drives, the compressor is run near its overload condition. By throttling either the inlet or the discharge valve, the flow is decreased in increments. Fine flow adjustments can be made using a bypass across the throttle valves. Data is recorded at each value of the flow until a minimum flow operating point is reached. Also, taking points over a large range of ambient temperatures will provide data for the operating speed line.

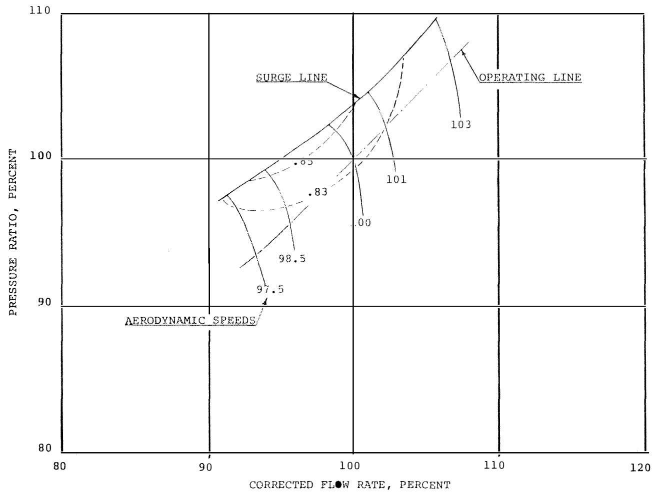

Figure 12. Typical Compressor Map.

For variable speed drives, the flow can be varied by changing the speed. Characteristics of the compressor conditions should be allowed to stabilize by holding the speed and inlet conditions constant. All the measured parameters should be monitored, and when they level-out, two sets of readings should be recorded. The procedure is repeated for all the operating points. Usually a speed line is determined by a minimum of three points; however, five points would be desirable. Surge line is determined also by a minimum of three speed lines and is usually parallel to the operating line. Care should be exercised to see that the compressor is not thrown into surge when operating near the minimum flow conditions.

Gas samples should be collected at regular intervals and analyzed for their properties. Samples collected at the inlet and exit of the compressor provide a meaningful insight to the changes in molecular weight and percentage of the component gases. These points at regular intervals can be used for statistical analysis also.

COMPUTATIONS

The calculations of the various parameters are based on the assumptions that the basic measurements followed the techniques outlined previously. The compressor is tested at various fractions x of design speed.

$$xN_{design} = \frac{xN_{test}}{\sqrt{\theta}}, \quad \theta = \frac{T_{t1}}{520} \tag{6}$$

The abscissa of the compressor performance characteristics is the corrected mass flow rate, give as:

$$\mathring{m}_c = \frac{\mathring{m}\sqrt{\theta}}{\delta} \tag{7}$$

where $\mathring{m}$ is the mass flow rate calculated based upon the type of flow measuring device used and:

$$\delta = \frac{P_{t1}}{14.7} \tag{8}$$

The adiabatic efficiency of the unit can be computed by the use of the following relationship:

$$\eta_{ad} = \frac{T_{in}\left\{\left(\frac{P_{t2}}{P_{t1}}\right)^{\frac{k-1}{k}} - 1\right\}}{\Delta T_{act}} \tag{9}$$

Polytropic index of compression for the gas mixture can be calculated using the following relationship:

$$\frac{n-1}{n} = \frac{\ln\left(\frac{T_{2t}}{T_{1t}}\right)}{\ln\left(\frac{P_{2t}}{P_{1t}}\right)} \tag{10}$$

This leads to a polytropic efficiency, calculated as follows, for a perfect gas:

$$\eta_p = \frac{\frac{(k-1)}{k}}{\frac{(n-1)}{n}} \tag{11}$$

Polytropic efficiencies being the truncated valves of the Taylor series agree closely with the adiabatic efficiency at low pressure ratios. At high pressure ratios, they are higher, as seen in Figure 13.

For real gases, the index of adiabatic compression, k, can be computed as follows:

$$\frac{k-1}{k} = \frac{R'}{M_{mix}\, C_{pmix}}\left[Z + T_{rmix}\left(\frac{\partial Z}{\partial T_{rmix}}\right)_{P_{rmix}}\right] \tag{12}$$

where the expression $R'\left[Z + T_{rmix}\left(\frac{\partial Z}{\partial T_{rmix}}\right)_{P_{rmix}}\right]$

has been represented as a function of T_{rmix} and P_{rmix}, the reduced temperature and pressure of the mixture, respectively. This is shown in Figures 14 and 15.

As an example, the following steps illustrate to determine $C_{p\,mix}$, M_{mix}, T_{rmix}, and P_{rmix}.

The specific heat at constant pressure for the gas mixture is estimated as:

$$C_{pmix} = \Sigma x_i c_{pi} \tag{13}$$

where x_i is the molecular fraction of the i_{th} component gas.

Consider natural gas being compressed to a pressure of 210 psig and 120°F. Methane, Ethane and Propane are the main components.

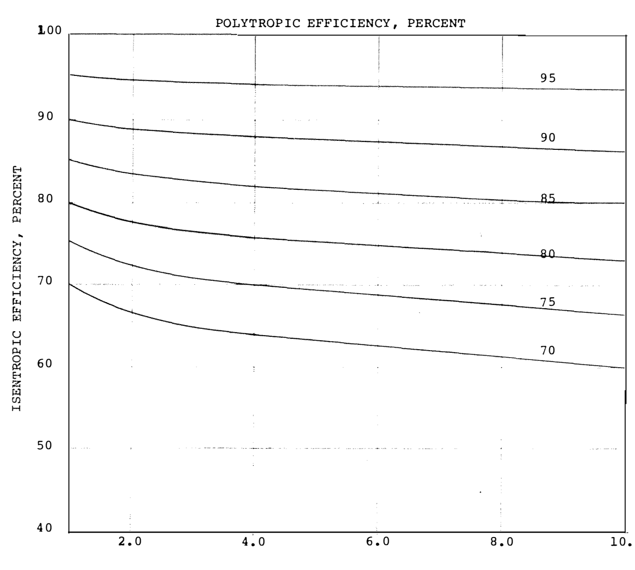

Figure 13. Polytropic and Isentropic Efficiency.

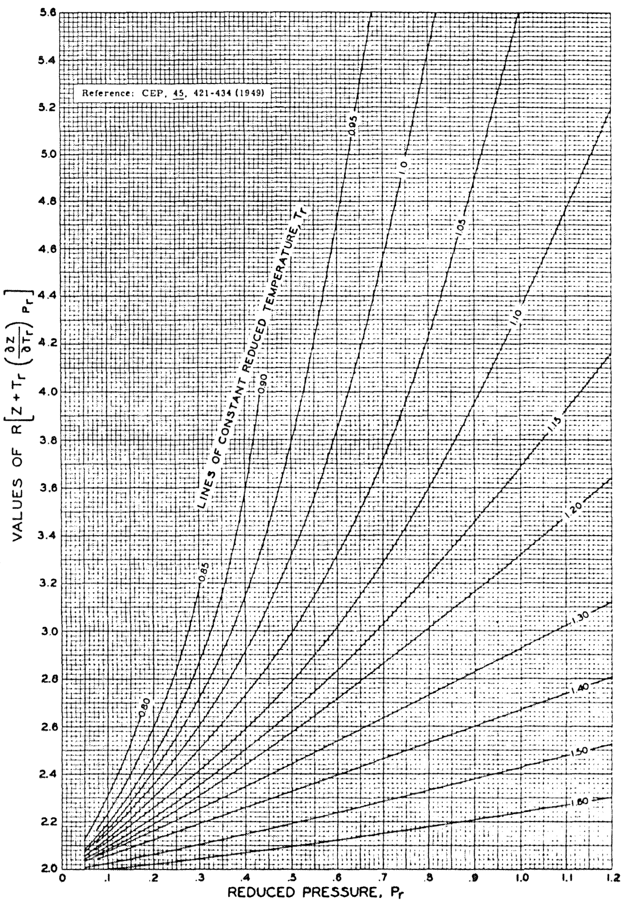

Figure 14. Isentropic Z Function, Low Range [2].

Component Gas*	Molecular Fraction*	Mol. Weight	Critical Pressure psia	Critical Temperature °R
Methane, CH_4	0.84	16.0	673	344
Ethane, C_2H_6	0.10	30.1	708	550
Propane, C_3H_8	0.06	44.1	617	666

*These are obtained from chemical analysis, and the rest are the intensive properties of the respective gases.

Then,

a) $M_{mix} = (.84 \times 16.0) + (.10 \times 30.1) + (.06 \times 44.1) = 19.10$

b) $T_{crmix} = (.84 \times 344) + (.10 \times 550) + (.06 \times 66)\,°R = 384°\,R$

Hence,

$$T_{rmix} = \frac{T_{mix}}{T_{crmix}} = \frac{120 + 460}{384} = 1.510$$

c) $P_{crmix} = (.84 \times 673) + (.10 \times 708) + (.06 \times 617)$ psia
$= 676.1$ psia.

Hence,

$$P_{rmix} = \frac{P_{mix}}{P_{crmix}} = \frac{210 + 14.7}{676.1} = 0.3324.$$

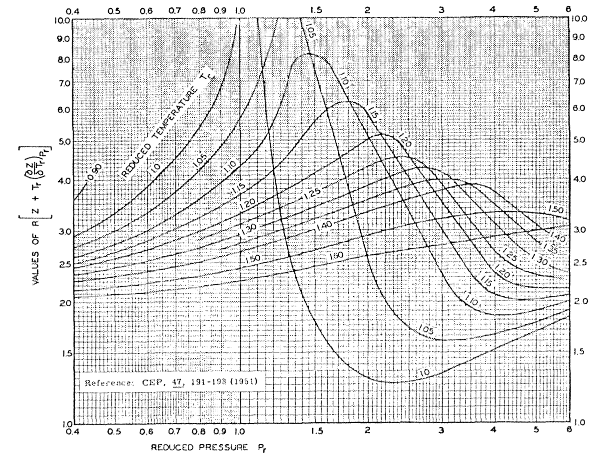

Figure 15. Isentropic Z Function, High Range [2].

With the help of these quantities, the adiabatic index, k, can be computed from equation (12).

The polytropic head developed by the compressor is calculated as:

$$H_p = \frac{Z_{av} R T_{1t}\left[\left(\frac{P_{2t}}{P_{1t}}\right)^{\frac{n-1}{n}} - 1\right]}{\left(\frac{n-1}{n}\right)} \tag{14}$$

where Z_{av} is evaluated as shown below, and **R** is the Universal gas constant.

The average compressibility factor, Z_{av}, necessary to calcalculate the head developed by the compressor is computed as follows:

$$Z_{av} = \frac{Z_{in} + Z_{ex}}{2} \tag{15}$$

$$Z_{in} = Z^{\circ}_{in} + \omega Z'_{in} \quad \text{and}$$

$$Z_{ex} = Z^{\circ}_{ex} + \omega Z'_{ex} \tag{15}$$

$(Z^{\circ}_{in}, Z'_{in})$ and $(Z^{\circ}_{ex}, Z'_{ex})$,

corresponding to inlet and exit conditions of temperature and pressure, have been plotted as functions of T_{rmix} and P_{rmix} (Figures 16, 17). The accentric factor can be evaluated as:

$$\omega = \frac{3}{7}\left[\frac{\ln (P_{cr.} \text{ in atm.})}{\left\{\frac{T_{cr.}}{T_{boil.}}\right\} - 1}\right] - 1 \tag{16}$$

T_{boil} is the normal boiling point of the component gas at 14.7 psia. For a mixture of gases,

$$\omega_{mix} = \Sigma x_i m_i \tag{17}$$

Alternately, the compressibility factor can be evaluated by means of the various empirical equations of state, such as the Benedict—Webb—Rubin equation. This equation in its virial form is given as:

$$\left.\begin{aligned} P = {} & \rho R'T + (B_0 R'T - A_0 - \frac{C_0}{T^2})\rho^2 \\ & + (bR'T - a)\rho^3 + a\alpha\rho^6 \\ & + \frac{c\rho^3}{T^2}(1 + \gamma\rho^2)e^{-\gamma\rho^2} \end{aligned}\right\} \tag{18}$$

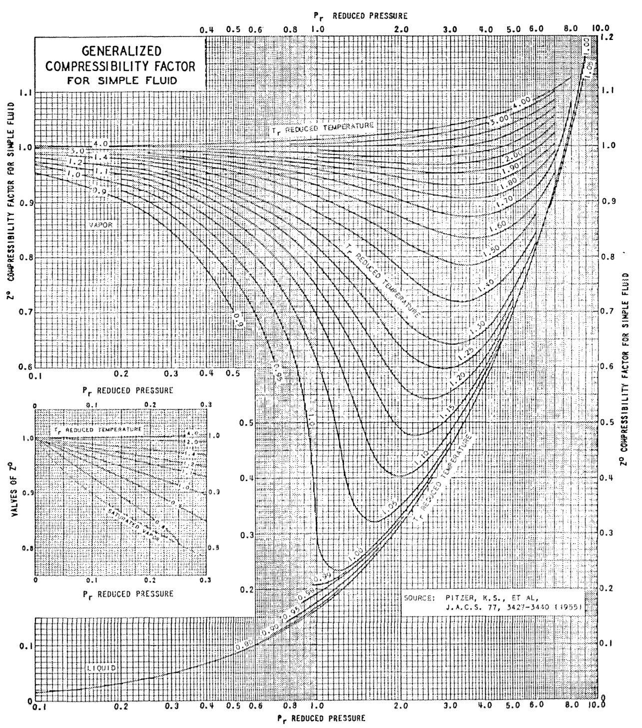

Figure 16. Generalized Compressibility Factor for Simple Fluid [2].

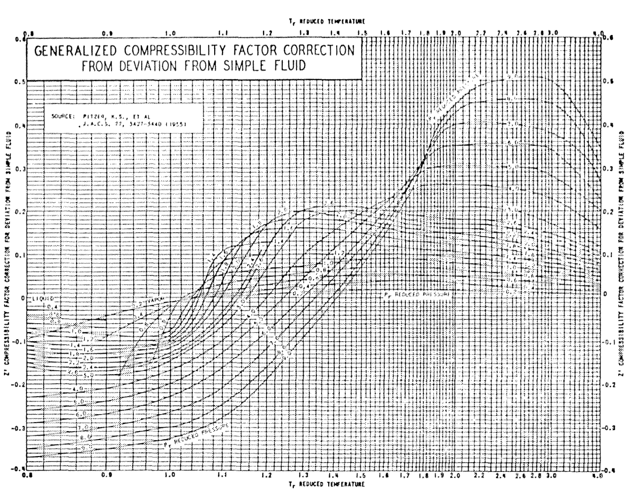

Figure 17. Generalized Compressibility Factor Correction for Deviation from Simple Fluid [2].

The eight empirical constants are B_o, A_o, C_o, b, a, α, c, and γ. Rearranging the above equation, the expression for the compressibility factor becomes:

$$\left.\begin{aligned} Z = 1 &+ (B_o - \frac{A_o}{R'T} - \frac{C_o}{R'T^3})\rho \\ &+ (b - \frac{a}{R'T})\rho^2 + \frac{a\alpha}{R'T}\rho^5 \\ &+ \frac{c\rho^2}{R'T^3}(1 + \gamma\rho^2)e^{-\gamma\rho^2} \end{aligned}\right\} \quad (19)$$

This equation of state has been widely used to evaluate the thermodynamic properties of a number of single-component,

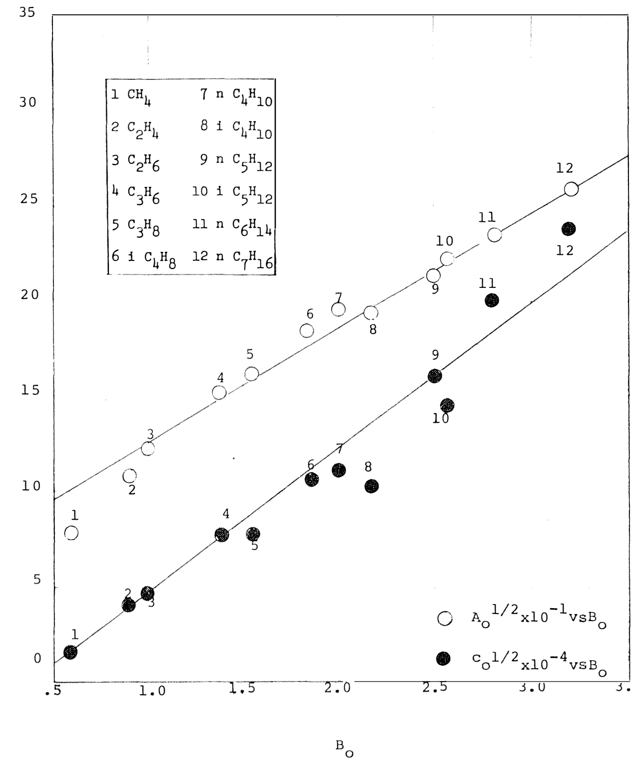

Figure 18. Constants of Benedict-Webb-Rubin Equation [3].

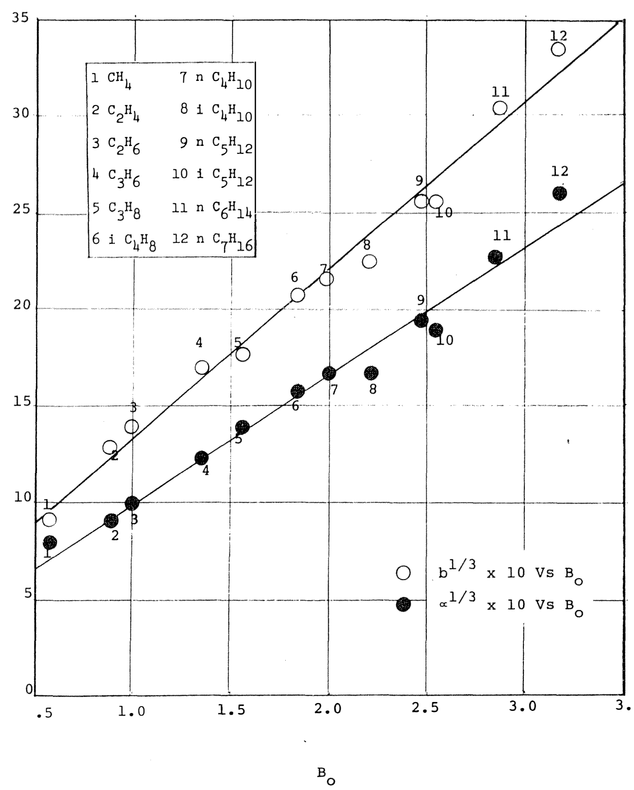

Figure 19. Constants of Benedict-Webb-Rubin Equation [3].

hydrocarbon systems. A system utilizing a combination of these hydrocarbons is handled with the help of the following set of relationships between the system- and the component-contants.

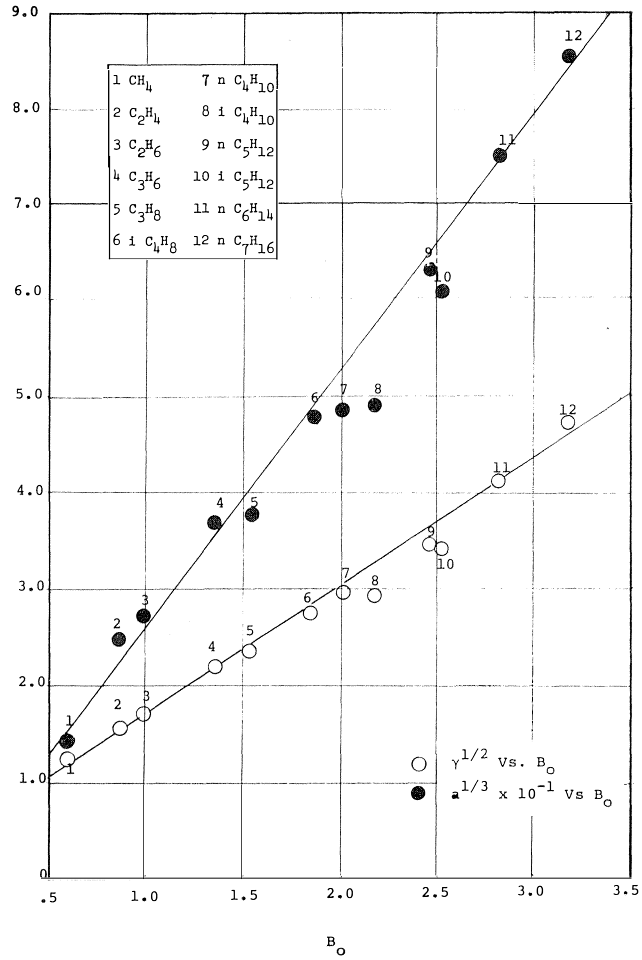

Figure 20. Constants of Benedict-Webb-Rubin Equation [3].

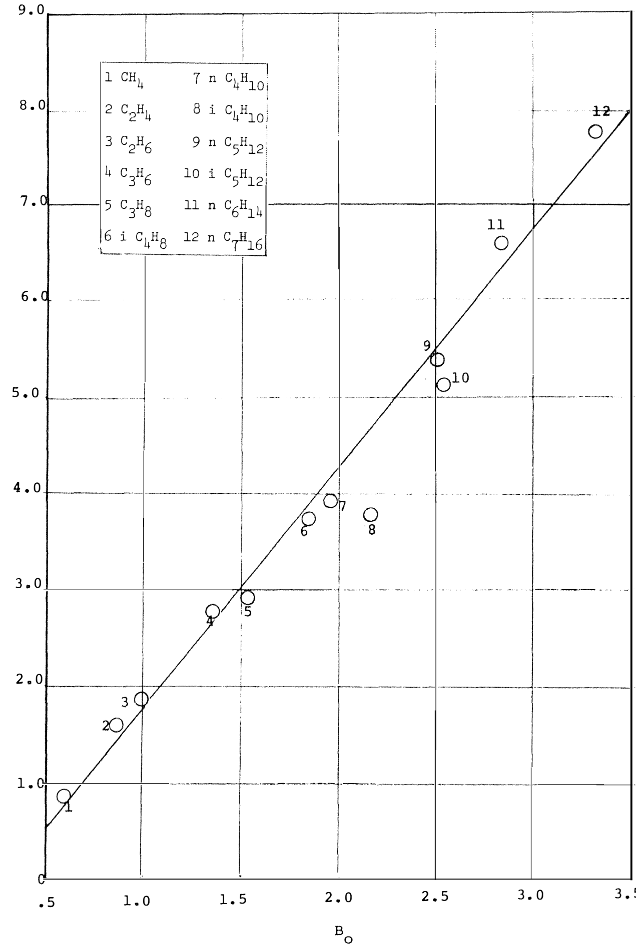

Figure 21. Constants of Benedict-Webb-Rubin Equation [3].

$$
\begin{aligned}
B_{o_{mix}} &= \Sigma x_i B_{o_i} \; ; A^{1/2}_{mix} = \Sigma x_i A^{1/2}_{o_i} \\
C_{o_{mix}} &= \Sigma x_i C^{1/2}_{o_i} ; b^{1/3}_{mix} = \Sigma x_i b_i^{1/3} \qquad (20) \\
a^{1/3}_{mix} &= \Sigma x_i a_i^{1/3} ; \alpha^{1/3}_{mix} = \Sigma x_i \alpha_i^{1/3} \\
c^{1/3}_{mix} &= \Sigma x_i c_i^{1/3} \text{ and } \gamma^{1/2}_{mix} = \Sigma x_i \gamma_i^{1/2}
\end{aligned}
$$

The relative variations of these constants, with regards to the constant B_o, for a number of hydrocarbons are shown in Figures 18, 19, 20, and 21. Use of these graphs, together with the equations 18 and 19 enables the computation of the compressibility factor. It is, however, of importance to note that the Benedict- Webb- Rubin equation of state fails to hold for systems with more than two component hydrocarbons.

Finally, the horsepower required to drive the compressor is estimated by the following relation:

$$HP = \frac{\mathring{m} H_p}{\eta_p} \qquad (21)$$

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2. Edmister, W. C., *Applied Hydrocarbon Thermodynamics*, Vol. 1, Gulf Publishing Co., Houston, Texas, 1961.
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