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Displacement Compressors, Vacuum Pumps and Blowers

PERFORMANCE
TEST
CODES

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
United Engineering Center
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Displacement Compressors, Vacuum Pumps and Blowers

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Foreword

The Test Code for Displacement Compressors, Vacuum Pumps and Blowers was originally issued in 1915. After further revision and extension, it was printed in tentative form in the January, 1922, issue of Mechanical Engineering, and was presented to the Society for discussion at a public hearing during the Spring Meeting in Atlanta in May, 1922. The revised version was approved by the Standing Test Codes Committee on December 1, 1924 and, on March 25, 1925 was approved and adopted by the Council as a standard practice of the Society. The first edition was exhausted in the Spring of 1927 and, acting under instructions from the Standing Committee, PTC Committee No. 9 made slight corrections and released it for reprinting.

In October, 1935, Council appointed a new Committee to undertake the complete rewriting of the Code. Prepared during the year 1938, the revised draft was approved at the December 9, 1938 meeting of the Standing Test Codes Committee. On May 9, 1939, it was approved and adopted by the Council.

When in 1947, the supply of this second edition, after several reprintings, again approached exhaustion, Committee No. 9 was instructed to draft a further revision primarily to bring the Code into conformity with the requirements as to scope, arrangement, and mandatory provisions summarized in the 1945 Code on General Instructions. The third edition included certain new material, particularly that having to do with gases other than air, but otherwise followed closely the procedures, methods and requirements of the second edition. It was approved by the Standing Committee on December 4, 1953, and approved and adopted by the Council as a standard practice of the Society by action of the Board on Codes and Standards on February 9, 1954.

In March, 1961, Council appointed a new PTC Committee No. 9 to again revise the Test Code for this class of equipment. The new Committee found that major advances in the procedures for indicating compressors made it necessary to completely revise affected paragraphs in the existing Code. Also, the Committee considered it not feasible to include sufficient detailed information on these advances, and decided instead to refer to the Supplement on Instruments and Apparatus, Part 8 on Measurement of Indicated Horsepower.

Where practicable, the same symbols and definitions are used in PTC 9-1970, as in PTC 10-1965. Material on the thermodynamic properties of gases has been eliminated, and instead typical references are given in the Appendix. Another important change is the admission of sharp-edged orifices as well as the standard ASME Flow Nozzle to measure air and gas flow.

The members of PTC Committee No. 9 wish to express sincere appreciation for the assistance of the ASME Headquarters staff, and also for the efforts of Chairman W.K. Newcomb, now retired, Acting Chairman W.F. Hartwick and Mr. Hays C. Mayo, who organized the present Committee and was its Chairman from 1961 to 1964. The cooperation of the organizations employing the the members of the Committee has likewise been of invaluable assitance.

This fourth edition, PTC 9-1970, was approved by the Performance Test Codes Committee on November 7, 1969. It was approved and adopted by the Council as a standard practice of the Society by action of the Policy Board, Codes and Standards on March 10, 1970.

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ASME PERFORMANCE TEST CODES
Test Code for Displacement Compressors, Vacuum Pumps and Blowers

CONTENTS

SECTION 0, INTRODUCTION

0.01 This Code applies to tests for determining the performance of positive displacement compressors, blowers and vacuum pumps, whether reciprocating or rotating. These machines are characterized by a pulsating or intermittent intake and delivery of the fluid. Throughout this Code the term compressor will be considered to mean any of these devices.

0.02 A study of the Code on General Instructions is recommended as an introduction to essential procedures necessary for proper use of all ASME Performance Test Codes. The mandatory requirements contained therein are incorporated here in Section 3.

0.03 Reference is made to the Performance Test Code Supplements on Instruments and Apparatus (abbreviated, I & A) for general instruction on instrumentation and the following Supplements should be available when using this Code:

(*a*) Pressure Measurement (PTC 19.2 – 1964)

(*b*) Temperature Measurement (PTC 19.3 – 1961)

(*c*) Flow Measurements (ASME's "Fluid Meters – Their Theory and Application," Sixth Edition, Part II, "Flow Measurement")

(*d*) Measurement of Shaft Horsepower (PTC 19.7 – 1961)

(*e*) Measurement of Indicated Power (PTC 19.8 – 1970)

The specific directions of this Code, which may differ from those given in other ASME publications, shall prevail for any instrument, procedure, or measurement.

0.04 Mandatory requirements for the application of instruments and methods of measurement for tests of displacement compressors and blowers are included herein.

0.05 This Code includes rules for measuring electrical input when the driving element is an electric motor. The test Codes applying to the driving element for various other types of drives are:

(*a*) Reciprocating Steam Engines (PTC 5 – 1940)

(*b*) Steam Turbines (PTC 6 – 1964)

(*c*) Internal-Combustion Engines (PTC 17 – 1957 under revision)

(*d*) Hydraulic Prime Movers (PTC 18 – 1949)

(*e*) Gas Turbine Power Plants (PTC 22 – 1966)

0.06 The Test Code for Speed-Governing Systems for Steam Turbine-Generator Units (PTC 20.1-1958) covers the test of all regulating devices. It may be used as a guide for testing pressure responsive regulators used for compressor control.

0.07 The tables for data and results included in this Code apply to complete units including compressor element and driving element.

SECTION 1, OBJECT AND SCOPE

1.01 The purpose of this Code is to establish rules for conducting tests of displacement compressors, vacuum pumps and blowers to determine the following:

(*a*) Capacity in relation to speed, inlet pressure, and discharge pressure.

(*b*) Power consumption in relation to speed, capacity, inlet pressure, discharge pressure, and intercooling. In addition, the Code provides rules for adjusting the test results to reconcile them with specified operating conditions.

1.02 This Code applies to the testing of compressor units when operated under conditions which permit discharging the gas compressed into the atmosphere or into pipe lines or receivers in which the pressure may be maintained essentially uniform and free from pulsations. It is intended to cover the compressor only, and is applicable only when the unit is operated without inlet pipe or duct, or when the magnitude and the over-all effect of pressure waves within an inlet duct fall within limits specified in in this Code.

1.03 Because of pulsating flow conditions inherent in displacement compressors, uniformly accurate methods of test are not available under all operating conditions. The Code specifically requires the parties to the test to choose from alternate methods described in this Code for conducting such tests.

1.04 This Code does not consider any over-all tolerances or margins which may, by contract or other agreement, be made applicable to any guaranteed or specified performance. Allowances for inaccuracy of measurements and thermodynamic data may be recognized as provided in Section 3, Par. 3.12 and Section 6, Par. 6B.01.

1.05 The procedures and the instrument specifications of Section 4, the formulas and methods for computing results of Section 5, and the indicated form of reporting the test of Section 6, are mandatory. For reasons of commercial expediency or otherwise, the parties to a test may, by agreement, substitute other instruments or procedure. However, only tests made in strict accordance with the mandatory provisions of this Code may be designated as complying with the ASME Test Code for Displacement Compressors, Vacuum Pumps and Blowers.

SECTION 2, DESCRIPTION AND DEFINITION OF TERMS

Pressures

2.01 Absolute pressure is the pressure measured from absolute zero, i.e., from an absolute vacuum. It is equal to the algebraic sum of the atmospheric pressure and the gage pressure.

2.02 Static pressure is the pressure of the gas measured in such a manner that no effect in measurement is produced by the velocity of the gas stream. It is the pressure that would be shown by a measuring instrument moving at the same velocity as the moving stream.

2.03 Stagnation (total) pressure is the pressure attained by converting the kinetic energy of a moving fluid stream isentropically to a state of zero velocity. In a stationary body of gas, the static and stagnation pressures are numerically equal. Refer to equation 6A.04.02 for converting from static to stagnation pressure.

2.04 Velocity pressure is the stagnation pressure minus the static pressure in a gas stream. It is the pressure generally measured by the differential pressure reading of a Pitot tube.

2.05 Inlet stagnation pressure is the stagnation pressure at a point near the inlet flange of the compressor as specified in Par. 4.14. *Unless specifically stated otherwise, this is the compressor inlet pressure as used in this Code.*

2.06 Inlet static pressure is the static pressure at or near the inlet flange of the compressor. In an air compressor without inlet pipe or duct, absolute inlet static pressure is equal to the barometric pressure. When inlet ducts are connected, it is represented by the integrated average of a pressure-time diagram for the period during which the inlet valves are open. See Par. 4.22.

2.07 Discharge stagnation pressure is the stagnation pressure at a point near the discharge flange of the compressor as specified in Par. 4.14. *Unless specifically stated otherwise, this is the compressor discharge pressure as used in this Code.*

2.08 Discharge static pressure is the static pressure at or near the discharge flange of the compressor. It is represented by the integrated time average of a pressure-time diagram for the period during which the discharge valves are open. See Par. 4.21.

2.09 Mean effective pressure is the mean net pressure on the piston during a complete cycle.

Temperatures

2.10 Absolute temperature is the temperature above absolute zero. It is equal to the degrees Fahrenheit plus 459.7 and is stated as degrees Rankine.

2.11 Static temperature is the temperature that would be shown by a measuring instrument moving at the same velocity as the fluid stream,

2.12 Stagnation (total) temperature is the temperature attained by converting the kinetic energy of a moving fluid stream adiabatically to a state of zero velocity. Refer to eq. 6A.04.1 for converting from static to stagnation temperature.

2.13 Inlet stagnation temperature is the stagnation temperature at a point near the inlet flange of a compressor as specified in Par. 4.24. *Unless specifically stated otherwise, this is the compressor inlet temperature as used in this Code.*

2.14 Inlet static temperature is the static temperature at a point near the inlet flange of the compressor as specified in Par. 4.24.

2.15 Discharge stagnation temperature is the stagnation temperature at a point near the discharge flange of the compressor as specified in Par. 4.26. *Unless specifically stated otherwise, this is the compressor discharge temperature as used in this Code.*

2.16 Discharge static temperature is the static temperature at a point near the discharge flange of the compressor as specified in Par. 4.26.

2.17 Intercooling is the removal of heat from the gas between stages by means of any heat transfer external to the compressor.

Gas Properties (other than Pressure and Temperature)

2.18 Density is the mass of the gas per unit volume.

2.19 Specific volume is the volume occupied by a unit mass of the gas.

2.20 Molecular weight is the relative weight of a molecule of a substance referred to that of an atom of Carbon-12 as 12.000. An apparent molecular weight of a gas mixture may be determined if the chemical composition is known.

2.21 Specific gravity is the ratio of the density of the gas at a pressure of 14.7 psia and a temperature of 68 F to the density of dry air at the same pressure and temperature and an apparent molecular weight of 28.970.

2.22 Standard air has several definitions depending on the particular society or industry defining it. The ASME standard is air or gas at 68 F and 14.7 psia, having a relative humidity of 36 per cent. (Density = 0.075 lbm/ft^3.)

Machine Characteristics

2.23 Displacement of compressor is the average volume displaced per unit time by the piston of a reciprocating type or by vanes, gear teeth and lobes of the rotary types.

When used to indicate size or rating, displacement must be related to a specified speed. For multistage machines, it refers to the first stage cylinder(s) only.

2.24 Capacity of a compressor is the volume rate of flow of gas compressed and delivered referred to conditions of stagnation pressure, stagnation temperature and gas composition prevailing at the compressor inlet.

2.25 Pressure (compression) ratio is the ratio of the absolute discharge pressure to the absolute inlet pressure. This ratio may be the over-all ratio of the final discharge pressure to the initial inlet pressure for a multistage compressor, or it may be expressed also as the ratio for each particular stage in a multistage compressor.

2.26 Isentropic power is defined as the power required to isentropically compress and deliver the volume rate of flow of gas represented by the capacity from the temperature and pressure at the compressor inlet to the discharge pressure of the compressor with no friction or leakage and with inlet and discharge pressure constant. For a multistage compressor, the isentropic power is the sum of the isentropic power of each of the stages.

2.27 Indicated power is the power calculated from compressor indicator diagrams.

2.28 Shaft power is the measured power input to the compressor.

2.29 Volumetric efficiency is the ratio of the capacity of the compressor cylinder to the displacement of the cylinder.

2.30 Stage compression efficiency is the ratio of the isentropic power to the indicated power of each stage.

2.31 Mechanical efficiency is the ratio of the summation of the compressor indicated power to the shaft power. In the case of integral steam or integral internal combustion engine driven compressors, the over-all mechanical efficiency is defined as the ratio of compressor indicated power to indicated power of the power cylinders. (See Par. 4.36 through Par. 4.41 for shaft power of electric motor drives).

2.32 Efficiency of the compressor is the ratio of the isentropic power to the shaft power.

SECTION 3, GUIDING PRINCIPLES

3.01 There are several pertinent items upon which agreement must be reached by the parties to the test prior to the beginning of the test. (See Par. 3.02.)

3.02 Specific items upon which agreement shall be reached prior to the beginning of the test are:

(*a*) Object of tests and methods of operation.

(*b*) The intent of the contract and/or specification as to operating conditions, guarantees, etc.

(*c*) Method of maintaining constant test conditions.

(*d*) Facilities for maintaining operating conditions required by the contract, such as constant power supply, cooling water, temperature control, and isolation of compressor during test.

(*e*) Instruments to be employed, (see Par. 4.02), and measurement techniques to be employed.

(*f*) Procedures for the calibration of all instruments.

(*g*) Organization of observers, arrangements for their direction, procedures for recording the readings and observations, and for calculating of the test results.

(*h*) Frequency of observations.

(*i*) Corrections, other than those defined in this Code, for deviation of test conditions from those specified.

(*j*) Tolerances required because of instrument errors.

(*k*) Duration of tests.

(*l*) Duration of operation at test load before readings are commenced.

(*m*) Procedure for examination of machine and for preliminary tests.

(*n*) Position of manually operated valves.

(*o*) Procedures for averaging and interpreting data.

(*p*) Procedure for condensate collection and measurement.

(*q*) Time signals.

(*r*) Allowances for different losses not chargeable to the equipment being tested.

(*s*) Sources and use of data on thermodynamic properties.

3.03 Unless otherwise provided by a contract, parties to the test shall designate a person to direct the test and serve as arbiter in event of dispute.

3.04 Preparation for Test. Before tests are commenced, the compressor shall be placed at the disposal of the supplier for examination in order to ascertain whether it is in suitable condition for the conduct of the test. The supplier may make any permanent adjustments that he may find necessary to place the unit in proper operating condition. The availability of the compressor and the time for these adjustments shall be a matter of mutual agreement.

3.05 Alternate arrangements of the primary metering element and other test apparatus described in Section 4 are provided for the various types of compressors. Selection of the arrangement to be used shall be determined according to the operating conditions.

3.06 Time of Acceptance Test. In the case of an acceptance test, unless otherwise agreed, the test shall be conducted within six months after the compressor is first put into commercial service and after inspection and necessary adjustments have been made by the supplier.

3.07 Pressure Pulsation. Prior to the test, the amplitude of pressure waves prevailing in the pipe system shall be measured at each of the stations described for inlet and discharge pressures. If the maximum amplitude (crest to trough) found exceeds 10 per cent of the average absolute pressure, methods of correction shall be mutually arranged. (See Fig. 1.)

3.08 Cooling Water. Alternate methods are provided for the control of cooling water quantity and temperature. The parties to the test may elect to test the compressor and the intercooler separately or as a complete unit. If tested separately, the control of cooling water temperature and flow rates may be adjusted to meet the permissible range of correction for intercooling.

3.09 Isolation of Compressor. All unused pipe connections from the compressor discharge shall be blanked. If this is not possible, the connections shall be broken at a suitable place, so that the outlets may be under constant observation for possible leakage.

3.10 Transmission Loss. If the compressor is driven through belts or gears, the procedure for ascertaining transmission losses shall be established by agreement. (Refer to Par. 4.42 through 4.44 and Par. 5.11.)

3.11 Power Measurements. If the compressor is driven by an internal combustion engine, a steam engine (or steam cylinders), steam turbine, or a gas turbine, the fuel rates, or the steam rates, shall be measured in accordance with ASME Performance Test Codes for these prime movers. Instructions for the measurement of electric power are given in Par. 5.10 of this Code.

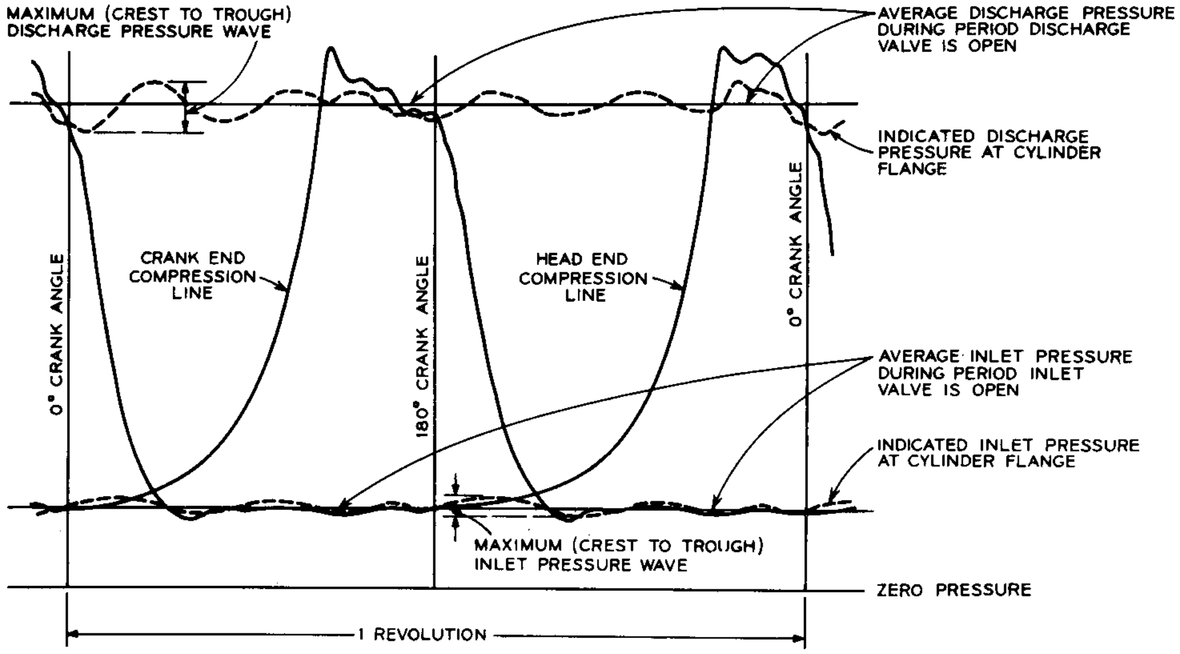

FIG. 1 PRESSURE TIME DIAGRAMS OF DOUBLE-ACTING COMPRESSOR, ILLUSTRATING CYLINDER PRESSURES AND PRESSURE WAVES AT MEASURING STATIONS FOR INLET AND DISCHARGE PRESSURES (SEE PARS. 3.07, 4.21, AND 4.22)

Cradle or torsion dynamometers may be used to measure power input in accordance with I & A, Part 7 on Measurement of Shaft Power.

3.12 **Over-all tolerances** or margins on power input, capacity, or pressure guarantees that may be part of the agreement are not within the scope of this Code. Allowances for error in measurement, however, are permissible provided they are agreed upon in writing by the parties to the test and a statement of the agreement is included in the test report.

3.13 **Preliminary Tests.** Preliminary tests may be run for the purpose of:

(*a*) Determining whether the compressor and associated pipe system is in a suitable condition for the conduct of the test.

(*b*) Checking of instruments.

(*c*) Training of personnel.

After a preliminary test has been made, this test may, by agreement, be considered a Code test, provided all requirements for a code test have been met.

3.14 **Duration of Test.** The controlling factor is the time required to record enough observations to demonstrate the uniformity of running conditions during the test. For compressors using constant speed drivers such as electric motors, the minimum duration of a test run shall be 30 min. with not less than five consecutive readings for each significant instrument. For compressors with variable speed drivers, the minimum duration shall be 60 min. with not less than eight consecutive readings for each instrument.

3.15 **Test Conditions.** The significant factors to be considered in planning a test or appraising the results obtained are:

(*a*) Pressure ratio

(*b*) Inlet pressure, each stage

(*c*) Inlet temperature, each stage

(*d*) Discharge pressure, each stage

(*e*) Discharge temperature, each stage

(*f*) Composition of the inlet gas including moisture content and adjusted composition of succeeding stages if condensation or removal of any constituent has occurred

(*g*) Capacity

(*h*) Speed

(*i*) Interstage removal or injection of gas or liquid

(*j*) Cooling water temperatures and flow rates

(*k*) Power input

3.16 When testing a compressor, every effort shall be made to have the operating conditions as near as possible to those specified. The maximum deviation for which adjustment may be applied to any of the variables is given in Table I. Under these conditions, the values of the variables, as calculated under the rules of this Code, shall be accepted as indicating the performance of the compressor, unless otherwise agreed.

Table I Maximum Allowable Variation in Operating Conditions

Variable	Deviation of test from value specified (plus or minus)	Fluctuations from average during any test run (plus or minus)
(*a*) Inlet pressure	2 per cent of abs pressure	1 per cent
(*b*) Pressure ratio[1]	1 per cent	
(*c*) Discharge pressure[1]		1 per cent
(*d*) Inlet temperature		1 F
(*e*) Inlet temperature deviation for any stage	15 F	
(*f*) Speed	3 per cent	1 per cent
(*g*) Cooling water inlet temperature		2 F
(*h*) Cooling water flow rate		3 per cent
(*i*) Metering temperature		3 F
(*j*) Primary element differential pressure		2 per cent
(*k*) Voltage	5 per cent	2 per cent
(*l*) Frequency	3 per cent	1 per cent
(*m*) Power factor	1 per cent	1 per cent
(*n*) Belt slip	3 per cent	None

[1] Discharge pressure shall be adjusted to maintain the pressure ratio within the limits stated.

3.17 The compressor shall remain in continuous steady operation throughout the test period. The readings recorded shall be consecutive and consistent within themselves. The maximum permissible fluctuation of any individual reading from the average shall be within the limits of Table I. Tests shall not be run during periods of sudden and widely changing weather conditions.

3.18 If inconsistencies become evident, either during a test or during the computation of results, the test or tests shall be rejected, appropriate adjustments made to avoid recurrence of the problem, and the test repeated in whole or in part.

3.19 Instruments. The instruments required to conduct tests under this Code are specified in Section 4. Spare instruments shall be available to replace those which are liable to failure or breakage in service. Initial calibration of the instruments shall be made prior to the test. Recalibration after test shall be made for those instruments of primary importance which are liable to variations in their calibration as a result of use during test. Any change in the instrument calibrations which will create a variation of more than ± 1 per cent in any calculated quantity listed in Par. 3.15 may be cause for the rejection of the test.

3.20 Records and Test Reports. Only such observations and measurements need be made as apply and are necessary to attain the object of the test. Instrument indications, or readings, shall be recorded as observed. Original data sheets shall remain in the custody of the engineer in charge of the test. Facsimile copies of all data sheets shall be furnished to each of the parties. Corrections and corrected values shall be entered separately in the test record. The test shall be reported according to the Report of Tests, Section 6D.

SECTION 4, INSTRUMENTS AND METHODS OF MEASUREMENT

4.01 The Performance Test Code Supplements to Instruments and Apparatus (hereinafter referred to as I & A) provide authoritative information concerning instruments and their use. The guidance given in these publications shall be followed.

4.02 Necessary Instruments. The instruments and testing apparatus which may be necessary for the measurement of the quantities required in tests covered by PTC 9 are included in the following list:

(*a*) Barometers (I & A, Part 2)

(*b*) Thermometers or thermocouples and associated instruments (I & A, Part 3, Chapters 1, 3, 5, and 9)

(*c*) Pressure gages or manometers (I & A, Part 2)

(*d*) Differential gages (I & A, Part 2)

(*e*) Flow Measurement (ASME's "Fluid Meters – Their Theory and Application," Sixth Edition, Part II, "Flow Measurement")

(*f*) Revolution counters, speed measuring devices (I & A, Part 13)

(*g*) Psychrometer, dewpoint measuring apparatus (National Bureau of Standards Circular 512, "Methods of Measuring Humidity and Testing Hygrometers")

(*h*) Gas analyzing apparatus (I & A, Part 10)

(*i*) Indicators measuring pressure (see Par. 4.20)

(*j*) Appropriate electrical instruments for motor driven compressors (I & A, Part 6)

(*k*) Fuel measuring apparatus where the driver is an internal-combustion engine or gas turbine (ASME publication "Fluid Meters – Their Theory and Application," Sixth Edition, Part II, "Flow Measurement")

(*l*) Condensate measuring tank, scale tank, or steam flow measuring device, where the driver is a steam engine or a steam turbine (PTC 6-1964)

(*m*) Calorimeter for steam (I & A, Part 11)

(*n*) Provision for measuring condensate in inter and aftercooler

4.03 The complete assembly of test apparatus will require, in addition to the meter run, suitable throttle valves for controlling pressures, receivers for controlling pressure waves, facilities for the measurement and control of cooling water, pipe taps located properly for the measurement of pressures and temperatures, and means for collecting and measuring condensate. (See Figs. 2 through 6.)

4.04 A receiver and a throttle valve shall be used between the flow measuring element and the compressor. The physical arrangement of this apparatus is illustrated in Fig. 2. It is important that pulsations in compressors piping be minimized at the orifice or nozzle to insure accurate flow measurements. This can be accomplished in a number of ways. In a small compressor the use of a receiver having a volume forty times the single stroke displacement is usually effective in reducing pulsations. In a large compressor installation the piping should be designed to avoid objectionable pulsations by avoiding reasonant lengths of piping or by using pulsation dampeners.

4.05 Test Arrangements. Figure 2 shows alternate arrangements for gas flow measurement to accommodate various compressor types and operation conditions.

(*a*) Arrangement "A" shall be used for air or gas compressors where the discharge pressure is sufficient to provide adequate throttling and sufficient differential head on the flow measuring element and where the gas compressed can be discharged to the atmosphere.

(*b*) Arrangement "B" shall be used for vacuum pumps and exhausters where the inlet pressure to vacuum pump or exhauster is low enough to provide sufficient differential head on the flow measuring element and where the test can be made on air.

(*c*) Arrangement "C" may be used for a closed system to test boosters where the supply pressure is sufficient to overcome any throttling losses and differential pressure across the flow measuring element. If part of a loop system, provisions should be made for cooling. A loop system is not recommended for air or gases containing oxygen because of the danger of fire or explosion.

(*d*) Arrangement "D" may be used for a closed system to test the boosters where the discharge pressure is sufficient to provide throttling and differential head on the flow measuring element. For boosters where the throttling requirement can be met at either the inlet or discharge, the choice of arrangements "C" and "D" shall be optional.

Refer to ASME publication "Fluid Meters – Their Theory and Application," Sixth Edition, Part II, "Flow Measurement" for information regarding the types of flow meters applicable to each of the foregoing arrangements. This source also defines necessary meter installation details.

4.06 The diagram of Fig. 3 indicates essential test apparatus for a multistage compressor where the gas is discharged to atmosphere through the flow measuring element. The facilities for measurement and control of cooling water may apply to any water-cooled compressor. The provisions for measuring condensate also apply to any compressor where

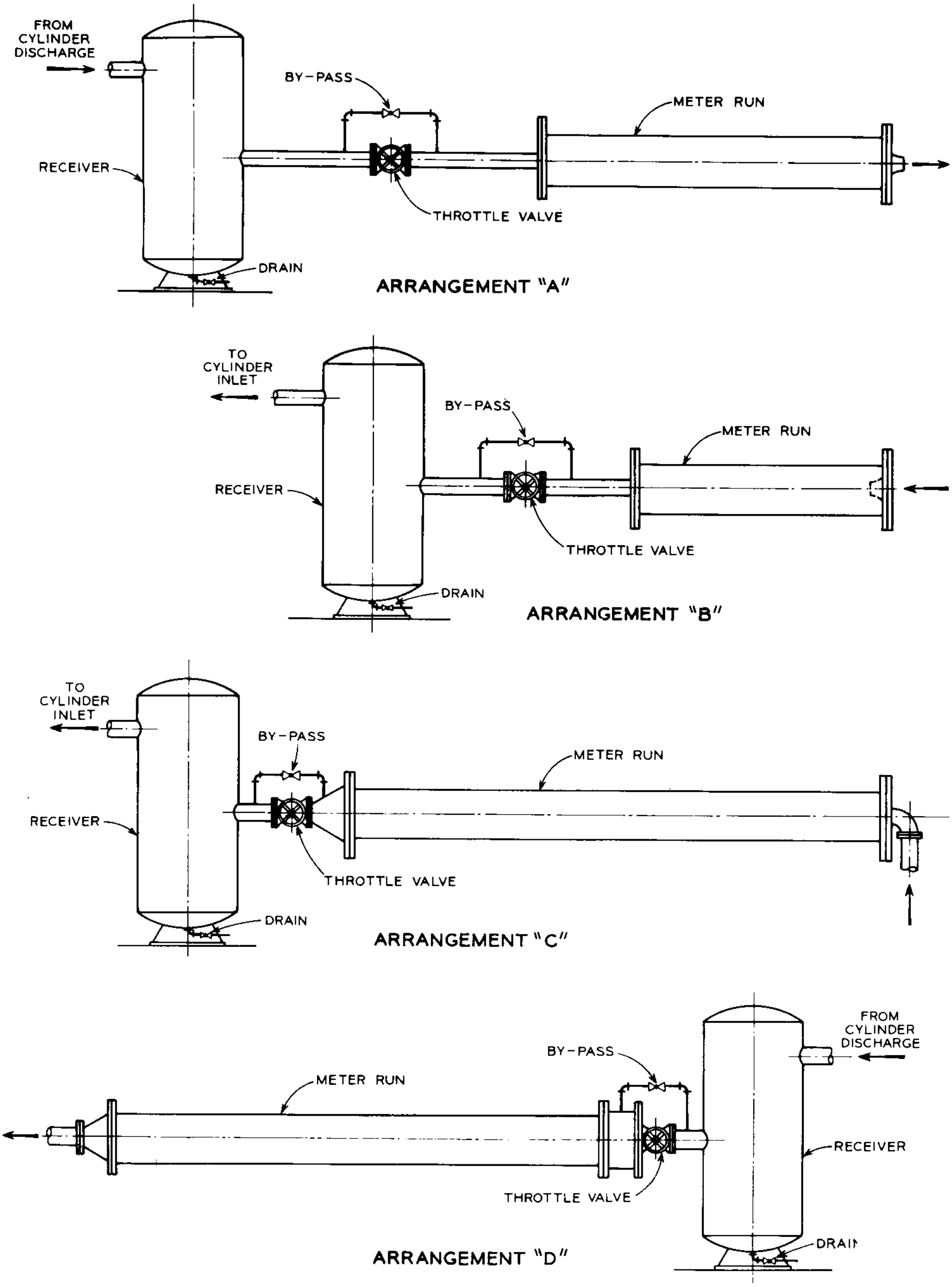

FIG. 2 METERING ARRANGEMENTS

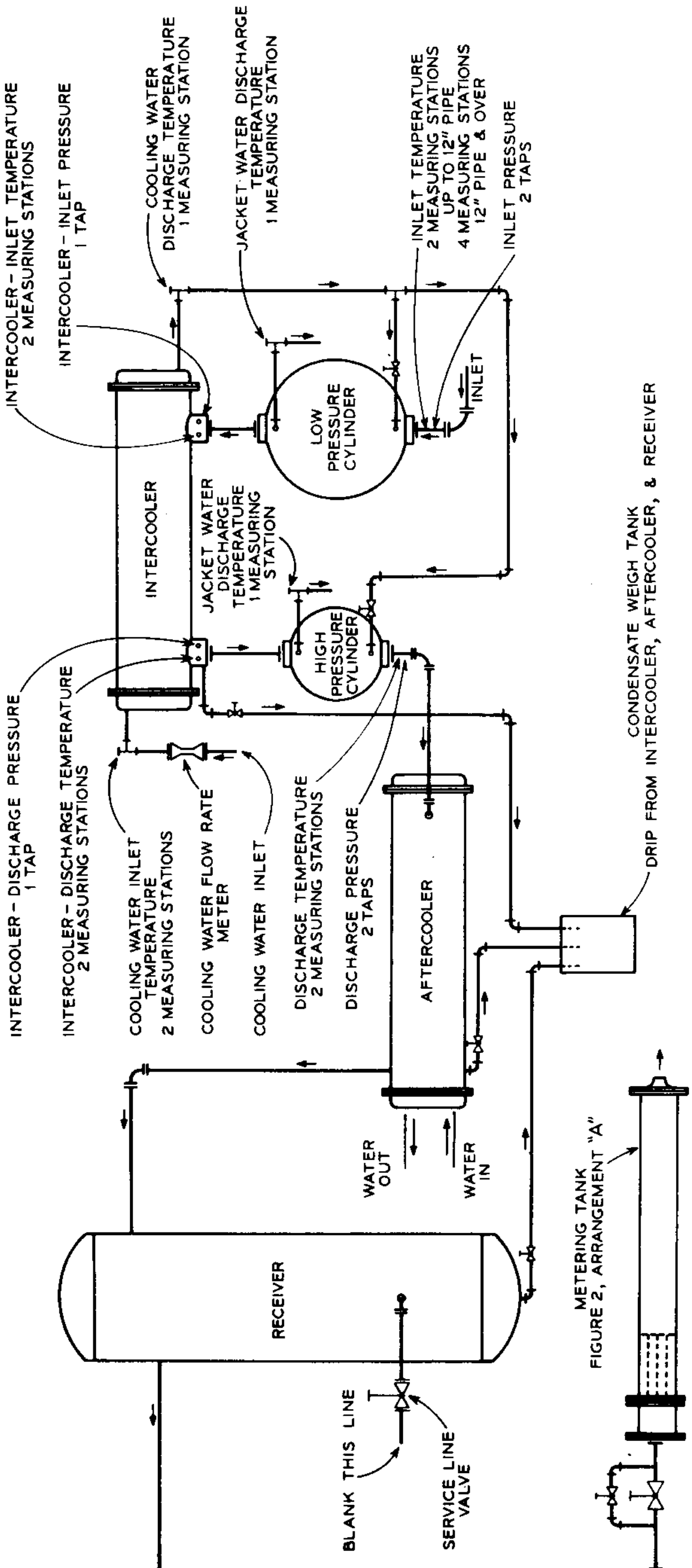

FIG. 3 TEST ARRANGEMENT FOR MULTISTAGE COMPRESSOR

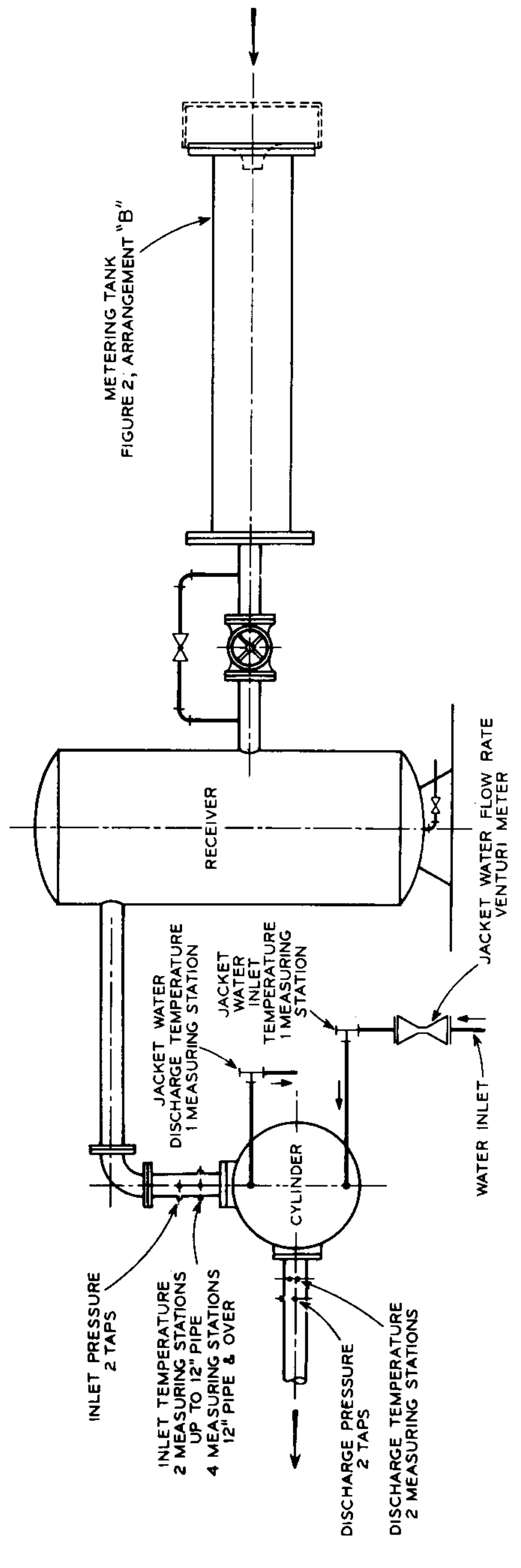

FIG. 4 TEST ARRANGEMENT FOR VACUUM PUMPS AND EXHAUSTERS

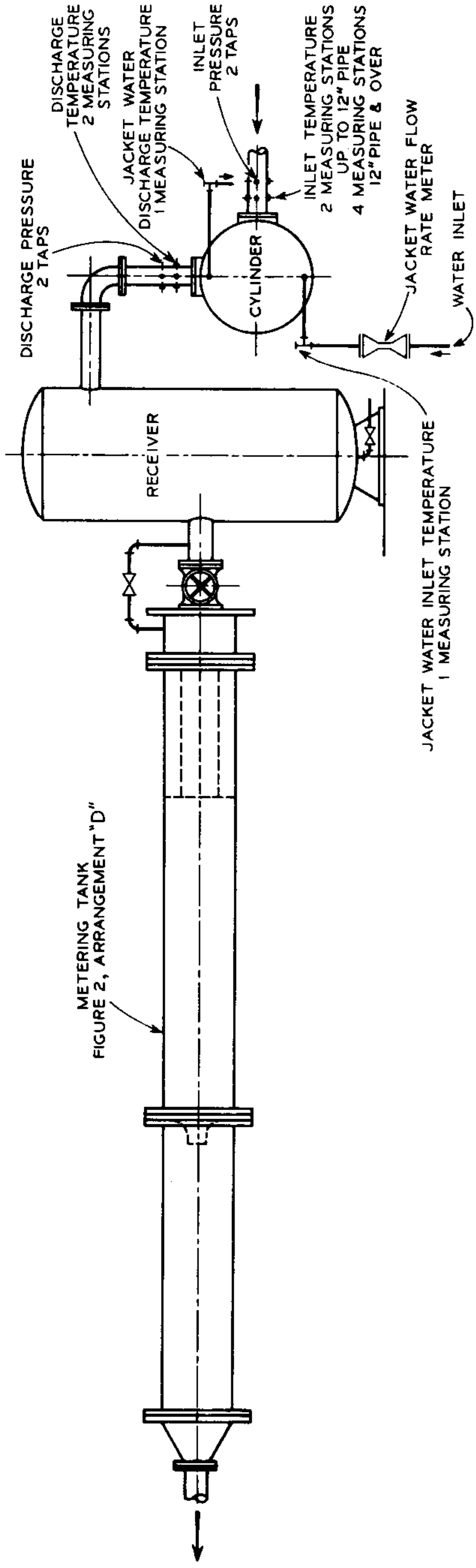

FIG. 5 TEST ARRANGEMENT FOR GAS BOOSTER

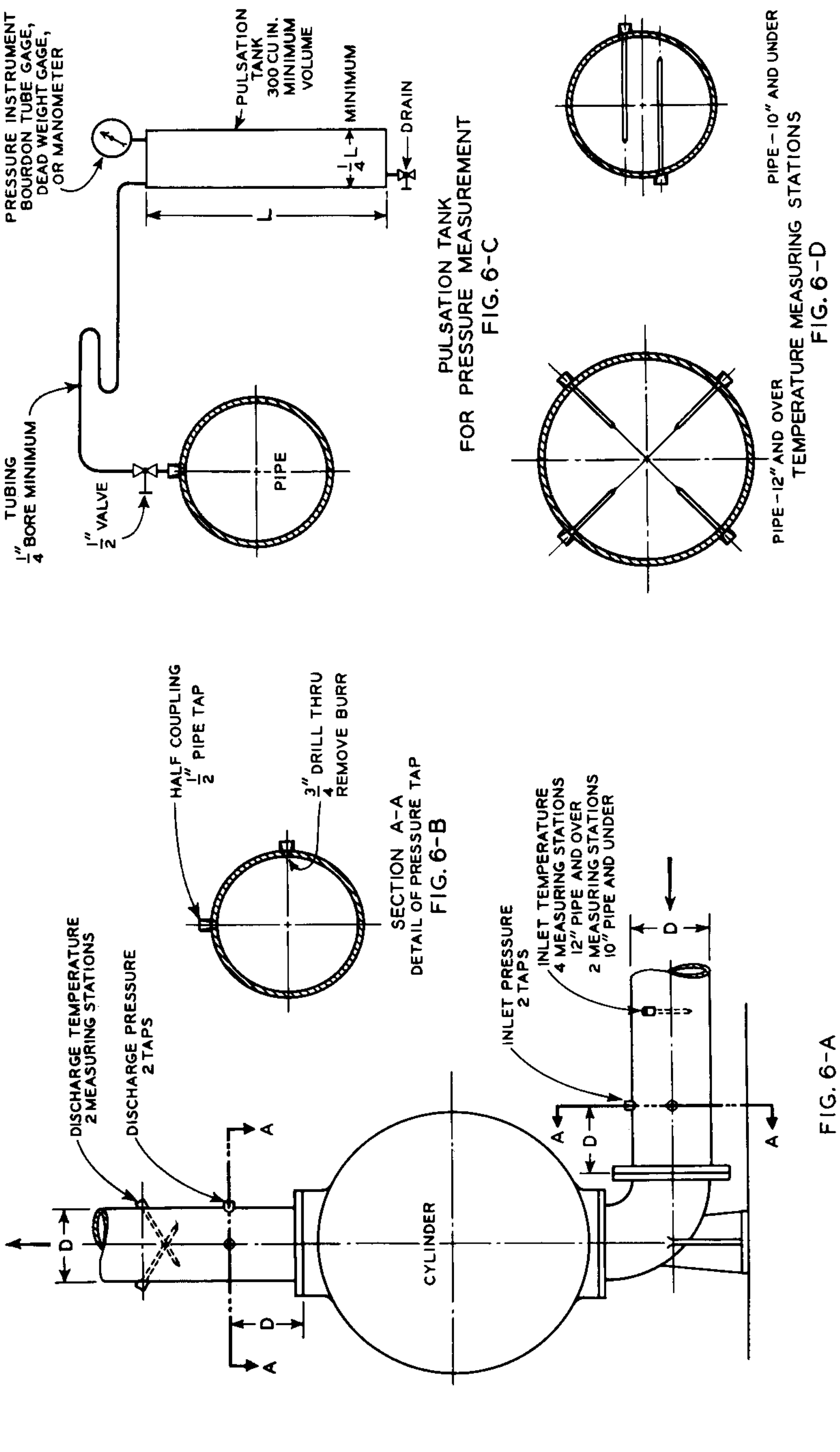

FIG. 6 DETAIL OF PRESSURE AND TEMPERATURE MEASURING STATIONS

intercoolers or aftercoolers are used. Figure 4 illustrates the arrangements used for testing vacuum pumps and exhausters.

4.07 If the nature of the gas handled is such that discharge into the atmosphere is impractical, the capacity may, subject to agreement, be measured by means of a flow measuring element in the discharge line, provided the pulsation is within the limits specified in Par. 4.12. The complete setup for a flow measuring element (nozzle) used in a closed system is shown in Fig. 5.

4.08 Prior to the test, measurement of pressure waves (see Par. 3.07) shall be made with the compressor operating at specified conditions of speed, inlet pressure, and discharge pressure. The pressure wave measurement shall be made with sensitive indicators of the type specified in Par. 4.20. If the pressure wave measured at cylinder inlet and discharge connections, Par. 4.14, is found to exceed 10 per cent, crest to trough, of the prevailing average absolute pressure, the cause of the pulsations shall be corrected before proceeding with the test because of their adverse influence on compressor performance. During these pressure wave tests the compressor control device shall be set for full load capacity and any other capacity which may be required. When the pressure waves exceed, 10 per cent crest to trough, of the specified average and cannot be corrected, the test shall not be undertaken under the provisions of this Code, unless agreed to in writing by all interested parties to the test.

4.09 Pulsations at the flowmeter shall be low enough to permit meeting the requirements of Par. 4.12.

4.10 Capacity Measurement. This Code provides for measurement of capacity by any of the means covered by ASME's "Fluid Meters – Their Theory and Application," Sixth Edition, Part II, "Flow Measurement." However, it must be recognized that a condition of this publication is a nonpulsating flow and that this is not necessarily achieved by the mere installation of a meter run that conforms to Part II of "Fluid Meters" with regard to location of taps, length of straight run, proximity of elbows, etc. A smooth flow condition is essential for accurate metering. See Par. 4.51 for a method of exposing the existence of gross flow meter error and an alternate method of capacity determination if gross error is found to exist. For alternate methods of capacity determination of a rotary type compressor see Pars. 4.52 and 4.53.

4.11 The differential pressure across the orifice, nozzle, or other primary element shall be measured from static taps with two independent instruments. The taps are spaced 90 deg circumferentially. Upstream stations shall be located one pipe diameter *(D)* from the inlet face of the primary element; downstream stations shall be $1/2\ D$ from the primary element face. Upstream pressure shall be measured for all arrangements. The differential pressure across the primary element shall be measured from static taps with two independent differential pressure gages. One gage is to be the mercury manometer type and the other the aneroid type so that instruments with different natural frequencies are utilized. The differential gages are to have a calibrated accuracy of 1/2 of 1 per cent at a differential reading within 10 per cent of expected reading at specified conditions, and at an ambient temperature within 10 F of test conditions. The differential pressure gages shall be installed in accordance with recommendations described in I & A, Part 5, Chap. 4.

4.12 The instruments sensing pressure differential shall indicate substantially steady pressure conditions across the primary element. If the differential pressure, as indicated by either instrument (Par. 4.11) varies more than 2 per cent from its mean differential pressure, the cause shall be investigated and the condition corrected. The mean differential pressure as indicated by each of the two differential instruments shall be in agreement within 1 per cent. Such agreement does not assure that accurate metering prevails in the presence of pulsating flow. (See Par. 4.51.)

4.13 Flow metering temperature measuring stations shall be located in accordance with "Fluid Meters – Their Theory and Application," Sixth Edition, Part II, "Flow Measurement." The precautions of Par. 4.28 shall apply.

4.14 Pressure Measurement. Measurements of the average static pressure are required:

(*a*) At the inlet of the compressor

(*b*) At the discharge of the compressor

(*c*) At the inlet and the discharge of each stage for multistage machines.

The pressure measuring stations shall consist of two taps spaced at 90 deg and located as shown in Fig. 6.

4.15 Wherever integrated time average pressure is required (steam pressures, discharge pressure, intake pressure, etc.), a suitable indicator should be used for taking such cards. In addition to that, and for matter of record, a Bourdon gage, a deadweight gage or manometer shall be connected at or near the same point.

4.16 Pressures below 20 psi shall be measured with U-tube manometers filled with stable liquids of suitable gravity. The bore of the glass tubes shall not be less than 3/8 in. The manometer scales shall be accurately graduated with minimum divisions preferably of 1/10 or 1/20 in. Manometer indications shall be interpolated to the nearest 1/50 in.

4.17 Pressures of 20 psi and above shall be measured by gages, either of the Bourdon tube, or the deadweight type. Bourdon tube gages shall be selected for an operating range above midpoint of the scale. The diameter of the scale and the arrangement of the graduations shall permit readings within 1/2 of 1 per cent of the expected pressure measurement. Deadweight gages shall have adjustable weights such that any error introduced in their use shall not exceed 1/2 of 1 per cent.

4.18 When Bourdon or deadweight gages are used for the measurement of inlet and/or discharge pressure, they shall be connected to the pressure measuring station through a receiver as illustrated in Fig. 6-C. This receiver shall have a minimum volume of 300 cu in. and the diameter must be at least 1/4 of the receiver length. Inasmuch as the receiver system comprises a Helmholtz resonator, care must be taken that its acoustic natural frequency be removed from the fundamental frequency of the compressor or any multiple thereof. Fundamental compressor frequency is defined as compressor rotative speed in rpm divided by 60.

The Helmholtz frequency of the receiver system can be computed from computed from the equation:

$$f_n = \frac{a}{2\pi} \sqrt{\frac{A}{VL}}$$

Where: f_n = resonant frequency in cycles/sec
a = acoustic velocity of the gas in ft/sec
A = internal cross-sectional area of the connecting tubing expressed in ft^2
L = length of connecting tubing in feet.
V = volume of receiver in ft^3

The receiver system should be proportioned so that f_n is less than one-half of the fundamental compressor frequency or midway between consecutive multiples of fundamental compressor frequency.

4.19 The pressure connections between the pulsation receiver and the compressor pipe shall be made with tubing having a minimum bore of 1/4 in. Throttle valves or snubbers shall not be used. Oscillation of the pressure measuring instrument shall be eliminated by increasing the volume of the receiver or the length of the tubing.

4.20 The amplitude of pressure waves shall be measured with indicators sensitive to pressure changes of 1/4 of 1 per cent of the average absolute pressure. These instruments shall have a minimum frequency of response of 20 times the compressor speed. Engine-type indicators may be used where the compressor speeds are 300 rpm or lower. For higher compressor speeds, indicators of the electrical type, or the balance pressure diaphragm type, shall be used.

4.21 Discharge Pressure. The flow through the discharge pipe to the receiver or compressor piping system is intermittent and pulsating in character for all displacement type compressors. If the acoustical resonances in the discharge piping approach the fundamental frequency of compressor speed or its multiples, pressure waves of considerable amplitude will be induced in the discharge piping making accurate determination of the average discharge pressure very difficult if not impossible. This Code defines discharge pressure as the average shown by a pressure-time indicator diagram, taken at a point on the discharge line immediately adjacent to the compressor cylinder, during that period which the discharge valves are open. Figure 1 shows a typical pipe indicator diagram and cylinder indicator diagram with that portion of the former marked to show the average discharge pressure as defined in this Code.

Upon agreement by the parties to the test, a Bourdon gage may be used to measure discharge pressure.

4.22 Inlet Pressure. The performance of a compressor is extremely sensitive to variations in inlet pressure, particularly to periodic fluctuations in pressure which approach resonance with the compression cycle or rotative speed of the unit. (See Par. 1.03.) When inlet pipes or ducts are connected, the intake pressure shall be regarded as the average pressure shown by a pressure-time indicator diagram, taken at a point adjacent to the compressor inlet, during that period which the inlet valves are open, see Fig. 1. The inlet pressure of an air compressor operating without inlet pipe shall be measured by a barometer.

4.23 Atmospheric Pressure. Atmospheric pressure shall be measured with a Fortin type mercurial barometer fitted with a vernier suitable for reading to the nearest 0.002 in. It shall have an attached thermometer for indicating the instrument temperature. It shall be located at the floor level of the compressor and supported on a structure free of mechanical vibrations.

4.24 Inlet Temperature. The inlet gas temperature shall be measured near the cylinder inlet flange or connection but sufficiently distant to avoid radiation and conduction errors from cold or hot surfaces. The temperature measuring stations shall not be less than one pipe diameter distant from the cylinder. Four temperature measuring instruments shall be used for pipe sizes of 12-in. diam and larger. For smaller pipes, not less than two instruments shall be used. (See Figs. 3, 4, 5, and 6.)

4.25 Precaution shall be taken to avoid stratified air temperature distribution within the gas stream. For air compressors with inlets or filters located near the cylinder, hot-air stream shall be effectively shielded from the inlet opening. If the thermometers do not agree within 2 F, the cause shall be investigated and the condition corrected.

4.26 **Discharge and Intercooler Temperatures.** The gas temperatures at the cylinder discharge flange or connection, or entering and leaving the intercoolers, shall be measured with the care and precaution described in Pars. 4.24 and 4.25. Not less than two instruments shall be used at each station.

4.27 **Flow Metering Temperature.** If the flow metering temperatures in the nozzle pipe differ from that of the surrounding atmosphere by more than 75 F, the meter run shall be thermally insulated. If the thermometers do not agree within 2 F, the cause shall be investigated and the discrepancy removed. (See Par. 4.28.)

4.28 **Temperatures** shall be measured by calibrated thermometers, thermocouples, or resistance temperature detectors in conformity with PTC 19.3 – 1961.

4.29 **Composition of the Gas.** The chemical composition and the physical properties of the gas entering the compressor during the test shall be determined.

4.30 **Psychrometric Measurements.** When the gas compressed is air, periodic readings of wet- and dry-bulb temperatures shall be made to determine the relative humidity of the air supply at the inlet of the compressor during the test. These readings shall be taken with a sling psychrometer. For tests with a closed system, humidity measurements shall be made through the use of a dew point instrument.

4.31 **Power consumption** of multistage compressors is commonly stated with specified cooling water temperatures and flows to the intercoolers. Compressor tests may be run with water temperatures other than specified but the water flows should be adjusted so that the intercooler outlet gas temperature is the same as those specified conditions. When the correct intercooler outlet gas temperature is not obtainable, adjustments shall be made in accordance with Par. 5.14.

4.32 **Intercooler Test.** The intercooler shall be tested separately. The specified water flow shall be maintained and the compressor shall be operated at the same speed, pressure, and inlet conditions which prevailed during the power and capacity test. Cooling water inlet and outlet temperatures, and gas inlet and and outlet temperatures shall be recorded for each intercooler. Pressures and speed shall also be recorded to show the compressor operating conditions. The cooling water flow rates shall be measured with meters accurate within 2 per cent as determined by calibration. The inlet water and gas temperatures shall not fluctuate more than 2 F. The performance of the intercoolers will be indicated by the temperature difference between the inlet water and the outlet gas.

4.33 **Speed measurement.** A revolution counter free of slip shall be used to record the total number of revolutions of the compressor during a test run (I & A, Part 13). The turnover rate shall be such that the register can be read at uniform time intervals throughout the test period to indicate fluctuation or change in speed. The fluctuation shall not exceed 1 per cent of the average value for a test run and the deviation from specified speed shall not exceed 3 per cent.

4.34 **Time Measurement.** The date and time of day at which each individual test reading is taken shall be recorded. The time pieces used by the individual observers shall have been coordinated with a master clock and shall be accurate to within 30 sec per day.

4.35 **Power Measurement.** The power input at the compressor may be measured directly by reaction mounted drivers, or a torquemeter, or indirectly determined from measurements of electrical input to a driving motor. Detailed instruction on the measurement of horsepower will be found in I & A, Part 7 on Measurement of Shaft Horsepower, PTC 19.7.

4.36 **Measurement of Shaft Input Power of Electric Motor-Driven Compressors.** The shaft power input to a motor driven compressor may be computed from measurements of the electrical input to the motor terminals. The output of a motor may be calculated by subtracting segregated losses from the measured electrical input, or as the product of input and efficiency. Usual practice in determining efficiency is to accept the motor manufacturer's load-efficiency curves, and this is acceptable for a Code test. However, efficiency may be determined by an input-output test where output is measured on an accurate dynamometer. For any efficiency determination, the supply line voltage shall be the same as that used for the compressor test.

4.37 Efficiency determination by input-output measurements is not practical for large motors of ratings above 500 hp. For these the loss method shall be used. The segregated losses of an induction motor shall include friction and windage, core loss, I^2R loss of the rotor and the stator, and a load loss. These measurements shall be made in accordance with USASI Standard C50.4 – 1965, published by the IEEE. Measurement of losses for large synchronous motors require special facilities and is usually done by the motor manufacturer.

4.38 The electric power input to the motor shall be measured by the instruments connected at the motor terminals. For three-phase ac motors the power input shall be measured by either the two- or three-wattmeter method. The power input of a dc motor shall be measured by the voltmeter-ammeter method. The detailed instructions for the measurement of electrical power as given in I & A, Part 6 on Electrical Measurements in Power Circuits, PTC 19.6 – 1955, shall be mandatory.

4.39 The indicating electric meters for ac power shall be of the portable type. They shall be matched with precision transformers of the correct capacity for Class III accuracy. The transformer ratios shall permit reading of the meters above 1/3 of the scale range. The accuracy of the meters shall be ±1/4 per cent of the indicated value.

4.40 Calculations of electrical power shall include calibration corrections for the meters and transformers. Electric meters shall have their calibrations checked against primary standards prior to installation and again after the test. The transformers shall be measured for ratio and phase angle at the burdens prevailing during the test. The results of these calibrations shall be used to establish ratio and phase angle correction factors.

4.41 Synchronous Motors. The performance of a compressor is generally expressed in terms of power input to the compressor shaft and the electric motor power output is required. This is the product of the ac input to the stator terminals and the motor efficiency. The latter term (power output/ac power input) is the efficiency shown on the motor manufacturer's curves. However, sometimes the compressor performance is also expressed in terms of electrical input, and the dc excitation power must be added to the "ac input to stator terminals" to get the total power supplied to the motor. The motor field rheostat losses are excluded from the exciter power. For belted or direct connected exciters, the exciter losses should be deducted, as they are a part of the ac power input readings. Refer to PTC 19.6.

4.42 Gear Losses. Where gears are used between a driver and the compressor, it is necessary to subtract the friction and windage loss of the gear to obtain a net power input to the compressor. For purposes of this paragraph it will be assumed that all the gear losses will appear as heat in the lubricating oil and as heat losses, primarily by convection, from the gear box. The gear loss appearing in the lubricating oil may be determined by measuring the flow rate by calibrated flowmeters and the temperature rise. The additional external loss to the atmosphere may be determined with accuracy from measurements of the exposed surface area, the average temperature of the surface, and the ambient temperature by the formula: *

$$q_r = \frac{S_r (t_r - t_a) h_r}{60} \tag{4.42.1}$$

where:

q_r = External heat loss, Btu/min
S_r = Heat transfer surface, ft^2
h_r = Coefficient of heat transfer, Btu/hr ft^2F (the value of h_r varies from about 3.5 for a gear box with 140 ft^2 area to 4.0 for a gear box with 20 ft^2)
t_r = Temperature of surface, F
t_a = Temperature, ambient F

The oil temperature, viscosity, and flow rates shall be uniform and duplicate the test condition where the loss measurements are made separately. The gear manufacturer's guaranteed efficiency may be used by mutual agreement.

4.43 The lubricating oil for a gear shall be checked to ascertain if the viscosity conforms with the specification. The temperatures of the gear and the oil shall be recorded.

4.44 Belt Losses. For belts, the tension and correct operating conditions shall be stated. Belt slip shall not exceed 3 per cent during any test. The slip shall be measured by revolution counters, independently driven from the motor and the compressor. The true pulley ratio shall be determined from counter readings taken at no load.

* Before calculating gear heat losses by this method, it is suggested that the following reference items be read:

"The Thermal Problem of Enclosed Gear Drives," E. J. Wellauer, AGMA Semi-Annual Meeting, Oct. 31, 1951.

American Gear Manufacturers Assn., Standard 420.02, Feb. 1951.

American Gear Manufacturers Assn., Standard 420.03, Dec. 1963. AGMA Standard Practice for Helical and Herringbone Gear Speed Reducers and Increasers.

4.45 Condensate. For test arrangements where vapor is condensed within the system before the gas is measured, as in Fig. 3, an adjustment shall be made to the capacity measurement for the amount of condensate removed.

4.46 Condensate drains shall be located to drain all separators, the receivers, and any other part of the system where liquid is likely to collect. These drain points shall be blown down at regular intervals or as often as necessary to prevent liquid carryover through the nozzle. The time interval between successive blowdowns shall be observed, and the amount of liquid drawn shall be weighed. These items shall be recorded in the data sheets, in a manner suitable for computing condensation rate for the period of the test run.

4.47 Control of Pressure Ratio. For tests where inlet pressure cannot be precisely controlled at the specified value, the discharge pressure shall be adjusted to keep the pressure ratio within the limits specified in Table I. However, power corrections as described in Par. 5.14 shall be made for deviations in inlet pressure and pressure ratio.

4.48 Density Determination. For tests with air, relative humidity shall be determined from measurements of the wet- and dry-bulb temperatures and the psychrometric tables or charts. For calculation of density refer to Par. 5.07.

4.49 For gases other than air and for gas mixtures, density shall be computed from values of specific gravity measured by the gas balance, or shall be calculated as indicated in Par. 5.07 in which case laboratory analysis must be made to determine mol fractions of the constituents of gas mixtures and the compressibility for the gas or gas mixture must be obtained from the thermodynamic data of the constituents.

4.50 Use of Indicators. Indicators may be used to determine compressor ihp, indicated power, compressor inlet and discharge pressures, and volumetric efficiency. For description and use of indicators see PTC 19.8.

4.51 Indicated Capacity. Indicated capacity may be determined from indicated volumetric efficiency. Indicated volumetric efficiency of an end of a compressor cylinder shall be determined from the *pressure-volume diagram* by projecting the diagram pressure at the end of the inlet stroke to its intersection with the expansion line. This intercepted length when divided by total diagram length in the volume direction is volumetric efficiency expressed as a fraction.

Indicated capacity may be used as a check to expose gross error in the flow meter determination of capacity. If this check shows the flow meter determination of capacity to be unreliable, indicated capacity may be taken as the actual flow by agreement of the parties to the test.

Alternate Method for Measuring Capacity of Low-Pressure Rotary-Type Blowers

4.52 Slip Test Method of Measuring Capacity of Dry Rotary Compressors. This is an alternate method of measuring capacity. When the absolute pressure in the nozzle pipe exceeds one-half of the absolute discharge pressure of the displacement blower or booster, the capacity for displacement blowers and boosters of the rotary type in which volumetric clearance is zero, may be determined by subtracting the leakage past the rotor from the gross displacement. The displacement is the product of the volume displaced per revolution and the normal speed in rpm. The volume displaced per revolution may be determined from measurements of the blower. The leakage past the rotor is a product of the displacement per revolution and the number of rpm required to maintain a predetermined pressure with the discharge pipe from the blower or booster closed and the inlet pipe opened to atmosphere. The leakage test may be conducted at the same pressure and temperature as the specified working conditions, or the test may be run with a differential of 1 lb/in^2 across the rotor and correction made for obtaining the slip at specified conditions by using the following formula:

Slip at specified condition may be determined from the equation

$$S_c = S_t \sqrt{\frac{\Delta p_c}{\Delta p_t} \times \frac{T_c}{T_t} \times \frac{G_t}{G_c} \times \frac{p_t}{p_c} \times \frac{Z_c}{Z_t}} \quad (4.58.1)$$

Where:

S_c = slip at specified conditions, cfm
S_t = slip at Δp_t approximately 1 psi differential, cfm = $(N_s \times CFR_B)$
N_s = rpm slip at Δp_t
CFR_B = displacement of blower, ft^3/rev
Δp_c = specified differential pressure, psi
Δp_t = test differential, approximately 1 psi
T_c = temperature (abs) at specified conditions, inlet temperature
T_t = temperature (abs) at test conditions, inlet temperature
G_t = specific gravity at test conditions

G_c = specific gravity at specified conditions
p_t = inlet pressure at test conditions, psia
p_c = inlet pressure at specified conditions, psia
Z_c = compressibility factor at specified condition
Z_t = compressibility factor at test condition
Inlet volume = $N_c \times CFR_B - S_c$
N_c = specified blower speed, rpm

This test is limited to *dry* Roots or screw-type rotary blowers. It cannot be applied to sliding vane or oil flooded machines.

4.53 Rotary Displacement Meter Test. Another means of testing rotary displacement blowers and boosters is by use of a calibrated rotary-type displacement meter located on the discharge side after the gas has passed through the restricting valve for loading the blower, so that the discharged gas will pass through the meter at approximately atmospheric pressure. In the case of vacuum air pumps, the meter should be located on the intake side with a restricting valve between the meter and the vacuum pump, so that the meter operates at near atmospheric pressure. The meter should be calibrated for accuracy.

SECTION 5, COMPUTATION OF RESULTS

5.01 The performance of any positive displacement compressor is expressed as follows:

(*a*) Motor or power-driven compressors: power input per unit of capacity.

(*b*) Steam or internal-combustion engine-driven compressors: quantity of steam at stated conditions or quantity of fuel at stated heating value respectively per unit of air or gas delivered.

The above performance characteristics shall be stated for specific conditions of operation as follows, for each stage, as applicable:

(*a*) Inlet pressure
(*b*) Discharge pressure
(*c*) Inlet temperature
(*d*) Discharge temperature
(*e*) Speed of rotation
(*f*) Gas composition, including moisture if any
(*g*) Cooling water flow rates to cylinders and coolers, individually
(*h*) Cooling water temperatures in and out of cylinders and coolers, individually
(*i*) Any other significant items such as auxiliary equipment driven from the compressor, lubricating oil conditions, etc.

5.02 The recorded data shall be scrutinized for consistency of operating conditions before final calculations are undertaken. The fluctuations of readings during any one test shall not exceed the limits presented in Table I. Readings showing excessive fluctuation may be discarded either at the beginning or end of a test run. Test results shall be calculated from the average values of the acceptable readings which shall be consecutive as described in Par. 3.14.

5.03 Capacity determinations shall be made with approved metering systems and procedures in full accordance with ASME's "Fluid Meters – Their Theory and Application," Sixth Edition, Part II, "Flow Measurement."

5.04 Capacity determinations in accordance with Par. 5.03 yield the flow through the metering element. If the metering element is located so that a part of the compressor flow stream can condense between the cylinder and the metering element, then a correction for the condensed fluid must be made to determine the capacity delivered by the compressor cylinder.

5.05 The flow of gas compressed and delivered is

$$w = w_m + w_f \qquad (5.05.1)$$

Where:

w = Flow rate delivered by compressor cylinder, lb/min
w_m = Flow rate measured by metering elements, lb/min
w_f = Condensate removed between compressor cylinder and metering element, lb/min

5.06 The flow (Q_1) at compressor inlet may be determined by:

$$Q_1 = \frac{w}{\rho_1} = \frac{w_m}{\rho_m} + \frac{w_f\, p_g\, Z_1}{\rho_g\, p_1\, Z_g} \quad \text{cfm} \qquad (5.06.1)$$

in which w, w_m and w_f are given in Par. 5.05 and

ρ_1 = density of gas mixture at inlet, lbm/ft^3
ρ_m = density of metered gas at inlet temperature and pressure, lbm/ft^3
ρ_g = density of saturated vapor at inlet temperature of the material represented by w_f, lbm/ft^3
p_g = saturation pressure at inlet temperature of material represented by w_f, psia
p_1 = inlet pressure, psia
Z_1 = compressibility factor at inlet pressure and temperature for material represented by w_f
Z_g = compressibility factor at saturation conditions corresponding to inlet temperature for material represented by w_f (Z_g may often equal unity)

5.07 Density is calculated from the following:

$$\rho_1 = \frac{144 p_1}{R T_1} \quad \text{for ideal gas} \qquad (5.07.1)$$

$$\rho_1 = \frac{144 p_1}{Z R T_1} \quad \text{for any gas} \qquad (5.07.2)$$

$$\rho_1 = \rho_{dg} \quad 1 - \left(1 - \frac{18.0}{M}\right) \frac{H_r\, p_g}{p_1} \qquad (5.07.3)$$

for gas containing water vapor

R = gas constant, $\frac{1545}{M}$ ft-lb per lb per F
p_1 = inlet pressure, psia
T_1 = inlet temperature, °R
ρ_{dg} = density of dry gas at inlet, lbm/ft^3
M = molecular weight
H_r = relative humidity at inlet, ratio
p_g = saturation pressure of water vapor at inlet temperature, psia
Z = compressibility factor

5.08 Isentropic power is calculated from

$$P_{\text{isen}} = \frac{w\,(h_2' - h_1)}{42.44} \qquad (5.08.1)$$

where:

P_{isen} = isentropic hp
w = mass flow rate, lbm/min
h_1 = enthalpy of gas at inlet pressure and temperature, Btu/lbm
h_2' = enthalpy of gas at discharge pressure and at the entropy corresponding to inlet pressure and temperature, Btu/lbm

Equation 5.08.1 is in suitable form for gases having charted or tabulated thermodynamic properties, and is correct for both real and perfect gases.

Another convenient form is as follows:

$$P_{isen} = \frac{w\left(\frac{n_s}{n_s-1}\right) Z_1 R T_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n_s-1}{n_s}} - 1\right]}{33000} \qquad (5.08.2)$$

where:

n_s = isentropic exponent relating pressure and specific volume in $(pv^n)_s$ = constant. n_s may be found from thermodynamic data, either directly or using the relation

$$n_s = \left[1 - \left(\frac{\partial\, l_{nz}}{\partial\, l_{nv}}\right)_T\right] k$$

where:

Z = compressibility at inlet
T = temperature at inlet, °R
R = gas constant, ft lbf per lbm °R
v = specific volume, ft³/lbm
k = ratio of specific heats of gas, c_p/c_v, at mean temperature and pressure of stage

5.09 For multistage compressors with or without intercooling between stages, the isentropic power is the sum of that of the individual stages. Proper values of temperature, pressure, and mass flow rate at the suction of each stage must be used; that is pressure, temperature and mass flow rate changes through the coolers must be accounted for.

5.10 Shaft Power. Shaft power may be measured by one of the methods given in Par. 4.35. Depending on the method used, the shaft power shall be computed in one of the following ways:

(a) Dynamometer tests

$$P_{sh} = \frac{\tau N}{5252} \qquad (5.10.1)$$

(b) Electric motor tests

$$P_{sh} = \frac{\text{Net kw input} \times \text{motor efficiency}}{0.7457} - \text{belt or gear loss hp} \qquad (5.10.2)$$

The net kw input and motor efficiency shall be determined as described in Par. 4.36 through Par. 4.41.

5.11 Transmission losses for belts and gears shall be used in accordance with provisions of the agreements required in Par. 3.10. Belt slip shall be computed from the ratio of revolution counter readings as provided in Par. 4.44.

Gear losses may be determined by the method described in Par. 4.42.

5.12 Adjustment of Results to Specified Conditions. It is unlikely that the actual test conditions will duplicate those specified. Corrections for deviations within limits given in Table I shall be made. Capacity shall be adjusted back to specified conditions for deviation in speed. Power shall be adjusted back to specified conditions for deviation in speed, inlet pressure and temperature, pressure ratio, intercooling and moisture content. It is assumed for the adjustments that the compressor efficiency as defined in Par. 5.15 remains unchanged for the deviations as limited by Table I.

5.13 Capacity Correction for Speed. The adjusted capacity, Q_{adj}, for speed variation as follows:

$$Q_{adj} = Q_m \times \frac{N_c}{N_{avg}} \qquad (5.13.1)$$

where:

Q_m = measured capacity calculated from observed results of test, cfm
N_c = specified speed, rpm
N_{avg} = average speed during test, rpm

5.14 Power Corrections for Inlet Conditions and Varied Interstage Conditions. The adjusted power, P_{adj}, may be calculated as follows:

$$P_{adj} = \frac{\Sigma P_{isen_c}}{\Sigma P_{isen_m}} \times P_m \qquad (5.14.1)$$

Where:

P_{isen_c} = Isentropic power for each stage calculated as in Par. 5.08 for specified conditions and summed for entire machine.
P_{isen_m} = isentropic power for each stage calculated as in Par. 5.08 for measured conditions
P_m = measured power

When vapor exists in the gas, P_{t_c} and P_{t_m} must be calculated with consideration given to the effects of condensation to flow and gas properties.

5.15 Other Performance Criteria. In addition to capacity, volumetric efficiency and mechanical efficiency (as defined in Section 2, Machine Characteristics) the following performance criteria, as applicable, may be determined as indicated:

(*1*) Where input power to compressor can be measured, as in motor driven compressors –

(*a*) Shaft power

(*b*) Compressor efficiency

$$\eta = \frac{\Sigma P_{isen_m}}{P_m} \times 100\%$$

(*c*) Power economy in units of power per selected unit of capacity

(*2*) Where input power to compressor cannot be measured, as in integral steam or internal combustion engine driven compressors

(*a*) Stage compression efficiency (each stage)

$$\eta = \frac{\text{Stage } P_{isen_m}}{\text{Stage Indicated } P} \times 100\%$$

(*b*) Power economy expressed as pounds of steam, pounds of liquid fuel or cubic feet of gaseous fuel for selected unit of capacity

(*c*) Unit thermal efficiency defined as

$$\eta = \frac{\Sigma P_{isen_m} \text{ (expressed in Btu/hr)}}{\text{Fuel/hr} \times \text{Heating value of fuel}}$$

(consistent units)

SECTION 6
APPENDIX A: USE OF STAGNATION PRESSURE AND TEMPERATURE TO DEFINE COMPRESSOR PERFORMANCE

6A.01 The performance characteristics of the positive displacement devices covered by this Code which depend upon thermodynamic properties for their determination are, under the provisions of this Code, based on stagnation conditions. However, it is recognized that the fluid velocities normally encountered at the inlet and discharge stations of such machines are so low as to make the contribution of the kinetic energy terms (as illustrated below) negligibly small. Where this is the case, static and stagnation conditions may be taken as numerically equivalent.

6A.02 The following will serve to indicate the distinction between the use of static and stagnation conditions. When conservation of energy is applied to a control volume analysis of a system comprising the machine contents from inlet to discharge flanges, the following equation results

$$h_{st_1} + \frac{u_1^2}{2g_cJ} + \frac{gy_1}{g_cJ} + W = h_{st_2} + \frac{u_2^2}{2g_cJ} + \frac{gy_2}{g_cJ} + q \qquad (6A.02.1)$$

Where:

W = work of compression, Btu

q = heat loss, and all terms are Btu/lb mass.

Subscripts 1 and 2 represent inlet and discharge respectively; subscript st indicates static enthalpies as specified by static pressures and temperatures.

In practice the elevations of inlet and discharge flanges are nearly the same, so, cancelling the elevation terms, eq. 6A.02.1 becomes

$$W = \left[h_{st} + \frac{u^2}{2g_cJ}\right]_2 - \left[h_{st} + \frac{u^2}{2g_cJ}\right]_1 + q \qquad (6A.02.2)$$

Since $h_{st} + \frac{u^2}{2g_cJ}$ = stagnation enthalpy, h_o, eq. (6A.02.2) may be written

$$W = h_{o_2} - h_{o_1} + q \qquad (6A.02.3)$$

6A.03 Hence, the external energy balance shown by eq. (6A.02.3) employs stagnation enthalpies which differ from static enthalpies by the appropriate kinetic energies of the fluid. (Note: that a fluid velocity of 4242 ft/min is required to result in a difference of 0.1 Btu/lb between static and stagnation enthalpies.)

6A.04 Stagnation enthalpy h_o, defined as $h_{st} + \frac{u^2}{2g_cJ}$, may be used to determine corresponding stagnation temperature and pressure from appropriate thermodynamic data bearing in mind that stagnation pressure is attained by adding kinetic energy conversion along an isentropic path.

When the assumption of perfect gas behavior is permissible (see 6B.03), the following equations yield stagnation temperature and pressure:

$$T_o = T_{st} + \frac{u^2}{2g_cJCp} \qquad (6A.04.1)$$

$$p_o = p_{st}\left(\frac{T_o}{T_{st}}\right)^{\frac{k}{k-1}} \qquad (6A.04.2)$$

Where T is expressed in °R

6A.05 The other use of the stagnation pressure and stagnation temperature in this Code is for the determination of capacity. Since capacity is the volumetric flow rate at inlet conditions, it is necessary to calculate the inlet density and convert the mass flow rate, which the fluid meter equations give, to the volumetric flow rate. However, capacity is defined herein in terms of the stagnation pressure and stagnation temperatures at inlet, thus requiring that the density be calculated in terms of those properties. This is convenient because it permits a clear definition of volume flow rate consistent with mass flow without referring to the design of the compressors.

SECTION 6
APPENDIX B: PROPERTIES OF GASES AND GAS MIXTURES

6B.01 Within the scope of this Code the primary intent relative to gases is that of determining enthalpy, entropy, specific heat, and compressibility at existing conditions. Written agreement between the parties involved in a Code test should delineate the data sources used for thermodynamic properties, the method of application and acceptable variations resulting from uncertainties in gas properties or thermal data. No specific recommendations are included in this Code as to permissible tolerances.

6B.02 Where the state of the gas is such that deviation from the perfect gas laws exists, methods must be used which recognize this deviation. If accurate thermodynamic properties for a gas based on experimental data or reliable mathematical or physical methods are available, these properties should be used with preference given to data based on experimental work. While there are several methods of approximating the thermodynamic state of gases, there is none that is sufficiently general to be made a mandatory procedure of this Code. Once the state of the gas is defined, presumably by pressure and temperature, the other properties of interest may be obtained from charts or tables, or calculated from references listed in Appendix E.

6B.03 Although it is recognized that the application of perfect gas laws and gas mixtures is justified for many cases of monatomic and diatomic gases and for other gases at high temperatures and/or low pressures, the parties should reach agreement based on acknowledged references on the applicability of perfect gas laws for their purposes.

Assuming perfect gas laws to apply, the mole fraction x_j, of any constituent gas "j" may be used to determine the partial pressure of that constituent by

$$p_j = x_j p_m \tag{6B.03.1}$$

The molal (volumetric) analysis of the mixture is one of the items of test data, and gives the mole fraction readily. With a homogeneous mixture, all constituent gases will have the same temperature as the mixture, thus providing the second of the two independent properties needed to define the gas state. With the state of each constituent thus defined, the individual property of interest may be determined, and the equivalent mixture properly calculated by the methods described below.

6B.04 With properties of the individual gases determined, the equivalent value of the property for the gas mixture may be calculated by summing the individual property value on a total basis, i.e., quantity of the gas times property value. The equations are summarized below.

Enthalpy:

$$m_m h_m = m_a h_a + m_b h_b + m_c h_c + \ldots\ldots m_j h_j \tag{6B.04.1}$$

$$n_m H_m = n_a H_a + n_b H_b + n_c H_c + \ldots\ldots n_j H_j \tag{6B.04.2}$$

$$H_m = x_a H_a + x_b H_b + x_c H_c + \ldots\ldots x_j H_j \tag{6B.04.3}$$

Entropy:

$$m_m s_m = m_a s_a + m_b s_b + m_c s_c + \ldots\ldots m_j s_j \tag{6B.04.4}$$

$$n_m S_m = n_a S_a + n_b S_b + n_c S_c + \ldots\ldots n_j S_j \tag{6B.04.5}$$

$$S_m = x_a S_a + x_b S_b + x_c S_c + \ldots\ldots x_j S_j \tag{6B.04.6}$$

Specific heats:

$$m_m c_m = m_a c_a + m_b c_b + m_c c_c + \ldots\ldots m_j c_j \tag{6B.04.7}$$

$$n_m C_m = n_a C_a + n_b C_b + n_c C_c + \ldots\ldots n_j C_j \tag{6B.04.8}$$

$$C_m = x_a C_a + x_b C_b + x_c C_c + \ldots\ldots x_j C_j \tag{6B.04.9}$$

In the preceding series of equations, (6B.04.1), (6B.04.4), and (6B.04.7), are on a mass basis; (6B.04.2), (6B.04.5), (6B.04.8) are on a mole basis; and (6B.04.3), (6B.04.6), and (6B.04.9) are on a mole fraction basis, It should be noted that the determination of the end point of the isentropic process starting at inlet conditions and ending at the discharge pressure and entropy value corresponding to inlet conditions will probably involve a trial-and-error solution.

6B.05 When perfect gas laws do not apply and adequate thermodynamic data for specific gases are not available, generalized charts of compressibility factor Z given in terms of reduced pressure and reduced temperature may be used. These are based on the law of corresponding states, which assumes that the behavior of all gases is nearly alike when their reduced pressures and reduced temperatures are alike. Sources of such charts are indicated in Appendix E.

The defining equations for the parameters involved are as follows:

$$Z = \frac{144\,pv}{RT} \qquad p_R = \frac{p}{p_{crit}} \qquad T_R = \frac{T}{T_{crit}}$$

It should be noted that the test results based on the use of these generalized charts cannot be considered to have the same reliability as those based on known thermodynamic properties.

SECTION 6
APPENDIX C: BASIC SYMBOLS AND UNITS

Symbol	Description	Units
a	Acoustic velocity	ft/sec
A	Internal cross-sectional area of tube	ft^2
c	Specific heat	Btu/lbm F
c_p	Specific heat at constant pressure	Btu/lbm F
c_v	Specific heat at constant volume	Btu/lbm F
C	Molal specific heat	Btu/lbm-mole F
CFR_B	Volume displaced per revolution of blower	ft^3/rev
f_n	Resonance frequency	cycles/sec
g	Gravitational acceleration	ft/sec^2
g_c	Dimensional constant	lbm ft/lbf sec^2
G	Specific gravity	dimensionless
h	Enthalpy	Btu/lbm
h_r	Heat transfer coefficient for gear casing	Btu/hr ft^2 F
H	Molal enthalpy	Btu/lbm-mole
H_r	Relative humidity	dimensionless
J	Mechanical equivalent of heat	ft lbf/Btu
k	Ratio of specific heats, c_p/c_v	dimensionless
L	Length	ft
m	Mass	lbm
M	Molecular weight	dimensionless
n	Molal quantity	lbm-mole
n_s	Isentropic exponent (nonideal gas)	dimensionless
N	Rotative speed	rpm
p	Pressure	psia
P	Power	hp
q	Heat loss	Btu/lbm
q_r	Heat loss from gear casing	Btu/min
Q	Capacity, volume flow rate	cfm
R	Gas constant for particular gas	ft lbf/lbm R
s	Entropy	Btu/lbm R

(Continued)

Symbol	Description	Units
S	Molal entropy	Btu/lbm-mole R
S_r	Heat transfer surface of gear box	ft^2
S_c	Slip at specified conditions	cfm
S_t	Slip at 1 psi differential	cfm
t	Temperature	F (Fahreinheit)
T	Absolute temperature	R (Rankine)
u	Velocity	ft/sec
v	Specific volume	ft^3/lbm
V	Volume of receiver	ft^3
w	Mass flow rate	lbm/min
W	Work	Btu/lbm
x	Mole fraction	dimensionless
Y	Elevation	ft
Z	Compressibility factor	dimensionless
Δp_c	Specified differential pressure	psi
Δp_t	Test differential (approx 1 psi)	psi
η	Efficiency	dimensionless
ρ	Density	lbm/ft^3
τ	Torque	lbf ft

Subscripts	
a	Ambient (gas component in 6B.03 and 6B.04)
b	Gas component
adj	Adjusted
avg	Average during test
c	Specified conditions (gas component in 6B.03 and 6B.04)
crit	Critical
dg	Dry gas
f	Condensate
g	Saturation (condensing vapor)
isen	Isentropic
j	Gas component
m	Measured or metered (mixture in 6B.03 and 6B.04)

(Continued)

Subscripts	
R	Reduced
r	Relates to heat transfer for exposed surface
sh	Shaft
st	Static
o	Stagnation Condition
1	Inlet
2	Discharge

Superscripts	
(′)	At entropy corresponding to inlet conditions (as h_2')

SECTION 6
APPENDIX D: REPORT OF TEST

6D.01 The report shall be a document prepared in suitable form to formally present clearly and concisely the data observed and computed. The report shall present sufficient information to demonstrate that all objectives of the tests have been obtained. The form of the report shall follow the general outline given in the following paragraphs.

6D.02 The title page shall present the following information:

(*a*) Report number (optional)
(*b*) Date(s) of test
(*c*) Title of the test
(*d*) Location of test
(*e*) Owner (purchaser)
(*f*) Manufacturer's name, engine designation and unit identification
(*g*) Test conducted by
(*h*) Report prepared by
(*i*) Report approved by
(*j*) Date of report

6D.03 The table of contents shall list the major subdivisions of the report.

6D.04 The summary shall present briefly the object, results and conclusions of the test.

6D.05 The detailed report shall include the following:

(*a*) Object of tests, guarantees, and stipulated agreements

(*b*) Description of installation

(1) Kind of gas handled by compressor: air, natural gas, coke oven gas, etc.

(2) Class of service, such as air for pneumatic tools, paint spray and drying, or gas pipe line booster, or vacuum exhauster for drying service, etc.

(3) Arrangement of discharge lines, aftercoolers, receivers, and important valves

(4) Arrangement of intake duct or pipe:

- For air compressors, with closed intake, give dimensions and sketch showing layout of duct; describe filter, if used
- For air boosters, give dimensions and layout of pipe
- If other compressors are served by the same intake system, give their respective speeds, displacement, and kind of drivers

(5) Type of drivers, electric motor, steam engine, steam turbine, or internal-combustion engine

(6) Method of drive, direct-connected by couplings, through reduction gears, belts, chains, or cylinders of engine in tandem with compressor cylinders, etc.

Description of Compressor

(1) Type of compressor, number of stages, number of cylinders, arrangement of cylinders

(2) Rated displacement

(3) Type of valves

(4) Method of regulation or volume control

(5) Type of intercoolers, air or water cooled, kind of cooling surface, fin tube or smooth tube, brass or iron, air or water through the tube, number of air passes, number of water passes, arranged countercurrent, or parallel

(6) Arrangement of cooling water circuits

(7) Area water-cooled surfaces, intercoolers, aftercoolers (list separately for each stage)

(8) Cylinder dimension (list separately for each stage):

- Diameter of cylinder
- Stroke of piston
- Diameter of piston and tail rods
- Clearance volume – head end
- Clearance volume – crank end
- Clearance volume – average

(9) Cylinder ratio (based on piston displacement) first stage to second, second stage to third, etc.

Description of Drivers

(1) Manufacturer
(2) Serial number and nameplate identification
(3) Power rating – full load
(4) Speed rating
(5) Type of dimensions, etc.

For Electric Motors:

(1) Ac or dc, shunt, compound, or squirrel cage, wound rotor, synchronous, etc.

(2) Current conditions – volts, full-load amperes, number of phases, frequency, and power factor

(3) For synchronous motors, give type and rating of exciter

(4) Arrangement and type of bearing, lubrication, method of cooling, etc.

For Steam Engines:

(1) Type – simple, multiple expansion, etc.
(2) Number and arrangement of cylinders
(3) Type of valves and governor
(4) Condensing or noncondensing, or back pressure
(5) Describe condensing equipment and other auxiliaries
(6) Dimensions – bore and stroke of each cylinder – diameter of piston and tail rods

(7) Clearance volume – head end and crank end each cylinder – average clearance each cylinder

For Steam Turbines:

(1) Impulse, or reaction, condensing or back pressure, extraction or mixed pressure, etc.

(2) Number of stages, and kind of flow, single or double

(3) Auxiliaries, type of governor, overspeed safety device, lubrication system, etc.

For Internal-Combustion Engines:

(1) Two- or four-stroke cycle, single or double acting, vertical or horizontal, cylinders in line or "V" arrangement, etc.

(2) Number of cylinders

(3) Diameter and stroke

(4) Kind and specification of fuel

(5) Method of introducing fuel, carburetor, air injection, solid injection

(6) Description of auxiliaries, type of fuel pump or carburetor, type of ignition system, scavenging compressor, lubrication system, cooling system, type of governor, and load control

For Gas Turbines:

(1) Type – Open, semiclosed, or closed cycle, number of shafts, regenerative or nonregenerative, etc.

(2) Kind and specification of fuel

(3) Description of auxiliaries

Specified Operating Conditions and Guarantees

(1) Average barometric pressure

(2) Altitude above sea level

(3) Inlet pressure:

- For compressors and boosters
- For vacuum pumps and exhausters

(4) Discharge pressure

(5) Speed

(6) Capacity referred to inlet temperature and pressure

(7) Cooling water temperature – degrees below temperature of inlet air or gas

(8) Flow of cooling water to intercoolers

(9) Flow of cooling water to jackets

(10) Absolute humidity of air or gas at inlet conditions in pounds of moisture per 1000 ft^3

(11) Power

For Electric Motor Drives:

(1) Input to motor

(2) Motor efficiency

(3) Transmission loss (belts, chain, or gears) per cent of motor output

(4) Net input to compressor

(5) For synchronous motors (power factor)

(6) Economy (bhp/1000 ft^3)

For Steam Engine Drives:

(1) Pressure at throttle

(2) Temperature at throttle

(3) Exhaust pressure

(4) Steam consumption

(5) Indicated power of steam cylinder

(6) Indicated power of air cylinder

(7) Mechanical efficiency

(8) Economy (lb steam/1000 ft^3)

For Steam Turbine Drives:

(1) Pressure at throttle

(2) Temperature at throttle

(3) Exhaust pressure

(4) Total steam consumption

(5) Turbine output

(6) Gear Loss

(7) Economy (lb steam/1000 ft^3)

For Internal-Combustion Engines:

(1) Kind and grade of fuel

(2) Cooling water

(3) Fuel rate

(4) Engine output

(5) Transmission losses (if gears or belts are used)

(6) Input to compressor

(7) Economy (lb fuel/1000 ft^3)

For Gas Turbines:

(1) Kind and grade of fuel

(2) Fuel rate

(3) Turbine inlet pressure

(4) Exhaust pressure

(5) Air extraction rate

(6) Power output

(7) Transmission losses

(8) Input to compressor

(9) Economy (lb fuel/1000 ft^3)

Mean Observations and Calculated Results, Test Conditions of Operation

Compressor Data:

(1) Duration of test run

(2) Barometric pressure corrected to 32 F

(3) Inlet pressure:

- For compressors without inlet pipe (equal to barometric pressure)
- For compressors with inlet pipe – by indicator
- For boosters (1) by gage
  (2) by indicator

- For vacuum pumps (1) vacuum
  (2) absolute pressure

(4) Amplitude of pressure waves in inlet pipe at cylinder

(5) Discharge pressure:
- By gage
- Indicator
- Absolute discharge pressure

(6) Amplitude of pressure waves in discharge pipe

(7) Intercooler pressure
- First intercooler
- Second intercooler

(8) Mean diameter of primary element and pipe size

(9) Pressure:
- For discharge nozzle differential pressure
- For suction nozzle
- Absolute nozzle differential pressure

(10) Nozzle temperature – upstream side

(11) Nozzle temperature – absolute

(12) Psychrometric readings:
- Wet bulb
- Dry bulb

(13) Relative humidity at inlet of compressor

(14) Absolute humidity at inlet

(15) Inlet air temperatures first stage

(16) Inlet temperatures second stage

(17) Room temperature

(18) Temperature at discharge of:
- First stage cylinder
- Second stage cylinder

(19) Cooling water temperatures – jackets:
- Inlet to jackets
- Discharge first stage jackets
- Discharge second stage jackets

(20) Cooling water temperatures – intercoolers:
- Inlet (each intercooler)
- Outlet (each intercooler)

(21) Flow of cooling water:
- Intercoolers
- Jackets (if separate circuit)

(22) Condensed moisture removed from coolers

(23) Elapsed time between initial and final blowdown on condensate

(24) Compressor speed by revolution counter

(25) Flow through nozzle

(26) Capacity at inlet temperature and pressure

(27) Correction to capacity for moisture condensed and removed

(28) Net capacity at test conditions

(29) Piston displacement:
- First stage
- Second stage (etc.)

(30) Cylinder ratios – first/second stage, second/third stage, etc.

(31) Volumetric efficiency (first stage cylinder)

(32) Indicated power of cylinders:
- First stage – head end
- First stage – crank end
- Second stage – head end
- Second stage – crank end
- Total

For Induction Motor Drives:

(1) Voltmeter readings (corrected) Phase 1..., Phase 2..., Phase 3....

(2) Ammeter readings (corrected) Phase 1..., Phase 2..., Phase 3....

(3) Wattmeter readings, W_1 Phase 1-2...... W_2 Phase 2-3

(4) Algebraic sum of wattmeter readings, $W_1 + W_2$

(5) Transformer ratios:
- Current (corrected for phase angle and ratio)
- Potential (corrected for ratio)

(6) Average phase voltage

(7) Average phase current

(8) Power input to stator

(9) Power factor

(10) Motor efficiency by input-output test

(11) Motor output

Additional for Synchronous Motor Drives:

(12) Excitation:
- Volts dc at slip rings
- Current dc
- Power

(13) Power factor (leading or lagging)

(14) Motor losses:
- Open circuit core loss
- Short circuit core loss
- Copper loss I^2R
- Friction windage and load loss
- Exciter and rheostat loss (if driven by main motor)

(15) Total motor input (add excitation if exciter is separately driven)

(16) Motor losses – total – including excitation and if exciter is driven by main motor, include also losses of exciter and field rheostat

(17) Net motor output

For DC Motor Drives:

(1) Volts at motor terminals (corrected for instrument error)

(2) Armature current – ammeter reading

(3) Shuntfield ammeter reading

(4) Multiplier for shunt ratio (a) Armature.....

(b) Field.....

(5) Power input to motor

(6) Motor efficiency:
- By loss method
- By input-output test

(7) Motor output

Additional if Motor Is Belted to Compressor:

(1) Motor speed by revolution counter

(2) Belt slip

Additional if Motor Is Geared to Compressor:

(1) Gear ratio

(2) Gear oil pressure

(3) Gear oil temperature

(4) Gear loss (in per cent of motor output power)

For Steam Engine Drives:

(1) Steam pressure at throttle:
- By gage
- By indicator
- Absolute pressure

(2) Steam pressure at exhaust:
- Vacuum (condensing)
- Absolute pressure (condensing)
- By gage (for back pressure engines)
- By indicator (for back pressure engines)
- Absolute exhaust pressure
- Calorimeter data

(3) Steam temperatures, at throttle

(4) Steam temperature at exhaust

(5) Steam pressure at jackets:
- First stage
- Second stage

(6) Steam temperature at jackets:
- First stage
- Second stage

(7) Steam temperature at reheater

(8) Steam pressure at reheater

(9) Receiver pressure

(10) Steam flow from engine, condensed and weighed

(11) Condenser leakage

(12) Condensate from jackets and reheater

(13) Net steam charged to engine – test conditions

(14) Total indicated horsepower

For Steam Turbine Drives:

(1) Steam pressure at throttle

(2) Steam temperature at throttle

(3) Exhaust pressure:
- For condensing turbine, vacuum gage corrected to 32 F
- Corresponding absolute pressure
- For back pressure turbines, by gage
- Corresponding absolute pressure

(4) Exhaust temperature

(5) Steam flow as condensed and weighed

(6) Condenser leakage

(7) Seal or gland loss

(8) Net steam consumption – test conditions

(9) Gear ratio

(10) Lubricating oil temperature

(11) Gear loss

For Internal Combustion Engines:

(1) Exhaust pressure

(2) Jacket water temperatures:
- Inlet
- Outlet

(3) Lubricating oil temperatures:
- Outlet from cooler
- Inlet to cooler

(4) Exhaust gas temperature

(5) Temperature of main air supply

(6) Temperature of air-fuel mixture at intake port (carburetor engine)

(7) Engine speed – average by revolution counter

(8) Gear ratio (if used)

(9) Speed variation (at full load by tachometer)

(10) Kind and specification of fuel used

(11) Heat value of fuel

(12) Fuel consumed

(13) Heat supplied

(14) Fuel consumed

(15) Fuel rate

(16) Heat rate

For Gas Turbines:

(1) Exhaust pressure

(2) Lubricating oil temperature
- Outlet from cooler
- Inlet to cooler

(3) Exhaust temperature

(4) Turbine speed

(5) Gear ratio

(6) Speed variation

(7) Kind and specification of fuel used

(8) Heat value of fuel

(9) Fuel consumed

(10) Heat supplied

(11) Fuel rate

(12) Heat rate

Corrections for Deviations from Specified Conditions

(1) Capacity, correction for speed (Par. 5.13)

(2) Power, correction for
- Inlet conditions (Par. 5.14)
- Inlet conditions

- Varied interstage conditions (Par. 5.14)
- Varied interstage conditions
- Total power corrections

(3) Steam rate, correction, steam engines, and steam turbines, for

- Throttle pressure
- Moisture or superheat
- Back pressure or vacuum
- Load

(4) Fuel rate, internal-combustion engine, correction for variation of compressor load

Calculated Results Corrected to Specified Conditions of Operations

For Motor Driven Compressors:

(1) Power input to driver
(2) Power input to compressor
(3) Capacity
(4) Mechanical efficiency
(5) Volumetric efficiency
(6) Compression efficiency
(7) Power economy

For Steam Engine or Steam Turbine Driven Compressors:

(1) Steam rates
(2) Capacity
(3) Power economy

For Internal Combustion Engine Driven Compressors:

(1) Fuel rate
(2) Capacity
(3) Power economy

For Gas Turbine:

(1) Fuel rate
(2) Capacity
(3) Power economy

(*c*) Tabular and graphical presentation of the test results

(*d*) Discussion of the test, its results and conclusions

6D.06 Appendixes and illustrations to clarify description of the equipment and methods and circumstances of the test, description of the methods of the calibration of instruments, outlines of details of calculations, descriptions and statements as to special testing apparatus, results of preliminary inspections and trials, and any supporting information required to make the report a complete self-contained document of the entire undertaking.

SECTION 6
APPENDIX E: REFERENCES

6E.01 References for Gas Properties and Test Methods. When calculating compressor performance under the provisions of this Code it will be necessary to use the properties of gases based on experimental work or resort to calculated property values. The values of the properties and the calculation methods should be those reported and published by reliable sources. *This Code does not pretent to authorize any of these references as being completely accurate.* They are given as a convenience and are, in general, thought to be acceptable. The latest edition of the listed reference is to be considered as more acceptable than an earlier edition even though the new edition may appear after publication of this Code.

6E.02 Where two or more references, concerning a stated gas, give property values or calculation methods that are not in agreement, the parties to the test shall agree in writing, prior to the test, which reference shall be used as the source of property data for the test. (Difference in value and methods may occur due to experimental error, differences in experimental technique or to differing mathematical techniques and assumptions.)

6E.03 Also included in the list of references are several documents pertaining to test correlation methods, test procedures and compressor performance. These will give a general insight into methods for handling the various problems connected with the testing of modern compressors and are included for their relation to these problems rather than for any detailed instructions. This Code does not authorize any of these references as part of the Code procedures or imply that they agree with Code methods.

6E.04 Thermodynamics of Gases

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6E.04.3 "Theory of Real Gases," J. E. Mayer, *Encyclopedia of Physics,* Vol. XII, ed. by S. Flugge, Springer-Verlag, Berlin, 1958.

6E.04.4 *Molecular Theory of Gases and Liquids,* J. O. Hirschfelder, C. F. Curtiss and R. R. Baird, John Wiley & Sons, New York, 1954.

6E.04.5 *Thermodynamics and Physics of Matter,* J. O. Hirschfelder, C. F. Curtiss, R. R. Baird and E. L. Spotz, Vol. I of *High Speed Aerodynamics and Jet Propulsion,* ed. by F. D. Rossini, Princeton University Press, 1955.

6E.04.6 "A Generalized Equation of State for Both Gases and Liquids (for digital computer)," J. O. Hirschfelder, J. J. Buehler, H. A. McGee and J. R. Sutton, University of Wisconsin Naval Research Laboratory Technical Report WIS-OOR-15, October 5, 1956.

6E.04.7 "Principles of the Kinetic Theory of Gases," Harold Grad, *Encyclopedia of Physics,* Vol. XII, ed. by S. Flugge, Springer-Verlag, Berlin, 1958.

6E.04.8 "Generalized Thermodynamic Properties of Pure Fluids," A. L. Lydersen, R. A. Greenkorn and O. A. Hougen, College of Engineering, University of Wisconsin Engineering Experiment Station Report 4, October, 1955.

6E.04.9 "Generalized Compressibility Charts for Industrial Problems," E. F. Obert and R. L. Young, Technical Institute, Northwestern University Research Report M101, 1951.

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PTC Committee No. 9 on Displacement Compressors, Vacuum Pumps and Blowers has revised the last equation in Par 5.08 located on page 26 of the Test Code. The corrected equation appears as follows:

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