8

General Reference Data

What is the capacity of an air compressor? How is the type and service limitation of a compressor defined? Will a drill bit purchased in New York fit a pneumatic tool purchased in San Francisco? Are compressors available to operate at pressures and capacities suitable for rock drills purchased from various sources of supply? If you order two compressors from two builders, will they work together in unison as a single source of air power? The answer is the adoption of *standards* – common language for performance and acceptability. These standards do not result in identical compressors, tools, or fittings, but they do make it possible for buyer and seller to be on a common ground of understanding. They are the "know-how" factors that reduce economic complexity to simple formula – solution to a recurring difficulty.

The Compressed Air and Gas Institute is heavily involved in the development of standards related to compressed air systems. The institute works with organizations, such as ASME, ANSI, ASTM, ISO, PNEUROP, and others providing input and expertise to establish equitable standards for the manufacturers and uses of pneumatic equipment.

Standardization on a voluntary basis, such as is advocated by the Compressed Air and Gas Institute, is more than an engineering function. Through a well-rounded program of sound standards, economy of operation and increased production inevitably result. By diminishing inventory and investment, by speeding up maintenance and shipments, by cutting down accidents, standards increase output, decrease cost, and are a benefit to buyer and seller alike.

Standard definitions, nomenclature, and terminology, which constitute a large portion of this section, represent either scientific fact or common usage. The pur-

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pose in presenting them is to make available a common language for the description of compressed air machinery and tools and to permit an accurate definition of performance. The acceptance of such definitions, nomenclature, and terminology by the builder and user of compressed air machinery and tools will avoid confusion, eliminate argument, and prevent misunderstanding, all in the public interest.

The Compressed Air and Gas Institute does not recommend standardization as to general design, appearance, performance, or overall interchangeability. Where interchangeability of parts used in connection with such machinery is desirable and necessary, they have for the most part followed the published recommendations of the ASA. For example, screw threads, pipe threads, companion flanges, pneumatic and rock-drill tool shanks are completely interchangeable for apparatus built by any manufacturer. Some of these standards are included in this section. These latter refer particularly to compressors and compressor practice.

A clear understanding of any subject depends primarily upon complete agreement on the definitions or all the important terms and values. This is true particularly in any consideration of such things as performance guarantees, test methods and procedure, and related subjects. The Compressed Air and Gas Institute has therefore adopted as standard the following definitions. They are in all cases rational, and do not violate scientific fact in any respect. In those cases where it has been necessary to choose between two or more possible definitions, each of which is valid, that definition which corresponds more nearly to established practice and usage has been adopted.

DEFINITIONS

See glossary.

Symbols

bhp	brake horsepower
c_p	specific heat at constant pressure
c_v	specific heat at constant volume
C	a constant; coefficient of discharge
D	deviation of horsepower from the isentropic value for a real gas
f	ratio of supercompressibility factors = Z_2/Z_1
ghp	gas horsepower
H	head of fluid in ft.-lb/lb
H_{ad}	adiabatic head for ideal gas, ft.-lb/lb
H_{ac}	adiabatic head for real gas, ft.-lb/lb
J	Joule's constant, the mechanical equivalent of heat. J = 778 ft.-lb/Btu
k	adiabatic exponent = ratio of specific heat at constant pressure to specific heat at constant volume = c_p/c_v
K	radius of gyration of a rotor

M	number of mols of a substance, that is, weight in pounds divided by the molecular weight, also in pounds
MCp	molal specific heat at constant pressure
MW	molecular weight
N	rotative speed, rpm; number of stages
n	polytropic exponent, in equation $Pt'' = C$ and related equations
p	pressure, lb/in.^{2} absolute (psia)
P	pressure, lb/ft^2
P_a	adiabatic horsepower for perfect gas
P_{ae}	adiabatic horsepower for real gas
P_c	critical pressure; the pressure required to liquefy a gas at its critical temperature
P_R	reduced pressure = p/p_c
d_v	partial pressure of any component (usually of the vapor component) in a gas mixture, psia
q	capacity, or flow; cubic feet per second, cfs
Q	capacity, or flow; cubic feet per minute, cfm
r	ratio of pressure = p_2/p_1; usually discharge pressure over intake pressure
R	gas constant = 1544/MW for perfect gases
RH	relative humidity

Subscripts 0, 1, 2, c refer to conditions of a gas in states 0, 1, 2, c

t	temperature in degrees Fahrenheit, °F, or Centigrade, °C
T	absolute temperature, equal to °F + 459.6, or °C + 273
T_c	critical temperature
T_R	reduced temperature, equal to T/T_c
v	specific volume, ft^3/lb
V	volume, ft^3, or m^3
w	weight flow, lb/minute
W	weight
x	mol fraction of a constituent of a mixture
X	adiabatic factor, equal to $\left(\frac{p_1}{p_2}\right)^{(k-1)/k} - 1$
y	weight fraction of a con- stituent in a mixture
z	supercompressibility factor; $pv = ZRT$
η	efficiency (identified with proper subscript)

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Subscripts

ad	adiabatic process
av	average
c	real gas
i	isothermal process
m	mixture
p	polytropic process
sv	saturated vapor
v	vapor
1	intake conditions
2	discharge conditions

SUPERCOMPRESSIBILITY

While a great many gases are well represented by perfect gas laws over fairly wide ranges of pressure and temperature, there are many others which show considerable variation from those laws. Even gases which are ordinarily treated as perfect gases require special representation in the neighborhood of the critical point.

Departure from perfect gas laws is referred to as supercompressibility, and is accounted for by means of a factor, Z, called the supercompressibility factor, introduced into the gas equations. Expressed mathematically,

$$Pv = ZRT \tag{8.1}$$

$$\frac{P_1 v_1}{T_1 Z_1} = \frac{P_2 v_2}{T_2 Z_2} \tag{8.2}$$

Stated simply, Z is a correcting factor which permits the application of ideal gas laws, with accuracy, to any known gas or gas mixture.

The numerical value of Z in Eqs. (8.1) and (8.2) is, of course, 1.0 for an ideal gas. In the case of actual gases, this value may be less than, equal to, or greater than 1.0, depending on the gas involved and the pressure and temperature conditions being considered.

In the measurement of air or gas passing through a nozzle it is often convenient to use equations of hydraulics, and to allow for the compressibility of ideal gases by a factor called *compressibility*. For real gases near the critical pressure, a further correction, called *supercompressibility*, allows for departure of real gases from ideal gases. Thus compressibility relates ideal gases to liquids, while supercompressibility relates real gases to ideal gases. The distinction is important, since compressor engineers may encounter both in the same problem.

The reader interested in pursuing this topic further is referred to the text *Thermodynamics* by Joseph Keenan for a discussion of supercompressibility. Compressibility factors, and a derivation of the compressibility equation, are given in the ASME *Fluid Meters Handbook.*

Determination of Supercompressiblilty Factor

The magnitude of Z depends upon the particular pressure and temperature at which the gas is being considered. Furthermore, the magnitude of Z at any stated pressure and temperature combination is different for practically every known gas; thus exact values for a given gas or gas mixture can be determined only by extensive laboratory tests of each gas or mixture. How can the compressor engineer determine the value of Z?

The most accurate source of graphical information is an authentic Mollier diagram applying to the particular gas or gas mixture being compressed. Unfortunately, this data is available for only a few of the many gases and mixtures which are commonly compressed.

An alternative approach is to approximate quite closely the value of Z for any gas or gas mixture by utilizing the law of corresponding states. This law or principle states that the magnitude of Z for a given gas at a specified pressure and temperature is definitely related to the critical pressure and temperature of that gas. The reduced pressure and reduced temperature of the gas may be stated mathematically:

$$P_R = \frac{p}{p_c} \qquad (8.3)$$

$$T_R = \frac{T}{T_c} \qquad (8.4)$$

The subscript R denotes the reduced function and p and T indicate the actual pressure (psia) and temperature (Rankine or Kelvin) of the gas for which the compressibility must be found.

T_c = Critical temperature, which is the maximum temperature (Rankine or Kelvin) at which a gas can be liquefied.

P_c = The critical pressure of the gas; that is, the absolute pressure, psia, required to liquefy a gas at its critical temperature.

Further, the law of corresponding states tells us that if various gases have their reduced pressures and reduced temperatures equal, then the supercompressibility factors of these gases are of about the same magnitude.

Consider three different gases, w, x and y, all existing at different pressures and temperatures. If this condition is true,

$$P_R = \frac{p_w}{p_{cw}} = \frac{p_x}{p_{cx}} = \frac{p_y}{p_{cy}}$$

and

$$T_R = \frac{T_w}{T_{cw}} = \frac{T_x}{T_{cx}} = \frac{T_y}{T_{cy}}$$

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then the magnitude of the Z factor of each of the three gases is practically the same, even though the gases exist at widely different pressures and temperatures.

This fact permits the development and use of generalized supercompressibility charts using P_R and T_R as coordinates and such charts may be applied to any known gas or gas mixture with a high degree of accuracy to determine the compressibility factor of the gas or mixture at any condition.

The general practice of the compressor industry is to use one of the established equations of state for the computation of Z internally in computer programs. One such equation is the Redlic-Kwong equation, which can be used for pure gases and mixtures, except near the critical point. Figures 8.2 to 8.6 were derived from the Redlic-Kwong equation. Figure 8.1 is a sketch showing how these charts overlap and is included only to make the later, accurate charts more usable.

The value of Z at the intake and discharge condition of each stage of compression is readily found from these charts since the intake pressure and temperature and the discharge pressure are known and the discharge temperature is easily calculated from the isentropic formula previously given.

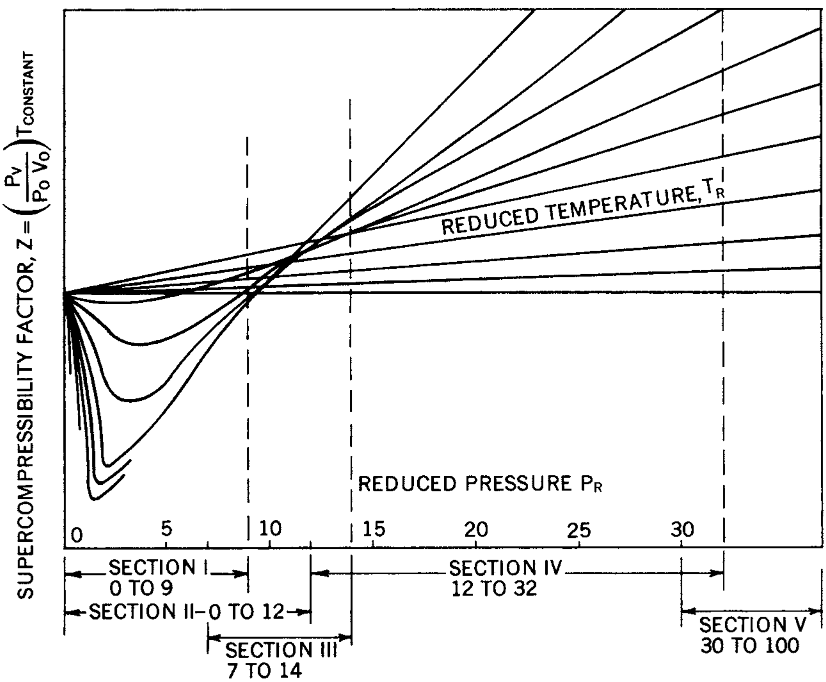

Figure 8.1 Supercompressibility factor vs. reduced pressure at varying reduced temperatures. The above curve is a composite sketch of all five sections of the compressibility plot. As shown above they overlap one another. The scales used on individual sections are arranged so as to maintain a consistent accuracy in reading.

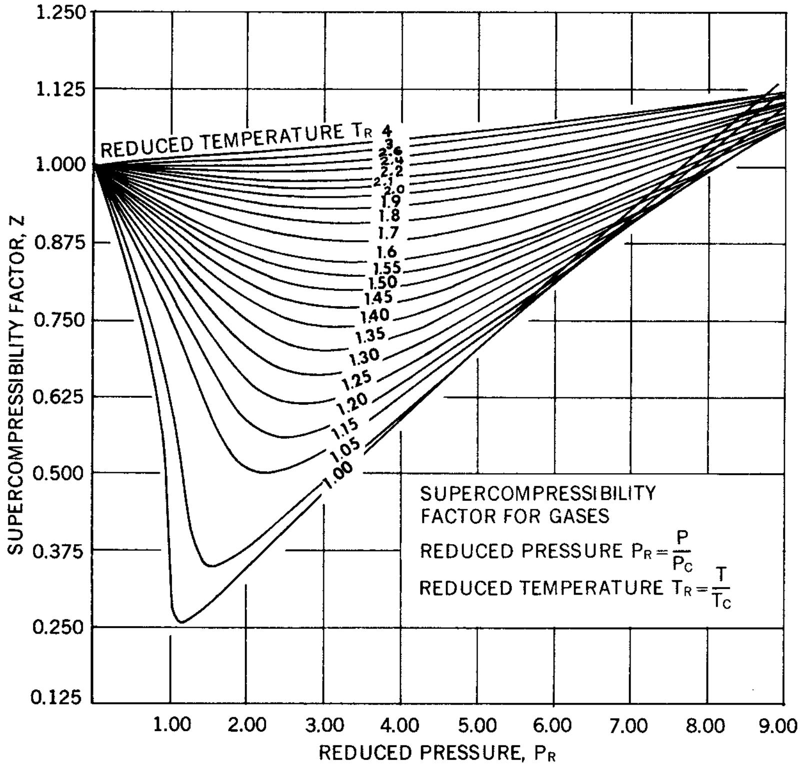

Figure 8.2 Supercompressibility factor for reduced pressure range, 0 to 9, inclusive.

Example 8.1

Find the volume of 1 lb of chlorine at 458 psig and 516~ Barometric pressure = 14.0 psia. From Table 8.39, p_c = 118 psia, T_c = 291°F, MW = 70.914.

$$T_R = \frac{516 + 459.6}{291 + 459.6} = 1.30, \quad P_R = \frac{458 + 14.0}{118} = 4.00, \qquad \text{See Figure 8.3}$$

$$v = \frac{ZRT}{P} = \frac{(0.67)(1544)(516 + 459.6)}{(144)(70.914)(458 + 14.0)} = 0.21 \text{ ft}^3$$

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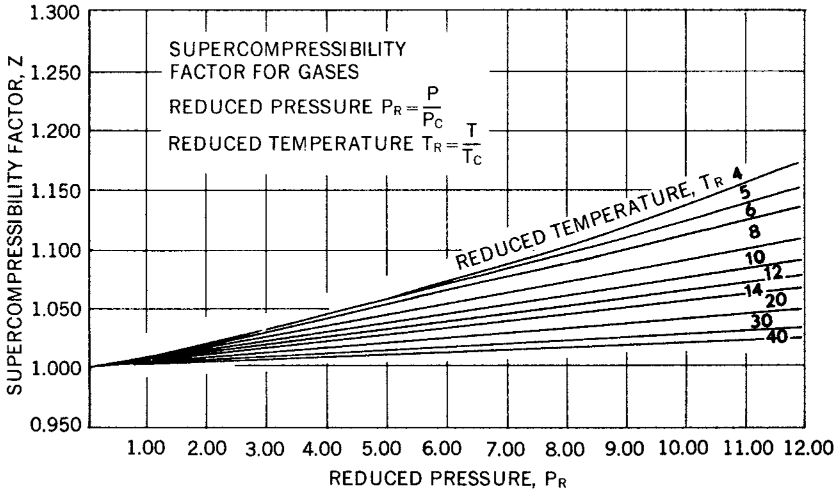

Figure 8.3 Supercompressibility factor for reduced pressure range, 0 to 12, inclusive. Note: In this range of reduced pressure, overlapping that of Figure. 8.2, the supercompressibility factor reaches a maximum at reduced temperature approximately equal 4. It then decreases with increasing values of reduced temperature. To avoid confusion in reading, which would result from the overlapping curves, values of Z for T_5 greater than 4 are shown in this separate graph.

PHYSICAL PROPERTIES OF GAS MIXTURES

If the chemical composition of a gas mixture is known, it becomes possible to determine the gas characteristics necessary to make compressor calculations through the application of the following relations:

$$W_m = W_1 + W_2 + W_3 + \cdots$$
$$M_m = M_1 + M_2 + M_3 + \cdots$$
$$x_1 = \frac{M_1}{M_m} \qquad x_2 = \frac{M_2}{M_m} \qquad x_3 = \frac{M_3}{M_m}$$
$$y_1 = \frac{W_1}{W_m} \qquad y_2 = \frac{W_2}{W_m} \qquad y_3 = \frac{W_3}{W_m}$$

And therefore,

$$x_1 + x_2 + x_3 + \cdots = 1.0 \text{ and } y_1 + y_2 + y_3 + \cdots = 1.0$$
$$MW_m = x_1MW_1 + x_2MW_2 + x_3MW_3 + \cdot$$
$$C_m = y_1c_1 + y_2c_2 + y_3c_3 + \cdots$$

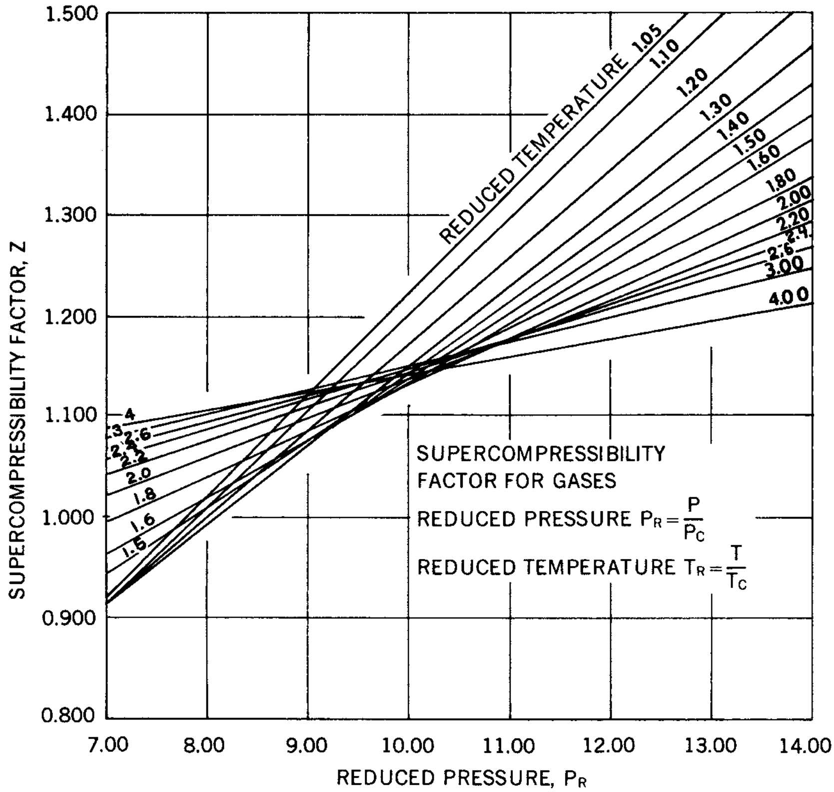

Figure 8.4 Supercompressibility factor for reduced pressure range, 7 to 14, inclusive.

where:

W_m, W_1, W_2, etc.	=	Weight of mixture and of constituents, respectively.
M_m, M_1, M_2, etc.	=	Number of mols of mixture and of constituents, respectively.
MW_m, MW_1, MW_2, etc	=	Molecular weight of mixture and of constituents, respectively.
C_m, C_1, C_2, etc.	=	Specific heats of mixture and of constituents, respectively.
x_1, x_2, x_3, etc.	=	Mol fraction of constituents in mixture.
y_1, y_2, y_3, etc.	=	Weight fraction of constituents in mixture.

Molal properties, such as molal specific heat, MC_m, are calculated on a molal basis.

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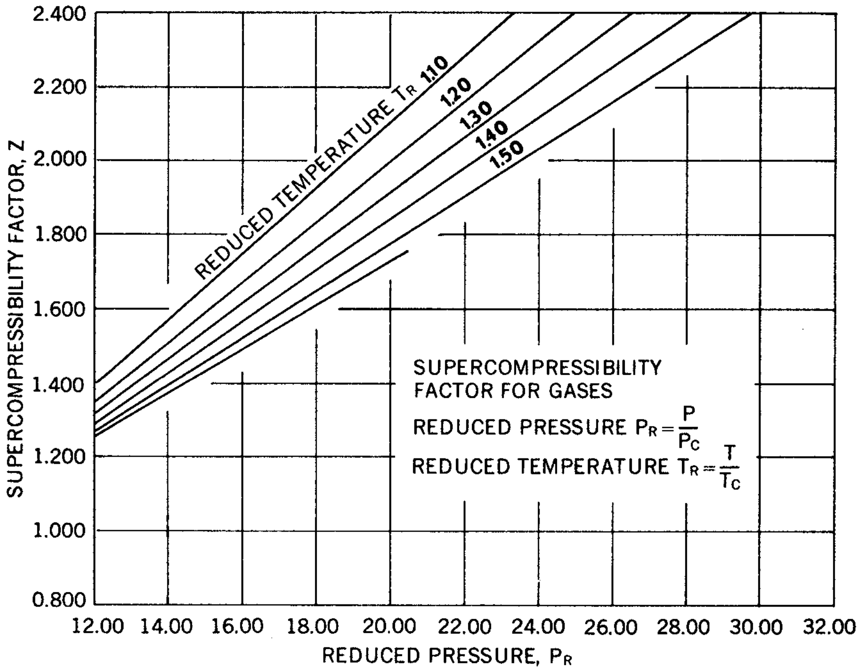

Figure 8.5 Supercompressibility factor for reduced pressure range, 12 to 32, inclusive.

An application of these relations is illustrated in the example, Table 8.1, which presents the computation of the physical characteristics of a typical natural gas, the composition of which is known on the volumetric basis. The molecular weight (MW_m) of this gas is found to be 19.75 and its specific gravity relative to air is

$$\text{sp.gr.} = \frac{19.75}{28.96} = 0.682$$

Example 8.2

Its other characteristics can be determined as follows:

$$R = \frac{1544}{MW} = \frac{1544}{19.75} = 78.18\text{, approximately}$$

$$c_v = c_p - \frac{R}{778} = 0.482 - \frac{78.78}{778} = 0.377$$

$$k = \frac{c_p}{c_v} = \frac{0.482}{0.377} = 1.279$$

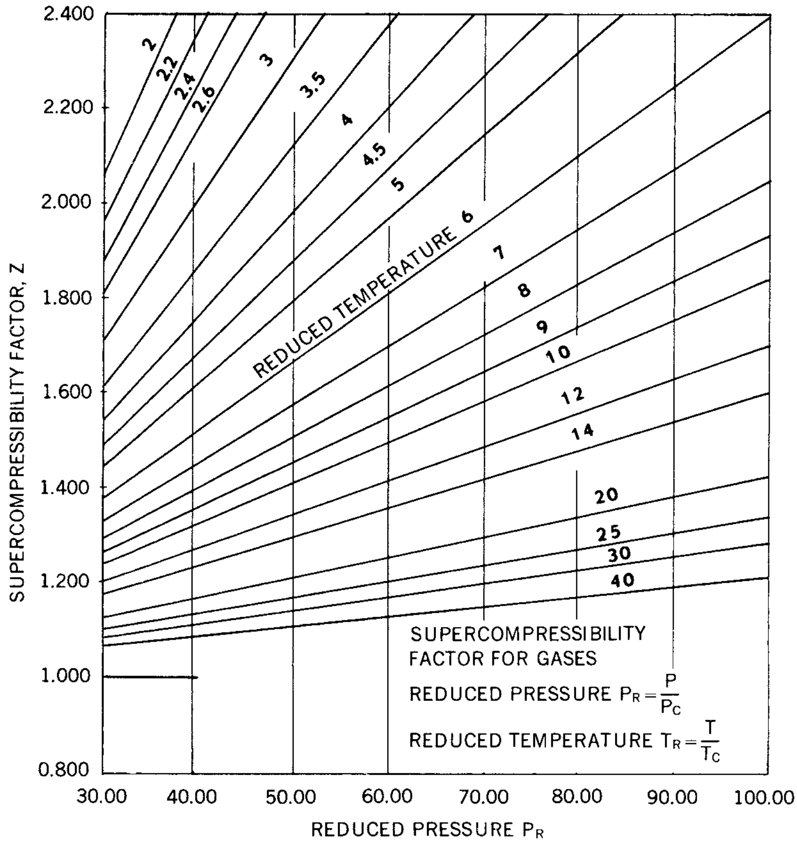

Figure 8.6 Compressibility factor for reduced pressure range, 30 to 100, inclusive.

Pseudo-critical Constants for Gas Mixtures

For a physical mixture of gases, not combined chemically, the usual method of determining a pseudo-critical constant presupposes a hypothetical gas having a critical constant equal to the sum of the products of the critical constants of the individual gases in the mixture times their respective mole fraction in the total mixture, i.e.,

$$p_c = p_{c1}x_1 + p_{c2}x_2 + p_{c3}x_3 \cdots + p_{cn}x_n$$

$$T_c + T_{c1}x_1 + T_{c2}x_2 + T_{c3}x_3 \cdots + T_{cn}x_n$$

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where p_c, T_c are critical constants of the mixture, p_{c1}, p_{c2} and so on, and T_{c1}, T_{c2}, and so on, are critical constants of the constituents, and x_1, x_2, and so on, are mol fractions of each constituent in the mixture. This calculation is illustrated in Table 8.2.

Table 8.1 Computation of Characteristics of a Typical Natural Gas

Gas Component	Chemical Formula	Fraction by Volume	Molecular Weight	Vol. Fraction × Mol. Wt.	Fraction by Weight	c_p	Wt. Fraction × c_p	MC_p at 150ºF	Vol. Fraction × MC_p
Methane	CH_4	0.832	16.04	13.35	0.675	0.526	0.355	8.97	7.46
Ethane	C_2H_6	0.085	30.07	2.56	0.130	0.409	0.053	13.78	1.17
Propane	C_3H_8	0.044	44.09	1.94	0.098	0.386	0.038	19.58	0.86
Butane	C_4H_{10}	0.027	58.12	1.57	0.080	0.397	0.032	26.16	0.71
Nitrogen	N_2	0.012	28.02	0.33	0.017	0.248	0.004	6.96	0.08
		1.000		M = 19.75	1.000		c_p = 0.482		MC_p = 10.28

Table 8.2 Computation of Pseudo-critical Temperature and Pressure of a Typical Natural Gas

Gas Component	Chemical Formula	Fraction by Volume	Critical Pressure	Mol Fraction × Crit Press.	Critical Temperature	Mol Fraction × Crit Temp.
Methane	CH_4	0.832	673.1	560	343.5	276
Ethane	C_2H_6	0.085	708.3	60	550.9	47
Propane	C_3H_8	0.044	617.4	27	666.26	29
Butane	C_4H_{10}	0.027	550.7	15	765.62	21
Nitrogen	N_2	0.012	492	6	227.2	3
		1.000		668 p_c of mixture		376 T_c of mixture

TEST PROCEDURE

Tests to determine the performance of air compressors and blowers or to establish compliance with performance guarantees can be of value only if they are conducted carefully and in strict conformity with accepted methods and standards. The Compressed Air and Gas Institute, therefore, endorses the ASME Test Code on Compressors and Exhausters (PTC 10) and for Displacement Compressors, Vacuum Pumps and Blowers (PTC 9), and recommends that all tests to establish performance he made according to the rules specified in these codes or according to the Institute's interpretation of these codes in this section. The Institute's endorsement of these codes includes acceptance of the ASME Code on General Instructions (PTC 1).

Copies of the ASME Test Codes may he purchased at a small cost from the American Society of Mechanical Engineers, New York.

The ASME Test Code on Compressors and Exhausters (PTC 10) and for Displacement Compressors (PTC 9) give complete instructions for testing compressors handling air. For compressors handling gases other than air, the codes are essentially complete, provided the gases conform to the perfect-gas laws or the generalized compressibility data are sufficiently accurate (see page 648). When a displacement-type air compressor is tested, the air must be discharged into the atmosphere; otherwise, reliable results cannot be obtained except under certain special circumstances specified in the code. When a centrifugal compressor is tested, air or gas may be discharged into the atmosphere or may be measured and retained within a closed system. Compressors handling gases for which none of the physical properties are known cannot be tested for capacity

Purpose

The first essential in any test is to establish its purpose. An air-compressor test is usually undertaken to determine the volume of air compressed and delivered in a given time under specified conditions or to determine the overall efficiency as a periodic check on operations, or for comparison with certain standards.

Capacity Measurement

The Compressed Air and Gas Institute has agreed to the methods described in ASME PTC-9 and PTC-10, also ISO 1217, as applicable. The form of the nozzle and all its associated dimensions for various sizes with throat diameters ranging from 1/8 to 24 in. are given in Figure. 8.7. The table in Figure. 8.7 gives approximate capacities for which each size of nozzle is suitable when discharging into or from the atmosphere. This table will be useful in selecting a nozzle for any particular test of a displacement-type compressor or of a centrifugal-type compressor discharging into the atmosphere. If the operating conditions are limited as specified on page 666, nozzles described in the ASME Code may be used. When the discharge pressure from a rotary displacement-type compressor is less than that required to meet the limitations given on page 678, the capacity may be determined by means of a slip test, page 678.

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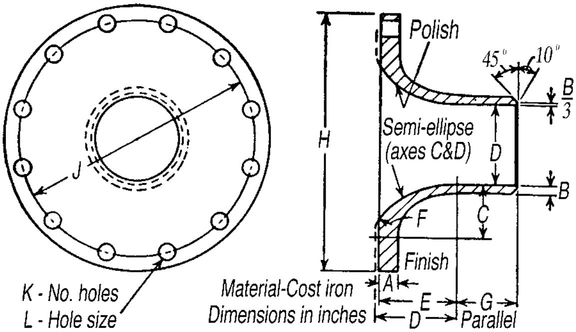

Long-radius low-ratio flange-type nozzle

D	A	B	C	E	F	G	H	J	K	L	Approx. flow rates, cfm	
											10" H_20	10" H_20
0.125	0.437	0.250	0.09	0.121	0.01	0.437	4.25	3.125	4	0.562	1	2
0.1875	0.437	0.250	0.13	0.181	0.01	0.468	4.25	3.125	4	0.562	2	4
0.250	0.437	0.250	0.17	0.242	0.01	0.500	4.25	3.125	4	0.562	4	8
0.375	0.625	0.250	0.25	0.363	0.02	0.562	7.50	6.00	4	0.750	9	18
0.500	0.625	0.250	0.34	0.484	0.03	0.625	7.50	6.00	4	0.750	16	32
0.750	0.625	0.250	0.50	0.726	0.04	0.750	7.50	6.00	4	0.750	36	71
1.000	0.937	0.250	0.67	0.699	0.05	0.875	9.00	7.50	8	0.750	62	127
1.375	1.000	0.250	0.92	1.332	0.07	1.063	11.00	9.50	8	0.875	119	239
2.000	1.000	0.313	1.33	1.938	0.10	1.500	11.00	9.50	8	0.875	253	506
2.500	1.000	0.375	1.67	2.422	0.13	1.875	11.00	9.50	8	0.875	397	790
3.000	1.000	0.375	2.00	2.906	0.15	2.500	11.00	9.50	8	0.875	565	1,127
4.000	1.125	0.438	2.67	3.875	0.20	3.000	13.50	11.75	8	0.875	1,010	2,020
5.000	1.188	0.500	3.33	4.844	0.25	3.750	16.00	14.25	12	1.000	1,590	3,160
6.000	1.250	0.500	4.00	5.812	0.30	4.500	19.00	17.00	12	1.000	2,260	4,510
8.000	1.438	0.625	5.33	7.750	0.40	6.000	23.50	21.25	16	1.125	4,050	8,100
10.000	1.688	0.625	6.67	9.688	0.50	7.500	27.50	25.00	20	1.250	6,350	12,600
12.000	1.875	0.750	8.00	11.625	0.60	9.000	32.00	29.50	20	1.375	9,100	18,200
18.000	2.375	1.000	12.00	17.438	0.90	13.500	46.00	42.75	32	1.625	18,000	36,000
24.000	2.750	1.125	16.00	23.250	1.20	18.000	59.50	56.00	44	1.625	39,500	78,000

Figure 8.7 Dimensions for standard nozzles.

Nozzle Coefficients

Coefficients for nozzles prescribed in the ASME Codes are nearly, but not quite, uniform for any particular nozzle diameter when used under limitations prescribed in the Codes. These coefficients vary with the differential pressure across the nozzle, the temperature of the air or gas flowing through the nozzle, and other factors. For arrangements A and B in Figure. 8.8 and for air, the coefficients are shown in Table 8.3 and must be selected with particular reference to the temperature and differential pressure across the nozzle as indicated by reference to the curves shown on Figure. 8.9, corresponding to the air temperature and differential pressure for any particular test. For arrangement C in Figure. 8.8, the nozzle coefficient may he taken as 0.993 to 0.995 when pressure before and after the nozzle may be maintained sensibly free from pressure and velocity pulsations.*

In general, test methods for compressors, whether of the displacement or centrifugal type, are essentially the same. The principal differences arise from the fact that in the former fluid flow is intermittent and pulsating while in the latter flow is steady and uniform, so that different rules covering the conduct of tests are required. As a matter of convenience and in order to avoid confusion, the two types of compressors are covered under separate headings in the Compressed Air and Gas Institute standards that are given in this section.

TESTS OF DISPLACEMENT COMPRESSORS, BLOWERS, AND VACUUM PUMPS

The paragraphs immediately following in this section constitute, in effect, a resume of the ASME Power Test Code (PTC 9), and afford an explanation of the methods used for air measurements in tests of displacement compressors, blowers, and vacuum pumps. Certain of the less important provisions of the code have been omitted in an effort to provide a simple exposition of the test methods employed. While the discussion outlined in this section has been directed primarily to two-stage air compressors, the same methods apply regardless of the number of stages of compression. Variations in setup and calculations for vacuum pump tests are included.

The ASME code applies to tests of complete compressor units when operated under conditions which permit discharging the gas compressed into the atmosphere or into pipe lines or reservoirs in which the pressure is maintained sensibly uniform and free from pressure or velocity pulsation. It is intended to cover the compressor only and is applicable for air compressors only when the unit is operated without an intake pipe or duct. The code provides that, when a compressor must be tested with an intake duct connected, an overall allowance must be made to compensate for the influence of the intake duct or pipe on the performance of the compressor.

*The exact value of the coefficient is stated as a function of the Reynolds number as follows:

Reynolds No.	Coefficient
200,000	0.987
250,000	0.990
300,000	0.992
350,000	0.993
400,000	0.994
450,000	0.995

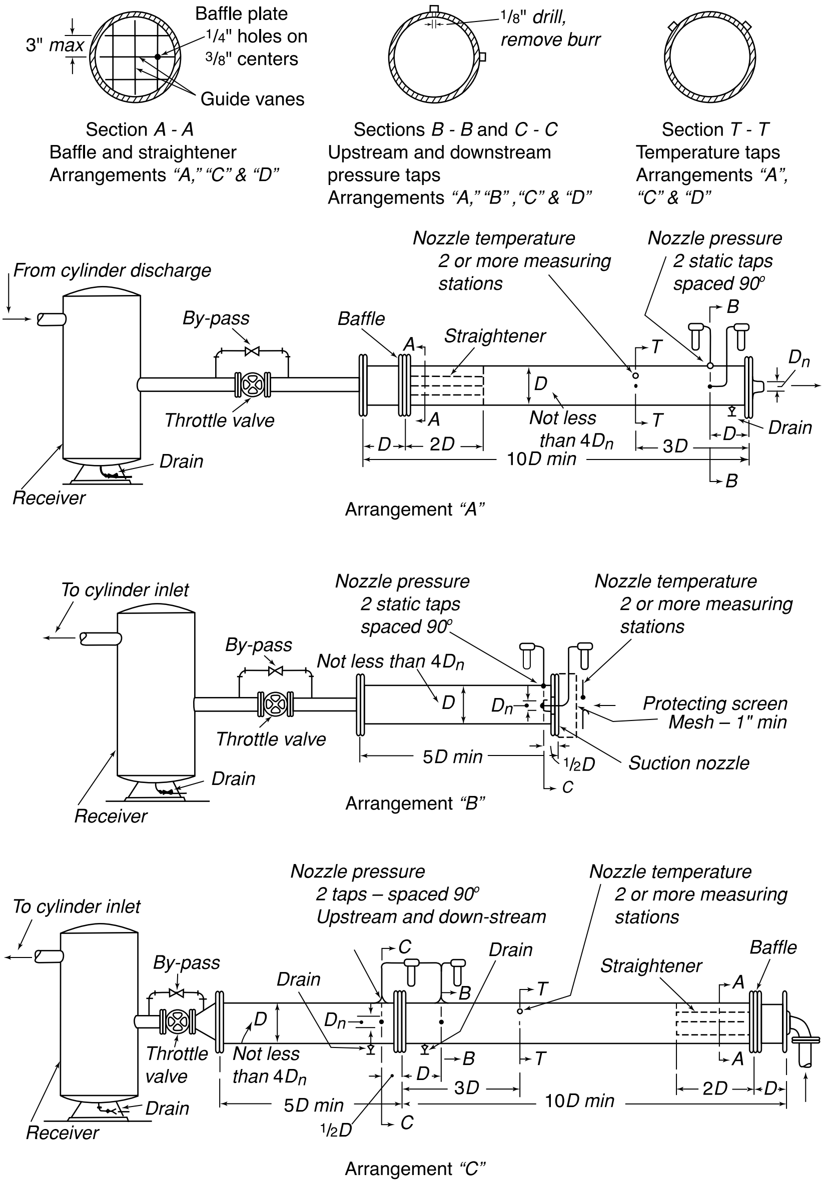

Figure 8.8 Coefficients for nozzles under various arrangements shown above are given in Table 8.3.

Table 8.3 Nozzle-flow Coefficients for air applicable to Arrangements A and B of Figure. 8.8.

	Nozzle Diameter, In.														
Curve	1/8	3/16	1/4	3/8	1/2	3/4	1	1 3/8	2	2 1/2	3	4	5	6	8
A	0.938	0.946	0.951	0.957	0.963	0.968	0.973	0.977	0.982	0.984	0.986	0.990	0.993	0.994	0.995
B	0.942	0.948	0.955	0.960	0.965	0.971	0.975	0.979	0.984	0.987	0.989	0.992	0.994	0.995	0.995
C	0.944	0.952	0.959	0.964	0.968	0.974	0.978	0.981	0.986	0.990	0.991	0.994	0.995	0.995	0.995
D	0.947	0.954	0.961	0.966	0.970	0.976	0.980	0.983	0.988	0.991	0.993	0.994	0.995	0.995	0.995
E	0.950	0.957	0.963	0.968	0.972	0.977	0.982	0.985	0.990	0.992	0.994	0.995	0.995	0.995	0.995
F	0.953	0.958	0.964	0.969	0.973	0.978	0.983	0.986	0.991	0.993	0.994	0.995	0.995	0.995	0.995
G	0.956	0.960	0.966	0.970	0.974	0.979	0.984	0.987	0.992	0.994	0.995	0.995	0.995	0.995	0.995
H	0.958	0.962	0.967	0.972	0.976	0.980	0.985	0.988	0.993	0.995	0.995	0.995	0.995	0.995	0.995
I	0.959	0.964	0.968	0.974	0.978	0.982	0.986	0.989	0.994	0.995	0.995	0.995	0.995	0.995	0.995
J	0.960	0.965	0.970	0.975	0.979	0.983	0.987	0.990	0.994	0.995	0.995	0.995	0.995	0.995	0.995
K	0.961	0.966	0.971	0.976	0.980	0.984	0.988	0.991	0.994	0.995	0.995	0.995	0.995	0.995	0.995
L	0.962	0.967	0.972	0.977	0.981	0.985	0.989	0.992	0.995	0.995	0.995	0.995	0.995	0.995	0.995
M	0.963	0.968	0.973	0.978	0.982	0.986	0.990	0.993	0.995	0.995	0.995	0.995	0.995	0.995	0.995
N	0.964	0.969	0.974	0.979	0.983	0.987	0.991	0.994	0.995	0.995	0.995	0.995	0.995	0.995	0.995

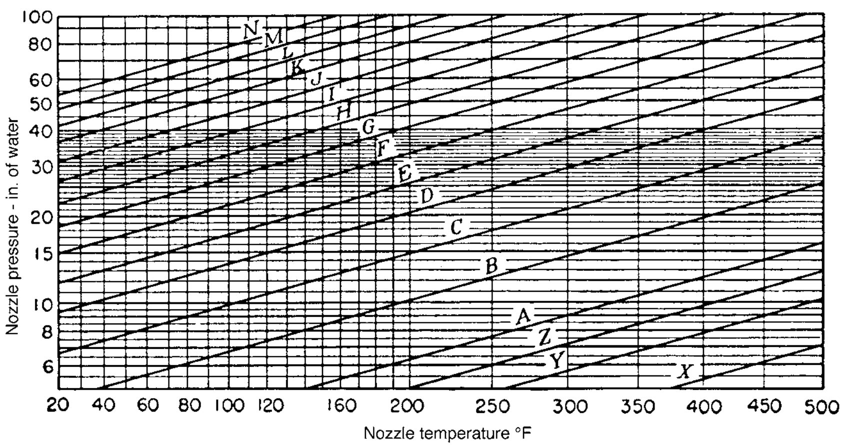

Figure 8.9 Curve for selecting nozzle coefficient from Table 8.3.

The power consumption of a multi-stage compressor depends not only on the operating conditions as to intake pressure, discharge pressure, speed, and so on, but also on the amount of heat removed in the intercoolers. Under winter conditions of operation, the cooling water temperature may be relatively high and the intake air temperature relatively low. Under this condition the degree of intercooling may be less than perfect. In the summertime the conditions may he reversed, and the degree of intercooling may he more than "perfect." The power consumption accordingly may vary 3 to 5 percent depending on the degree of intercooling obtained. Manufacturers' power-consumption statements are usually based on perfect intercooling; therefore, since cooling water conditions are extremely variable and it may be impossible to obtain perfect intercooling, a correction must he applied to the horsepower data to compensate for the variation. This correction applies only when testing a machine having two or more stages of compression with intercooling.

The barometric pressure is not subject to control by the test engineers. The discharge pressure is subject to control within certain limits, but it is usually difficult to hold it at exactly the desired test point.

If the test is made in humid summer weather or with the compressor intake in a location where warm, moist air is taken into the compressor, considerable moisture may be condensed out of the air between the compressor intake and the measuring nozzle. This may cause an error of as much as 1 or 2 percent in the capacity calculations.

To secure comparable results, the corrections for all of the preceding variables are discussed later.

Maximum Deviation from Specified Conditions

For each variable the maximum permissible deviation of the average operating conditions from conditions specified in the contract shall fall within the limits stated in column (2), Table 8.4.

Required Constancy of Test Operating Conditions

During any one test, no single value for any operating conditions shall fluctuate from the average for the test by any amount more than that shown in column (3), Table 8.4.

Method of Conducting the Test

Figure 8.10 shows diagrammatically the general scheme of the test setup for a two-stage displacement-type compressor. The compressor is isolated from the regular service line so that there can be no leakage into or out of the test system, and the entire output of the machine is throttled from the receiver to the low-pressure nozzle tank, for measurement. The actual delivered capacity of the compressor is calculated by a suitable formula using measured values of nozzle pressure, barometric pressure, air temperature at the nozzle, and air temperature at the compressor intake. The compressor is operated at constant speed, and the discharge pressure in the receiver is controlled by the rate of throttling into the nozzle tank.

Table 8.4 Maximum Allowable Variation in Operating Conditions

Variable	Deviation of Test From Value Specified	Fluctuation From Average During Any Test Run
Intake pressure	2 per cent of abs press	1 per cent
Compression ratio*	1 per cent	
Discharge pressure*	. . . .	1 per cent
Intake temperature	. . . .	1°F
Temperature difference, low-pressure in-take and exit air or gas from intercooler	15°F	
Speed	3 per cent	1 per cent
Cooling water inlet temperature	. . . .	2°F
Cooling water flow rate	. . . .	2 per cent
Nozzle temperature	. . . .	3°F
Nozzle differential pressure	. . . .	2 per cent
Voltage	5 per cent	2 per cent
Frequency	3 per cent	1 per cent
Power factor	1 per cent	1 per cent
Belt slip	3 per cent	None

*Discharge pressure shall be adjusted to maintain the compression ratio within the limits stated.

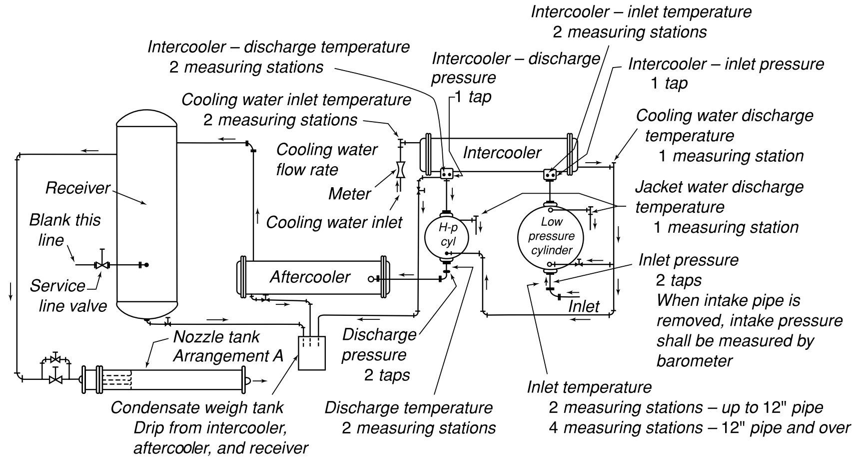

Figure 8.10 Arrangement of nozzle tank and other essential apparatus for test of multi-stage air compressor.

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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 discharged pressures. If the amplitude is found to exceed 10% of the average absolute pressure, methods for correction shall be mutually arranged.

For air compressors with atmospheric intake, removal of the intake pipe is mandatory, except by mutual agreement to the contrary. For all compressors where the intake pipe is used, an allowance must be agreed upon to compensate for pipe effects on capacity and horsepower.

Nozzle-tank Design

The accuracy normally expected from a low-pressure nozzle test is based on the assumption that the air stream approaching the nozzle flows straight and is free from turbulence, whirls, and spiral motions. It is practically impossible to measure the correct nozzle pressure without these conditions of flow. It is, therefore, extremely important to give detailed attention to the design and construction of the nozzle tank.

A conservative design of nozzle tank is shown in Figure. 8.11. The recommended minimum nozzle tank diameter is four times the maximum nozzle diameter, and the length of the tank should be at least 10 times its diameter or 40 times the maximum nozzle diameter.

The valve through which the air is throttled from the receiver to the nozzle tank frequently sets up a spiral flow, which necessitates the use of the baffle plate and guide vanes shown in Figure. 8.11. When the nozzle size is not more than 2 in., it is often satisfactory to use a long section of 8- or 10-in. pipe for the nozzle tank, and if the length of this pipe is made from two to three times that recommended in Figure. 8.11, the baffle plate and guide vanes are not necessary.

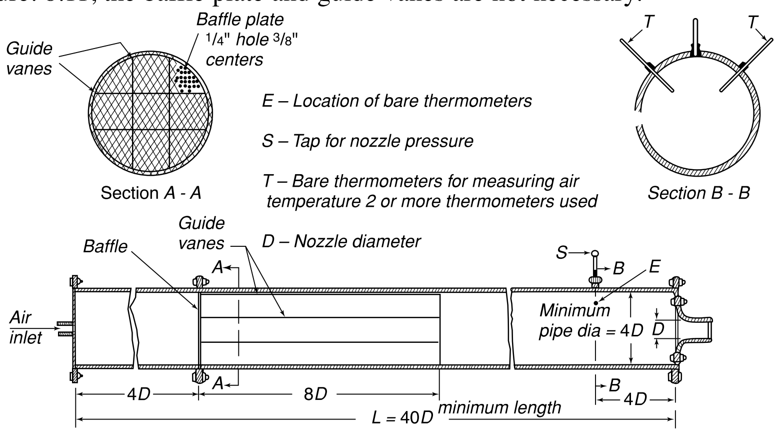

Figure 8.11 Details of nozzle tank for compressor test.

Nozzle Design and Selection

The ASME Test Code limits the use of nozzles for measuring the capacity of a displacement-type air compressor to those applications where the air may be discharged into the atmosphere from a reservoir or nozzle tank in which the pressure does not exceed 40 in. of water and is not less than 10 in. of water. These limiting conditions are such as to minimize the requirements for close tolerance as to nozzle contour. As measuring instruments under these conditions of operation, the accuracy depends primarily on (1) absolute roundness and uniformity of bore along the throat or straight portion of the nozzle and (2) smoothness and character of machine finish for the rounded entrance, with the throat of the nozzle tangent where it joins the rounded nozzle entrance.

Figure 8.7 gives dimensions for convenient nozzle sizes suitable for a large range of air capacity. One nozzle size can usually be selected to measure the output of a given compressor, at loads down to about 50% capacity.

In every case, the nozzle throat should he accurately measured in several directions to the nearest one-thousandth of an inch, and the average diameter used in the calculations.

Nozzle Pressure

When the ratio of gaging tank diameter to nozzle diameter is limited as specified in this code ($D_1/D_2 \geq 4$), the static pressure and total pressure in the gaging tank are sensibly identical. A manometer applied as shown in Figure. 8.11 shall be used for measuring this pressure. A tap connection for the manometer shall be located upstream one pipe diameter from the face of the nozzle. The inner bore of the manometer and the connecting tube shall never be less than 1/4 in., preferably 1/2 in. The inner bore shall not be contracted by projecting gaskets or other imperfections of manufacture or assembly. Nozzle pressures shall be measured in inches of water. The measuring scale used for the manometer shall be engine-divided with divisions of 0.1 in. or less, and accurate within 0.0100 in.

Manometer and Connections

A well-made manometer with an accurately graduated scale is an essential part of the apparatus for measuring nozzle pressure. Glass tubes with small bores are not suitable for manometers because of the error caused by capillarity. A glass tube 48 in. long, 5/8 in. outside diameter, and with a 3/8-in. bore is recommended.

Small air leaks in the manometer connections often cause serious errors, and after setup all joints should he carefully tested with a soap solution to detect small leaks.

Nozzle Temperature

In the calculation of air flow through the nozzle, the air temperature is of equal importance with the nozzle pressure. This temperature is measured on the upstream side of the nozzle with bare-bulb thermometers (no thermometer bulbs) located as shown in Figure. 8.11, and the test should not be started until the nozzle temperature has become approximately uniform.

Engraved stem thermometers of a good laboratory grade are held in the nozzle tank by small packing glands. The bulb of the thermometer should project into the tank as far as possible. To guard against accidental errors, two or more thermometers should be used, and they should be moved in and out of the tank to make sure that they are located to get the maximum temperature reading. The nozzle temperature should be determined to the nearest 1/2°F.

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 natural period of the discharge pipe approaches resonance with the speed of the compressor, or if the discharge pipe is too small in diameter or too long, pressure waves of considerable amplitude are induced in the discharge pipe and the discharge receiver, and it becomes impossible to measure accurately the average discharge pressure by any suitable means for damping the gage. The code defines discharge pressures 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 corresponds to the delivery line on an indicator diagram taken from the compressor cylinder, that is, for the period during which the discharge valve is open (A to B in Figure. 8.12). This figure shows respectively a typical pipe indicator diagram and a cylinder indicator diagram with that portion of the former marked to show the average discharge pressure as defined in the Code. Upon agreement by the parties to the test, a Bourdon gage may be used to measure the discharge pressure.

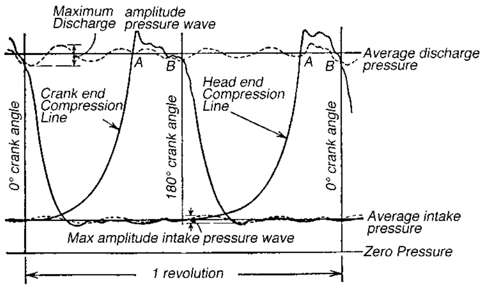

Figure 8.12 Pressure-time diagrams of double-acting compressor, illustrating cylinder pressures and pressure waves at measuring stations for inlet and discharge pressure.

For rotary-type displacement compressors, discharge pressure cannot be measured as previously specified. It shall be measured by determining the mean ordinate of a pressure curve obtained from a card drawn by an indicator operated with the drum rotated at approximately uniform speed (hand-pulled).

Intercooler pressure is not ordinarily an essential reading, but it should be taken as a check on the mechanical condition of the machine. The gage must be connected so that there is no needle vibration due to pulsations.

Intake Pressure

The performance of a compressor is extremely sensitive to variations in intake pressure, particularly to periodic fluctuations in pressure which approach resonance with the compression cycle or rotative speed of the unit (see pages 659-664). When intake pipes or ducts are connected, the intake pressure shall be regarded as the average pressure shown by a pressing-time indicator diagram taken at a point adjacent to the compressor intake during the intake portion of the stroke for the period during which the inlet valve is open (Figure. 8.12). The inlet pressure of an air compressor operating without an intake pipe shall be measured by the barometer.

Atmospheric pressure shall be measured with a Fortin-type mercurial barometer fitted with a vernier suitable for reading to the nearest 0.002 part of an inch. 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.

Air Intake, Temperature, and Pressure

The actual delivery capacity of an air compressor must always be referred to the actual conditions of temperature and pressure at the compressor intake. The procedure to be followed in measuring intake temperature and pressure therefore is extremely important.

Air Temperature at Intercooler

Thermometer wells should be located at the inlet and outlet of the intercooler, and temperature readings should be taken at these points. The temperature of the air leaving the intercooler (entering the high-pressure cylinder) is particularly necessary. The cooling water should be adjusted to obtain as near perfect cooling as possible and should be adjusted to give approximately the same relation between the low-pressure inlet temperature and the high-pressure inlet temperature on all test runs.

At the same time it should be recognized that the cooling water required is a part of the compressor performance and is chargeable to the cost of operation.

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Measurement of Cooling Water

Where a complete test is desired, the cooling water from the cylinder jackets and intercooler can best be measured by leading the water from the cooler outlet into a tank or tanks fitted with an orifice, preferably of the rounded-inlet type, and equipped with a gage glass and scale for actually measuring the head on the orifice. For most installations a suitable tank can be made from a section of 8- or 10-in. pipe about 4 ft. high, with a 3 or 4 in. coupling welded into the side. The orifice or nozzle used is exactly similar to that recommended for measuring the air. The coefficient of flow will be approximately 0.970, and the complete formula for calculation can be found in any standard engineers' handbook.

Air Temperature at Compressor Discharge

The temperature at the discharge of the high-pressure cylinder should be measured in a thermometer well. This reading is not used in calculations, but serves as an additional check on the mechanical condition of the compressor.

Speed

An accurate measurement of the total revolutions during the period of the test is required for calculating the revolutions per minute and the piston displacement of the unit. A mechanical counter geared to the compressor shaft or operated by a cam and indicating total revolutions is the best instrument to use. Tachometers and in-termittent counters should be avoided. Counter readings should be carefully controlled to eliminate changes in the discharge pressure and the nozzle-tank pressure, and also to eliminate inaccuracies in power-input readings.

Barometric Pressure

The barometric pressure must be known in order to make proper calculations of the air flow through the nozzle and for the determination of the absolute intake pressure. A barometer that has been carefully checked and compared with a standard barometer should he used, and if possible, it should be located in the near vicinity of the compressor. Simultaneous readings of the barometer and room temperature at the barometer should be taken at 1/2 -hour intervals throughout the test. If a reliable barometer is not available, an approximation may be obtained by using records of the nearest Weather Bureau station and correcting for the difference in altitude between the government station and the compressor.

Condensation

If the test is run during humid summer weather, or if the compressor intake is warm and moist, corrections to the capacity calculations will be required to compensate for the shrinkage in the volume of air due to condensation of moisture between the compressor intake and the measuring nozzle. It is necessary to drain all points, such as the intercooler, aftercooler, and receiver, and any low points in the piping between the compressor inlet and nozzle tank where moisture might collect. It is frequently difficult to obtain consistent values for the condensate, and it may be necessary to prolong the test to get dependable measurements. In cool weather or when the relative humidity of the atmosphere is below 30%, the correction for condensation is so small that it may be neglected. In summer weather, however, the correction may amount to as much as 2% of the total capacity.

Technique of Taking Readings

Readings should be taken as nearly simultaneously as possible and with sufficient frequency to ensure average results. This is important in obtaining the air-capacity data, but it is even more important when taking electrical power-input data.

CALCULATION OF TEST RESULTS: DISPLACEMENT COMPRESSORS

Actual Compressor Capacity

The ASME Code gives two sets of formulas for calculating the air quantity discharged through the nozzle. The first, known as adiabatic formulas, are complicated in form and rather difficult to use. The second set are simple approximations of the adiabatic formulas, and for conditions of flow permissible under the test code they give nearly identical results. The code specifies that the approximate formula shall be used for calculating the results of all acceptance tests. Two forms of this approximate formula are presented in these standards as follows:

$$Q_3 = 19.16 \frac{CD^2T_3}{p_3} \sqrt{\frac{(p_1 - p_2)p_2}{T_1}}$$

$$Q_3 = 2.552 \frac{CD^2T_3}{p_3} \sqrt{\frac{B_i}{T_1}}$$

where:

T_1 = Absolute Fahrenheit temperature, upstream side of nozzle.

T_3 = Absolute Fahrenheit temperature at which volume of flowing air is to be expressed; usually compressor intake temperature.

p_1 = Absolute static pressure, psi, upstream side of nozzle.

p_2 = Absolute total pressure, psi, downstream side of nozzle.

p_3 = Absolute pressure, psi, at which volume of flowing air is to be expressed; usually compressor intake.

B = Value p_2 expressed in in. Hg (32°F); in the usual case of a nozzle discharging into atmosphere, this is the barometer reading, corrected for the temperature of the mercury column, and the barometer calibration constant.

i = Differential pressure ($p_1 - p_2$), the pressure drop through the nozzle,expressed in inches of water column.

Q_3 = Cubic feet of air flowing per minute, expressed at pressure p_3 and temperature T_3, which in air-compressor testing are usually the conditions of the compressor intake.

D = Diameter of smallest part of nozzle throat, inches.

C = Coefficient of discharge.

Moisture Correction for Capacity

In the case of a compressor handling moist air, if some of the moisture is condensed and removed during the process of intercooling and aftercooling, the weight of the air and water vapor mixture which passes through the measuring nozzle will be less than that taken in by the compressor. In reducing the amount of air as shown by the nozzle measurements to terms of air at intake pressure and temperature, it is necessary to correct for the water thus removed, and this correction can most readily be made by reducing the capacity to terms of air at intake total pressure and temperature and discharge specific humidity. This correction may be made in either of two ways: through vapor pressure or through vapor density.

A simple method of approximating the correction is as follows. Having measured the condensation rate as discussed previously, the correction to be added to Q_3 is the product of the condensation rate and the specific volume for superheated steam at intake pressure and temperature. Equivalent volumes may be obtained from the curves given in Figure. 8.13.

Power Correction for Imperfect Intercooling

To correct for imperfect intercooling, there shall be added to the measured horsepower input calculated from the observed results of the test a horsepower correction as shown by the following formula:

$$\text{Horsepower correction} = \frac{1}{N} \frac{T_1 - T_1}{T_1} \text{ x } P$$

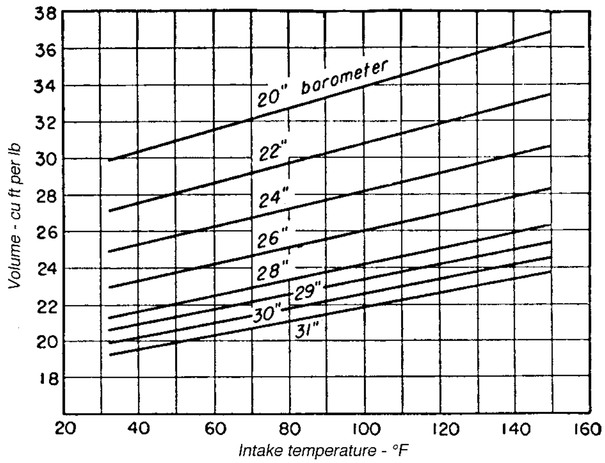

Figure 8.13 Equivalent volume condensate at atmospheric pressures and temperatures.

where:

P = Measured horsepower input.
N = Number of stages.
T_1 = Absolute temperature of air or gas at compressor intake (Rankine).
T_t = Absolute temperature of air or gas leaving the intercooler in question (Rankine).

If the compressor in question is a multi-stage unit with more than one intercooler, a correction must be made for each intercooler, and in each case T_1 will be the temperature of the air or gas at the compressor intake and T_1 will be the temperature of the air or gas leaving the intercooler in question.

Power Correction for Variation in Intake Pressure

To correct for intake pressure, the horsepower correction as given by the following formula is added to the horsepower input calculated from the observed results of the test.

$$\frac{p_1 - p_a}{P_a} \text{ x } P$$

where:

P = Measured horsepower input.
P_1 = Absolute contract intake pressure.
P_a = Absolute intake pressure observed during test.

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Power Correction for Variation In Compression Ratio

To correct for variations in compression ratio, there shall be added to the measured horsepower input calculated from the observed results of the test a horsepower correction as shown by the following formula:

$$\text{Horsepower correction} = \frac{\left(r_1^{(k-1)/Nk} - 1\right) - \left(r_t^{(k-1)/Nk} - 1\right)}{\left(r_t^{(k-1)/Nk} - 1\right)}$$

where:

P = Measured horsepower input.
N = Number of stages.
k = Ratio of specific heats (1.395 for air with some moisture as commonly employed in engineering).
r_1 = Ratio of compression under conditions guaranteed in contract.
r_t = Ratio of compression observed during test.

The formulas giving the correction for variation in compression ratio and also the correction for imperfect intercooling assume that the cylinder ratio of the compressor is such as to divide the work between the cylinders equally. These corrections will in all cases be small, so that, if for the compressor tested the division of work between cylinders is not exactly equal, the effect of such deviation on the correction will be negligible.

Power Correction for Speed

To correct for variations in speed, there shall be added to the measured horsepower input calculated from observed results of the test a horsepower correction as shown by the following formula:

$$\frac{N_c - N_t}{N_t} \text{ x } P_c$$

where:

P_c = Corrected horsepower input.
N_c = Contract speed, rpm.
N_t = Average speed during test, rpm.

Capacity Correction for Speed

To correct for variations in speed, a capacity correction as shown by the following formula shall be added to the measured capacity calculated from the observed results of the test.

$$\frac{N_c - N_t}{N_t} \text{ x } q$$

where:

q = Measured capacity calculated from observed results of test, cfm.
N_c = Contract speed, rpm.
N_t = Average speed during test, rpm.

Power Measurements

The following codes should be observed in determining the power input to the compressor:

Steam-engine drive: ASME Test Code for Reciprocating Steam Engines
Steam-turbine drive: ASME Test Code for Steam Turbines
Oil- or gas-engine drive: ASME Test Code for Internal Combustion Engines
Electric-motor drive:
Direct-current motor: AIEE Standard No.5
Induction motor: AIEE Standard No.9
Synchronous motor: AIEE Standard No.7

It is, of course, essential that all power test apparatus be carefully calibrated and the readings taken with care corresponding to that used in obtaining the air-capacity results.

Example

The following is an example of the essential calculations and corrections for the test of a two-stage air compressor with electric-motor drive:

SPECIFIED CONDITIONS OF OPERATION:

1. Speed = 225 rpm.
2. Discharge pressure = 100 psig.
3. Intake pressure, atmospheric at sea-level elevation, normal barometric pressure = 14.7 psia.
4. Intercooling is perfect.

OBSERVED DATA:

5. Duration of test = 60 minutes.
6. Speed, average, by revolution counter = 222.5 rpm.
7. Intake pressure (barometer) = 14.41 psia.
8. Discharge pressure = 97.6 psig.
9. Barometer, corrected to 32°F = 29.348 in. Hg.
10. Gaging tank
    a. Nozzle diameter = 4.062 in.
    b. Nozzle pressure = 13.91 in. water
    c. Nozzle temperature = 83.0°F
11. Intake temperature = 84.0°F.
12. Psychrometric readings
    a. Wet bulb = 72.0°F
    b. Dry bulb = 84.0°F
13. Intercooler pressure = 25.2 psig.
14. Air temperature at discharge of intercooler = 98.0°F.
15. Total condensate collected from intercooler, aftercooler, and receiver = 45.8 lb.

MOTOR-INPUT METER READINGS, CORRECTED:

16. AC input to stator, wattmeter = 205.80 kW.
17. DC input to field = 8.48 kW.
18. Total motor losses from calibrations, including excitation = 18.05 kW.
19. Volume discharged through nozzle referred to intake pressure and temperature, using the Equation on page 766, nozzle coefficient C, Table 8.3 and Figure 8.8.

$$Q = \frac{2.552 \times 4.062^2 \times 544 \times 0.995}{14.41} \times \sqrt{\frac{29.348 \times 13.91}{543}} = 1371.6\,\text{cfm}$$

20. Correction for moisture condensed and removed
    a. Condensation rate = $\dfrac{45.8\text{ lb}}{60\text{ min}}$ = 0.76333 lb/min
    b. Equivalent volume at intake conditions
       (Figure 8.13) = 22.5 x 0.76333 = 17.2 cfm
21. Capacity as run = 1371.6 + 17.2 = 1388.8 cfm.
22. Bhp as run = [(205.80 + 8.48) – 18.05] ÷ 0.746 = 263.04.

TEST RESULTS CORRECTED TO SPECIFIED CONDITIONS:

23. Capacity correction for speed = $\frac{225\text{-}222.5}{222.5}$ x 1388.8 = 15.6 cfm

24. Horsepower corrections for

a. Intake pressure $\frac{14.7-14.41}{14.41}\times 263.04 = 5.29\text{ hp}$

b. Compression ratio = $\frac{\left(r_1^{(k-1)/Nk}-1\right)-\left(r_t^{(k-1)/Nk}-1\right)}{r_t^{(k-1)/Nk}-1}\times 263.04$

$$r_1=\frac{114.7}{14.7}=7.8027, \qquad k=1.3947$$

$$r_t\ \frac{112.3}{14.41}=7.7731, \qquad \frac{(k-1)}{Nk}=0.1415$$

$$\frac{\left|(7.8027)^{0.1415}-1\right|-\left|(7.7731)^{0.1415}-1\right|}{\left(7.7731^{0.1415}-1\right)}\times 263.04$$
$$=\frac{(1.3374-1)-(1.3367-1)}{1.3367-1}\times 263.04 = 0.002079\times 263.04$$
$$=0.05\text{hp}$$

c. Intercooling = $\left(\frac{1}{2}\times\frac{544-558}{544}\right)\times 263.04 = -3.38\text{ hp}$

d. Speed = $(263.04+a+b+c)\times\frac{225-222.5}{222.5} = 2.98\text{ hp}$

25. Total bhp correction = $a + b + c + d$ = 5.44 hp.
26. Bhp corrected to contract conditions = 263.04 + 5.44 = 268.48 hp.
27. Capacity corrected to contract conditions = 1388.8 + 15.6 = 1404.4 cfm.
28. Bhp/100 cfm corrected to contract conditions.

$$=\frac{268.48 \text{ x } 100}{1404.4} = 19.117$$

Normally the air ihp/100 cfm is not the ultimate desired result. To show the complete performance, the overall cost or the overall efficiency to the basic source of power should be shown. This means that results should be expressed by kW/100, ehp/l00, pounds of steam/100, pounds of oil/100, or cubic feet of gas/100, depending on the source of power.

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DISPLACEMENT-TYPE VACUUM PUMPS

For accurately measuring the capacity of a vacuum pump, the same general procedure should be followed as outlined for compressors. The apparatus, however, should be placed at the intake of the machine rather than at the discharge. The setup for a nozzle test of a dry vacuum pump is shown in Figure 8.14.

The nozzle tank used in Figure 8.l5 is similar to that shown in Figure 8.12, but since the air enters as stream-line flow, the use of baffle plates and guide vanes is not essential.

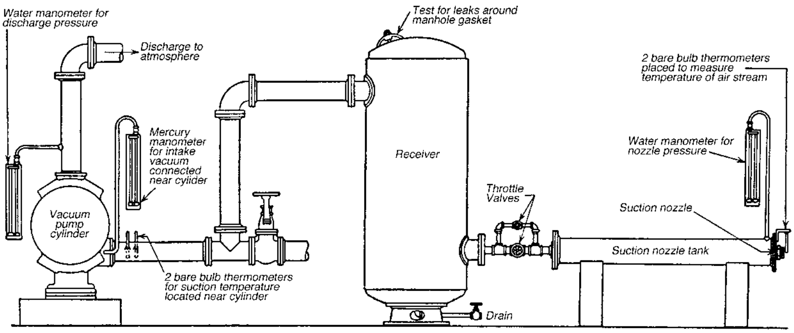

Figure 8.14 Setup for test of vacuum pump with nozzle in intake.

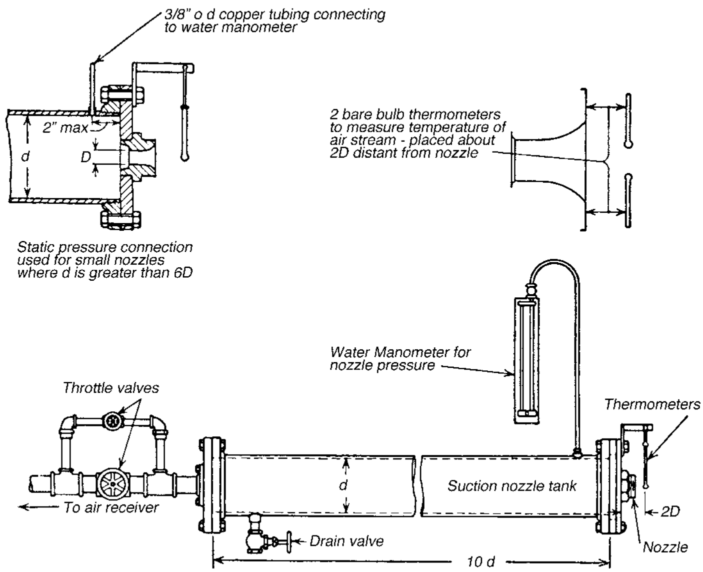

Figure 8.15 Assembly of intake nozzle tank and methods for measuring nozzle pressures.

The nozzle-tank diameter and length will be determined by the maximum nozzle size used, and a tank diameter at least four times the maximum nozzle diameter is satisfactory except that the smallest nozzle tank recommended is 6 in. in diameter by 10 ft. in length. It will be necessary to estimate the actual free air capacity of the vacuum pump at the desired test vacuum to choose the proper size of nozzle. This capacity can best be obtained from the manufacturer.

It is desirable to test a vacuum pump over a considerable range on either side of the test vacuum So that a curve will be procured from which the performance at the desired vacuum may be obtained.

It is essential that leakage into the system be absolutely eliminated. Extreme care is therefore necessary in making the setup. All pipe joints should be painted under vacuum.

The first equation for calculating the flow Q_3 through a nozzle is now applicable. The flow Q_3 is usually expressed at the temperature and pressure of the vacuum pump intake; p_1 will now be the barometric pressure, and p_2 the absolute pressure inside the nozzle tank. Since all the air entering the vacuum pump passes first through the nozzle, no moisture corrections are necessary for the flow or capacity calculations.

ROTARY COMPRESSORS, BLOWERS, AND VACUUM PUMPS

For displacement blowers and boosters of the type in which volumetric clearance is zero, when the discharge pressure is insufficient to provide throttling to the extent specified in the ASME Code, capacity may be determined by subtracting the leakage past the impellers from the gross displacement. This method does not lead to greater accuracy than nozzle measurements, but may be more convenient as a means of determining the approximate net capacity. The displacement may be determined from the measurements of the blower, and as defined on page 741, it is the product of the volume displaced per revolution and the normal speed in revolutions per minute. The leakage past the impellers is the product of the displacement per revolution and the number of revolutions per minute required to maintain the predetermined rated pressure with the discharge pipe from the blower or booster closed and the inlet pipe open to the atmosphere. The leakage test or tests may be conducted at the same pressure and temperature as the contract conditions, or the test may be run with a differential of 1 psi across the impellers and correction made for obtaining the slip at contract conditions by using the following formula.

$$S_c = S_t \sqrt{\Delta_c \frac{T_c G_t P_t}{T_t G_c p_c}}$$

where:

Δ_c = contract differential pressure, psi
S_t = slip at ~ differential, cfm
S_c = slip at contract conditions, cfm
T_t = Rankine temperature at test conditions
T_c = Rankine temperature at contract conditions
G_t = specific gravity at test conditions
G_c = specific gravity at contract conditions
p_t = discharge pressure at test conditions, psia
p_c = discharge pressure at contract conditions, psia

TESTS OF CENTRIFUGAL BLOWERS, COMPRESSORS AND EXHAUSTERS

The paragraphs that follow constitute a resume of the ASME Power Test Code (PTC 10). Certain of the less important provisions of the code have been omitted in order to provide a simple exposition of the test methods employed.

The ASME code provides standard directions for conducting and reporting tests on

centrifugal compressors or exhausters of the radial-flow, mixed-flow, or axial-flow types (hereafter inclusively covered by the term compressors) in which the gas specific weight change produced exceeds 7%. Apparatus of the centrifugal type for compressing or exhausting service in which the gas specific weight change is 7% or less shall be tested in accordance with the ASME Power Test Code for Fans (PTC11).

The capacity and power consumption of a centrifugal compressor depend on the composition of the gas, the density at intake, and the pressure rise. In the case of multi-stage group machines with intercooling between stage groups, the power also depends on the degree of intercooling. Manufacturers' statements of capacity and power are based on stipulated conditions of temperature, total pressure, and composition of gas prevailing at the compressor inlet, as well as on speed, pressure rise, and degree of intercooling (where intencoolers are used). Since these conditions cannot be subject to independent control by the test engineers, it is necessary to correct or adjust the test results to account for any deviation from specified conditions. To permit a direct comparison between test results and a manufacturer's guarantee, the correction for all of these variables is discussed later.

Where intercoolers are used between groups of stages, the conditions of heat removal and the drop in pressure in the intercooler and associated piping must be included as part of the operating conditions in any complete statement of performance guarantee.

In a multi-stage group compressor with intercooling and with a common driver, the contract performance must be determined for each stage group separately when the inlet conditions at the first stage and the degree of intercooling differ from the specified conditions. If the degree of intercooling is such that the deviations of density, pressure ratio, and *q/n* (ratio of capacity to revolutions per minute) at the inlets to both stage groups are within the limits stated in Table 8.5, corrections shall be applied to the performance of each stage group to reduce it to contract conditions as if each group were a single machine.

When testing a compressor, every effort should be made to have the operating conditions and composition of the gas as near as possible to those specified in the contract. A single contract condition or guarantee shall be established either by a complete characteristic curve or by not less than two test points that bracket the contract capacity by not more than ± 3 percent. The maximum deviation for which adjustment may be applied to any of the variables is given in Table 8.5. Under these conditions, the values of these variables, as calculated under the rules of this code, shall be accepted as indicating the performance under specified operating conditions.

Before starting a test, the machine shall be run for a sufficient length of time to assure steady conditions. The duration of a test shall be long enough to record sufficient data to demonstrate the uniformity of test operation, but in any event it shall be not less than 30-minute duration, but longer if the test code requirements of the driving element specify additional time. During the test, readings of each instrument having an important bearing on the calculation of results shall be taken at 5-minute intervals, the readings of each set being as nearly simultaneous as practicable. Throughout the test period, the machine shall be in continuous steady operation, the observed readings shall be consistent, and the maximum permissible fluctuation of any individual reading from the average shall be within the limits shown in column (3), Table 8.5.

Table 8.5 Maximum Allowable Variations in Operating Conditions, Centrifugal Compressors

Variable (1)	Deviation of Test From Value Specified (2)	Fluctuation From Average During Any One Test (3)
(a) Inlet pressure*	. .	2%
(b) Inlet temperature* (abs)	6%	4°F
(c) Intercooling, deg	30°F	4°F
(d) Discharge of pressure (abs)	. .	2%
(e) Capacity	4%	
(f) Molecular weight of gas*	10%	1%
(g) Ratio of specific heats	5%	
(h) Voltage	5%	2%
(i) Frequency	2%	1%
(j) Speed	5%	1%
(k) Power factor (synchronous motor)	1%	1%
(l) Nozzle pressure	. .	2%
(m) Nozzle temperature	. .	3°F
(n) Capacity speed ratio q/n	4%	
(o) Inlet specific weight*	10%	

*The combined effect of variables a, b, and f shall not produce a deviation greater than specified for inlet specific weight.

Test Arrangements

Four alternate test arrangements are provided in this code, and selection of the arrangement to be used for any particular test will depend on the type of compressor to be tested and upon the operating conditions.

Test arrangement 1 (Figure 8.16). Compressor with atmospheric inlet. The arrangement of the flow nozzle and the location of the instruments for measuring temperature, pressure, and so on, shall be as shown in arrangement A, Figure 8.8.

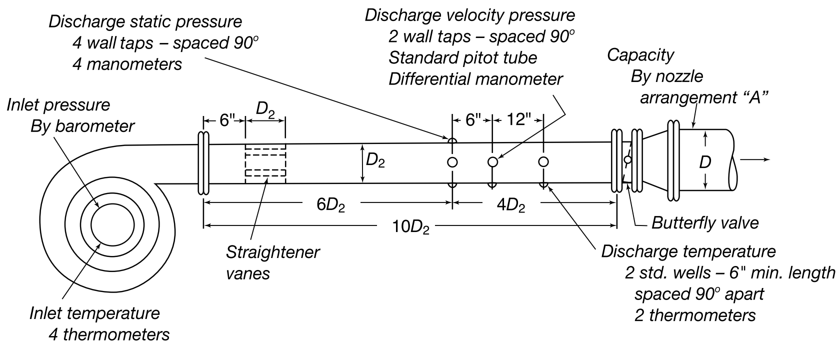

Figure 8.16 Test setup No. 1. volute-type compressor, atmospheric inlet.

Test arrangement 2 (Figure 8.17). Exhauster with atmospheric discharge. The arrangement of the flow nozzle and the location of instruments for measuring temperature, pressure, and so on, shall be as shown in arrangement B, Figure 8.8.

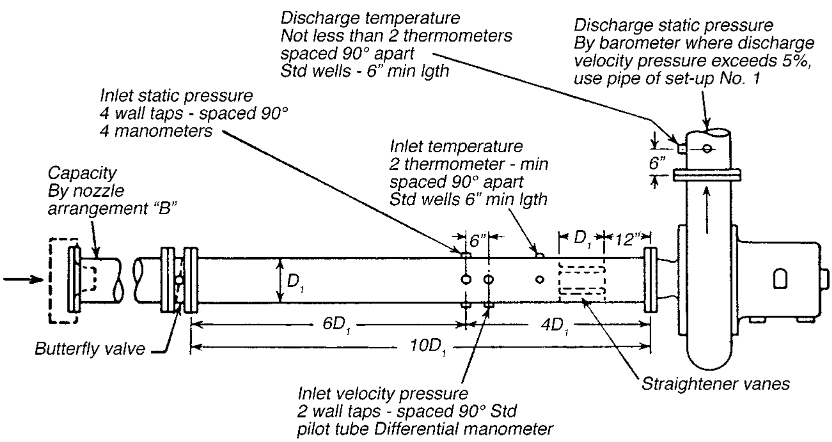

Figure 8.17 Test setup No.2. single-stage compressor, atmospheric discharge.

Test arrangement 3 (Figure 8.18). Compressor without intercooler. The arrangement of the flow nozzle and the location of instruments for measuring temperature, pressure, and so on, shall be as shown in arrangements A or C, Figure 8.8.

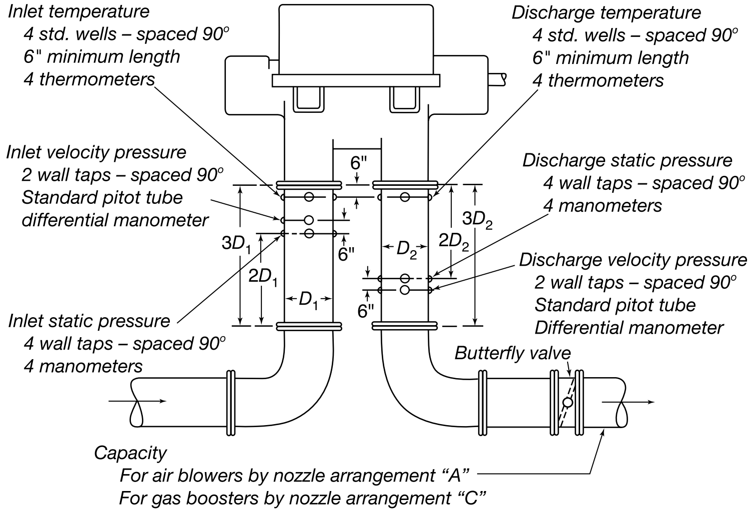

Figure 8.18 Test setup No.3. Multi-stage compressor.

Test arrangement 4 (Figure 8.19). Multi-stage group compressor with intercooler between stage groups. The arrangement of the flow nozzle and the location of instruments for measuring temperature, pressure, and so on, shall be as shown in arrangements A or C, Figure 8.8.

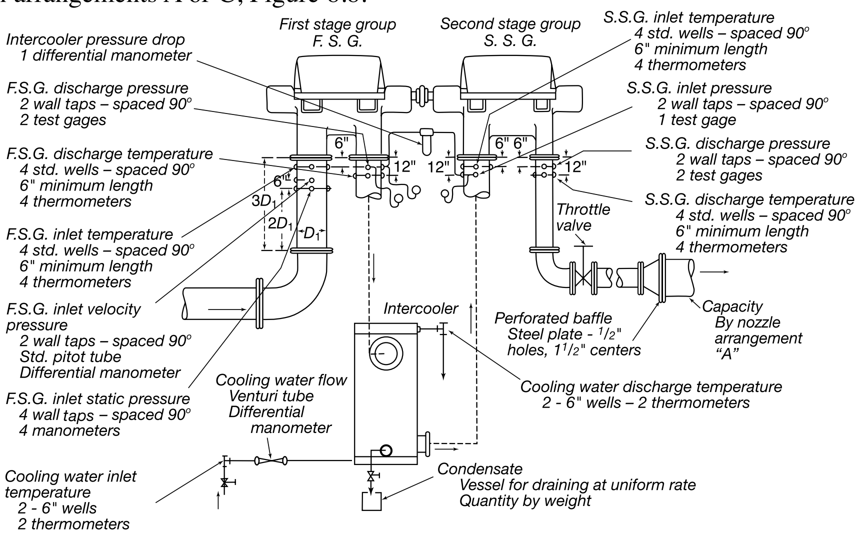

Figure 8.19 Test setup No. 4. Multi-stage groups with intercooler.

Barometric Pressure

The barometric pressure shall be read at intervals of 30 minutes, during the test from a mercury barometer of the type used by the U.S. Weather Bureau.

Inlet Pressure

When the inlet flange of the compressor is open to the atmosphere (Figure 8.16), the inlet pressure shall be taken as the barometric pressure adjacent to the compressor intake. When the inlet flange to the compressor is piped, the inlet pressure shall be the sum of the static and velocity pressure as computed from measurements made with instruments placed as shown in Figures 8.17, 8.18, or 8.19.

Discharge Pressure

When the discharge flange of the compressor is open to the atmosphere (Figure 8.17), the discharge pressure shall be taken as the sum of the barometric pressure and the velocity pressure at the plane of the discharge flange. If the velocity pressure is greater than 5 percent of the total pressure, the test shall be conducted with the outlet flange piped. When the outlet flange of the compressor is piped, the discharge pressure shall be the sum of the static and the velocity pressure as computed from measurements made with instruments placed as shown in Figures 8.16, 8.18, or 8.19.

Static Pressure

The static pressure shall be taken as the arithmetic average of the readings obtained by means of four wall taps, each connected to a separate manometer. The four taps shall be disposed at intervals of 90° around the circumference of the pipe. The diameter of the holes shall not be greater than one thirtieth of the pipe diameter (nor less than 1/8 in.), and they shall be drilled perpendicular to the pipe wall, with their inner edges free of burrs. Where the individual readings of the four wall taps differ from their mean by more than 1 percent, the cause shall be determined and corrected. If the cause is traceable to the flow pattern at the measuring section, and this cannot be corrected, a reliable test cannot be obtained. The pressure-measuring stations shall be located in a region where the flow is essentially parallel to the pipe wall. For the measurement of pressure or pressure differences in excess of 35 psig, dead-weight gages, or their equivalent, or calibrated gages shall be employed. For lower pressures or pressure differences, liquid manometers shall be used. Whichever of the above means of pressure measurement is employed, the instruments shall be so graduated that readings can be made within 1/2 percent of the absolute pressure.

Velocity Pressure

When the velocity pressure is not more than 5 percent of the total pressure, it shall be calculated on the basis of average velocity. The velocity shall be computed as the ratio of the quantity at the measuring section to the pipe area.

When the velocity pressure is more than 5 percent of the total pressure, it shall be determined by a pitot-tube traverse. The traverse shall consist of readings made at 10 traverse points across each of two diameter disposed at 90 degrees to each other. The traverse points shall be spaced at equal area positions. In a round pipe, the spacing shall be as defined in Figure 8.20. The pitot tube shall be of a type and design shown in Figure 8.21.

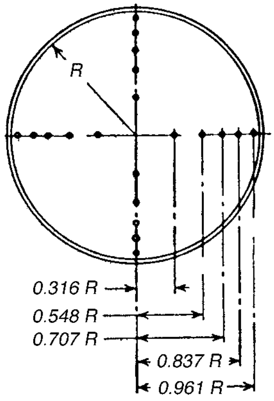

Figure 8.20 Traverse points in pipe.

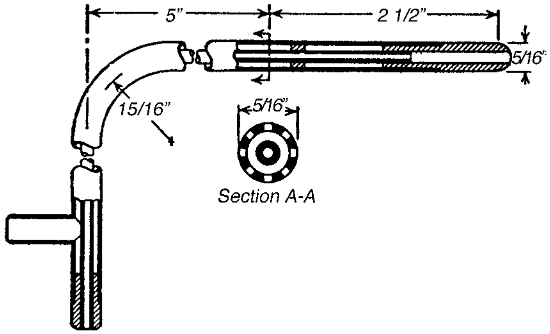

Figure 8.21 Pitot tube.

Inlet Temperature

The inlet temperature shall be measured by four temperature-measuring devices. When the compressor has a piped inlet flange, the instruments shall be placed as shown in Figures 8.17, 8.18, and 8.19 and shall be disposed symmetrically and at 45° to the inlet-pressure-measuring location. For machines assembled for test with an atmospheric intake (Figure 8.16), the inlet total temperature is the atmospheric temperature measured in a region of substantially zero velocity (less than 125 fps) in the vicinity of the inlet flange. For machines assembled for test with an intake pipe, the intake temperature shall be the sum of the measured stream temperature and the velocity recovery effect. Thus the total temperature is

$$t_{1_t} = t_1 + \frac{1}{Jc_p}\frac{V^2}{2g}(1-\alpha)$$

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where:

c_p = Specific heat at constant pressure.
g = Acceleration due to gravity (32.17 fps^2 at sea level and 45 degrees latitude).
t_1 = Measured temperature, °F.
V = Velocity at temperature-measuring station, fps.
α = Recovery factor of temperature-measuring device.
J = Mechanical equivalent of heat = 778 ft-lb/Btu.

For temperature-measuring devices, such as bare thermometer, wells, or thermocouples installed perpendicular to the stream flow, the recovery factor a equals 0.6. For thermocouples installed in such a fashion that their junction points essentially upstream, the recovery factor α equals 1.0.

Discharge Temperature

For an exhauster assembled for test with an open exhaust (Figure 8.17), the discharge temperature shall be the total temperature as measured at not less than four stations symmetrically disposed around the discharge flange. For a compressor or exhauster assembled for test with an exit pipe attached, four discharge temperatures shall be measured approximately in the plane of the discharge static pressure stations and disposed at 45° to them, as shown in Figures 8.16, 8.18, or 8.19. Velocity corrections to the measured discharge temperatures shall be made as explained on page 684.

Temperature Measurements

Depending on the operating conditions or on convenience, the temperatures may be measured by certified thermometers or calibrated thermocouples inserted into the pipe or into wells. The installation of the temperature-measuring device directly into the pipe without the addition of a well is desirable for temperatures below 300°F.

Whichever means is employed, the temperature device shall be so chosen that it can be read to within an accuracy of 0.2 percent of the absolute temperature. The average of the four readings at each measuring station shall be taken as the temperature of the fluid. If discrepancies between the individual readings and the average are greater than 0.2 percent of the absolute temperature, they shall be investigated and eliminated.

Capacity

The ASME Code provides for the measurements of capacity by means of a long-radius low-ratio nozzle, page 658, located (A) in the discharge pipe of the compressor and discharging to the atmosphere, (B) in the intake to the compressor and discharging into the compressor from the atmosphere, or (C) in either the intake or discharge pipe in a closed system (Figure 8.22). For arrangement (A) or (B), the nozzle diameter shall be such that the drop in pressure across the nozzle will not be less than 10 in. of water or greater than 100 in. of water. For arrangement (C) the nozzle diameter shall be such that the Reynolds number (Part 5, Chapter 4, equation 6, Par. 23, ASME Power Test Codes) will not be less than 300,000. For information regarding test nozzles and nozzle coefficients, see pages 658-659.

Nozzle Pressure

The nozzle pressure designates the differential pressure Δp across the nozzle used in the flow formulas (see page 690-691). In arrangements A and B, nozzle pressure is measured directly by the differential manometer when located as indicated in Figure 8.22, and must be used in Eq. (8.13) or (8.14). In arrangement C, Δp is measured by a differential manometer located as shown in Figure 8.22 and is to be used in Eq. (8.13) for calculating flow.

Differential Pressure

The differential pressure across the nozzle and the temperature ahead of the nozzle shall be measured with duplicate instruments independently connected at the locations shown in Figure 8.22. For arrangements B and C, the pressure alter the nozzle shall be read by two independent manometers connected to the downstream pressure taps as shown in Figure 8.22. For arrangement B (Figure 8.22), the upstream pressure is equal to the barometric pressure, while the downstream pressure is the difference between the barometric pressure and the differential pressure. For arrangement A (Figure 8.22), the downstream pressure is equal to the barometric pressure, while the upstream pressure is equal to the sum of the barometric and the differential pressures. For arrangement C (Figure 8.22), the upstream pressure is equal to the sum of the downstream pressure and the differential pressure. When arrangement A or C is employed, upstream straightening vanes shall be installed. The flow being measured must be sensibly steady, and the manometers must not show pulsations greater than 2 percent of the differential pressure. Any greater pulsation in the flow is to be corrected at its source; attempts to reduce the pulsations by damping the correcting piping to the manometers are not permissible.

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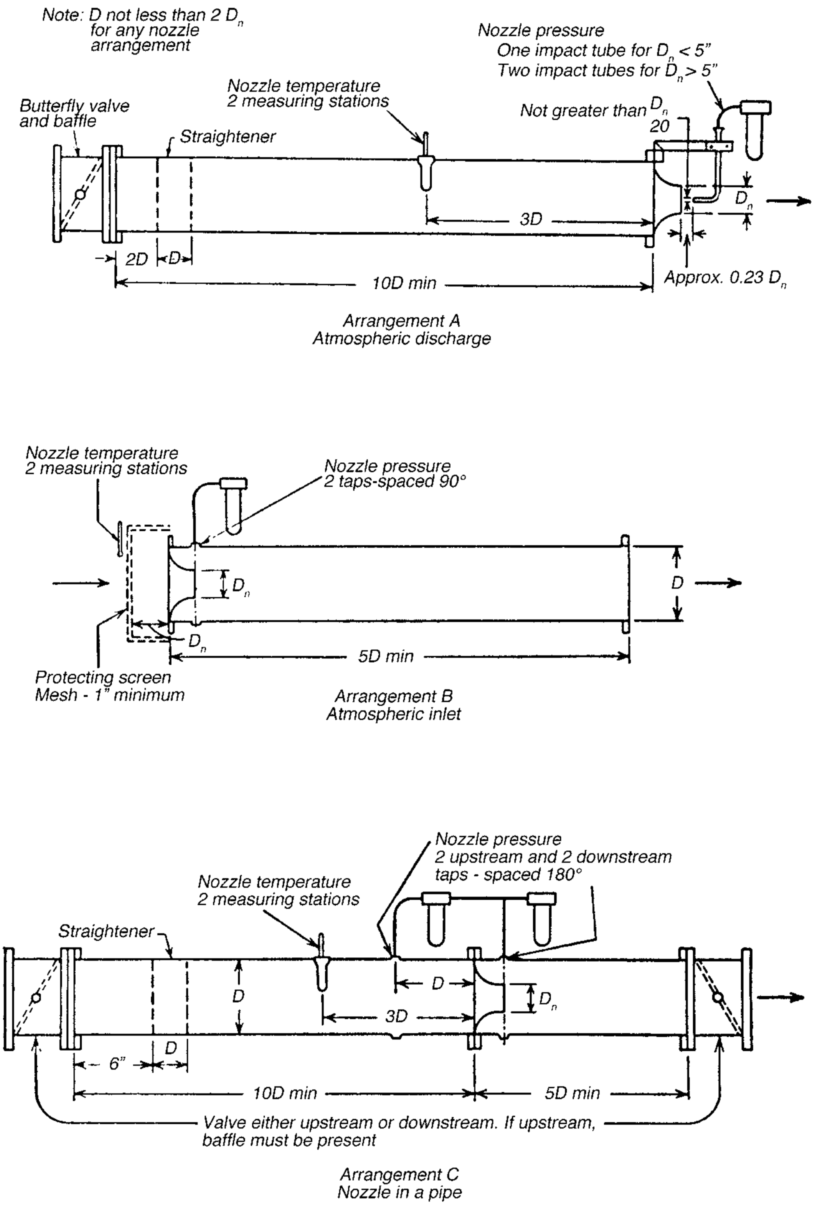

Figure 8.22 Various arrangements of flow nozzles for compressor tests.

Nozzle Temperature

The nozzle temperature shall be measured on the upstream side of the nozzle by instruments located as shown in Figure 8.22. No fewer than two instruments shall be used.

Cooling Water

When intercoolers are used, the flow of cooling water shall be measured by indicating-type meters accurate within 2 percent as shown by calibration under flow conditions corresponding to those obtaining during the test. The flow of cooling water through the intercooler may be adjusted to regulate the degree of intercooling.

Speed Measurement

An integrating revolution counter directly connected to a geared rotating shaft shall be used to record the total number of revolutions of the compressor during a test. The rate of speed shall be computed from the total number of revolutions during successive periods and the time of those periods.

Time Measurement

The date and time of day at which each individual test reading is taken and the time of day during which the test is conducted shall be recorded. The time of day may be determined by observation of timepieces by the individual observers, which timepieces have been compared with a master clock and are accurate to within 30 seconds per day.

Technique of Taking Readings

Readings should be taken as nearly simultaneously as possible and with sufficient frequency to ensure average results. This is important in obtaining the air-capacity data but is even more important when taking electrical power-input data.

Computation of Results

A complete presentation of the performance of a compressor must include a statement of the following significant quantities: capacity, pressure ratio, and power consumption. These quantities shall be stated for specific conditions of operation including pressure, inlet temperature, discharge pressure, rotative speed, and degrees of intercooling, including the temperature and quantity of the circulating water entering the intercoolers and the pressure drop across the intercoolers.

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Before final calculations are undertaken, the observed data recorded during each test run shall be scrutinized. Readings at the beginning of any test run may be discarded provided the time interval covered by the acceptable data is not less than 30 minutes. If a sufficient number of consecutive test readings meeting the conditions on pages 679-682 do not cover the minimum time specified for the test, the test shall be repeated.

Velocity Pressure

When the velocity pressure is not more than 5 percent of the total pressure, the velocity shall be computed as the ratio of the quantity at the measuring section to the pipe area. The velocity pressure shall then be

$$\text{Velocity pressure, psi} = \frac{V_{av}{}^2\gamma}{2g \times 144} \qquad (8.5)$$

When the velocity pressure (page 745) is greater than 5 percent of the total pressure, it shall be computed from a pitot-tube traverse in accordance with the following equations:

$$\text{Velocity pressure, psi} = \frac{\Sigma V_i{}^3\gamma}{144N2gV_{av}} \qquad (8.6)$$

where:

V_i = Velocity at each traverse point (i) as determined by the pitot tube, fps.

Σ = Indicates that the sum is to be taken of the third powers of the velocity at each traverse point.

γ = Specific weight of the gas at the measuring section, lb/ft^3.

N = Number of traverse points (20).

g = Gravitational constant = 32.17 fps^2.

V_{av} = Average velocity of gas at the traverse section as determined by dividing the volume rate of flow at the section by the area of the section, fps.

Specific Weight

Computations of specific weight y shall be made from measured values of temperature, pressure, specific gravity, relative humidity, molecular weight, or chemical composition. Alternate formulas are given to facilitate the direct use of the measurable properties.

For any dry gas, where R is the gas constant and is defined as 1544/M:

For dry air:

$$\gamma = \frac{p_1 \text{ x } 144}{ZT_1R} \qquad (8.7)$$

$$\gamma = \frac{p_1}{ZT_1} 2.70 \qquad (8.8)$$

For any gas or air containing water vapor, where *G* is based on dry air:

$$\gamma = \frac{p_1}{ZT_1} G \text{ x } 2.70 \qquad (8.8)$$

For any gas or air containing water vapor, where MW_o is the molecular weight of dry air:

$$\gamma = \frac{p_1 MW_1}{ZT_1 MW_o} 2.70 \qquad (8.10)$$

For any gas or air containing water vapor, where *RH* is the relative humidity and Z= 1:

$$\gamma_m = \gamma_o \frac{1 - RHp_s}{p_1} + RH\gamma \qquad (8.11)$$

For conversion of any known values of γ_x , to conditions p_1 and T_1, where Z = 1:

$$\gamma_m = \gamma_x \frac{p_1 T_x}{p_x T_1} \qquad (8.12)$$

where:

Z	=	Supercompressibility factor as defined in the equation of state.
pv	=	ZRT; for a perfect gas and the common diatomic gases in the low-pressure range, Z is approximately 1; for air at pressures below 115 psia, Z shall be taken as 1.
R	=	Gas constant 1544/M, ft-lb/°F.
G	=	Specific gravity with respect to dry air.
MW	=	Molecular weight.
MW_o	=	Molecular weight of dry air = 28.96.
γ	=	Specific weight at point of measurement, lb/ft³.
γ_o	=	Specific weight of dry gas at p_1 and T_1, lb/ift³.
p_s	=	Saturation pressure of water vapor at T_1, psia.
γ_s	=	Specific weight of saturated water vapor at T_1, lb/ft³.
RH	=	Relative humidity at measuring section.
p_1	=	Pressure at compressor inlet, psia.
T_1	=	Temperature at compressor inlet, °F abs.

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For tests with air, relative humidity shall be determined from measurements of the wet- and dry-bulb temperatures and from the psychrometric tables (published by the Department of Agriculture, U.S. Weather Bureau). Table 8.6 gives values of specific gravity for moist air throughout the usual range of temperatures and degrees of saturation. The use of Table 8.6 shall be limited to a barometric pressure range of 28 to 30.5 in. Hg.

Table 8.6 Specific Gravity of Moist Air at Standard Sea-level Pressure

Temperature deg F	Relative Humidity, Per Cent									
	10	20	30	40	50	60	70	80	90	100
0	.9999	.9999	.9999	.9998	.9998	.9997	.9997	.9996	.9996	.9995
10	.9999	.9998	.9998	.9997	.9996	.9995	.9994	.9994	.9993	.9992
20	.9999	.9997	.9996	.9995	.9994	.9992	.9991	.9990	.9988	.9987
30	.9998	.9996	.9994	.9992	.9990	.9987	.9985	.9983	.9981	.9979
40	.9997	.9994	.9991	.9987	.9984	.9981	.9978	.9975	.9972	.9969
50	.9995	.9991	.9986	.9982	.9977	.9972	.9968	.9963	.9959	.9954
60	.9993	.9987	.9980	.9974	.9967	.9960	.9954	.9947	.9941	.9934
70	.9991	.9981	.9972	.9963	.9953	.9944	.9935	.9925	.9916	.9907
80	.9987	.9974	.9961	.9948	.9935	.9922	.9909	.9896	.9883	.9870
90	.9982	.9964	.9946	.9928	.9910	.9892	.9875	.9857	.9839	.9821
100	.9976	.9951	.9927	.9903	.9878	.9854	.9830	.9805	.9781	.9756
110	.9967	.9935	.9902	.9869	.9837	.9804	.9771	.9738	.9706	.9673
120	.9957	.9913	.9870	.9827	.9783	.9740	.9697	.9653	.9610	.9567
130	.9943	.9886	.9830	.9773	.9716	.9659	.9602	.9545	.9488	.9432

For gases other than air, where the chemical composition is likely to be variable, m shall be computed from values of G, measured directly by the gas balance or indirectly through chemical analysis.

Capacity

For tests with air, providing no condensation occurs and using the nozzle arrangement A or B of Figure 8.22 as described on pages 686-687, the capacity may be conveniently computed by the formula

$$Q_1 = \frac{36.0 C D_n^2 p_{2n} T_1 \sqrt{X(X+1)}}{p_1 \sqrt{T_n} \sqrt{G}} \qquad (8.13)$$

For tests with gases (including air), providing no condensation occurs and using the nozzle arrangement C of Figure 8.22, the following formula shall be used [for nozzle arrangements A and B, either Eq. (8.13) or (8.14) may be used]:*

$$Q_1 \frac{31.5CD_n^2 Y^1 \sqrt{\gamma_n \Delta p}}{\gamma_1} \qquad (8.14)$$

where:

Q_1 = Capacity, volume rate of flow at inlet conditions p_1 and T_1, cfm.
D_n = Nozzle-throat diameter, in.
C = Flow coefficient (Table 8.3).
T_n = Total temperature at upstream side of nozzle, °F abs.
D_1 = Diameter of nozzle pipe, in.
p_1 = Total pressure at compressor inlet, psia.
T_1 = Absolute temperature at compressor inlet, °F abs.
p_{2n} = Static pressure, downstream side of nozzle, psia.
p_{1n} = Total pressure, upstream side of nozzle, psia.
X = $(p_{1n}/p_{2n})^{(k-1)/k} - 1$; for standard air, $(k - 1)/k = 0.283$ (see Table 8.7).
r = p_{1n}/p_{2n}
γ_n = Specific weight of gas, upstream side of nozzle, lb/ft^3.
γ_1 = Specific weight at compressor inlet, lb/ft^3.
Δp = Differential pressure across nozzle $(p_{1n} - p_{2n})$, psi.

$$Y^1 = \left[\frac{k}{k-1}\left(\frac{p_{2n}}{p_{1n}}\right)^{2/k}\frac{1-p_{2n}/p_{1n}{}^{(k-1)/k}}{1-(p_{2n}/p_{1n})}\right]^{1/2} \div \left[1-\left(\frac{D_n}{D_1}\right)^4\left(\frac{p_{2n}}{p_{1n}}\right)^{2/k}\right]^{1/2}$$

*It shall be note that p_{1n}, when used in the Y^1 factors of Eq. (8.14), is static pressure and that the velocity of approach effect is accounted for by selecting values of Y^1 for the correct ratio of D_n/D_1, in Table 8.8.

However, when p_{1n} becomes total pressure, as in the case of nozzle arrangement A or B, the correct value of Y_1 is found under $D_n/D_1 = 0$.

To facilitate the use of Eq. (8.13), values of X have been computed for standard air, and are arranged in Table 8.7. In like manner, values of Y^1, for Eq. (8.14), are given in Table 8.8. Either of these tables may be used for air and gases in which the value of k lies between 1.39 and 1.40.

Values of the flow coefficient are given in Table 8.3. Selection of the values for C is made through the use of the curves of Figure 8.9, which serve to integrate the relation of nozzle pressure and nozzle temperature, and thereby avoid the necessity of computing Reynolds number.

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When an intercooled compressor operates with moist air or gas, and the flow is measured on the discharge side of the compressor, a correction to the measured flow shall be made when any moisture is removed by the intercooler. This correction can be based on either the vapor pressure or the vapor density, and since the correction is small, any one of the generally used methods is acceptable (see page 671).

Theoretical Power for Compression

The theoretical power to compress the gas delivered by a compressor shall be computed for an isentropic compression. For a single-stage group and for diatomic gases, including air, where c_p is known, P_t shall be computed by:

$$P_t = \frac{wc_pT_{1t}}{42.42}\left[\left(\frac{p_2}{p_1}\right)^{(k-1)/k} - 1\right] \qquad (8.15)$$

When used for air or gases having a value of k between 1.39 and 1.40, Eq. (8.15) may be stated in terms of volume rate of flow and of X to permit the use of Table 8.7.

$$P_t = \frac{Q_1 p_1 X}{64.85} \qquad (8.16)$$

where:

c_p = Arithmetic average of specific heats at constant pressure between initial and final condition, Btu/lb/°F.

k = Arithmetic average of specific heat ration c_p/c_v between initial and final conditions.

p_1 = Inlet pressure of blower, psia.

p_2 = Discharge pressure of blower, psia.

Q_1 = Volume rate of flow at inlet conditions, cfm.

$$X = \left[\left(\frac{p_2}{p_1}\right)^{0.283} - 1\right] \quad \text{(values from Table 8.7)}$$

Table 8.7 Values of X for Standard Air and Perfect Diatomic Gases*

$$X = \left(\frac{p_1}{p_2}\right)^{0.283} - 1$$

r		0	1	2	3	4	5	6	7	8	9	Proportional Parts	
1.00	0.00	000	028	057	085	113	141	169	198	226	254	28	
1.01		282	310	338	366	394	422	450	478	506	534	1	2.8
1.02		562	590	618	646	673	701	729	757	785	812	2	5.6
1.03		840	868	895	923	951	978	006	034	061	089	3	8.4
1.04	0.01	116	144	171	199	226	253	281	308	336	363	4	11.2
1.05		390	418	445	472	500	527	554	581	608	636	5	14.0
1.06		663	690	717	744	771	798	825	852	879	906	6	16.8
1.07		933	960	987	014	041	068	095	122	148	175	7	19.6
1.08	0.02	202	229	255	282	309	336	362	389	416	442	8	22.4
1.09		469	495	522	549	575	602	628	655	681	708	9	25.2
1.10		734	760	787	813	840	866	892	919	945	971	27	
1.11		997	024	050	076	102	129	155	181	207	233	1	2.7
1.12	0.03	259	285	311	337	363	389	415	441	467	493	2	5.4
1.13		519	545	571	597	623	649	675	700	726	752	3	8.1
1.14		778	804	829	855	881	906	932	958	983	009	4	10.8
1.15	0.04	035	060	086	111	137	162	188	213	239	264	5	13.5
1.16		290	315	341	366	391	417	422	467	493	518	6	16.2
1.17		543	569	594	619	644	670	695	720	745	770	7	18.9
1.18		796	821	846	871	896	921	946	971	996	021	8	21.6
1.19	0.05	046	071	096	121	146	171	196	221	245	270	9	24.3
1.20		295	320	345	370	394	419	444	469	493	518	26	
1.21		543	567	592	617	641	666	691	715	740	764	1	2.6
1.22		789	813	838	862	887	911	936	960	985	009	2	5.2
1.23	0.06	034	058	082	107	131	155	180	204	228	253	3	7.8
1.24		277	301	325	350	374	398	422	446	470	495	4	10.4
1.25		519	543	567	591	615	639	663	687	711	735	5	13.0
1.26		759	783	807	831	855	879	903	927	951	974	6	15.6
1.27		998	022	046	070	094	117	141	165	189	212	7	18.2
1.28	0.07	236	260	283	307	331	354	378	402	425	449	8	20.8
1.29		472	496	520	543	567	590	614	637	661	684	9	23.4
1.30		708	731	754	778	801	825	848	871	895	918	25	
1.31		941	965	988	011	035	058	081	104	128	151	1	2.5
1.32	0.08	174	197	220	243	267	290	313	336	359	382	2	5.0
1.33		405	428	451	474	497	520	543	566	589	612	3	7.5
1.34		635	658	681	704	727	750	773	795	818	841	4	10.0
1.35		864	887	910	932	955	978	001	023	046	069	5	12.5
1.36	0.09	092	114	137	160	182	205	228	250	273	295	6	15.0
1.37		318	341	363	386	408	431	453	476	498	521	7	17.5
1.38		543	566	588	611	633	655	678	700	723	745	8	20.0
1.39		767	790	812	834	857	879	901	923	946	968	9	22.5
1.40		990	012	035	057	079	101	123	145	168	190	24	
1.41	0.10	212	234	256	278	300	322	344	366	389	411	1	2.4
1.42		433	455	477	499	521	542	564	586	608	630	2	4.8
1.43		652	674	696	718	740	761	783	805	827	849	3	7.2
1.44		871	892	914	936	958	979	001	023	045	066	4	9.6
1.45	0.11	088	110	131	153	175	196	218	239	261	283	5	12.0
1.46		304	326	347	369	390	412	433	455	476	498	6	14.4
1.47		520	541	562	584	605	627	648	669	691	712	7	16.8
1.48		734	755	776	798	819	840	862	883	904	925	8	19.2
1.49		947	968	989	010	032	053	074	095	116	138	9	21.6

Table 8.7 (continued)

r		0	1	2	3	4	5	6	7	8	9	Proportional Parts	
1.50	0.12	159	180	201	222	243	264	286	307	328	349	23	
1.51		370	391	412	433	454	475	496	517	538	559	1	2.3
1.52		580	601	622	643	664	685	706	726	747	768	2	4.6
1.53		789	810	831	852	872	893	914	935	956	977	3	6.9
1.54		997	018	039	060	080	101	122	142	163	184	4	9.2
1.55	0.13	205	225	246	266	287	308	328	349	370	390	5	11.5
1.56		411	431	452	472	493	513	534	554	575	595	6	13.8
1.57		616	636	657	677	698	718	739	759	780	800	7	16.1
1.58		820	841	861	881	902	922	942	963	983	003	8	18.4
1.59	0.14	024	044	064	085	105	125	145	165	186	206	9	20.7
1.60		226	246	267	287	307	327	347	367	387	408	22	
1.61		428	448	468	488	508	528	548	568	588	608	1	2.2
1.62		628	648	668	688	708	728	748	768	788	808	2	4.4
1.63		828	848	868	888	908	928	948	968	988	007	3	6.6
1.64	0.15	027	047	067	087	107	126	146	166	186	206	4	8.8
1.65		225	245	265	284	304	324	344	363	383	403	5	11.0
1.66		423	442	462	481	501	521	540	560	580	599	6	13.2
1.67		619	638	658	678	697	717	736	756	775	795	7	15.4
1.68		814	834	853	873	892	912	931	951	970	990	8	17.6
1.69	0.16	009	028	048	067	087	106	125	145	164	184	9	19.8
1.70		203	222	242	261	280	299	319	338	357	377	21	
1.71		396	415	434	454	473	492	511	531	550	569	1	2.1
1.72		588	607	626	646	665	684	703	722	741	760	2	4.2
1.73		780	799	818	837	856	875	894	913	932	951	3	6.3
1.74		970	989	008	027	046	065	084	103	122	141	4	8.4
1.75	0.17	160	179	198	217	236	255	274	292	311	330	5	10.5
1.76		349	368	387	406	425	443	462	481	500	519	6	12.6
1.77		538	556	575	594	613	631	650	669	688	706	7	14.7
1.78		725	744	762	781	800	818	837	856	874	893	8	16.8
1.79		912	930	949	968	986	005	023	042	061	079	9	18.9
1.80	0.18	098	116	135	153	172	191	209	228	246	265	20	
1.81		283	302	320	339	357	376	394	412	421	449	1	2.0
1.82		468	486	505	523	541	560	578	596	615	630	2	4.0
1.83		652	670	688	707	725	743	762	780	798	816	3	6.0
1.84		835	853	871	890	908	926	944	962	981	999	4	8.0
1.85	0.19	017	035	054	072	090	108	126	144	163	181	5	10.0
1.86		199	217	235	253	271	289	308	326	344	362	6	12.0
1.87		380	398	416	434	452	470	488	506	524	542	7	14.0
1.88		560	578	596	614	632	650	668	686	704	722	8	16.0
1.89		740	758	776	794	811	829	847	865	883	901	9	18.0
1.90		919	937	954	972	990	008	026	044	061	079	19	
1.91		097	115	133	150	168	186	204	221	239	257	1	1.9
1.92		275	292	310	328	345	363	381	399	416	434	2	3.8
1.93		452	469	487	504	522	540	557	575	593	610	3	5.7
1.94		628	645	663	681	698	716	733	751	768	786	4	7.6
1.95		804	821	839	856	874	891	909	926	944	961	5	9.5
1.96		979	996	013	031	048	066	083	101	118	135	6	11.4
1.97	0.20	153	170	188	205	222	240	257	275	292	309	7	13.3
1.98		327	344	361	079	396	413	431	448	465	482	8	15.2
1.99		500	517	534	552	569	586	603	620	638	655	9	17.1
2.00	0.21	672	689	707	724	741	758	775	792	810	827		
2.01		844	861	878	895	913	930	947	964	981	998		
2.02	0.22	015	032	049	066	084	101	118	135	152	169	18	
2.03		186	203	220	237	254	271	288	305	322	339	1	1.8
2.04		356	373	390	407	424	441	458	474	491	508	2	3.6
2.05		525	542	559	576	593	610	627	644	660	677	3	5.4
2.06		694	711	728	745	762	778	795	812	829	846	4	7.2
2.07		863	879	896	913	930	946	953	980	997	013	5	9.0

Table 8.7 (continued)

r		0	1	2	3	4	5	6	7	8	9	Proportional Parts	
2.08	0.23	030	047	064	080	097	114	130	147	164	181	6	10.8
2.09		197	214	231	247	264	281	297	314	331	347	7	12.6
												8	14.4
2.10		364	380	397	414	430	447	463	480	497	513	9	16.2
2.11		530	546	563	579	596	613	629	646	662	679		
2.12		695	712	728	745	761	778	794	811	827	844		
2.13		860	877	893	909	926	942	959	975	992	008		
2.14	0.24	024	041	057	074	090	106	123	139	155	172	17	
2.15		188	204	221	237	253	270	286	302	319	335	1	1.7
2.16		351	368	384	400	416	433	449	465	481	498	2	3.4
2.17		514	530	546	563	579	595	511	627	644	660	3	5.1
2.18		676	692	708	724	741	757	773	789	805	821	4	6.8
2.19		838	854	870	886	902	918	934	950	966	983	5	8.5
												6	10.2
2.20		999	015	031	047	063	079	095	111	127	143	7	11.9
2.21	0.25	159	175	191	207	223	239	255	271	287	303	8	13.6
2.22		319	335	351	367	383	399	415	431	447	463	9	15.3
2.23		479	495	511	526	542	558	574	590	606	622		
2.24		638	654	669	685	701	717	733	749	765	780		
2.25		796	812	828	844	859	875	891	907	923	938		
2.26		954	970	986	001	017	033	049	064	080	096	16	
2.27	0.26	112	127	143	159	175	190	206	222	237	253	1	1.6
2.28		269	284	300	316	331	347	363	378	394	409	2	3.2
2.29		425	441	456	472	488	503	519	534	550	566	3	4.8
												4	6.4
2.30		581	597	612	628	643	659	675	690	706	721	5	8.0
2.31		737	752	768	783	799	814	830	845	861	876	6	9.6
2.32		892	907	923	938	954	969	984	000	015	031	7	11.2
2.33	0.27	046	062	077	092	108	123	139	154	169	185	8	12.8
2.34		200	216	231	246	262	277	292	308	323	338	9	14.4
2.35		354	369	284	400	415	430	446	461	476	492		
2.36		507	522	538	553	568	583	599	614	629	644		
2.37		660	675	690	705	721	736	751	766	781	797		
2.38		812	827	842	857	873	888	903	918	933	948	15	
2.39		964	979	994	009	024	039	054	070	085	100	1	1.5
												2	3.0
2.40	0.28	115	130	145	160	175	190	205	220	236	251	3	4.5
2.41		266	281	296	311	326	341	356	371	386	401	4	6.0
2.42		416	431	446	461	476	491	506	521	536	551	5	7.5
2.43		566	581	596	611	626	641	656	671	686	701	6	9.0
2.44		716	730	745	760	775	790	805	820	835	850	7	10.5
2.45		865	879	894	909	924	939	954	969	984	993	8	12.0
2.46	0.29	013	028	043	058	073	087	102	117	132	147	9	13.0
2.47		162	176	191	206	221	235	250	265	280	295		
2.48		309	324	339	353	368	383	398	412	427	442		
2.49		457	471	486	501	515	530	545	559	574	589		
2.50	0.29	604	618	633	647	662	677	691	706	721	735		
2.51		750	765	779	794	808	823	838	852	867	881		
2.52		896	911	925	940	954	969	984	998	013	027		
2.53	0.30	042	056	071	085	100	114	129	144	158	173		
2.54		187	202	216	231	245	260	274	289	303	318		
2.55		332	346	361	375	390	404	419	433	448	462		
2.56		476	491	505	520	534	548	563	577	592	606		
2.57		620	635	649	663	678	692	707	721	735	750		
2.58		764	778	793	807	821	836	850	864	879	893		
2.59		907	921	936	950	964	979	993	007	021	036	14	
												1	1.4
2.60	0.31	050	064	079	093	107	121	136	150	164	178	2	2.8

Table 8.7 (continued)

r		0	1	2	3	4	5	6	7	8	9	Proportional Parts	
2.61		193	207	221	235	249	264	278	292	306	320	3	4.2
2.62		335	349	363	377	391	405	420	434	448	462	4	5.6
2.63		476	490	505	519	533	547	561	575	589	603	5	7.0
2.64		618	632	646	660	674	688	702	716	730	744	6	8.4
2.65		759	773	787	801	815	829	843	857	871	855	7	9.5
2.66		899	913	927	941	955	969	983	997	011	025	8	11.2
2.67	0.32	039	053	067	081	095	109	123	137	151	165	9	12.6
2.68		179	193	207	221	235	249	262	278	290	304		
2.69		318	332	346	360	374	388	402	416	429	443		
2.70		457	471	485	499	513	527	540	554	568	582	13	
2.71		596	610	624	637	651	665	679	693	707	720	1	1.3
2.72		734	748	762	776	789	803	817	831	845	858	2	2.6
2.73		872	886	900	913	927	941	955	968	982	996	3	3.9
2.74	.033	010	023	037	051	065	078	092	106	119	133	4	5.2
2.75		147	161	174	188	202	215	229	243	256	270	5	6.5
2.76		284	297	311	325	338	352	366	379	393	407	6	7.8
2.77		420	434	448	461	475	488	502	516	529	543	7	8.4
2.78		556	570	584	597	611	624	638	651	665	679	8	10.4
2.79		692	706	719	733	746	760	773	787	801	814	9	11.7
2.80		828	841	855	868	882	895	909	922	936	949		
2.81		963	976	990	003	017	030	044	057	070	084		
2.82	0.34	097	111	124	138	151	165	178	191	205	218	12	
2.83		232	245	259	272	285	299	312	326	339	352	1	1.2
2.84		366	379	393	406	419	433	446	459	473	486	2	2.4
2.85		500	513	526	540	553	566	580	593	606	620	3	3.6
2.86		633	645	660	673	686	700	713	726	739	753	4	4.8
2.87		766	779	793	806	819	832	846	859	872	886	5	6.0
2.88		899	912	925	939	952	965	978	991	005	018	6	7.2
2.89	0.35	031	044	058	071	084	097	110	124	137	150	7	8.4
												8	9.6
2.90		163	176	190	203	216	229	242	255	269	282	9	16.8
2.91		295	308	321	334	347	361	374	387	400	413		
2.92		426	439	452	466	479	492	505	518	531	544		
2.93		557	570	584	597	610	623	636	649	662	675		
2.94		688	701	714	727	740	753	767	780	793	806		
2.95		819	832	845	858	871	884	897	910	923	936		
2.96		949	962	975	988	001	014	027	040	053	066		
2.97	0.36	079	092	105	118	131	144	157	169	182	195		
2.98		208	221	234	247	260	273	286	299	312	324		
2.99		337	350	363	376	389	402	415	428	440	453		

r	0	1	2	3	4	5	6	7	8	9
3.0	0.3647	0.3659	0.3672	0.3685	0.3698	0.3711	0.3723	0.3736	0.3749	0.3761
3.1	0.3774	0.3786	0.3799	0.3811	0.3824	0.3836	0.3849	0.3861	0.3874	0.3886
3.2	0.3898	0.3911	0.3923	0.3935	0.3947	0.3959	0.3971	0.3984	0.3996	0.4008
3.3	0.4020	0.4032	0.4044	0.4056	0.4068	0.4080	0.4091	0.4103	0.4115	0.4127
3.4	0.4139	0.4150	0.4162	0.4174	0.4186	0.4197	0.4209	0.4220	0.4232	0.4244
3.5	0.4255	0.4267	0.4278	0.4290	0.4301	0.4313	0.4324	0.4335	0.4347	0.4358
3.6	0.4369	0.4380	0.4392	0.4403	0.4414	0.4425	0.4437	0.4448	0.4459	0.4470
3.7	0.4481	0.4492	0.4503	0.4514	0.4525	0.4536	0.4547	0.4558	0.4569	0.4580
3.8	0.4591	0.4602	0.4612	0.4623	0.4634	0.4645	0.4656	0.4666	0.4677	0.4688
3.9	0.4698	0.4709	0.4720	0.4730	0.4741	0.4752	0.4762	0.4773	0.4783	0.4794
4.0	0.4804	0.4815	0.4825	0.4835	0.4846	0.4856	0.4867	0.4877	0.4887	0.4898

For nozzles, $r = p_1/p_2$. For compressors and exhausters, $r = p_2 p_1$.

*Taken torn Moss and Smith, Engineering Computations for Air and Gases, Trans. ASME, vol.52, Paper APM-52-8.

For a multi-stage group with intercooling between N stage groups, and with the same restrictions specified for Eq. (8.15), the theoretical power for compression is:

$$P_t = \frac{Nwc_pT_{1t}}{42.42}\left[\left(\frac{p_2}{p_1}\right)^{(k-1)/Nk} - 1\right] \qquad (8.17)$$

Table 8.8 Values for *Y*'

	D2/D1														
	k = 1.40					k = 1.35					k = 1.30				
P_2/P_1	0	0.2	0.3	0.4	0.5	0	0.2	0.3	0.4	0.5	0	0.2	0.3	0.4	0.5
1.00	1.000	1.001	1.004	1.013	1.033	1.000	1.001	1.004	1.013	1.033	1.000	1.001	1.004	1.013	1.033
0.99	0.995	0.995	0.999	1.007	1.027	0.994	0.995	0.999	1.007	1.027	0.994	0.995	0.998	1.007	1.026
0.98	0.989	0.990	0.993	1.002	1.021	0.989	0.990	0.993	1.001	1.020	0.988	0.989	0.992	1.001	1.020
0.97	0.984	0.985	0.988	0.996	1.015	0.983	0.984	0.987	0.995	1.014	0.983	0.983	0.986	0.995	1.013
0.96	0.978	0.979	0.982	0.990	1.009	0.978	0.978	0.981	0.990	1.008	0.977	0.977	0.980	0.989	1.007
0.95	0.973	0.974	0.977	0.985	1.002	0.972	0.973	0.976	0.984	1.001	0.971	0.972	0.974	0.982	1.000
0.94	0.967	0.968	0.971	0.979	0.996	0.966	0.967	0.970	0.978	0.995	0.965	0.966	0.968	0.976	0.993
0.93	0.962	0.963	0.965	0.973	0.990	0.961	0.961	0.964	0.972	0.989	0.959	0.960	0.962	0.970	0.987
0.92	0.956	0.957	0.960	0.967	0.984	0.955	0.955	0.958	0.966	0.982	0.953	0.954	0.956	0.964	0.980
0.91	0.951	0.951	0.954	0.961	0.978	0.949	0.950	0.952	0.960	0.976	0.947	0.948	0.950	0.957	0.973
0.90	0.945	0.946	0.948	0.956	0.971	0.943	0.944	0.946	0.953	0.969	0.941	0.942	0.944	0.951	0.966
0.89	0.939	0.940	0.943	0.950	0.965	0.937	0.938	0.940	0.947	0.963	0.935	0.935	0.938	0.945	0.959
0.88	0.934	0.934	0.937	0.944	0.959	0.931	0.932	0.934	0.941	0.956	0.929	0.929	0.932	0.938	0.953
0.87	0.928	0.928	0.931	0.938	0.953	0.925	0.926	0.926	0.935	0.950	0.922	0.923	0.926	0.932	0.946
0.86	0.922	0.923	0.925	0.932	0.946	0.919	0.920	0.922	0.929	0.943	0.916	0.917	0.919	0.926	0.939
0.85	0.916	0.917	0.919	0.926	0.940	0.913	0.914	0.916	0.923	0.936	0.910	0.911	0.913	0.923	0.932
0.84	0.910	0.911	0.913	0.920	0.933	0.907	0.908	0.910	0.916	0.930	0.904	0.904	0.907	0.919	0.925
0.83	0.904	0.905	0.907	0.913	0.927	0.901	0.902	0.904	0.910	0.923	0.897	0.898	0.900	0.916	0.918
0.82	0.898	0.899	0.901	0.907	0.920	0.895	0.895	0.898	0.904	0.917	0.891	0.891	0.894	0.900	0.911
0.81	0.892	0.893	0.895	0.901	0.914	0.889	0.889	0.891	0.897	0.910	0.885	0.885	0.887	0.893	0.904
0.80	0.886	0.887	0.889	0.895	0.907	0.883	0.883	0.885	0.891	0.903	0.878	0.879	0.880	0.886	0.897
0.79	0.880	0.881	0.883	0.889	0.901	0.876	0.877	0.879	0.884	0.896	0.872	0.872	0.874	0.880	0.890
0.78	0.874	0.875	0.877	0.882	0.894	0.870	0.870	0.872	0.878	0.889	0.865	0.865	0.868	0.873	0.883
0.77	0.868	0.869	0.871	0.876	0.887	0.864	0.864	0.866	0.871	0.882	0.859	0.859	0.861	0.866	0.876
0.76	0.862	0.862	0.864	0.869	0.881	0.857	0.858	0.859	0.865	0.876	0.852	0.852	0.854	0.859	0.869
0.75	0.856	0.856	0.858	0.863	0.874	0.851	0.851	0.853	0.858	0.869	0.845	0.846	0.848	0.852	0.862
0.74	0.849	0.850	0.852	0.857	0.867	1.844	0.845	0.846	0.851	0.862	0.839	0.839	0.841	0.845	08.55
0.73	0.843	0.844	0.845	0.850	0.860	0.838	0.838	0.840	0.845	0.855	0.832	0.832	0.834	0.838	0.848
0.72	0.837	0.837	0.839	0.844	0.854	0.831	0.831	0.833	0.838	0.848	0.825	0.825	0.827	0.831	0.841
0.71	0.830	0.831	0.832	0.837	0.847	0.825	0.825	0.827	0.831	0.840	0.818	0.819	0.820	0.824	0.834
0.70	0.824	0.824	0.826	0.830	0.840	0.818	0.818	0.820	0.824	0.833	0.811	0.812	0.813	0.817	0.826

If the velocity of approach is zero (as with a nozzle taking in air from the outside), D_1 is infinite, and D_2/D_1 is zero.

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Shaft Power

Where compressors are driven by electric motors, shaft horsepower may be computed from measured values of the electrical input. For induction motors of the squirrel-cage type, the shaft horsepower shall be:

$$P_s = \frac{\text{electrical input x efficiency}}{0.746}$$

in which the efficiency has been determined by test.

For synchronous motors, shaft horsepower shall be computed as:

$$P_s = \frac{\text{electrical input - sum of the losses in kW}}{0.746}$$

where the losses are based on the prevailing voltage, armature current, and field current. The losses shall he established by test measurements of armature resistance, open-circuit core losses, short-circuit core losses, and the friction and windage losses. The complete loss shall be the sum of I^2R + core loss + load loss + excitation + friction and windage. (See PTC 10 for the definition of electrical input.)

The shaft horsepower output of a wound-rotor type of induction motor may be calculated in the same manner as outlined for the squirrel-cage type when the secondary winding is short-circuited and where measured efficiency data for this condition of operation are available. This Code does not provide for shaft horsepower determination with a wound-rotor type of motor when it is operated with external resistance in the secondary circuit.

Computations of shaft horsepower by the motor input method shall not be acceptable in the case of the induction-type motor when the output is less than one-half of the motor rating.

Compressor Efficiency

The compressor efficiency shall be computed as the ratio of the theoretical power to the shaft power.

$$\eta = \frac{P_t}{P_s}$$

Adjustment of Results to Specified Conditions

Tests that are made in accordance with this Code and that have deviations between test and specified conditions within the limits prescribed in column (2), Table 8.5, may be adjusted to the operating conditions specified by the following equations.

Adjustment of Capacity

When the speed at test conditions deviates from the specified speed by not more than 5 percent, capacity shall he adjusted by the equation

$$Q_{mc} = Q_m \frac{N_c}{N_m}$$

where:

N = speed, rpm
Q = capacity, cfm
subscript *m* refers to measured quantity
subscript *c* refers to specified quantity
subscript *mc* refers to adjusted value

Adjustment of Pressure Ratio

The adjustment of pressure ratio shall be in accordance with the relation

$$\left(r_{mc}{}^{(k_c-1)/k_c}-1\right)=\left(\frac{N_c}{N_m}\right)^2\frac{\mathrm{MW}_c k_m(k_c-1)T_{1m}}{\mathrm{MW}_m k_c(k_m-1)T_{1c}}\left(r_{mc}{}^{(k_m-1)/k_m}-1\right)$$

For tests with air or with other gases in which the values of k and MW are the same for both test and contract conditions, and in which the value of k lies between 1.39 and 1.40, the following relation may be used to simplify the computations:

$$X_{mc} = X_m\left(\frac{N_c}{N_m}\right)^2\frac{T_{1m}G_c}{T_{1c}G_m} \qquad (8.18)$$

where:

X = $r^{0.283} - 1$ (see Table 8.7)
r = p_2/p_1
T_1 = inlet temperature, °F abs
MW = molecular weight
G = specific gravity

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Having found the value of X_{mc} by Eq. (8.18), the corresponding value of r_{mc} is found from Table 8.7. The adjusted pressure is:

$$p_{2mc} = r_{mc}\, p_{1c}$$

Adjustment of Power

The shaft horsepower shall be adjusted by the equation:

$$P_{mc} = P_m \left(\frac{N_c}{N_m} \right)^2 \frac{T_{1m}\, p_{1c}\, M_c}{T_{1c}\, p_{1m}\, M_m}$$

For tests with air in which values of X have already been determined, the computations may be facilitated by the equivalent equation:

$$P_{mc} = P_m \frac{p_{1c}\, N_c\, X_{mc}}{p_{1m}\, N_m\, X_m}$$

When the compressor is driven by an electric motor, the adjusted kilowatt input shall be 0.746 P_{mc}/e, where e is the measured motor efficiency at the power output of P_{mc} . If the motor is of the induction type, the speed value of N_c used in the pressure and power correction formula shall be the actual speed at the power output P_{mc} as determined by plotting slip against kilowatt input.

If the compressor is driven by a steam turbine, the steam consumption shall be corrected in accordance with the Power Test Code for Steam Turbines. This Code prescribes limited correction for deviation in initial steam pressure, superheat, exhaust pressure, and load. In view of their complexity, the values for these corrections are preferably included as a part of the contract.

Examples

A complete presentation of observed data, computations, and adjustments is illustrated by Examples 8.3 and 8.4 for air compressors. In each case the test consists of two points which bracket the “guarantee” or specified point within the limits of capacity and speed given in Table 8.5. The adjusted results are compared with the specified pressure and power in the form of curves (Figures 8.23 and 8.24).

Example 8.3 illustrates the test of a multi-stage compressor driven by a direct-connected steam turbine of the straight condensing type. The test setup is arranged as shown in Figure 8.16, without pipe on the inlet, so that the inlet total pressure is measured by the barometer. The air output is measured by the nozzle setup in accordance with arrangement A of Figure 8.22. Steam is condensed and measured either by a pair of weigh tanks or by calibrated volume tanks. In accordance with the requirements of the Power Test Code for Steam Turbines, the correction values for deviation in operating conditions were established by an agreement in which the correction for superheat, initial steam pressure, and exhaust pressure was to be based on the ratio of available enthalpy, and in which the steam flow was to be directly proportioned to the load.

Table 8.9 shows the capacities, pressures, speeds, and steam flows for the two test points, "as run" and "after adjustment." The last column shows corresponding values for the specified point. The exact values of discharge pressure and steam flow to be compared with the guarantee are given in the curves of Figure 8.23.

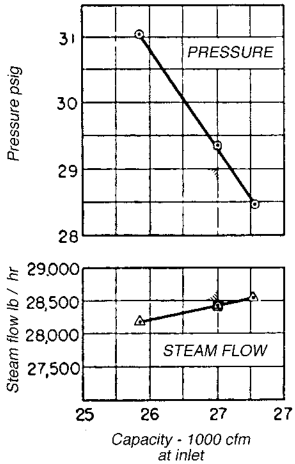

Figure 8.23 Values of discharge pressure and steam flow.

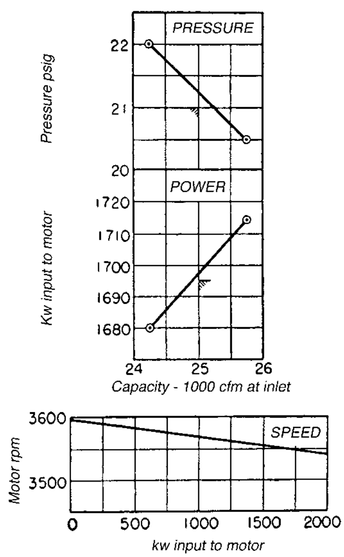

Figure 8.24 Corrected test values of pressure and power.

Example 8.3

Table 8.9 Summary of Computed Results

	Unit	As run (1)	As run (2)	Adjusted to specified conditions (1)	Adjusted to specified conditions (2)	Specified
Speed	rpm	4,763	4,746	4,700	4,700	4,700
Barometer	psia	14.50	14.47	14.7	14.7	14.7
Gas composition	....	Air	Air	Air	Air	Air
Specific Gravity of gas (dry air = 1)	....	0.9855	0.9853	0.9962	0.9962	0.9962
Inlet conditions:						
Pressure	psia	14.50	14.47	14.7	14.7	14.7
Temperature	deg F	90.0	89.4	70	70	70
Relative humidity	per cent	80.0	82	40	40	40
Capacity	cfm	26,340	27,970	25,990	27,700	27,000
Discharge pressure	psig	29,705	26,936	31.05	28.40	29.00
Turbine steam conditions:						
Steam pressure of throttle	psig	398.3	401.9	400	400	400
Total steam temperature	deg F	556.2	557.4	548	548	548
Exhaust pressure, lb	In. Hg	3.259	2.986	3.0	3.0	3.0
Total steam flow, including seal leakage	lb per hr	27,726	27,348	28,180	28,470	28,500

Average of Observed Readings

	Unit	Symbol	Test 1	Test 2
Speed	rpm	N_m	4,763	4,746
Inlet temperature of compressor	deg F	t_{1m}	90.0	89.4
Barometric pressure, corrected to 32F	in. Hg	....	29.52	29.461
Barometric pressure	psia	....	14.50	14.470
Pressure at compressor inlet flange	psia	p_{1m}	14.500	14.470
Psychrometer reading at inlet:				
Dry bulb	deg F	....	89.6	89.2
Wet bulb	deg F	....	84.3	84.5
Static pressure at compressor discharge	in. Hg	....	60.50	54.779
Temperature at compressor discharge	deg F	t_{2m}	357	344
Air measuring nozzle differential pressure measured by impact tube	in. H_20	....	26.22	29.34
Nozzle upstream temperature	deg F	t_n	221.3	215.8
Nozzle-throat diameter	in.	D_n	16.000	16.000
Room temperature at gage board	deg F	....	88.7	87.9
Compressor-discharge (circular section) diameter	in.	....	20	20
Compressor-inlet (circular section) diameter	in.	....	30	30
Steam pressure at turbine throttle valve, corrected for gage error and head of water in tubing	psig	....	398.3	401.9
Steam temperature at turbine throttle valve	deg F	....	556.2	557.4
Exhaust steam temperature	deg F	....	118.0	114.8
Vacuum at turbine exhaust flange	in. Hg	....	26.409	26.617
Vacuum corrected to 32F	in. Hg	....	26.264	26.475
Absolute pressure at turbine exhaust flange	in. Hg	....	3.259	2.986
Steam flow, condensate measurement	lb per hr	....	27,463	27,089
Turbine steam seal leakage	lb per hr	....	264	259

Table 8.9 continued

Calculation of Results

	Unit	Symbol	Test 1	Test 2
Capacity (by equation 8.13)				
Density of water at room temperature	lb/cu in	....	0.035952	0.035958
Differential pressure across nozzle	psig	Δ_p	0.9425	1.0549
Pressure ratio across nozzle [(barometer + diff. press.) ÷ barometer], p_{ni}/p_{n2}	....	r_m	1.0650	1.0729
X value from Table 8.7	....	....	0.01798	0.02011
Nozzle diameter squared	sq in.	D_n^2	256.000	256.000
Nozzle coefficient of discharge	....	C	0.995	0.995
Nozzle upstream temperature	deg F abs	T_{nm}	681.00	675.5
Temperature to which flow is referred	deg F abs	T_{lm}	549.7	549.1
Pressure to which flow is referred	psig	p_{lm}	14.500	14.470
$\text{Cap.} = \dfrac{36.0\text{x}0\times 0.995\times 256\times T_{lm}\sqrt{X(X+1)}}{\sqrt{T_n G}}$	cfm	Q_m	26,340	27,970
Inlet specific gravity:				
Relative humidity from psychrometer reading and psychrometric tables	per cent	....	80	82
Inlet temperature	deg F	t_{lm}	90	89.4
Specific gravity at inlet conditions from Table 8.6	....	G_{lm}	0.9855	0.9853
Total discharge pressure:				
Static pressure corrected to 32F	in. Hg	....	60.175	54.486
Static pressure (in Hg at 32F x 0.49115)	psig	....	29.555	26.716
Velocity-pressure calculations:				
Specific weight of air in discharge pipe:				
$(1)\ 0.0717\times\dfrac{29.555+14.500}{14.500}\times\dfrac{459.6+90.0}{459.6+357}$	lb per cu ft	....	0.1434	
$(2)\ 0.07009\times\dfrac{26.716+14.470}{14.47}\times\dfrac{459.6+89.4}{459.6+344}$	lb per cu ft	....	....	0.1368
Approximate flow rate in discharge pipe:				
$(1)\ \dfrac{26{,}340}{60}\times\dfrac{14.50}{44.1}\times\dfrac{816.7}{549.7}$	cfs	....	214.5	
$(2)\ \dfrac{27{,}810}{60}\times\dfrac{14.47}{41.3}\times\dfrac{803.7}{549.1}$	cfs	....	....	238.8
Area of discharge pipe	sq ft	....	2.182	2.182
Velocity in discharge pipe = cfs ÷ area	fps	V_2	98.30	109.4
Velocity pressure:				
$(1)\ 0.1434\times\dfrac{(98.30)^2}{2\times 32.174}\times\dfrac{1}{144}$	psi	....	0.150	
$(2)\ 0.1363\times\dfrac{(109.4)^2}{2\times 32.174}\times\dfrac{1}{144}$	psi	....	....	0.176
Total discharge pressure	psig	....	29.703	26.892

Table 8.9 continued

	Unit	Symbol	Test 1	Test 2
Capacity adjustment:				
(1) $26{,}340 \times \dfrac{4{,}700}{4{,}763}$	cfm	Q_{mc}	25,990	
(2) $27{,}970 \times \dfrac{4{,}700}{4{,}746}$	cfm	Q_{mc}	. . . .	27,700
Steam-flow adjustment:				
Enthalpy of steam at turbine throttle, as run	Btu per lb	. . . .	1279.7	1280.0
Enthalpy of exhaust steam by isentropic expansion to exhaust pressure	Btu per lb	. . . .	894.1	889.4
Enthalpy of steam at turbine throttle, at specified conditions	Btu per lb	. . . .	1274.4	1274.4
Enthalpy of exhaust steam by isentropic expansion to specified exhaust pressure	Btu per lb	. . . .	886.6	886.6
Theoretical enthalpy drop, as measured	Btu per lb	. . . .	385.6	390.7
Theoretical enthalpy drop at specified conditions	Btu per lb	. . . .	387.8	387.8
Steam flow adjusted to specified compressor intake conditions, speed and turbine steam conditions:				
(1) $27{,}462 \times \dfrac{14.7}{14.5} \times \dfrac{0.37886}{0.3709} \times \dfrac{4{,}700}{4{,}763} \times \dfrac{385.6}{387.8}$	lb per hr	. . . .	27.917	
(2) $27{,}089 \times \dfrac{14.7}{14.47} \times \dfrac{0.35581}{0.34654} \times \dfrac{4{,}700}{4{,}746} \times \dfrac{390.7}{387.8}$	lb per hr	. . . .	. . . .	28,206
Turbine steam seal leakage	lb per hr	. . . .	264	259
Total steam flow, adjusted to specified conditions	lb per hr	. . . .	28,180	28,470

Adjustment of Test Results to Specified Inlet Conditions

	Unit	Symbol	Test 1	Test 2
Pressure adjustment:				
Pressure ratio as run:				
(1) (29.705 + 14.500)/14.500	. . . .	r_m	3.0486	
(2) (26.892 + 14.470)/14.470	. . . .	r_m	. . . .	2.8585
X_m value from Table 8.7	. . . .	X_m	0.3709	0.34613
X_{mc}:				
(1) $= 0.3709 \left(\dfrac{549.7}{529.7}\right)\left(\dfrac{0.9962}{0.9855}\right)\left(\dfrac{4{,}700}{4{,}763}\right)^2$	. . . .	X_{mc}	0.37886	
(2) $= 0.34613 \left(\dfrac{549.1}{529.7}\right)\left(\dfrac{0.9962}{0.9853}\right)\left(\dfrac{4{,}700}{4{,}746}\right)^2$	. . . .	X_{mc}	. . . .	0.35581
Pressure ratio for X_{mc} from Table 8.7	. . . .	r_{mc}	3.112	2.9318
Adjusted discharge pressure = $r_{mc} \times 14.696$	psia	P_{2mc}	45.746	43.097
Adjusted discharge pressure	psig	. . . .	31.05	28.40

Example 8.4 illustrates the test of a multi-stage air compressor driven by a direct-connected induction motor. The test setup is given in Figure 8.18 except for the intake pipe. In this case no pipe is used, and the inlet total pressure is measured by the barometer.

The nozzle setup is in accordance with arrangement A of Figure 8.22. Motor output is based on the measured input and efficiency curves. The two test points, which bracket the specified capacity within the limits of Table 8.5, are corrected to the actual motor speed and not the specified speed, which is usually a nominal figure or that stamped on the motor name plate. The actual speed-load curve is established from the test measurements. The corrected test values of pressure and power to be compared with the specified values are shown in the curves of Figure 8.24.

Testing with Substitute Gases

The foregoing discussion of testing centrifugal compressors assumes that the test gas will be the same as the gas for which the machine is designed, or at least differ only to a minor degree. In factory tests, however, such duplication of the design gas is frequently impractical. State or local safety laws, insurance limitations or labor agreements, for example, often prevent the introduction of combustible or toxic gases onto the test floor of machinery manufacturers.

In such circumstances, tests must be conducted with substitute gases. Fortunately, the procedures for correlating the performance so obtained are well known, and quite accurate tests can be conducted despite this handicap. The essential step is to establish a test speed that will recreate dynamic similarity between test and design gases, within the compressor.

The test speed on the substitute gas is generally referred to as "equivalent speed" – it produces equivalent aerodynamics in the machines. Two factors must be considered in selecting this speed: (1) Mach number and (2) density ratio.

The equivalent test speed for a substitute gas at which the design Mach numbers obtain is determined from the ratio of the sonic velocity in the substitute gas to that in the design gas. It is usually sufficient to compute this ratio only at inlet conditions. The relation is:

$$N_{eq} = N_d \frac{\sqrt{(kgZRT_1)_s}}{\sqrt{(kgZRT_1)_d}}$$

The density ratio is the ratio of gas density at a given point in the compression to the density at inlet conditions. If equivalence of the density ratio (test versus design) is maintained throughout the compression cycle, the gas volume will be consistent with the intended use of the machine.

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Example 8.4

Table 8.10 Summary of Computed Results

	Unit	As run (1)	As run (2)	Adjusted to specified conditions (1)	Adjusted to specified conditions (2)	Specified
Speed	rpm	3,554	3,551	3,551	3,549	3,550
Barometer	psia	14.580	14.620	14.7	14.7	14.7
Gas composition	....	Air	Air	Air	Air	Air
Specific Gravity of gas (dry air = 1)	....	0.9878	0.9881	0.9962	0.9962	0.9962
Inlet conditions:						
Pressure	psia	14.580	14.620	14.7	14.7	14.7
Temperature	deg F	87.4	88.5	70	70	70
Relative humidity	per cent	73	69	40	40	40
Capacity	cfm	24,363	25,869	24,340	25,850	25,000
Discharge pressure	psig	20.695	19.303	21.99	20.50	21.00
Electrical conditions:						
Line voltage	volts	2,215	2,208	....	....	2,200
Line current	amp	445.2	454.9			
Power input to motor	kw	1,605.5	1,638.7	1,680	1,715	1,695

Averages of Observed Readings

		Symbol	Test 1	Test 2
Speed	rpm	N_m	3,554	3,551
Inlet temperature of compressor	deg F	t_{1m}	87.4	88.5
Barometric pressure, corrected to 32F	in. Hg	....	29.685	29.767
Barometric pressure	psia	....	14.580	14.620
Inlet Pressure at compressor inlet flange	psia	p_{1m}	14.580	14.620
Psychrometer reading at inlet:				
Dry bulb	deg F	....	87.2	87.9
Wet bulb	deg F	....	80.0	79.4
Static pressure at compressor discharge	in. Hg	....	42.049	39.150
Temperature at compressor discharge	deg F	t_{2m}	249.6	287.8
Air measuring nozzle differential pressure measured by impact tube	in. H_2O	....	22.661	25.527
Nozzle upstream temperature	deg F	t_n	218.4	217.9
Nozzle-throat diameter	in.	D_n	16.000	16.000
Room temperature at gage board	deg F	....	86.7	87.2
Compressor-inlet (circular section) diameter	in.	....	30	30
Compressor-discharge (circular section) diameter	in.	....	20	20
Power input to motor	kw	P_m	1,605.5	1,638.7
Line voltage at motor terminal (average 3 phases)	volts	....	2,215	2,208
Line current (average 3 phases)	amp	....	445.2	454.9

Calculation of Results

	Unit	Symbol	Test 1	Test 2
Capacity (by equation 8.13)				
Density of water at room temperature	lb per cu in	....	0.035965	0.035962
Differential pressure across nozzle	psig	Δ_p	0.8150	0.9180
Pressure ratio across nozzle [(barometer + diff. press.) + barometer]	....	r_n	1.0559	1.0628
X value from Table 8.7	....	....	0.01552	0.01739
Nozzle diameter squared	sq in.	D_n^2	256.0	256.0
Nozzle coefficient of discharge from Table 8.3	....	C	0.995	0.995
Nozzle upstream temperature	deg F abs	T_{nm}	678.1	677.6

Table 8.10 continued

	Unit	Symbol	Test 1	Test 2
Temperature to which flow is referred	deg F abs	T_{lm}	547.1	548.2
Pressure to which flow is referred	psig	p_{lm}	14.580	14.620
$\text{Cap.} = \dfrac{36.0 \times 0.995 \times 256 \times T_{lm}\sqrt{X(X+1)}}{\sqrt{T_n G}}$	cfm	Q_m	24,363	25,869
Inlet specific gravity:				
Relative humidity from psychrometer reading and psychrometric tables	per cent	....	73	69
Inlet temperature	deg F	....	87.4	88.5
Specific gravity at inlet conditions from Table 8.6	....	G_{lm}	0.9878	0.9881
Total discharge pressure:				
Static pressure corrected to 32 F	in. Hg	....	41.828	38.943
Static pressure (in Hg at 32F x 0.49115)	psig	....	20.544	19.127
Velocity-pressure calculations:				
Specific weight of air in discharge pipe:				
$(1)\ 0.07106 \times \dfrac{20.544+14.580}{14.580} \times \dfrac{547.1}{754.3}$	lb per cu ft	....	0.1243	....
$(2)\ 0.07113 \times \dfrac{19.127+14.620}{14.620} \times \dfrac{548.2}{747.5}$	lb per cu ft	....	....	0.1205
Approximate flow rate in discharge pipe:				
$(1)\ \dfrac{24{,}363}{60} \times \dfrac{14.580}{20.544+14.580} \times \dfrac{754.3}{547.1}$	cfs	....	231.3	....
$(2)\ \dfrac{25{,}869}{60} \times \dfrac{14.620}{19.127+14.620} \times \dfrac{747.5}{548.2}$	cfs	....	....	253.5
Area of discharge pipe	sq ft	....	2.182	2.182
Velocity in discharge pipe = cfs ÷ area	fps	V_2	106.5	116.8
Velocity pressure:				
$(1)\ 0.1243 \times \dfrac{(106.5)^2}{2 \times 32.174} \times \dfrac{1}{144}$	psi	....	0.152	....
$(2)\ 0.1205 \times \dfrac{(116.8)^2}{2 \times 32.174} \times \dfrac{1}{144}$	....	....	....	0.177
Total discharge pressure	psig	....	20.695	19.303

Adjustment of Test Results to Specified Inlet Conditions

	Unit	Symbol	Test 1	Test 2
Pressure adjustment:				
Pressure ratio as run:				
(1) (20.695 + 14.580)/14.580	....	r_m	2.4194	
(2) (19.303 + 14.620)/14.620	....	r_m	....	2.3203
X_m value from Table 8.7	....	X_m	0.28407	0.26898
Estimation of probable speed at specified inlet conditions, from motor rpm vs. kw curve. The kw will vary approximately as the inlet specific weight:				

Table 8.10 continued

	Unit	Symbol	Test 1	Test 2
(1) $1{,}605.5 \times \frac{0.07463}{0.07106} = 1686$ kW; corresp. rpm = rpm	....	....	3,551	
(2) $1{,}638.7 \times \frac{0.07470}{0.07113} = 1719$ kW; corresp. rpm = rpm	....	....	....	3,549
X_{mc}:				
(1) $= 0.28407 \frac{547.1}{529.7} \times \frac{0.9962}{0.9878} \times \left(\frac{3{,}551}{3{,}554}\right)^2$	....	X_{mc}	0.29544	
(2) $= 0.26898 \frac{548.2}{529.7} \times \frac{0.9962}{0.9881} \times \left(\frac{3{,}549}{3{,}551}\right)^2$	....	X_{mc}	....	0.28037
Pressure ratio for X_{mc} from Table 8.7	....	r_{mc}	2.4960	2.3949
Adjusted discharge pressure = $r_{mc} \times 14.7$	psia	p_{2mc}	36.691	35.205
Adjusted discharge pressure	psig	....	21.99	20.50
Capacity adjustment:				
(1) $24{,}363 \times \frac{3{,}551}{3{,}554}$	cfm	Q_{mc}	24,340	
(2) $24{,}869 \times \frac{3{,}549}{3{,}551}$	cfm	Q_{mc}	....	25,850
Power adjustment:				
Motor efficiency as run	per cent	....	95.3	95.4
Shp as run = kW × eff ÷ 746	hp	....	2,051.00	2,095.6
Shp adjusted to specified inlet conditions and probable speed:				
(1) $2051.0 \times \frac{14.7}{14.58} \times \frac{0.29540}{0.28407} \times \frac{3{,}551}{3{,}554}$	hp	....	2,148.8	
(2) $2095.6 \times \frac{14.7}{14.62} \times \frac{0.28032}{0.26896} \times \frac{3{,}549}{3{,}551}$	hp	....	....	2,196.0
Motor efficiency (data from motor manufacturer)	per cent	e	95.4	95.5
Power input to motor, adjusted to specified conditions. The speeds corresponding to the values of P_{mc} are sufficiently close to the estimated speeds for specified intake conditions.	kw	P_{mc}	1,680	1,715

The equivalent test speed for a substitute gas at which the design density ratio obtains is determined from the relations for polytropic compression. The computations are made with the temperatures and pressures at the inlet and discharge of the compressor. The relations are:

$$(r_s)^{1/n_s} = \left(\frac{\gamma_2}{\gamma_1}\right)_s = \left(\frac{\gamma_2}{\gamma_1}\right)_d = (r_d)^{1/n_d}, \quad r_s = (r_d)^{n_s/n_d}$$

$$N_{eq} = N_d \sqrt{\frac{\left[ZR^{n/(n-1)}T_2\left(r^{(n-1)/n} - 1\right)\right]_s}{\left[ZR^{n/(n-1)}T_1\left(r^{(n-1)/n} - 1\right)\right]_d}}$$

where:

N_{eq} = equivalent speed
N_d = design speed
r = compressor pressure ratio
γ_1 = inlet density
γ_2 = discharge density
Z = supercompressibility factor
g = acceleration due to gravity
k = ratio of specific heats
R = gas constant = 1544/mo1. wgt.
T_1 = inlet temperature, °R
n = polytropic compression exponent
s = substitute gas, subscript
d = design gas, subscript

For perfect dynamic similarity, both the Mach number and the density ratio with the substitute gas must be exactly the same as the design values, and the equivalent speeds calculated by relations (8.11) and (8.12) must be identical. In practice this is seldom possible and difference of a few percent between the two computed equivalent speeds is accepted.

The most desirable substitute gas for test purposes is, of course, air. For many light gas compressors both the design density ratio and the design Mach numbers can be very nearly obtained when compressing air at an equivalent speed below design speed. For gases heavier than air, the equivalent test speed is above the design speed and unsafe stresses may be encountered. In addition, properties of the heavier gases frequently preclude simulation of both the design density ratio and design Mach numbers at the same or nearly the same equivalent speeds. In these cases, a substitute gas with more suitable properties is selected, such as Freon.

To translate test points to design conditions, the following relations are applied:

$$Q_d = Q_s \frac{N_d}{N_s} \tag{8.22}$$

$$H_d = H_s \left(\frac{N_d}{N_s}\right)^2 \tag{8.23}$$

$$\eta_d = \eta_s \tag{8.24}$$

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where:

Q = inlet volume
H = polytropic head
N_d = design speed
N_{eq} = equivalent test speed
η = polytropic efficiency
d = design gas, subscript
S = substitute gas, subscript

Although similar to the fan laws, these relations are valid only for translating data to the design speed from a proper equivalent speed where dynamic similarity was achieved; that is, the design density ratio and Mach numbers existed in the compressor. In addition, the relations apply only to the polytropic head and efficiency, and are not accurate for isentropic head and efficiency where the ratio of specific heats of the substitute gas differs from that of the design gas.

The pressure level at which compressor tests are run does not enter the dynamic similarity relations, and performance tests may ordinarily be run at a convenient pressure level. Where the test pressure is greatly different from the design pressure, however, some error will appear because of the difference in Reynolds numbers and friction factors within the compressor.

A compressor designed to operate at 1000 psi will show poorer performance at atmospheric pressure than at design pressure because the friction factors at the test conditions are higher than at design conditions. Where it is not practicable to test such compressors at the design pressure level, low pressure test results are corrected by the calculated ratio of the friction losses at test pressure to those at design pressure.

LUBRICATION

For satisfactory performance and freedom from wear, any machine or tool having moving parts with rubbing surfaces depends on adequate and efficient lubrication. Not only must the lubricant itself, whether grease, oil, or other liquid, be carefully chosen to meet the required conditions of service, but adequate means, including lubricators, oil ducts, and feeding mechanisms, must be provided to ensure dependable application wherever lubrication is needed. These are the two necessary requirements for good lubrication, but they are not sufficient without conscientious attention on the part of the operator, who must assume responsibility for maintaining the supply of lubricant, for guarding against contamination, and for adjusting the rate of feed when necessary. Even so-called "fully automatic" systems of lubrication require occasional attention on the part of the operator.

Table 8.11 Cost of Air Leaks

Size of Opening in.	Cu. ft. Air Wasted per Month at 100 Lb. Pressure, Based on an Orifice Coefficient of .65	Cost of Air Wasted per Month, Based on 13.677 cents per 1,000 Cu. ft.*
3/8	6,671,890	$912.52
1/4	2,290,840	$399.50
1/8	740,210	$101.22
1/16	182,272	$ 24.91
1/32	45,508	$ 6.24

*At 22 bhp/100 cfm (16.412 kW/100 cfm) and 5 cents/kWh.

Thus, it will be seen that good lubrication depends on three principal factors:

1. Type of lubricant.
2. Effective application.
3. Attention from the operator.

In the preceding chapters, specific information is given in regard to these items as applied to various types of compressors and compressed-air equipment.

It should be borne in mind that lubricants, and particularly greases and oils manufactured from petroleum, are so complex in nature and vary so widely as to physical properties, that detailed specifications describing any desired grade or quality cannot be written with any assurance that the specification will define exactly what is wanted. Manufacturers of compressed-air machinery limit their specifications as to lubrication requirements to cover only the more important physical characteristics, such as viscosity at one or more temperatures, fire point, pour point, etc. In each case they specify the particular kind of service intended, in what atmosphere the oil must operate, what are the minimum and maximum temperatures it must withstand, and they indicate and provide the means for applying the oil to bearings or rubbing surfaces. However, they must leave to the oil refiner and those who sell the lubricant the responsibility of furnishing an oil suitable in all other respects for the service intended. The necessity of following this practice as a matter of policy is obvious, since many important qualities of the lubricant depend entirely on the origin of the oil, on the processes used for refining it, and on other items over which only the oil supplier has control.

Oils and greases are usually known by their trade names, but as the oil-refining art progresses, improvements or variations in the quality of the oil may result, so that the complex qualities of any particular brand of oil may change from time to time without corresponding change in the brand name. For this and similar reasons, it is against the policy of the Compressed Air and Gas Institute to specify the kind and quality of lubricant required by brand or trade name. The Institute recommends that the user purchase oil and greases only from reputable oil companies and that he require the oil companies to guarantee the quality of their lubricant for the use intended.

LOSS OF AIR PRESSURE IN PIPING DUE TO FRICTION

All these data are based on nonpulsating flow and apply to clean and smooth pipe. The data included in Figure 8.25 and Tables 8.12 to 8.16 are calculated from the following formulas by E.G. Harris:*

**University of Missouri Bulletin*, vol. no. 4.

$$f = \frac{CLq^2}{rd^5}$$

$$C = \frac{0.1025}{d^{0.31}}$$

$$f = \frac{0.1025Lq^2}{rd^{0.31}d^5} = \frac{0.1025Lq^2}{rd^{(0.31+5)}} = \frac{0.1025Lq^2}{rd^{5.31}}$$

where:

f	=	pressure drop, psi
L	=	length of pipe, ft.
q	=	cubic feet of free air per second
r	=	ratio of compression (from free air) at entrance of pipe
d	=	actual internal diameter of pipe, in.
C	=	experimental coefficient

Figure 8.28 gives directly the pressure drop in pipes of up to 12-in. diameter, capacities up to 10,000 ft^3 of free air per minute (atmospheric), for initial pressures up to 400 psig.

Tables 8.12 to 8.15 show directly the pressure drop in pipes of up to 12-in. diameter, for capacities up to 30,000 ft^3 of free air per minute, and for initial pressures of 60, 80,100, and 125 lb.

Table 8.16 gives factors that can be used conveniently to determine the pressure drops in pipes of up to 12-in. diameter, for capacities up to 30,000 ft^3 of free air per minute and for any initial pressure.

$$D = \frac{F}{P_m}\,\frac{L}{1000}$$ (formula based on data of Fritzsche for steel pipe)

where:

D	=	pressure drop in pipe line, psi
F	=	friction factor from Figure 8.25
P_m	=	mean pressure in pipe, psia
L	=	length of pipe, ft.
d	=	actual inside diameter of pipe, in.

Note: For first approximation, use known terminal pressures at either end of pipe. Add barometric pressure to gage pressure to get absolute pressure.

Table 8.17 shows the loss of pressure through screw pipe fittings, expressed in equivalent lengths of straight pipe.

Figure 8.26 will be found convenient in determining pressure drop due to pipe friction where comparatively large volumes are handled at low initial pressures, such as are encountered in centrifugal-blower applications.

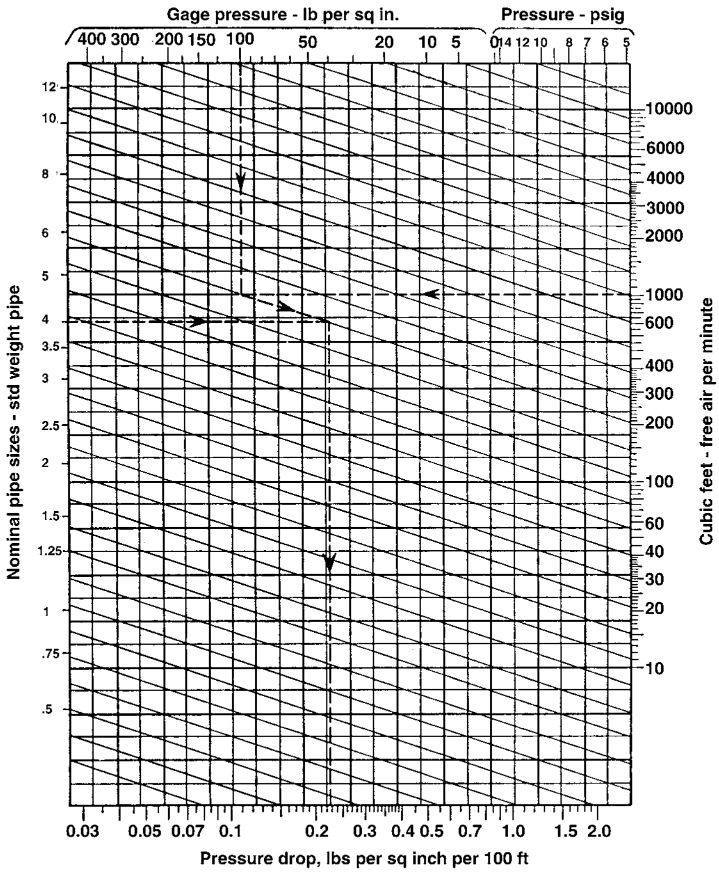

Figure 8.25 Loss of air pressure due to pipe friction for initial pressures up to 400 lb. *Problem:* 1,000 cfm free air (standard air) is to be transmitted at 100 psig pressure through a 4-in. standard weight pipe. What will be the pressure drop due to friction? *Solution by chart:* Enter the chart at the top, at the point representing 100 psig pressure, and proceed vertically downward to the intersection with a horizontal line representing 1,000 cfm, then parallel to the diagonal guide lines to the right (or left) to the intersection with a horizontal line representing a 4-in. pipe, then vertically downward to the pressure-loss scale at the bottom of the chart, where it is observed that the pressure loss would be 0.225 psi per 100 ft. of pipe. (Reprinted by permission from Walworth Co.)

Table 8.12 Loss of Air Pressure Due to Friction

Cu ft Free Air Per Min	Equivalent Cu ft Compressed Air Per Min	Nominal Diameter, In.											
		1/2	3/4	1	1 1/4	1 1/2	2	3	4	6	8	10	12
10	1.96	10.0	1.53	0.43	0.10								
20	3.94	39.7	5.99	1.71	0.39	0.18							
30	5.89	....	13.85	3.86	0.88	0.40							
40	7.86	....	24.7	6.85	1.59	0.71	0.19						
50	9.84	....	38.6	10.7	2.48	1.10	0.30						
60	11.81	....	55.5	15.4	3.58	1.57	0.43						
70	13.75	....	....	21.0	4.87	2.15	0.57						
80	15.72	....	....	27.4	6.37	2.82	0.75						
90	17.65	....	....	34.7	8.05	3.57	0.57	0.37					
100	19.60	....	....	42.8	9.95	4.40	1.18						
125	19.4	....	....	46.2	12.4	6.90	1.83	0.14					
150	29.45	....	....	....	22.4	9.90	2.64	0.32					
175	34.44	....	....	....	30.8	13.40	3.64	0.43					
200	39.40	....	....	....	39.7	17.60	4.71	0.57					
250	49.20	....	....	....	....	27.5	7.37	0.89	0.21				
300	58.90	....	....	....	....	39.6	10.55	1.30	0.31				
350	68.8	....	....	....	....	54.0	14.4	1.76	0.42				
400	78.8	....	....	....	....	....	18.6	2.30	0.53				
450	88.4	....	....	....	....	....	23.7	2.90	0.70				
500	98.4	....	....	....	....	....	29.7	3.60	0.85				
600	118.1	....	....	....	....	....	42.3	5.17	1.22				
700	137.5	....	....	....	....	....	57.8	7.00	1.67				
800	157.2	....	....	....	....	....	....	9.16	2.18				
900	176.5	....	....	....	....	....	....	11.6	2.76				
1,000	196.0	....	....	....	....	....	....	14.3	3.40				
1,500	294.5	....	....	....	....	....	....	32.3	7.6	0.87	0.29		
2,000	394.0	....	....	....	....	....	....	57.5	13.6	1.53	0.36		
2,500	492	....	....	....	....	....	....	....	21.3	2.42	0.57	0.17	
3,000	589	....	....	....	....	....	....	....	30.7	3.48	0.81	0.24	
3,500	688	....	....	....	....	....	....	....	41.7	4.68	1.07	0.33	
4,000	788	....	....	....	....	....	....	....	54.5	6.17	1.44	0.44	
4,500	884	....	....	....	....	....	....	....	....	7.8	1.83	0.55	0.21
5,000	984	....	....	....	....	....	....	....	....	9.7	2.26	0.67	0.27
6,000	1,181	....	....	....	....	....	....	....	....	13.9	3.25	0.98	0.38
7,000	1,375	....	....	....	....	....	....	....	....	18.7	4.43	1.34	0.51
8,000	1,572	....	....	....	....	....	....	....	....	24.7	5.80	1.73	0.71
9,000	1,765	....	....	....	....	....	....	....	....	31.3	7.33	2.20	0.87
10,000	1,960	....	....	....	....	....	....	....	....	38.6	9.05	2.72	1.06
11,000	2,165	....	....	....	....	....	....	....	....	46.7	10.9	3.29	1.28
12,000	2,362	....	....	....	....	....	....	....	....	55.5	13.0	3.90	1.51
13,000	2,560	....	....	....	....	....	....	....	....	....	15.2	4.58	1.77
14,000	2,750	....	....	....	....	....	....	....	....	....	17.7	5.32	2.07
15,000	2,945	....	....	....	....	....	....	....	....	....	20.3	6.10	2.36
16,000	3,144	....	....	....	....	....	....	....	....	....	23.1	6.95	2.70
18,000	3,530	....	....	....	....	....	....	....	....	....	29.2	8.80	3.42
20,000	3,940	....	....	....	....	....	....	....	....	....	36.2	10.8	4.22
22,000	4,330	....	....	....	....	....	....	....	....	....	43.7	13.2	5.12
24,000	4,724	....	....	....	....	....	....	....	....	....	51.9	15.6	5.92
26,000	5,120	....	....	....	....	....	....	....	....	....	....	18.3	7.15
28,000	5,500	....	....	....	....	....	....	....	....	....	....	21.3	8.3
30,000	5,890	....	....	....	....	....	....	....	....	....	....	24.4	9.5

In psi in 1000-ft of pipe, 60-lb gage initial pressure. For longer or shorter lengths of pipe the friction loss is proportional to the length, i.e., for 500 ft, one-half of the above; for 4,000 ft, four times the above, etc.

Table 8.13 Loss of Air Pressure Due to Friction

Cu ft Free Air Per Min	Equivalent Cu ft Compressed Air Per Min	Nominal Diameter, In.											
		1/2	3/4	1	1 1/4	1 1/2	2	3	4	6	8	10	12
10	1.55	7.90	1.21	0.34									
20	3.10	31.4	4.72	1.35	0.31								
30	4.65	70.8	10.9	3.04	0.69	0.31							
40	6.20	....	19.5	5.40	1.25	0.56							
50	7.74	....	30.5	8.45	1.96	0.87							
60	9.29	....	43.8	12.16	2.82	1.24	0.34						
70	10.82	....	59.8	16.6	3.84	1.70	0.45						
80	12.40	....	78.2	21.6	5.03	2.22	0.59						
90	13.95	....	....	27.4	6.35	2.82	0.75						
100	15.5	....	....	33.8	7.85	3.74	0.93						
125	19.4	....	....	46.2	12.4	5.45	1.44						
150	23.2	....	....	76.2	17.7	7.82	2.08						
175	27.2	....	....	....	24.8	10.6	2.87						
200	31.0	....	....	....	31.4	13.9	3.72	0.45					
250	38.7	....	....	....	49.0	21.7	5.82	0.70					
300	46.5	....	....	....	70.6	31.2	8.35	1.03					
350	54.2	....	....	....	....	42.5	11.4	1.39	0.33				
400	62.0	....	....	....	....	55.5	14.7	1.82	0.42				
450	69.7	....	....	....	....	....	18.7	2.29	0.55				
500	77.4	....	....	....	....	....	23.3	2.84	0.67				
600	92.9	....	....	....	....	....	33.4	4.08	0.96				
700	108.2	....	....	....	....	....	45.7	5.52	1.32				
800	124.0	....	....	....	....	....	59.3	7.15	1.72				
900	139.5	....	....	....	....	....	....	9.17	2.18				
1,000	155	....	....	....	....	....	....	11.3	2.68				
1,500	232	....	....	....	....	....	....	25.5	6.0	0.69			
2,000	310	....	....	....	....	....	....	45.3	10.7	1.21	0.29		
2,500	387	....	....	....	....	....	....	70.9	16.8	1.91	0.45		
3,000	465	....	....	....	....	....	....	....	24.2	2.74	0.64	0.19	
3,500	542	....	....	....	....	....	....	....	32.8	3.70	0.85	0.26	
4,000	620	....	....	....	....	....	....	....	43.0	4.87	1.14	0.34	
4,500	697	....	....	....	....	....	....	....	54.8	6.15	1.44	0.43	
5,000	774	....	....	....	....	....	....	....	67.4	7.65	1.78	0.53	0.21
6,000	929	....	....	....	....	....	....	....	....	11.0	2.57	0.77	0.29
7,000	1,082	....	....	....	....	....	....	....	....	14.8	3.40	1.06	0.40
8,000	1,240	....	....	....	....	....	....	....	....	19.5	4.57	1.36	0.54
9,000	1,395	....	....	....	....	....	....	....	....	24.7	5.78	1.74	0.69
10,000	1,550	....	....	....	....	....	....	....	....	30.5	7.15	2.14	0.84
11,000	1,710	....	....	....	....	....	....	....	....	36.8	8.61	2.60	1.01
12,000	1,860	....	....	....	....	....	....	....	....	43.8	10.3	3.08	1.19
13,000	2,020	....	....	....	....	....	....	....	....	51.7	12.0	3.62	1.40
14,000	2,170	....	....	....	....	....	....	....	....	60.2	14.0	4.20	1.63
15,000	2,320	....	....	....	....	....	....	....	....	68.5	16.0	4.82	1.84
16,000	2,480	....	....	....	....	....	....	....	....	78.2	18.2	5.48	2.13
18,000	2,790	....	....	....	....	....	....	....	....	....	23.0	6.95	2.70
20,000	3,100	....	....	....	....	....	....	....	....	....	28.6	8.55	3.33
22,000	3,410	....	....	....	....	....	....	....	....	....	34.5	10.4	4.04
24,000	3,720	....	....	....	....	....	....	....	....	....	41.0	12.3	4.69
26,000	4,030	....	....	....	....	....	....	....	....	....	48.2	14.4	5.6
28,000	4,350	....	....	....	....	....	....	....	....	....	55.9	16.8	6.3
30,000	4,650	....	....	....	....	....	....	....	....	....	64.2	19.3	7.5

In psi in 1000-ft of pipe, 80-lb gage initial pressure. For longer or shorter lengths of pipe the friction loss is proportional to the length, i.e., for 500 ft, one-half of the above; for 4,000 ft, four times the above, etc.

Table 8.14 Loss of Air Pressure Due to Friction

Cu ft Free Air Per Min	Equivalent Cu ft Compressed Air Per Min	Nominal Diameter, In.											
		1/2	3/4	1	1 1/4	1 1/2	2	3	4	6	8	10	12
10	1.28	6.50	.99	0.28									
20	2.56	25.9	3.90	1.11	0.25	0.11							
30	3.84	58.5	9.01	2.51	0.57	0.26							
40	5.12	....	16.0	4.45	1.03	0.46							
50	6.41	....	25.1	9.96	1.61	0.71	0.19						
60	7.68	....	36.2	10.0	2.32	1.02	0.28						
70	8.96	....	49.3	13.7	3.16	1.40	0.37						
80	10.24	....	64.5.	17.8	4.14	1.83	0.49						
90	11.52	....	82.8	22.6	5.23	2.32	0.62						
100	12.81	....	....	27.9	6.47	2.86	0.77						
125	15.82	....	....	48.6	10.2	4.49	1.19						
150	19.23	....	....	62.8	14.6	6.43	1.72	0.21					
175	22.40	....	....	....	19.8	8.72	2.36	0.28					
200	25.62	....	....	....	25.9	11.4	3.06	0.37					
250	31.64	....	....	....	40.4	17.9	4.78	0.58					
300	38.44	....	....	....	58.2	25.8	6.85	0.84	0.20				
350	44.80	....	....	....	....	35.1	9.36	1.14	0.27				
400	51.24	....	....	....	....	45.8	12.1	1.50	0.35				
450	57.65	....	....	....	....	58.0	15.4	1.89	0.46				
500	63.28	....	....	....	....	71.6	19.2	2.34	0.55				
600	76.88	....	....	....	....	....	27.6	3.36	0.79				
700	89.60	....	....	....	....	....	37.7	4.55	1.09				
800	102.5	....	....	....	....	....	49.0	5.89	1.42				
900	115.3	....	....	....	....	....	62.3	7.6	1.80				
1,000	128.1	....	....	....	....	....	76.9	9.3	2.21				
1,500	192.3	....	....	....	....	....	....	21.0	4.9	0.57			
2,000	256.2	....	....	....	....	....	....	37.4	8.8	0.99	0.24		
2,500	316.4	....	....	....	....	....	....	58.4	13.8	1.57	0.37		
3,000	384.6	....	....	....	....	....	....	84.1	20.0	2.26	0.53		
3,500	447.8	....	....	....	....	....	....	....	27.2	3.04	0.70	0.22	
4,000	512.4	....	....	....	....	....	....	....	35.5	4.01	0.94	0.28	
4,500	576.5	....	....	....	....	....	....	....	45.0	5.10	1.19	0.36	
5,000	632.8	....	....	....	....	....	....	....	55.6	6.3	1.47	0.44	0.17
6,000	768.8	....	....	....	....	....	....	....	80.0	9.1	2.11	0.64	0.24
7,000	896.0	....	....	....	....	....	....	....	....	12.2	2.88	0.87	0.33
8,000	1,025	....	....	....	....	....	....	....	....	16.1	3.77	1.12	0.46
9,000	1,153	....	....	....	....	....	....	....	....	20.4	4.77	1.43	0.57
10,000	1,280	....	....	....	....	....	....	....	....	25.1	5.88	1.77	0.69
11,000	1,410	....	....	....	....	....	....	....	....	30.4	7.10	2.14	0.83
12,000	1,540	....	....	....	....	....	....	....	....	36.2	8.5	2.54	0.98
13,000	1,668	....	....	....	....	....	....	....	....	42.6	9.8	2.98	1.15
14,000	1,795	....	....	....	....	....	....	....	....	49.2	11.5	3.46	1.35
15,000	1,923	....	....	....	....	....	....	....	....	56.6	13.2	3.97	1.53
16,000	2,050	....	....	....	....	....	....	....	....	64.5	15.0	4.52	1.75
18,000	2,310	....	....	....	....	....	....	....	....	81.5	19.0	5.72	2.22
20,000	2,560	....	....	....	....	....	....	....	....	....	23.6	7.0	2.74
22,000	2,820	....	....	....	....	....	....	....	....	....	28.5	8.5	3.33
24,000	3,080	....	....	....	....	....	....	....	....	....	33.8	10.0	3.85
26,000	3,338	....	....	....	....	....	....	....	....	....	39.7	11.9	4.65
28,000	3,590	....	....	....	....	....	....	....	....	....	46.2	13.8	5.40
30,000	3,850	....	....	....	....	....	....	....	....	....	53.0	15.9	6.17

In psi in 1000-ft of pipe, 100-lb gage initial pressure. For longer or shorter lengths of pipe the friction loss is proportional to the length, i.e., for 500 ft, one-half of the above; for 4,000 ft, four times the above, etc.

Table 8.15 Loss of Air Pressure Due to Friction

Cu ft Free Air Per Min	Equivalent Cu ft Compressed Air Per Min	Nominal Diameter, In.											
		1/2	3/4	1	1 1/4	1 1/2	2	3	4	6	8	10	12
10	1.05	5.35	0.82	0.23									
20	2.11	21.3	3.21	0.92	0.21								
30	3.16	48.0	7.42	2.07	0.47	0.21							
40	4.21	....	13.2	3.67	0.85	0.38							
50	5.26	....	20.6	5.72	1.33	0.59							
60	6.32	....	29.7	8.25	1.86	0.84	0.23						
70	7.38	....	40.5	11.2	2.61	1.15	0.31						
80	8.42	....	53.0	14.7	3.41	1.51	0.40						
90	9.47	....	68.0	18.6	4.30	1.91	0.51						
100	10.50	....	....	22.9	5.32	2.36	0.63						
125	13.15	....	....	39.9	8.4	3.70	0.98						
150	15.79	....	....	51.6	12.0	5.30	1.41	0.17					
175	18.41	....	....	....	16.3	7.2	1.95	0.24					
200	21.05	....	....	....	21.3	9.4	2.52	0.31					
250	26.30	....	....	....	33.2	14.7	3.94	0.48					
300	31.60	....	....	....	47.3	21.2	5.62	0.70					
350	36.80	....	....	....	....	28.8	7.7	0.94	0.22				
400	42.10	....	....	....	....	37.6	10.0	1.23	0.28				
450	47.30	....	....	....	....	47.7	12.7	1.55	0.37				
500	52.60	....	....	....	....	58.8	15.7	1.93	0.46				
600	63.20	....	....	....	....	....	22.6	2.76	0.65				
700	73.80	....	....	....	....	....	30.0	3.74	0.89				
800	84.20	....	....	....	....	....	40.2	4.85	1.17				
900	94.70	....	....	....	....	....	51.2	6.2	1.48				
1,000	105.1	....	....	....	....	....	63.2	7.7	1.82				
1,500	157.9	....	....	....	....	....	....	17.2	4.1	0.47			
2,000	210.5	....	....	....	....	....	....	30.7	7.3	0.82	0.19		
2,500	263.0	....	....	....	....	....	....	48.0	11.4	1.30	0.31		
3,000	316	....	....	....	....	....	....	69.2	16.4	1.86	0.43		
3,500	368	....	....	....	....	....	....	....	22.3	2.51	0.57	0.18	
4,000	421	....	....	....	....	....	....	....	29.2	3.30	0.77	0.23	
4,500	473	....	....	....	....	....	....	....	37.0	4.2	0.98	0.24	
5,000	526	....	....	....	....	....	....	....	45.7	5.2	1.21	0.36	
6,000	632	....	....	....	....	....	....	....	65.7	7.5	1.74	0.52	0.20
7,000	738	....	....	....	....	....	....	....	....	10.0	2.37	0.72	0.27
8,000	842	....	....	....	....	....	....	....	....	13.2	3.10	0.93	0.38
9,000	947	....	....	....	....	....	....	....	....	16.7	3.93	1.18	0.47
10,000	1,051	....	....	....	....	....	....	....	....	20.6	4.85	1.46	0.57
11,000	1,156	....	....	....	....	....	....	....	....	25.0	5.8	1.76	0.68
12,000	1,262	....	....	....	....	....	....	....	....	29.7	7.0	2.09	0.81
13,000	1,368	....	....	....	....	....	....	....	....	35.0	8.1	2.44	0.95
14,000	1,473	....	....	....	....	....	....	....	....	40.3	9.7	2.85	1.11
15,000	1,579	....	....	....	....	....	....	....	....	46.5	10.9	3.26	1.26
16,000	1,683	....	....	....	....	....	....	....	....	53.0	12.4	3.72	1.45
18,000	1,893	....	....	....	....	....	....	....	....	66.9	15.6	4.71	1.83
20,000	2,150	....	....	....	....	....	....	....	....	....	19.4	5.8	2.20
22,000	2,315	....	....	....	....	....	....	....	....	....	23.4	7.1	2.74
24,000	2,525	....	....	....	....	....	....	....	....	....	27.8	8.4	3.17
26,000	2,735	....	....	....	....	....	....	....	....	....	32.8	9.8	3.83
28,000	2,946	....	....	....	....	....	....	....	....	....	37.9	16.4	4.4
30,000	3,158	....	....	....	....	....	....	....	....	....	43.5	13.1	5.1

In psi in 1000-ft of pipe, 125-lb gage initial pressure. For longer or shorter lengths of pipe the friction loss is proportional to the length, i.e., for 500 ft, one-half of the above; for 4,000 ft, four times the above, etc.

Table 8.16 Factor for Calculating Loss of Air Pressure Due to Pipe Friction Applicable for any Initial Pressure*

Cu ft Free Air Per Min	Nominal Diameter, In.												
	1/2	3/4	1	1 1/4	1 1/2	1 3/4	2	3	4	6	8	10	12
5	12.7	1.2	0.5										
10	50.7	7.8	2.2	0.5									
15	114.1	17.6	4.9	1.1									
20	202	30.4	8.7	2.0	0.9								
25	316	50.0	13.6	3.2	1.4	0.7							
30	456	70.4	19.6	4.5	2.0	1.1							
35	811	95.9	26.2	6.2	2.7	1.4							
40	....	125.3	34.8	8.1	3.6	1.9							
45	....	159	44.0	10.2	4.5	2.4	1.2						
50	....	196	54.4	12.6	5.6	2.9	1.4						
60	....	282	78.3	18.2	8.0	4.2	2.2						
70	....	385	106.6	24.7	10.9	5.7	2.9						
80	....	503	139.2	32.3	14.3	7.5	3.8						
90	....	646	176.2	40.9	18.1	9.5	4.8						
100	....	785	217.4	50.5	22.3	11.7	6.0						
110	....	950	263	61.2	27.0	14.1	7.2						
120	....	....	318	72.7	32.2	16.8	8.6						
130	....	....	369	85.3	37.8	19.7	10.1	1.2					
140	....	....	426	98.9	43.8	22.9	11.7	1.4					
150	....	....	490	113.6	50.3	26.3	13.4	1.6					
160	....	....	570	129.3	57.2	29.9	15.3	1.9					
170	....	....	628	145.8	64.6	33.7	17.6	2.1					
180	....	....	705	163.3	72.6	37.9	19.4	2.4					
190	....	....	785	177	80.7	42.2	21.5	2.6					
200	....	....	870	202	89.4	46.7	23.9	2.9					
220	....	....	....	244	108.2	56.5	28.9	3.5					
240	....	....	....	291	128.7	67.3	34.4	4.2					
260	....	....	....	341	151	79.0	40.3	4.9					
280	....	....	....	395	175	91.6	46.8	5.7					
300	....	....	....	454	201	105.1	53.7	6.6					
320	....	....	....	....	....	....	61.1	7.5					
340	....	....	....	....	....	....	69.0	8.4	2.0				
360	....	....	....	....	....	....	77.3	9.5	2.2				
380	....	....	....	....	....	....	86.1	10.5	2.5				
400	....	....	....	....	....	....	94.7	11.7	2.7				
420	....	....	....	....	....	....	105.2	12.9	3.1				
440	....	....	....	....	....	....	115.5	14.1	3.4				
460	....	....	....	....	....	....	125.6	15.4	3.7				
480	....	....	....	....	....	....	137.6	16.8	4.0				
500	....	....	....	....	....	....	150.0	18.3	4.3				
525	....	....	....	....	....	....	165.0	20.2	4.8				
550	....	....	....	....	....	....	181.5	22.1	5.2				
575	....	....	....	....	....	....	197	24.2	5.7				
600	....	....	....	....	....	....	215	26.3	6.2				
625	....	....	....	....	....	....	233	28.5	6.8				
650	....	....	....	....	....	....	253	30.9	7.3				
675	....	....	....	....	....	....	272	33.3	7.9				
700	....	....	....	....	....	....	294	35.8	8.5				
750	....	....	....	....	....	....	337	41.4	9.7				
800	....	....	....	....	....	....	382	46.7	11.1				
850	....	....	....	....	....	....	433	52.8	12.5				
900	....	....	....	....	....	....	468	59.1	14.0				
950	....	....	....	....	....	....	541	65.9	15.7				
1,000	....	....	....	....	....	....	600	73.0	17.3	1.9			
1,050	....	....	....	....	....	....	658	80.5	19.1	2.1			

Table 8.16 (continued)

Cu Ft Free Air Per Min	Nominal Diameter, In.												
	1/2	3/4	1	1 1/4	1 1/2	1 3/4	2	3	4	6	8	10	12
1,100	....	....	....	....	....	....	723	88.4	21.0	2.4			
1,150	....	....	....	....	....	....	790	96.6	22.9	2.6			
1,200	....	....	....	....	....	....	850	105.2	25.0	2.8			
1,300	....	....	....	....	....	....	....	123.4	29.3	3.3			
1,400	....	....	....	....	....	....	....	....	33.9	3.8			
1,500	....	....	....	....	....	....	....	....	39.0	4.4			
1,600	....	....	....	....	....	....	....	....	44.3	5.1			
1,700	....	....	....	....	....	....	....	....	50.1	5.7			
1,800	....	....	....	....	....	....	....	....	56.1	6.4			
1,900	....	....	....	....	....	....	....	....	62.7	7.1	1.6		
2,000	....	....	....	....	....	....	....	....	69.3	7.8	1.8		
2,100	....	....	....	....	....	....	....	....	76.4	8.7	2.0		
2,200	....	....	....	....	....	....	....	....	83.6	9.5	2.2		
2,300	....	....	....	....	....	....	....	....	91.6	10.4	2.4		
2,400	....	....	....	....	....	....	....	....	99.8	11.3	2.6		
2,500	....	....	....	....	....	....	....	....	108.2	12.3	2.9		
2,600	....	....	....	....	....	....	....	....	117.2	13.3	3.1		
2,700	....	....	....	....	....	....	....	....	126	14.3	3.3		
2,800	....	....	....	....	....	....	....	....	136	15.4	3.6		
2,900	....	....	....	....	....	....	....	....	146	16.5	3.9		
3,000	....	....	....	....	....	....	....	....	156	17.7	4.1		
3,200	....	....	....	....	....	....	....	....	177	20.1	4.7		
3,400	....	....	....	....	....	....	....	....	200	22.7	5.3		
3,600	....	....	....	....	....	....	....	....	224	25.4	5.6	1.8	
3,800	....	....	....	....	....	....	....	....	250	28.4	6.6	2.0	
4,000	....	....	....	....	....	....	....	....	277	31.4	7.3	2.2	
4,200	....	....	....	....	....	....	....	....	305	34.6	8.1	2.4	
4,400	....	....	....	....	....	....	....	....	335	38.1	8.9	2.7	
4,600	....	....	....	....	....	....	....	....	366	41.5	9.7	2.9	
4,800	....	....	....	....	....	....	....	....	399	45.2	10.5	3.2	
5,000	....	....	....	....	....	....	....	....	433	49.1	11.5	3.4	
5,250	....	....	....	....	....	....	....	....	477	54.1	12.6	3.4	
5,500	....	....	....	....	....	....	....	....	524	59.4	13.9	4.2	1.6
5,750	....	....	....	....	....	....	....	....	....	64.9	15.2	4.6	1.8
6,000	....	....	....	....	....	....	....	....	....	70.7	16.5	5.0	1.9
6,500	....	....	....	....	....	....	....	....	....	82.9	19.8	5.9	2.3
7,000	....	....	....	....	....	....	....	....	....	96.2	22.5	6.8	2.6
7,500	....	....	....	....	....	....	....	....	....	110.5	25.8	7.8	3.0
8,000	....	....	....	....	....	....	....	....	....	125.7	29.4	8.8	3.6
9,000	....	....	....	....	....	....	....	....	....	159	37.2	10.2	4.4
10,000	....	....	....	....	....	....	....	....	....	196	45.9	13.8	5.4
11,000	....	....	....	....	....	....	....	....	....	237	55.5	16.7	6.5
12,000	....	....	....	....	....	....	....	....	....	282	66.1	19.8	7.7
13,000	....	....	....	....	....	....	....	....	....	332	77.5	23.3	9.0
14,000	....	....	....	....	....	....	....	....	....	387	89.9	27.0	10.5
15,000	....	....	....	....	....	....	....	....	....	442	103.2	31.0	12.0
16,000	....	....	....	....	....	....	....	....	....	503	117.7	35.3	13.7
18,000	....	....	....	....	....	....	....	....	....	636	148.7	44.6	17.4
20,000	....	....	....	....	....	....	....	....	....	....	184	55.0	21.4
22,000	....	....	....	....	....	....	....	....	....	....	222	66.9	26.0
24,000	....	....	....	....	....	....	....	....	....	....	264	79.3	30.1
26,000	....	....	....	....	....	....	....	....	....	....	310	93.3	36.3
28,000	....	....	....	....	....	....	....	....	....	....	360	108.0	42.1
30,000	....	....	....	....	....	....	....	....	....	....	413	123.9	48.2

*To determine the pressure drop in psi, the factor listed in the table for a given capacity and pipe diameter should be divided by the ratio of compression (from free air) at entrance of pipe, multiplied by the actual length of the pipe in feet, and divided by 1000.

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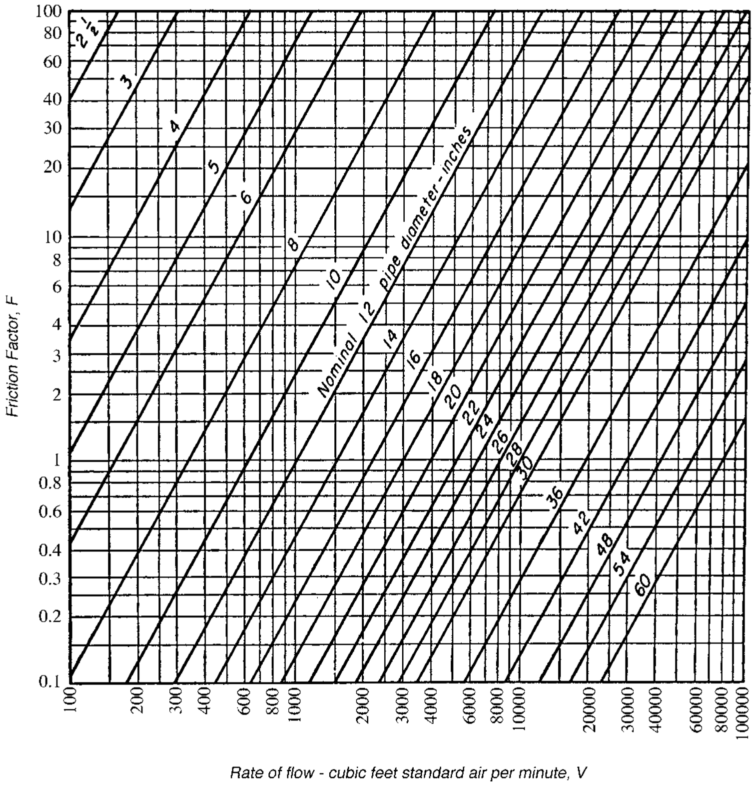

Figure 8.26 Loss of air pressure due to pipe friction measured at standard conditions of 14.7 psia and 60°F.

In all blower installations where a length of pipe is used to deliver air, either to the blower inlet or from the blower discharge or both, a certain amount of pressure is used up in forcing the air through these pipes. Ordinarily, when the combined length of the intake and discharge pipes is greater than ten pipe diameters, the drop in pressure is great enough to make a difference between the generated pressure and the pressure at the delivery end of the discharge pipe. This drop must be taken into consideration, especially if the pressure generated by the blower is to be very little in excess of that required to force the desired volume of air through the particular apparatus or system alone.

The formula for calculating this drop is:

$$D = 0.7 \frac{V^{1.85}}{d^5 p_m} \frac{L}{1000} = \frac{F}{p_m} \frac{L}{1000} \tag{8.25}$$

where:

- D = Pressure drop, psi.
- V = Volume of air flowing through the pipe in cfm measured at standard conditions (14.7 psia and 60°F).
- d = Actual inside diameter of pipe, in.
- L = Length of pipe, ft.*
- F = Friction factor from Figure 8.25.
- p_m = Mean pressure in pipe, psia. If p_1 is the initial pressure and p_2 the final pressure, then:

$$p_m = p_1 - \frac{D}{2} = p_2 + \frac{D}{2} \qquad (8.26)$$

Substitute first in Eq. (8.25) the known pressure (whether initial or final) for p_m and solve for the drop D. If this is less than 1 lb, it will not be necessary to calculate the mean pressure as this drop will be sufficiently accurate for most work. If the drop as calculated is greater than 1 lb, calculate the mean pressure p_m by substituting Eq (8.26) the value of D as first calculated. This will give a close value for the drop D so that it will not be necessary to refigure the mean pressure again. Continued trial will give any accuracy desired.

The use of Eq. (8.25) has been simplified by Figure 8.26, which gives the value of the friction factor F to be substituted in the right-hand member of Eq. (8.25) for various rates of flow (cubic feet per minute standard air).

Equation (8.25) is based on a flowing air temperature in the pipe of 60°F. For other flowing temperatures, multiply D already found by $T/520$, where T = 460 + actual "flowing" temperature (°F).

* Where there are bends in the pipe line, to derive L the linear length should be increased in accordance with the following data:

For each 90-degree bend with radius equal to:

(a) 1 pipe diameter, L should be increased 17.5 pipe diameters.
(b) 1 1/2 pipe diameters, L should be increased 10.4 pipe diameters.
(c) 2 pipe diameters, L should be increased 9.0 pipe diameters.
(d) 3 pipe diameters, L should be increased 8.2 pipe diameters.

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TABLE 8.17 Loss of Pressure through Screw Pipe Fittings, Steam, Air, Gas*

Nominal Pipe Size, In.	Actual Inside Diameter, In.	Gate Valve	Long Radius Ell or On Run of Standard Tee	Standard Ell or On Run of Tee Reduced in Size 50 per cent	Angle Valve	Close Return Bend	Tee through Side Outlet	Globe Valve
1/2	0.622	0.36	0.62	1.55	8.65	3.47	3.10	17.3
3/4	0.824	0.48	0.82	2.06	11.4	4.60	4.12	22.9
1	1.049	0.61	1.05	2.62	14.6	5.82	5.24	29.1
1 1/4	1.380	0.81	1.38	3.45	19.1	7.66	6.90	38.3
1 1/2	1.610	0.94	1.61	4.02	22.4	8.95	8.04	44.7
2	2.067	1.21	2.07	5.17	28.7	11.5	10.3	57.4
2 1/2	2.469	1.44	2.47	6.16	34.3	13.7	12.3	68.5
3	3.068	1.79	3.07	6.16	42.6	17.1	15.3	85.2
4	4.026	2.35	4.03	7.67	56.0	22.4	20.2	112
5	5.047	2.94	5.05	10.1	70.0	28.0	25.2	140
6	6.065	3.54	6.07	15.2	84.1	33.8	30.4	168
8	7.981	4.65	7.98	20.0	111	44.6	40.0	222
10	10.020	5.85	10.00	25.0	139	55.7	50.0	278
12	11.940	6.96	11.0	29.8	166	66.3	59.6	332

* Adapted from Sabin Crocker, *Piping Handbook*, 4th ed., McGraw-Hill Book Company, Inc., New York, 1945. Given in equivalent lengths (feet) of straight pipe, schedule 40.

DATA, TABLES, FORMULAS

TABLE 8.18 Friction of Air in Hose, Pulsating Flow*

Size of Hose, Coupled Each End In.	Gage Pressure at Line, Lb	Cu ft. Air Per Min Passing through 50-ft Lengths of Hose													
		20	30	40	50	60	70	80	90	100	110	120	130	140	150
		Loss of Pressure (psi) in 50-ft Lengths of Hose													
	50	1.8	5.0	10.1	18.1										
	60	1.3	4.0	8.4	14.8	23.4									
1/2	70	1.0	3.4	7.0	12.4	20.0	28.4								
	80	0.9	2.8	6.0	10.8	17.4	25.2	34.6							
	90	0.8	2.4	5.4	9.5	14.8	22.0	30.5	41.0						
	100	0.7	2.3	4.8	8.4	13.3	19.3	27.2	36.6						
	110	0.6	2.0	4.3	7.6	12.0	17.6	24.6	33.3	44.5					
	50	0.4	0.8	1.5	2.4	3.5	4.4	6.5	8.5	11.4	14.2				
	60	0.3	0.6	1.2	1.9	2.8	3.8	5.2	6.8	8.6	11.2				
3/4	70	0.2	0.5	0.9	1.5	2.3	3.2	4.2	5.5	7.0	8.8	11.0			
	80	0.2	0.5	0.8	1.3	1.9	2.8	3.6	4.7	5.8	7.2	8.8	10.6		
	90	0.2	0.4	0.7	1.1	1.6	2.3	3.1	4.0	5.0	6.2	7.5	9.0		
	100	0.2	0.4	0.6	1.0	1.4	2.0	2.7	3.5	4.4	5.4	6.6	7.9	9.4	11.1
	110	0.1	0.3	0.5	0.9	1.3	1.8	2.4	3.1	3.9	4.9	5.9	7.1	8.4	9.9
	50	0.1	0.2	0.3	0.5	0.8	1.1	1.5	2.0	2.6	3.5	4.8	7.0		
	60	1.1	0.2	0.3	0.4	0.6	0.8	1.2	1.5	2.0	2.6	3.3	4.2	5.5	7.2
	70	.....	0.1	0.2	0.4	0.5	0.7	1.0	1.3	1.6	2.0	2.5	3.1	3.8	4.7
	80	.....	0.1	0.2	0.3	0.5	0.7	0.8	1.1	1.4	1.7	2.0	2.4	2.7	3.5
	90	.....	1.1	0.2	0.3	0.4	0.6	0.7	0.9	1.2	1.4	1.7	2.0	2.4	2.8
	100	.....	1.1	0.2	0.2	0.4	0.5	0.6	0.8	1.0	1.2	1.5	1.8	2.1	2.4
	110	.....	0.1	0.2	0.2	0.3	0.4	0.6	0.7	0.9	1.1	1.3	1.5	1.8	2.1
	50	.....	.....	0.1	0.2	0.2	0.3	0.4	0.5	0.7	1.1				
	60	.....	.....	.....	0.1	0.2	0.3	0.3	0.5	0.6	0.8	1.0	1.2	1.5	
	70	.....	.....	.....	0.1	0.2	0.2	0.3	0.4	0.4	0.5	0.7	0.8	1.0	1.3
1 1/4	80	.....	.....	.....	.....	0.1	0.2	0.2	0.3	0.4	0.5	0.6	0.7	0.8	1.0
	90	.....	.....	.....	.....	0.1	0.2	0.2	0.3	0.3	0.4	0.5	0.6	0.7	0.8
	100	.....	.....	.....	.....	.....	0.1	0.2	0.2	0.3	0.4	0.4	0.5	0.6	0.7
	110	.....	.....	.....	.....	.....	0.1	0.2	0.2	0.3	0.3	0.4	0.5	0.5	0.6
	50	.....	.....	.....	.....	.....	0.1	0.2	0.2	0.2	0.3	0.3	0.4	0.5	0.6
	60	.....	.....	.....	.....	.....	.....	0.1	0.2	0.2	0.2	0.3	0.3	0.4	0.5
1 1/2	70	.....	.....	.....	.....	.....	.....	.....	0.1	0.2	0.2	0.2	0.3	0.3	0.4
	80	.....	.....	.....	.....	.....	.....	.....	.....	0.1	0.2	0.2	0.2	0.3	0.4
	90	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.1	0.2	0.2	0.2	0.3
	100	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.1	0.2	0.2	0.2
	110	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.1	0.2	0.2	0.2

*For longer or shorter lengths of hose the friction loss is proportional to the length, i.e., for 25 ft one-half of the above; for 150 ft, three times the above, etc.

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TABLE 8.19 Effect of Altitude on Capacity of Single-stage Compressors

	25 psig		40 psig		60 psig		80 psig		90 psig		100 psig		125 psig	
Altitude, ft.	Compressor Ratio	Factor	Compressor Ratio	Factor	Compressor Ratio	Factor	Compressor Ratio	Factor	Compressor Ratio	Factor	Compressor Ratio	Factor	Compressor Ratio	Factor
Sea level	2.70	1.0	3.72	1.0	5.08	1.0	6.44	1.0	7.12	1.0	7.81	1.0	9.51	1.0
1,000	2.76	0.996	3.82	0.993	5.23	0.992	6.64	0.992	7.34	0.988	8.05	0.987	9.81	0.982
2,000	2.84	0.992	3.94	0.987	5.42	0.984	6.88	0.977	7.62	0.972	8.35	0.972	10.20	0.962
3,000	2.91	0.987	4.06	0.981	5.59	0.974	7.12	0.967	7.87	0.959	8.63	0.957	10.55	0.942
4,000	2.99	0.982	4.18	0.974	5.76	0.963	7.36	0.953	8.15	0.944	8.94	0.942	10.92	0.923
5,000	3.07	0.977	4.31	0.967	5.96	0.953	7.62	0.940	8.44	0.931	9.27	0.925	11.32	
6,000	3.14	0.972	4.42	0.961	6.13	0.945	7.84	0.928	8.69	0.917	9.55	0.908	11.69	
7,000	3.23	0.967	4.57	0.953	6.36	0.936	8.14	0.915	9.03	0.902	9.93	0.890	12.17	
8,000	3.32	0.962	4.71	0.945	6.56	0.925	8.42	0.900	9.33	0.886	10.26	0.873	12.58	
9,000	3.41	0.957	4.85	0.938	6.77	0.915	8.70	0.887	9.65	0.868	10.62	0.857	13.02	
10,000	3.50	0.951	5.00	0.931	7.00	0.902	9.00	0.872	10.00	0.853	11.00	0.340	13.50	
11,000	3.61	0.945	5.17	0.923	7.25	0.891	9.34	0.858	10.38	0.837	11.42	....	14.03	
12,000	3.72	0.938	5.35	0.914	7.53	0.878	9.70	0.839	10.79	0.818	11.88	....	14.60	
14,000	3.94	0.927	5.71	0.897	8.06	0.852	10.42	0.805	11.60	....	12.78	....	15.71	
15,000	4.09	0.918	5.94	0.887	8.42	0.836	10.88	0.784	12.12	....	13.36	....	16.43	

Factor for estimating, based on 7% cylinder clearance.
Note: To find the capacity of a compressor when it is used at an altitude, multiply the sea-level capacity of the unit by the factor corresponding to the altitude and the discharge pressure. The result will be the actual capacity of the unit at the altitude.

TABLE 8.20 Multipliers to Determine Air Consumption of Rock Drills at Altitudes and for Various Number of Drills*

	Number of Drills																
	1	2	3	4	5	6	7	8	9	10	12	15	20	25	30	40	50
Altitude, ft.	Multiplier- Assuming 90 psig Air Pressure																
0	1.000	2.0	3.0	4.0	5.0	6.0	6.3	7.2	8.1	9.0	10.8	12.0	16.0	20.0	22.5	30.0	37.5
1,000	1.032	2.1	3.1	4.1	5.2	6.2	6.5	7.4	8.4	9.3	11.1	12.4	16.5	20.6	23.2	31.0	38.7
2,000	1.065	2.1	3.2	4.3	5.3	6.4	6.7	7.7	8.6	9.6	11.5	12.8	17.0	21.3	24.0	31.9	40.0
3,000	1.100	2.2	3.3	4.4	5.5	6.6	6.9	7.9	8.9	9.9	11.9	13.2	17.6	22.0	24.8	33.0	41.3
4,000	1.136	2.3	3.4	4.5	5.7	6.8	7.1	8.2	9.2	10.2	12.3	13.7	18.2	22.7	25.6	34.1	42.6
5,000	1.174	2.3	3.5	4.7	5.9	7.0	7.4	8.5	9.5	10.5	12.7	14.1	18.8	23.5	26.4	35.2	44.0
6,000	1.213	2.4	3.6	4.9	6.1	7.3	7.6	8.7	9.8	10.9	13.0	14.6	19.4	24.2	27.4	36.4	45.5
7,000	1.255	2.5	3.8	5.0	6.3	7.5	7.9	9.0	10.2	11.3	13.6	15.1	20.0	25.1	28.2	37.6	47.1
8,000	1.298	2.6	3.9	5.2	6.4	7.8	8.2	9.3	10.5	11.7	14.0	15.6	20.8	26.0	29.2	38.9	48.7
9,000	1.343	2.7	4.0	5.4	6.7	8.1	8.5	9.7	10.9	12.1	14.5	16.1	21.5	26.9	30.3	40.3	50.4
10,000	1.391	2.8	4.2	5.6	7.0	8.3	8.8	10.0	11.3	12.5	15.0	16.7	22.3	27.8	31.3	41.7	52.2
12,500	1.520	3.0	4.6	6.1	7.6	9.1	9.6	10.9	12.3	13.7	16.4	18.2	24.3	30.4	34.2	45.6	57.0
15,000	1.665	3.3	5.0	6.7	8.3	10.0	10.5	11.9	13.5	15.0	18.0	20.0	26.6	33.3	37.5	49.9	62.4

*Air consumption for various number of drills based on fact that all drills will not operate at once. It will vary with rock, type of work, etc.

TABLE 8.21 Theoretical Horsepower Required at Altitude to Compress 100 Cubic Feet

Altitude,	Isothermal Compression Single- and Two-stage Gage Pressure					Adiabatic Compression Single-stage Gage Pressure			Adiabatic Compression Two-stage Gage Pressure				
ft.	60	80	100	125	150	60	80	100	60	80	100	125	150
0	10.4	11.9	13.2	14.4	15.5	13.4	15.9	18.1	11.8	13.7	15.4	17.1	18.7
1,000	10.2	11.7	12.9	14.1	15.1	13.2	15.6	17.8	11.6	13.5	15.1	16.8	18.3
2,000	10.0	11.4	12.6	13.8	14.8	13.0	15.4	17.5	11.4	13.2	14.8	16.4	17.9
3,000	9.8	11.2	12.3	13.5	14.4	12.8	15.2	17.2	11.2	13.0	14.5	16.1	17.5
4,000	9.6	11.0	12.1	13.2	14.1	12.6	14.9	16.9	11.0	12.7	14.2	15.7	17.1
5,000	9.4	10.7	11.8	12.8	13.7	12.4	14.7	16.5	10.8	12.5	13.9	15.4	16.7
6,000	9.2	10.5	11.5	12.5	13.4	12.2	14.4	16.2	10.6	12.2	13.6	15.1	16.4
7,000	9.0	10.3	11.2	12.2	13.0	12.0	14.2	16.0	10.4	12.0	13.4	14.8	16.0
8,000	8.9	10.0	11.0	11.9	12.7	11.8	14.0	15.7	10.2	11.8	13.1	14.5	15.6
9,000	8.7	9.8	10.7	11.6	12.4	11.6	13.7	15.4	10.0	11.6	12.8	14.1	15.3
10,000	8.5	9.6	10.4	11.4	12.1	11.5	13.5	15.1	9.8	11.3	12.6	13.8	15.0

Table 8.22 Approximate Brake Horsepower Required by Air Compressors

Altitude,	Single-stage psig			Two-stage psig			
ft.	60	80	100	60	80	100	125
0	16.3	19.5	22.1	14.7	17.1	19.1	21.3
1,000	16.1	19.2	21.7	14.5	16.8	18.7	20.9
2,000	15.9	18.9	21.3	14.3	16.5	18.4	20.5
3,000	15.7	18.6	20.9	14.0	16.1	18.0	20.0
4,000	15.4	18.2	20.6	13.8	15.8	17.7	19.6
5,000	15.2	17.9	20.3	13.5	15.5	17.3	19.2
6,000	15.0	17.6	20.0	13.3	15.2	17.0	18.8
7,000	14.7	17.3	19.6	13.0	14.9	16.6	18.4
8,000	14.5	17.1	19.3	12.7	14.6	16.2	18.0
9,000	14.3	16.8	18.9	12.5	14.3	15.9	17.6
10,000	14.1	16.5	18.6	12.3	14.1	15.6	17.2
12,000	13.6	15.9	17.9	11.8	13.5	15.0	16.5
14,000	13.1	15.2	17.2	11.3	12.9	14.3	15.7

Figures given are bhp per 100 ft^3 of free air per minute actually delivered. Bhp per 100 ft^3 of free air per minute will vary considerably with the size and type of compressor.

Table 8.23 Theoretical Horsepower Required to Compress Air from Atmospheric Pressure to Various Pressures, Mean Effective Pressures (mep)

Discharge Pressure			Isothermal Compression, Single or Multi-stage		Adiabatic Compression*					
					Single-stage		Two-stage			Per Cent of Power Saved by Two-stage over Single-stage Adiabatic Compression
psig	psig	Atm Abs	Mep	Theo-rectical hp Per 100 Cu. ft.	Mep	Theo-rectical hp Per 100 Cu. ft.	Mep, Psi Referred to Low-press. Air Cylinder	Theo-rectical hp Per 100 Cu. ft.	Theo-rectical Intercooler Gauge Pressure	
5	19.7	1.34	4.13	1.8	4.48	1.96				
10	24.7	1.68	7.57	3.3	8.21	3.58				
15	29.7	2.02	10.31	4.5	11.4	5.0				
20	34.7	2.36	12.62	5.5	14.3	6.2				
25	39.7	2.70	14.68	6.4	16.9	7.4				
30	44.7	3.04	16.30	7.1	19.2	8.4				
35	49.7	3.38	17.90	7.8	21.4	9.3				
40	54.7	3.72	19.28	8.4	23.4	10.2				
45	59.7	4.06	20.65	9.0	25.2	11.0				
50	64.7	4.40	21.80	9.5	27.0	11.8				
55	69.7	4.74	22.95	10.0	28.7	12.6				
60	74.7	5.08	23.90	10.4	30.3	13.3				
65	79.7	5.42	24.80	10.8	31.9	13.9				
70	84.7	5.76	25.70	11.2	33.3	14.6	29.2	12.8	20.6	12.3
75	89.7	6.10	26.62	11.6	34.7	15.2	30.2	13.3	21.6	12.5
80	94.7	6.44	27.52	12.0	36.0	15.7	31.3	13.7	22.7	12.7
85	99.7	6.78	28.21	12.3	37.3	16.3	32.3	14.1	23.6	13.5
90	104.7	7.12	28.93	12.6	38.6	16.9	33.2	14.5	24.5	14.2
95	109.7	7.46	29.60	12.9	39.8	17.4	34.2	14.9	25.5	14.4
100	114.7	7.80	30.30	13.2	40.9	17.9	35.0	15.3	26.3	14.5
110	124.7	8.48	31.42	13.7	43.2	18.9	36.7	16.1	28.1	14.8
120	134.7	9.16	32.60	14.2	45.2	19.8	38.3	16.8	29.8	15.1
130	144.7	9.84	33.75	14.7	47.2	20.7	39.6	17.3	31.5	16.4
140	154.7	10.52	34.67	15.1	49.2	21.5	40.8	17.9	32.9	15.7
150	164.7	11.20	35.59	15.5	51.0	22.3	42.3	18.5	34.5	17.1
160	174.7	11.88	36.30	15.8	.....	.....	43.6	19.0	36.1	
170	184.7	12.56	37.20	16.2	.....	.....	44.7	19.5	37.3	
180	194.7	13.24	38.10	16.6	.....	.....	45.8	20.0	38.8	
190	204.7	13.92	38.80	16.9	.....	.....	46.8	20.4	40.1	
200	214.7	14.60	39.50	17.2	.....	.....	47.8	20.9	41.4	
250	264.7	18.00	42.70	18.6	.....	.....	52.5	22.7	47.6	
300	314.7	21.40	45.30	19.7	.....	.....	56.5	24.5	53.4	
350	364.7	24.81	47.30	20.6	.....	.....	59.6	26.1	58.5	
400	414.7	28.21	49.20	21.4	.....	.....	62.7	27.4	63.3	
450	464.7	31.61	51.20	22.3	.....	.....	65.3	28.6	67.8	
500	514.7	35.01	52.70	22.9	.....	.....	67.8	29.6	71.2	
550	564.7	38.41	53.75	23.4	.....	.....	70.0	30.6	76.3	
600	614.7	41.81	54.85	23.9	.....	.....	72.3	31.3	80.5	

* Based on a value for *n* of 1.3947.

TABLE 8.24 Displacement of a Double-acting Piston at Various Piston Speeds (Piston Rods Not Deducted)

Diameter of cylinder, In.	Piston Speed, Fpm = 2 × Stroke (Ft) × Rpm (for Double-acting Cylinder)																		
	200	225	250	275	300	325	350	375	400	425	450	475	500	550	600	650	700	750	800
	Displacement, Cfm																		
3	9.8	11.1	12.3	13.5	14.7	16	17.2	18.5	19.6	21	22.2	23.4	24.6	27	29.4	31.8	34.4	36.8	39.2
4	17.4	19.7	21.8	24	26.2	28.5	30.7	32.9	34.8	37.2	39.4	41.5	43.6	48	52.4	56.7	61.4	65.2	69.5
5	27.2	30.8	34.2	37.5	41	44.5	48	51.4	54.4	58	61.6	65	68.4	75	82	88.7	96	102	109
6	39.2	44.4	49	54	59	64	69	74	78.4	84	88.8	93	98	108	118	128	138	148	157
7	53.4	60.5	67	74	80	87	94	100.6	106.8	114	121	127	134	148	160	174	188	201	214
8	70	79	87.5	96	105	114	123	132	140	149	158	166	175	192	210	227	246	262	280
9	88	100	110	122	133	144	155	166	176	188	200	210	220	244	266	287	310	332	352
10	109	123	136	150	164	178	192	206	218	233	246	260	272	300	328	354	384	409	436
11	132	149	165	182	198	215	232	250	264	282	298	315	330	364	396	430	464	496	528
12	156	177	196	216	236	256	276	296	312	335	354	374	392	432	472	514	552	585	625
13	184	208	230	255	276	300	325	345	368	392	416	440	460	510	552	601	650	690	736
14	212	240	266	295	320	348	375	400	424	455	480	510	532	590	640	695	750	795	850
15	244	276	305	337	368	400	432	460	488	521	552	580	610	674	736	798	864	915	976
16	278	315	350	385	420	455	490	525	556	595	630	660	700	790	840	910	980	1,010	1,110
17	314	355	395	435	472	512	550	591	628	672	710	750	790	870	944	1,020	1,100	1,180	1,260
18	352	400	440	485	530	575	620	660	704	755	800	840	880	970	1,060	1,150	1,240	1,320	1,410
19	392	445	490	540	590	640	690	740	784	840	890	930	980	1,080	1,180	1,280	1,380	1,470	1,570
20	436	490	545	600	656	710	765	820	872	930	980	1,040	1,090	1,200	1,312	1,420	1,530	1,640	1,750
22	530	595	660	725	790	860	930	990	1,060	1,120	1,190	1,260	1,320	1,450	1,580	1,720	1,860	1,990	2,040
24	628	710	790	865	940	1,020	1,100	1,180	1,256	1,340	1,420	1,500	1,580	1,730	1,880	2,040	2,200	2,360	2,520
26	738	830	920	1,020	1,100	1,200	1,300	1,390	1,476	1,570	1,660	1,750	1,840	2,040	2,200	2,400	2,600	2,770	2,970
28	854	900	1,070	1,180	1,280	1,390	1,500	1,600	1,780	1,820	1,920	2,030	2,140	2,360	2,560	2,780	3,000	3,200	3,420
30	980	1,110	1,230	1,350	1,480	1,600	1,720	1,810	1,960	2,100	2,220	2,340	2,460	2,700	2,960	3,200	3,440	3,680	3,920
32	1,120	1,260	1,400	1,540	1,670	1,820	1,960	2,100	2,240	2,380	2,520	3,660	2,800	2,080	3,340	3,630	3,920	4,200	4,480
34	1,260	1,420	1,570	1,740	1,890	2,050	2,220	2,360	2,520	2,680	2,840	3,000	3,140	3,480	3,780	4,090	4,440	4,720	5,040
36	1,420	1,600	1,770	1,950	2,120	2,300	2,500	2,650	2,840	3,000	3,200	3,360	3,540	4,240	4,240	4,620	5,000	5,320	5,680
38	1,560	1,780	1,960	2,170	2,360	2,560	2,750	2,950	3,120	3,350	3,560	3,750	3,920	4,720	4,720	5,110	5,500	5,850	6,250
40	1,740	1,970	2,180	2,400	2,620	2,850	3,060	3,280	3,480	3,720	3,940	4,150	4,360	5,240	5,240	5,680	6,120	6,520	6,960

A convenient formula for calculating the displacement of one double-acting cylinder is square of diameter (in.) x stroke (in.) x rpm x .0009 = piston displacement (cfm). This provides for a reasonable allowance for piston rod on all piston sizes from about 5 in. diameter to about 30 in. diameter. If there were no allowance for the piston rod, the multiplier would be .0009090.

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TABLE 8.25 Discharge of Air Through an Orifice

Gage Pressure before Orifice, psi	Nominal Diameter, In.										
	1/64	1/32	1/16	1/8	1/4	3/8	1/2	5/8	3/4	7/8	1
	Discharge, Cu. ft. Free Air Per Min.										
1	.028	0.112	0.450	1.80	7.18	16.2	28.7	45.0	64.7	88.1	115
2	.040	0.158	0.633	2.53	10.1	22.8	40.5	63.3	91.2	124	162
3	.048	0.194	0.775	3.10	12.4	27.8	49.5	77.5	111	152	198
4	.056	0.223	0.892	3.56	14.3	32.1	57.0	89.2	128	175	228
5	.062	0.248	0.993	3.97	15.9	35.7	63.5	99.3	143	195	254
6	.068	0.272	1.09	4.34	17.4	39.1	69.5	109	156	213	278
7	.073	0.293	1.17	4.68	18.7	42.2	75.0	117	168	230	300
9	.083	0.331	1.32	5.30	21.1	47.7	84.7	132	191	260	339
12	.095	0.379	1.52	6.07	24.3	54.6	97.0	152	218	297	388
15	.105	0.420	1.68	6.72	26.9	60.5	108	168	242	329	430
20	.123	0.491	1.96	7.86	31.4	70.7	126	196	283	385	503
25	.140	0.562	2.25	8.98	35.9	80.9	144	225	323	440	575
30	.158	0.633	2.53	10.1	40.5	91.1	162	253	365	496	648
35	.176	0.703	2.81	11.3	45.0	101	180	281	405	551	720
40	.194	0.774	3.10	12.4	49.6	112	198	310	446	607	793
45	.211	0.845	3.38	13.5	54.1	122	216	338	487	662	865
50	.229	0.916	3.66	14.7	58.6	132	235	366	528	718	938
60	.264	1.06	4.23	16.9	67.6	152	271	423	609	828	1,082
70	.300	1.20	4.79	19.2	76.7	173	307	479	690	939	1,227
80	.335	1.34	5.36	21.4	85.7	193	343	536	771	1,050	1,371
90	.370	1.48	5.92	23.7	94.8	213	379	592	853	1,161	1,516
100	.406	1.62	6.49	26.0	104	234	415	649	934	1,272	1,661
110	.441	1.76	7.05	28.2	113	254	452	705	1,016	1,383	1,806
120	.476	1.91	7.62	30.5	122	274	488	762	1,097	1,494	1,951
125	.494	1.98	7.90	31.6	126	284	506	790	1,138	1,549	2,023

Based on 100% coefficient of flow. For well-rounded entrance multiply values by 0.97. For sharp-edged orifices a multiplier of 0.65 may be used.

This table will give approximate results only. For accurate measurements see ASME Power Test Code, Velocity Volume Flow Measurement.

Values for pressures from 1 to 15 psig calculated by standard adiabatic formula.

Values for pressures above 15 psig calculated by approximate formula proposed by S. A. Moss: $w = 0.5303\ aCp_1 \quad T_1$ where w = discharge in lb per sec, a = area of orifice in sq. in., C = coefficient of flow, p_1 = upstream total pressure in psia, and T_1 = upstream temperature in deg F abs.

Values used in calculating above table were C = 1.0, p_1 = gage pressure + 14.7 psi, T_1 = 530 F abs.

Weights (w) were converted to volumes using density factor of 0.07494 lb. per cu. ft. This is correct for dry air at 14.7 psia and 70°F.

Formula cannot be used where p_1 is less than two times the barometric pressure.

Table 8.26 Standard Weight of Welded and Seamless Steam, Air, Gas and Water Pipe

Nominal Pipe Size, In.	Diameter In.		Thickness, In.	Circumference, In.		Transverse Area, Sq In.			Length of Pipe, Ft Per Sq Ft		Length of Pipe Containing 1 Cu Ft
	External	Internal		External	Internal	External	Internal	Metal	External Surface	Internal Surface	
1/8	0.405	0.269	.068	1.272	0.845	0.129	0.057	0.072	9.431	14.199	2,533.775
¼	0.540	0.364	.088	1.696	1.144	0.229	0.104	0.125	7.073	10.493	1,383.789
3/8	0.675	0.493	.091	2.121	1.549	0.358	0.191	0.167	5.658	7.748	754.360
½	0.840	0.622	.109	2.639	1.954	0.554	0.304	0.250	4.547	6.141	473.906
¾	1.050	0.824	.113	3.299	2.589	0.866	0.533	0.333	3.637	4.635	270.034
1	1.315	1.049	.133	4.131	3.296	1.358	0.864	0.494	2.904	3.641	166.618
1 ¼	1.660	1.380	.140	5.215	4.335	2.164	1.495	0.669	2.301	2.768	96.275
1 ¼	1.900	1.610	.145	5.969	5.058	2.835	2.036	0.799	2.010	2.372	70.733
2	2.375	2.067	.154	7.461	6.494	4.430	3.355	1.075	1.608	1.847	42.913
2 ½	2.875	2.469	.203	9.032	7.757	6.492	4.788	1.704	1.328	1.547	30.077
3	3.500	3.068	.216	10.996	9.638	9.621	7.393	2.228	1.091	1.245	19.479
3 ½	4.000	3.548	.226	12.566	11.146	12.566	9.886	2.680	0.954	1.076	14.565
4	4.500	4.026	.237	14.137	12.648	15.904	12.730	3.174	0.848	0.948	11.312
5	5.563	5.047	.258	17.477	15.856	23.306	20.066	4.300	0.686	0.756	7.198
6	6.625	6.065	.280	20.813	19.054	34.472	28.891	5.581	0.576	0.629	4.984
8	8.625	8.071	.277	27.096	25.356	58.426	51.161	7.265	0.443	0.473	2.815
8	8.625	7.981	.322	27.096	25.073	58.426	50.027	8.399	0.443	0.478	2.878
10	10.750	10.020	.365	33.772	31.479	90.763	78.855	11.908	0.355	0.381	1.826
12	12.750	12.000	.375	40.055	37.699	127.676	113.097	14.579	0.299	0.318	1.273
14 OD	14.000	13.250	.375	43.982	41.626	153.938	137.886	16.052	0.272	0.288	1.044
15 OD	15.000	14.250	.375	47.124	44.768	176.715	159.485	17.230	0.254	0.268	0.903
16 OD	16.000	15.250	.375	50.265	47.909	201.062	182.654	18.408	0.238	0.250	0.788
17 OD	17.000	16.214	.393	53.407	50.938	226.980	206.476	20.504	0.224	0.235	0.697
18 OD	18.000	17.182	.409	56.549	53.979	254.469	231.866	22.603	0.212	0.222	0.621
20 OD	20.000	19.182	.409	62.832	60.262	314.159	288.986	25.173	0.191	0.199	0.498

Based on ASTM Standard Specification A53-33.

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Table 8.27 Weight of Water Vapor in One Cubic Foot of Air and Various Temperatures and Percentages of Saturation

Temperature deg F	Per cent of Saturation									
	10	20	30	40	50	60	70	80	90	100
	Weight, Grains									
– 10	0.028	0.057	0.086	0.114	0.142	0.171	0.200	0.228	0.256	0.285
0	0.048	0.096	0.144	0.192	0.240	0.289	0.337	0.385	0.433	0.481
10	0.078	0.155	0.233	0.210	0.388	0.466	0.543	0.621	0.698	0.776
20	0.124	0.247	0.370	0.494	0.618	0.741	0.864	0.988	1.112	1.235
30	0.194	0.387	0.580	0.774	0.968	1.161	1.354	1.548	1.742	1.935
32	0.211	0.422	0.634	0.845	1.056	1.268	1.479	1.690	1.902	2.113
35	0.237	0.473	0.710	0.947	1.183	1.420	1.656	1.893	2.129	2.366
40	0.285	0.570	0.855	1.140	1.424	1.709	1.994	2.279	2.564	2.749
45	0.341	0.683	1.024	1.366	1.707	2.048	2.390	2.731	3.073	3.414
50	0.408	0.815	1.223	1.630	2.038	2.446	2.853	3.261	3.668	4.076
55	0.485	0.970	1.455	1.940	2.424	2.909	3.394	3.879	4.364	4.849
60	0.574	1.149	1.724	2.298	2.872	3.447	4.022	4.596	5.170	5.745
62	0.614	1.228	1.843	2.457	3.071	3.685	4.299	4.914	5.528	6.142
64	0.656	1.313	1.969	2.625	3.282	3.938	4.594	5.250	5.907	6.563
66	0.701	1.402	2.103	2.804	3.504	4.205	4.906	5.607	6.208	7.009
68	0.748	1.496	2.244	2.992	3.740	4.488	5.236	5.974	7.732	7.480
70	0.798	1.596	2.394	3.192	3.990	5.788	5.586	6.384	7.182	7.980
72	0.851	1.702	2.552	3.403	4.254	5.105	5.956	6.806	7.657	8.508
74	0.907	1.813	2.720	3.626	4.553	5.440	6.346	7.523	8.159	9.066
76	0.966	1.931	2.896	3.862	4.828	5.793	6.758	7.724	8.690	9.655
78	1.028	2.055	3.083	4.111	5.138	6.166	7.194	8.222	9.249	10.277
80	1.093	2.187	3.280	4.374	5.467	6.560	7.654	8.747	9.841	10.934
82	1.163	2.325	3.488	4.650	5.813	6.976	8.138	9.301	10.463	11.626
84	1.236	2.471	3.707	4.942	6.178	7.414	8.649	9.885	11.120	12.356
86	1.313	2.625	3.938	5.251	6.564	7.877	9.189	10.502	11.814	13.127
88	1.394	2.787	4.181	5.575	6.968	8.362	9.576	11.150	12.543	13.937
90	1.479	2.958	4.437	5.916	7.395	8.874	10.353	11.832	13.311	14.790
92	1.569	3.138	4.707	7.276	7.844	9.413	10.982	12.551	13.120	15.689
94	1.663	3.327	4.990	6.654	8.317	9.980	11.644	13.307	14.971	16.634
96	1.763	3.525	5.288	7.050	8.813	10.576	12.338	14.101	15.863	17.626
98	1.867	3.734	5.601	7.468	9.336	11.203	13.070	14.937	16.804	18.671
100	1.977	3.953	5.930	7.906	9.883	11.860	13.836	15.813	17.789	19.766

Table 8.28 Atmospheric Pressure and Barometer Readings at Different Altitudes

Altitude above Sea Level, ft	Atmospheric Pressure, Psi	Barometer Reading, In. Hg	Altitude above Sea Level, ft	Atmospheric Pressure, Psi	Barometer Reading, In. Hg
0	14.69	29.92	7,500	11.12	22.65
500	14.42	29.38	8,000	10.91	22.22
1,000	14.16	28.86	8,500	10.70	21.80
1,500	13.91	28.33	9,000	10.50	21.38
2,000	13.66	27.82	9,500	10.30	20.98
2,500	13.41	27.31	10,000	10.10	20.58
3,000	13.16	26.81	10,500	9.90	20.18
3,500	12.92	26.32	11,000	9.71	19.75
4,000	12.68	25.84	11,500	9.52	19.40
4,500	12.45	25.36	12,000	9.34	19.03
5,000	12.22	24.89	12,500	9.15	18.65
5,500	11.99	24.43	13,000	8.97	18.29
6,000	11.77	23.98	13,500	8.80	17.93
6,500	11.55	23.53	14,000	8.62	17.57
7,000	11.33	23.09	14,500	8.45	17.22
			15,000	8.28	16.88

TABLE 8.29 Average Gas Compositions Composition, Per Cent

		Composition, Per Cent							
Gas	By	H_2	CO	CO_2	CH_4	C_2H_6	O_2	N_2	Illuminating
Air (dry)	Volume	. . . .	. . . .	. . . .	. . . .	. . . .	21.0	79.0	
	Weight	. . . .	. . . .	. . . .	. . . .	.. . . .	23.2	76.8	
Natural gas	Volume	. . . . .	. . . .	. . . .	85.0	14.0	. . . .	1.00	
	Weight	. . . .	. . . .	. . . .	75.20	23.25	. . . .	1.55	
Blast-furnace gas	Volume	3.00	25.00	12.00	2.00	. . . .	. . . .	58.00	
	Weight	0.21	24.22	18.27	1.11	. . . .	. . . .	56.19	
Coke-oven gas	Volume	56.00	6.00	1.50	30.00	0.50	0.50	2.50	3.00
	Weight	11.27	14.90	6.64	48.32	1.51	1.61	4.28	11.47
Illuminating gas	Volume	35.00	25.00	5.50	12.50	2.50	0.50	5.00	14.00
	Weight	3.59	35.95	12.43	10.27	3.85	0.82	7.20	25.89

At 32°F and 29.92 in. Hg.

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TABLE 8.30 K Value and Properties of Various Gases at 60°F and 14.7 Pounds Absolute*

Gas	Symbol	*Cp/Cv* = k	Sp. Gr. Air 1.00	Mol. Wt.	Lb Per Cu ft	Cu ft Per Lb	Boiling Point at Atmos. Press. Deg F	Crit. Temp., Deg F	Crit. Pressure, Lb Abs
Acetylene	C_2H_2	1.3	0.9073	26.0156	0.06880	14.534	- 118	96	910
Air	. . . .	1.395	1.000	28.9752	0.07658	13.059	- 317	- 221	546
Ammonia	NH_3	1.317	0.5888	17.0314	0.04509	22,178	- 28	270	1,638
Argon	A	1.667	1.379	39.944	0.10565	0.467	- 302	- 187	705
Benzene	C_6H_6	1.08	2.6935	78.0468	0.20640	4.845	176	551	700
Butane	C_4H_{10}	1.11	2.067	58.078	0.15350	6.514	31	307	528
Butylene	C_4H_8	1.11	1.9353	56.0624	0.14826	6.7452	21	291	621
Carbon dioxide	CO_2	1.30	1.529	44.000	0.11637	8.593	- 109	88	1,072
Carbon disulfide	CS_2	1.20	2.6298	76.120	0.20139	4.965	115	523	1,116
Carbon monoxide	CO	1.403	0.9672	28.000	0.07407	13.503	- 313	- 218	514
Carbon tetrachloride	CCl_4	1.18	5.332	153.828	0.40650	2.4601	170	541	661
Carbureted water gas	. . . .	1.35	0.4090						
Chlorine	Cl_2	1.33	2.486	70.914	0.18750	5.333	-30	291	118
Dichloromethane	CH_2Cl_2	1.18	3.005	84.9296	0.22450	4.458	105	421	1,490
Ethane	C_2H_6	1.22	1.049	30.0468	0.07940	12.594	- 127	90	717
Ethyl chloride	C_2H_5Cl	1.13	2.365	64.4960	0.17058	5.866	54	370	764
Ethylene	C_2H_4	1.22	0.9748	28.0312	0.07410	13.495	- 155	50	747
Flue gas	. . . .	1.40							
Freon (F-12)	CCl_2F_2	1.13	4.520	120.9140	0.31960	3.129	- 21	233	580
Helium	He	1.66	0.1381	4.002	0.01058	94.510	- 452	- 450	33
Hexane	C_6H_{14}	1.08	2.7395	86.1092	0.22760	4.393	156	454	433
Hexylene	C_6H_{12}	. . . .	2.9201	84.0936	0.22250	4.4951			
Hydrogen	H_2	1.41	0.06952	2.0156	0.00530	188.62	- 423	- 400	188
Hydrogen chloride	HCl	1.41	1.268	36.4648	0.09650	10.371	- 121	124	1,198
Hydrogen sulfide	H_2S	1.30	1.190	34.0756	0.09012	11.096	- 75	212	1,306
Isobutane	C_4H_{10}	1.11	2.0176	58.078	0.15365	6.5135	14	273	543
Isopentane	C_5H_{12}	. . . .	2.5035	72.0936	0.19063	5.2451			
Methane	CH_4	1.316	0.5544	16.0312	0.04234	23.626	- 258	- 116	672
Methyl chloride	CH_3Cl	1.20	1.785	50.4804	0.13365	7.491	- 11	289	966
Naphthalene	$C_{10}H_8$	. . . .	4.423	128.0624	0.33870	2.952			
Natural gas† (app. avg.)	. . . .	1.269	0.6655	19.463	0.05140	19.451	. . . .	- 80	670
Neon	Ne	1.642	0.6961	20.183	0.05332	18.784	- 410	- 380	389
Nitric oxide	NO	1.40	1.037	30.008	0.07935	12.605	- 240	- 137	954
Nitrogen	N_2	1.40	0.9672	28.016	0.07429	13.460	-320	- 232	492
Nitrous oxide	N_2O	1.311	1.530	44.016	0.11632	8.595	- 129	98	1,053
Oxygen	O_2	1.398	1.105	32.000	0.08463	11.816	- 297	- 182	730
Pentane	C_5H_{12}	1.06	2.471	72.0936	0.19055	5.248	97	387	485
Phenol	C_6J_5OH	. . . .	3.2655	94.0486	0.24870	4.022	360	786	889
Propane	C_3H_8	1.15	1.562	44.0624	0.11645	8.587	- 48	204	632
Propylene	C_3H_6	. . . .	1.4505	42.0468	0.11115	8.997	- 52	198	661
Refinery gas† (app. avg.)	. . . .	1.20							
Sulfur oxide	SO_2	1.256	2.264	64.060	0.16945	5.901	14	315	1,141
Water vapor (steam)	H_2O	1.33‡	0.6217	18.0156	0.04761	21.004	212	706	3,206

* From "Plain Talks on Air and Gas Compression."

† To obtain exact characteristics of natural gas and refinery gas, the exact constituents must be known.

‡ This k value is given at 212ºF. All others are at 60ºF. Authorities differ slightly; hence above data are average results.

Table 8.31 Dynamic Viscosity of Gases at Atmospheric Pressure

	50F	100F	150F	200F	300F	400F	500F
Air	.37	.40	.42	.44	.49	.53	.56
Carbon Dioxide	.29	.31	.33	.36	.40	.45	.50
CO and N_2	.36	.38	.40	.43	.48	.52	.56
Ethane	.19	.20	.22	.23	.26	.29	.32
Hydrogen	.20	.20	.215	.23	.25	.27	.28
Methane	.23	.24	.25	.27	.29	.32	.35
Oxygen	.41	.44	.46	.49	.53	.58	.63

Slugs per ft. sec. x 10^6.

Table 8.32 Approximate Coefficient of Friction for Clean Commercial Iron and Steel Pipe, Steady Flow

Reynold's Number	Coefficient of Friction
20,000	.0077
50,000	.0063
100,000	.0055
200,000	.0048
500,000	.0041
1,000,000	.0036
5,000,000	.0032
10,000,000	.0026

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TABLE 8.33 Number of Tools That Can Be Operated by One Compressor

	Compressor Capacity, Cu Ft Per Min																	
	75		85		125		150		250		365		600		900		1200	
	Air Pressure, Psi																	
	70	90	70	90	70	90	70	90	70	90	70	90	70	90	70	90	70	90
Rock drills:																		
Very light, wet or dry	2	2	3	2	4	3	5	4	8	7	11	10	19	16	30	24	38	32
Very light, blower style	2	1	2	1	3	2	4	3	6	5	9	8	16	14	24	21	32	38
Light, wet or dry	1	1	1	1	2	1	3	2	4	3	6	5	10	8	15	12	20	16
Light, blower style	1	1	1	1	1	1	2	1	3	2	5	4	9	7	13	11	18	14
Medium, wet or dry	1	..	1	..	1	1	2	1	3	2	5	3	7	6	10	9	14	12
Medium, blower style	1	..	1	..	1	1	2	1	3	2	4	3	7	6	10	9	14	12
Heavy, blower	..	..	1	..	1	1	1	1	2	2	4	3	6	4	9	7	12	9
Wagon & crawler drills:																		
Lightweight	..	..	..	..	..	..	1	..	1	1	2	1	3	2	4	3	6	4
Medium-weight	..	..	..	..	..	..	..	..	1	..	1	1	2	1	3	2	4	3
Heavyweight	..	..	..	..	..	..	..	..	..	..	1	..	1	1	2	1	2	2
Paving breaker:																		
Light, horizontal and light work	2	1	2	1	3	2	4	3	7	5	10	8	15	13	22	20	30	26
Medium, light and general work	1	1	1	1	2	2	3	2	5	4	8	6	14	10	21	15	27	20
Heavy, general work	1	1	1	1	2	1	2	2	4	3	6	5	10	7	15	11	20	16
Heavy, concrete breaking	1	1	1	1	2	1	2	2	4	3	6	5	10	8	15	12	20	16
Heavy, all-purpose	1	1	1	1	2	1	2	2	4	3	6	5	10	8	15	12	20	16
Sheeting drivers:																		
Light, gravel	1	1	2	1	2	1	4	2	6	4	10	6	14	12	21	18	28	24
Light, general	1	1	2	1	2	1	4	2	6	4	10	6	14	12	21	18	28	24
Light, stiff ground	1	1	2	2	3	2	4	3	7	5	13	8	15	14	22	21	30	28
Drill-steel cutter	..	..	1	1	1	1	2	2	3	2	4	3	*	*	*	*	*	*
Sharpeners:																		
Lightweight	1	1	2	2	3	3	4	4	6	6	*	*	*	*	*	*	*	*
Medium-weight	1	1	1	1	1	1	2	2	3	3	*	*	*	*	*	*	*	*
Heavyweight, high production	..	..	..	..	..	..	1	1	1	1	2	2	4	4	6	6	8	8
Heavyweight, greater production	..	..	..	..	..	..	1	1	1	1	2	2	3	3	4	4	6	6
Furnaces:																		
Light	3	3	5	5	6	6	10	10	*	*	*	*	*	*	*	*	*	*
Heavy	2	2	4	4	5	5	8	8	*	*	*	*	*	*	*	*	*	*
Heating bits, rods and steels	2	2	4	4	5	5	8	8	*	*	*	*	*	*	*	*	*	*
Grinder for detachable rock-drill bits	1	1	1	1	2	2	3	2	5	4	7	6	12	10	18	15	24	20
Diggers:																		
Light, medium-duty and trench digger	2	1	3	2	4	3	5	4	8	5	10	9	16	13	24	20	32	27
Medium- and heavy-duty	2	1	2	1	3	2	4	3	7	5	10	8	16	13	24	20	32	27
Heavy-duty	1	1	1	1	2	2	3	3	5	4	7	7	12	11	17	15	27	25
Riveting hammers:																		
4-in. stroke, ⅜-in. capacity	3	2	4	3	6	5	9	7	13	11	24	18	*	*	*	*	*	*
5-in. stroke, ¾ in. capacity	3	2	4	3	6	5	8	6	12	10	20	15	*	*	*	*	*	*
6-in. stroke, ⅞-in. capacity	2	2	3	2	5	4	8	5	11	9	19	14	*	*	*	*	*	*
8-in. stroke, 1⅛-in. capacity	2	2	3	2	5	4	7	5	10	8	18	14	*	*	*	*	*	*
9-in. stroke, 1¼-in. capacity	2	1	3	2	4	3	7	5	9	7	17	13	*	*	*	*	*	*

TABLE 8.33 continued

	Compressor Capacity, Cu Ft Per Min																	
	75		85		125		150		250		365		600		900		1200	
	Air Pressure, Psi																	
	70	90	70	90	70	90	70	90	70	90	70	90	70	90	70	90	70	90
Jam rivets, ⅞-in. capacity	1	1	1	1	2	1	3	2	4	3	7	5	*	*	*	*	*	*
Short jam rivets, 1⅛-in. capacity	1	1	1	1	2	1	3	2	4	3	7	5	*	*	*	*	*	*
Comp. jam rivets, ¾ in.	4	2	4	3	6	5	9	6	14	10	18	12	*	*	*	*	*	*
Rivet buster up to ⅞-in. cap.	2	1	3	2	4	4	6	5	8	7	14	11	*	*	*	*	*	*
Caulking and chipping hammers:																		
1-in. stroke, medium chipping	5	3	6	4	9	6	10	7	18	11	26	17	42	28	64	47	85	57
2-in. stroke, medium and heavy chipping	5	3	6	4	9	6	10	7	17	11	26	17	42	28	64	47	85	57
3-in. stroke, heavy chipping	5	3	6	4	9	6	10	7	17	11	26	17	42	28	64	47	85	57
3-in. stroke, general chipping	5	3	6	4	9	6	10	7	17	11	26	17	42	28	64	47	85	57
Light caulking and scaling	5	3	6	4	9	6	10	7	17	11	26	17	42	28	64	47	85	57
Light and medium caulking and scaling	5	3	6	4	9	6	10	7	17	11	26	17	42	28	64	47	85	57
Scaling tools:																		
Light-duty and boiler scaling tool	11	9	13	10	19	15	23	18	38	31	56	45	*	*	*	*	*	*
Heavy-duty scaling hammer	8	7	10	8	15	12	18	15	31	25	45	36	*	*	*	*	*	*
Grinders:																		
Die grinder	6	5	7	5	10	8	12	10	20	16	30	24	*	*	*	*	*	*
Grinder 2-in. diameter wheel cap	2	2	2	2	4	3	5	4	8	6	12	10	20	16	*	*	*	*
Grinder 6-in. diameter wheel cap	1	1	2	1	3	2	3	3	6	5	9	7	15	12	*	*	*	*
Grinder 8-in. diameter wheel cap	1	1	1	1	2	1	2	2	4	3	6	5	11	9	17	13	*	*
Wire brushing machine, 8-in. radial brush cap	1	1	2	1	3	2	3	3	6	5	9	7	15	12	*	*	*	*
For squaring shanks on rock-drill steels	2	2	3	2	4	3	4	4	9	7	15	11	*	*	*	*	*	*
Grinders:																		
5-in. cup wheel or 7-in. sanding pad cap	1	1	2	1	3	2	3	3	6	5	9	7	15	12	*	*	*	*
6-in. cup wheel or 9-in. sanding pad cap	1	1	1	1	2	1	2	2	4	3	6	5	11	8	*	*	*	*
Drills, drilling, reaming, tapping:																		
Heavy-duty drilling machine up to ⅜-in. cap	3	2	3	3	5	4	6	5	11	8	16	13	*	*	*	*	*	*
Drilling and reaming up to ⅞-in.	1	1	2	1	3	2	3	3	6	5	9	7	*	*	*	*	*	*
Close-quarter drilling and reaming machine up to 1 1/16-in.	2	2	3	2	5	4	7	5	11	9	18	14	*	*	*	*	*	*
Machines up to 1 ¼ in.	..	..	..	..	..	..	..	..	6	5	8	7	*	*	*	*	*	*
Machines up to 2 in.	..	..	..	..	..	..	..	..	3	2	5	3	*	*	*	*	*	*
Drilling and reaming machines up to 1 ¼ in. cap.	1	1	1	1	2	2	3	2	4	3	8	5	*	*	*	*	*	*
Machines up to 2-in. cap.	..	..	..	..	1	1	2	1	3	2	5	4	*	*	*	*	*	*

TABLE 8.33 continued

	Compressor Capacity, Cu Ft Per Min																	
	75		85		125		150		250		365		600		900		1200	
	Air Pressure, Psi																	
	70	90	70	90	70	90	70	90	70	90	70	90	70	90	70	90	70	90
Wood borers:																		
Wood-boring machines up to 1-in. cap.	1	1	2	1	3	2	3	3	6	5	9	7	15	12	*	*	*	*
Up to 4-in cap.	2	2	3	2	5	4	7	5	10	8	16	13	*	*	*	*	*	*
Torque wrenches:																		
Right-angle nut running machines up to ¾ in. bolt cap. torque type	1	1	2	1	3	2	3	3	6	5	9	7	15	12	22	18	30	24
Reversible nut running machine, impact type:																		
Capacity up to ⅜-in, bolt size nuts	5	4	5	4	8	6	10	8	17	13	25	20	41	33	62	50	83	66
Up to ⅝ in.	3	2	3	2	5	4	6	5	10	8	15	12	25	20	37	30	50	40
Up to ¾ in.	2	2	3	2	4	3	5	4	9	7	13	10	21	17	32	26	43	35
Up to 1 ¼ in.	2	1	2	2	4	3	4	3	8	6	11	9	19	15	29	23	38	31
Up to 1 ¾ in.	..	..	1	1	1	1	3	2	3	3	6	4	8	6	12	9	18	14
Port. circ. saw, 12-in. blade	2	1	2	2	4	3	6	4	9	7	16	12	*	*	*	*	*	*
Concrete vibrators:																		
Light-duty concrete vibrator	2	2	3	2	6	4	9	7	14	10	22	17	*	*	*	*	*	*
Medium, heavy-duty	1	..	1	1	2	1	3	2	5	3	8	6	*	*	*	*	*	*
Port. sing. drum hoists:																		
Capacity up to 2,000 lb pull, slow rope speed on single line	..	..	..	1†	..	1†	..	2†	..	3†	..	6†	..	7†	..	9†	..	14†
High rope speed	..	..	..	..	..	..	..	..	..	1†	..	2†	..	2†	..	3†	..	4†
Up to 3,500 lb	..	..	..	..	..	..	..	..	..	..	..	1†	..	1†	..	2†	..	3†
Heavy-duty backfill-tamping machine	1	1	3	2	5	3	5	4	9	7	11	9	14	11	21	16	28	22
Sump pump:																		
Portable centrifugal-type, low heads	1	..	1	..	2	1	2	2	3	2	5	4	9	7	14	11	19	15
Two tandem connected pumps, higher heads	..	..	..	..	..	..	1	1	2	1	2	1	3	3	5	4	8	6
Portable centrifugal-type, high heads	..	..	..	..	..	..	1	1	1	1	1	1	2	2	3	3	4	4

* Compressor capacity more than that needed for usual number of tools.
† Hoist figures based on 80-lb pressure.
This table is based upon a factor of intermittent use, that is, that all tools are not operated simultaneously. Many conditions may develop in which more or fewer tools may be operated than shown above.

GLOSSARY

Absolute Pressure – Total pressure measured from zero.

Absolute Temperature – See **Temperature, Absolute.**

Absorption – The chemical process by which a hygroscopic desiccant, having a high affinity with water, melts and becomes a liquid by absorbing the condensed moisture.

Actual Capacity – Quantity of gas actually compressed and delivered to the discharge system at rated speed and under rated conditions. Also called Free Air Delivered (FAD).

Adiabatic Compression – See **Compression, Adiabatic.**

Adsorption – The process by which a desiccant with a highly porous surface attracts and removes the moisture from compressed air. The desiccant is capable of being regenerated.

Air Receiver – See **Receiver.**

Air Bearings – See **Gas Bearings.**

Aftercooler – A heat exchanger used for cooling air discharged from a compressor. Resulting condensate may be removed by a moisture separator following the aftercooler.

Atmospheric Pressure – The measured ambient pressure for a specific location and altitude.

Automatic Sequencer – A device which operates compressors in sequence according to a programmed schedule.

Brake Horsepower (bhp) – See **Horsepower, Brake.**

Capacity – The amount of air flow delivered under specific conditions, usually expressed in cubic feet per minute (cfm).

Capacity, Actual – The actual volume flow rate of air or gas compressed and delivered from a compressor running at its rated operating conditions of speed, pressures, and temperatures. Actual capacity is generally expressed in actual cubic feet per minute (acfm) at conditions prevailing at the compressor inlet.

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Capacity Gauge – A gauge that measures air flow as a percentage of capacity, used in rotary screw compressors

Check Valve – A valve which permits flow in only one direction.

Clearance – The maximum cylinder volume on the working side of the piston minus the displacement volume per stroke. Normally it is expressed as a percentage of the displacement volume.

Clearance Pocket – An auxiliary volume that may be opened to the clearance space, to increase the clearance, usually temporarily, to reduce the volumetric efficiency of a reciprocating compressor.

Compressibility – A factor expressing the deviation of a gas from the laws of thermodynamics. (See also **Supercompressibility**)

Compression, Adiabatic – Compression in which no heat is transferred to or from the gas during the compression process.

Compression, Isothermal – Compression in which the temperature of the gas remains constant.

Compression, Polytropic – Compression in which the relationship between the pressure and the volume is expressed by the equation PV^n is a constant.

Compression Ratio – The ratio of the absolute discharge pressure to the absolute inlet pressure.

Constant Speed Control – A system in which the compressor is run continuously and matches air supply to air demand by varying compressor load.

Critical Pressure – The limiting value of saturation pressure as the saturation temperature approaches the critical temperature.

Critical Temperature – The highest temperature at which well-defined liquid and vapor states exist. Sometimes it is defined as the highest temperature at which it is possible to liquify a gas by pressure alone.

Cubic Feet Per Minute (cfm) – Volumetric air flow rate.

cfm, Free Air – cfm of air delivered to a certain point at a certain condition, converted back to ambient conditions.

Actual cfm (acfm) – Flow rate of air at a certain point at a certain condition at that point.

Inlet cfm (icfm) – Cfm flowing through the compressor inlet filter or inlet valve under rated conditions.

Standard cfm – Flow of free air measured and converted to a standard set of reference conditions (14.5 psia, 68^{O}F, and 0% relative humidity).

Cut-In/Cut-Out Pressure – Respectively, the minimum and maximum discharge pressures at which the compressor will switch from unload to load operation (cut in) or from load to unload (cut out).

Cycle – The series of steps that a compressor with unloading performs; 1) fully loaded, 2) modulating (for compressors with modulating control), 3) unloaded, 4) idle.

Cycle Time – Amount of time for a compressor to complete one cycle.

Degree of Intercooling – The difference in air or gas temperature between the outlet of the intercooler and the inlet of the compressor.

Deliquescent – Melting and becoming a liquid by absorbing moisture.

Desiccant – A material having a large proportion of surface pores, capable of attracting and removing water vapor from the air.

Dew Point – The temperature at which moisture in the air will begin to condense if the air is cooled at constant pressure. At this point the relative humidity is 100%.

Demand – Flow of air at specific conditions required at a point or by the overall facility.

Diaphragm – A stationary element between the stages of a multi-stage centrifugal compressor. It may include guide vanes for directing the flowing medium to the impeller of the succeeding stage. In conjunction with an adjacent diaphragm, it forms the diffuser surrounding the impeller.

Diaphragm cooling – A method of removing heat from the flowing medium by circulation of a coolant in passages built into the diaphragm.

Diffuser – A stationary passage surrounding an impeller, in which velocity pressure imparted to the flowing medium by the impeller is converted into static pressure.

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Digital Controls – See **Logic Controls.**

Discharge Pressure – Air pressure produced at a particular point in the system under specific conditions.

Discharge Temperature – The temperature at the discharge flange of the compressor.

Displacement – The volume swept out by the piston or rotor(s) per unit of time, normally expressed in cubic feet per minute.

Droop – The drop in pressure at the outlet of a pressure regulator, when a demand for air occurs.

Dynamic Type Compressors – Compressors in which air or gas is compressed by the mechanical action of rotating impellers imparting velocity and pressure to a continuously flowing medium. (Can be centrifugal or axial design.)

Efficiency – Any reference to efficiency must be accompanied by a qualifying statement which identifies the efficiency under consideration, as in the following definitions of efficiency:

> **Efficiency, Compression –** Ratio of theoretical power to power actually imparted to the air or gas delivered by the compressor.
>
> **Efficiency, Isothermal –** Ratio of the theoretical work (as calculated on a isothermal basis) to the actual work transferred to a gas during compression.
>
> **Efficiency, Mechanical –** Ratio of power imparted to the air or gas to brake horsepower (bhp).
>
> **Efficiency, Polytropic –** Ratio of the polytropic compression energy transferred to the gas, to the actual energy transferred to the gas.
>
> **Efficiency, Volumetric –** Ratio of actual capacity to piston displacement.

Exhauster – A term sometimes applied to a compressor in which the inlet pressure is less than atmospheric pressure.

Expanders – Turbines or engines in which a gas expands, doing work, and undergoing a drop in temperature. Use of the term usually implies that the drop in temperature is the principle objective. The orifice in a refrigeration system also performs this function, but the expander performs it more nearly isentropically, and thus is more effective in cryogenic systems.

Filters – Devices for separating and removing particulate matter, moisture or entrained lubricant from air.

Flange connection – The means of connecting a compressor inlet or discharge connection to piping by means of bolted rims (flanges).

Fluidics – The general subject of instruments and controls dependent upon low rate of flow of air or gas at low pressure as the operating medium. These usually have no moving parts.

Free Air – Air at atmospheric conditions at any specified location, unaffected by the compressor.

Full-Load – Air compressor operation at full speed with a fully open inlet and discharge delivering maximum air flow.

Gas – One of the three basic phases of matter. While air is a gas, in pneumatics the term gas normally is applied to gases other than air.

Gas Bearings – Load carrying machine elements permitting some degree of motion in which the lubricant is air or some other gas.

Gauge Pressure – The pressure determined by most instruments and gauges, usually expressed in psig. Barometric pressure must be considered to obtain true or absolute pressure.

Guide Vane – A stationary element that may be adjustable and which directs the flowing medium approaching the inlet of an impeller.

Head, Adiabatic – The energy, in foot pounds, required to compress adiabatically to deliver one pound of a given gas from one pressure level to another.

Head, Polytropic – The energy, in foot pounds, required to compress polytropically to deliver one pound of a given gas from one pressure level to another.

Horsepower, Brake – Horsepower delivered to the output shaft of a motor or engine, or the horsepower required at the compressor shaft to perform work.

Horsepower, Indicated – The horsepower calculated from compressor indicator diagrams. The term applies only to displacement type compressors.

Horsepower, Theoretical or Ideal – The horsepower required to isothermally compress the air or gas delivered by the compressor at specified conditions.

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Humidity, Relative – The relative humidity of a gas (or air) vapor mixture is the ratio of the partial pressure of the vapor to the vapor saturation pressure at the dry bulb temperature of the mixture.

Humidity, Specific – The weight of water vapor in an air vapor mixture per pound of dry air.

Hysteresis – The time lag in responding to a demand for air from a pressure regulator.

Impeller – The part of the rotating element of a dynamic compressor which imparts energy to the flowing medium by means of centrifugal force. It consists of a number of blades which rotate with the shaft.

Indicated Power – Power as calculated from compressor-indicator diagrams.

Indicator Card – A pressure – volume diagram for a compressor or engine cylinder, produced by direct measurement by a device called an indicator.

Inducer – A curved inlet section of an impeller.

Inlet Pressure – The actual pressure at the inlet flange of the compressor.

Intercooling – The removal of heat from air or gas between compressor stages.

Intercooling, Degree of – The difference in air or gas temperatures between the inlet of the compressor and the outlet of the intercooler.

Intercooling, Perfect – When the temperature of the air or gas leaving the intercooler is equal to the temperature of the air or gas entering the inlet of the compressor.

Isentropic Compression – See **Compression, Isentropic.**

Isothermal Compression – See **Compression, Isothermal.**

Leak – An unintended loss of compressed air to ambient conditions.

Liquid Piston Compressor – A compressor in which a vaned rotor revolves in an elliptical stator, with the spaces between the rotor and stator sealed by a ring of liquid rotating with the impeller.

Load Factor – Ratio of average compressor load to the maximum rated compressor load over a given period of time.

Load Time – Time period from when a compressor loads until it unloads.

Load/Unload Control – Control method that allows the compressor to run at full-load or at no load while the driver remains at a constant speed.

Modulating Control – System which adapts to varying demand by throttling the compressor inlet proportionally to the demand.

Multi-casing Compressor – Two or more compressors, each with a separate casing, driven by a single driver, forming a single unit.

Multi-stage Axial Compressor – A dynamic compressor having two or more rows of rotating elements operating in series on a single rotor and in a single casing.

Multi-stage Centrifugal Compressor – A dynamic compressor having two or more impellers operating in series in a single casing.

Multi-stage Compressors – Compressors having two or more stages operating in series.

Perfect Intercooling – The condition when the temperature of air leaving the intercooler equals the of air at the compressor intake.

Performance Curve – Usually a plot of discharge pressure versus inlet capacity and shaft horsepower versus inlet capacity.

Piston Displacement – The volume swept by the piston; for multi-stage compressors, the piston displacement of the first stage is the overall piston displacement of the entire unit.

Pneumatic Tools – Tools that operate by air pressure.

Polytropic Compression – See **Compression, Polytropic.**

Polytropic Head – See **Head, Polytropic.**

Positive Displacement Compressors – Compressors in which successive volumes of air or gas are confined within a closed space, and the space mechanically reduced, results in compression. These may be reciprocating or rotating.

Power, theoretical (polytropic) – The mechanical power required to compress polytropically and to deliver, through the specified range of pressures, the gas delivered by the compressor.

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Pressure – Force per unit area, measured in pounds per square inch (psi).

Pressure, Absolute – The total pressure measured from absolute zero (i.e. from an absolute vacuum).

Pressure, Critical – See **Critical Pressure.**

Pressure Dew Point – For a given pressure, the temperature at which water will begin to condense out of air.

Pressure, Discharge – The pressure at the discharge connection of a compressor. (In the case of compressor packages, this should be at the discharge connection of the package)

Pressure Drop – Loss of pressure in a compressed air system or component due to friction or restriction.

Pressure, Intake – The absolute total pressure at the inlet connection of a compressor.

Pressure Range – Difference between minimum and maximum pressures for an air compressor. Also called cut in-cut out or load-no load pressure range.

Pressure Ratio – See **Compression Ratio.**

Pressure Rise – The difference between discharge pressure and intake pressure.

Pressure, Static – The pressure measured in a flowing stream in such a manner that the velocity of the stream has no effect on the measurement.

Pressure, Total – The pressure that would be produced by stopping a moving stream of liquid or gas. It is the pressure measured by an impact tube.

Pressure, Velocity – The total pressure minus the static pressure in an air or gas stream.

Rated Capacity – Volume rate of air flow at rated pressure at a specific point.

Rated Pressure – The operating pressure at which compressor performance is measured.

Required Capacity – Cubic feet per minute (cfm) of air required at the inlet to the distribution system.

Receiver – A vessel or tank used for storage of gas under pressure. In a large compressed air system there may be primary and secondary receivers.

Reciprocating Compressor – Compressor in which the compressing element is a piston having a reciprocating motion in a cylinder.

Relative Humidity – The ratio of the partial pressure of a vapor to the vapor saturation pressure at the dry bulb temperature of a mixture.

Reynolds Number – A dimensionless flow parameter ($\vartheta v \rho / \mu$), in which ϑ is a significant dimension, often a diameter, v is the fluid velocity, ρ is the mass density, and μ is the dynamic viscosity, all in consistent units.

Rotor – The rotating element of a compressor. In a dynamic compressor, it is composed of the impeller(s) and shaft, and may include shaft sleeves and a thrust balancing device.

Seals – Devices used to separate and minimize leakage between areas of unequal pressure.

Sequence – The order in which compressors are brought online.

Shaft – The part by which energy is transmitted from the prime mover through the elements mounted on it, to the air or gas being compressed.

Sole Plate – A pad, usually metallic and embedded in concrete, on which the compressor and driver are mounted.

Specific Gravity – The ratio of the specific weight of air or gas to that of dry air at the same pressure and temperature.

Specific Humidity – The weight of water vapor in an air-vapor mixture per pound of dry air.

Specific Power – A measure of air compressor efficiency, usually in the form of bhp/100 acfm.

Specific Weight – Weight of air or gas per unit volume.

Speed – The speed of a compressor refers to the number of revolutions per minute (rpm) of the compressor drive shaft or rotor shaft.

Stages – A series of steps in the compression of air or a gas.

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Standard Air – The Compressed Air & Gas Institute and PNEUROP have adopted the definition used in ISO standards. This is air at 14.5 psia (1 bar); 68˚F (20˚C) and dry (0% relative humidity).

Start/Stop Control – A system in which air supply is matched to demand by the starting and stopping of the unit.

Supercompressibility – See **Compressibility.**

Surge – A phenomenon in centrifugal compressors where a reduced flow rate results in a flow reversal and unstable operation.

Surge Limit – The capacity in a dynamic compressor below which operation becomes unstable.

Temperature, Absolute – The temperature of air or gas measured from absolute zero. It is the Fahrenheit temperature plus 459.6 and is known as the Rankine temperature. In the metric system, the absolute temperature is the Centigrade temperature plus 273 and is known as the Kelvin temperature.

Temperature, Critical – See **Critical Temperature.**

Temperature, Discharge – The total temperature at the discharge connection of the compressor.

Temperature, Inlet – The total temperature at the inlet connection of the compressor.

Temperature Rise Ratio – The ratio of the computed isentropic temperature rise to the measured total temperature rise during compression. For a perfect gas, this is equal to the ratio of the isentropic enthalpy rise to the actual enthalpy rise.

Temperature, Static – The actual temperature of a moving gas stream. It is the temperature indicated by a thermometer moving in the stream and at the same velocity.

Temperature, Total – The temperature which would be measured at the stagnation point if a gas stream were stopped, with adiabatic compression from the flow condition to the stagnation pressure.

Theoretical Power – The power required to compress a gas isothermally through a specified range of pressures.

Torque – A torsional moment or couple. This term typically refers to the driving couple of a machine or motor.

Total Package Input Power – The total electrical power input to a compressor, including drive motor, belt losses, cooling fan motors, VSD or other controls, etc.

Unit Type Compressors – Compressors of 30 bhp or less, generally combined with all components required for operation.

Unload – (No load) Compressor operation in which no air is delivered due to the intake being closed or modified not to allow inlet air to be trapped.

Vacuum Pumps – Compressors which operate with an intake pressure below atmospheric pressure and which discharge to atmospheric pressure or slightly higher.

Valves – Devices with passages for directing flow into alternate paths or to prevent flow.

Volute – A stationary, spiral shaped passage which converts velocity head to pressure in a flowing stream of air or gas.

Water-cooled Compressor – Compressors cooled by water circulated through jackets surrounding cylinders or casings and/or heat exchangers between and after stages.

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TABLE 8.34 Preferred SI (Metric) Units

Quantity	Unit	Symbol	Metric-U.S. Customary Unit Equivalents	Remarks
LENGTH	meter	m	1 m = 1000 mm = 39.37 in. = 3.281 ft.	Use mm for dimensions on product engineering drawings, µm for surface finish, clearance & vibration amplitude.
	millimeter	mm	25.4 mm = 1 inch	
	micrometer	µm	1 µm = 10^{-6} m	
			25.4 µm = 1 mil = .001 inch	
AREA	square meter	m^2	1 m^2 = 10.764 $ft.^2$	
	square millimeter	mm^2	645.16 mm^2 = 1 $inch^2$	
			100 m^2 = 119.6 yd. $inch^2$ 10,000 m^2 = 2.47 acres	
MASS	kilogram	kg	1 kg =2.205 lb_m	
VOLUME FLOW RATE-GASES	cubic meter per second	m^3/s	1 m^3/s = 2118.9 ft^3/min	Allow time to vary to provide suitable numbers.
	cubic meter per minute	m^3/min	1 m^3/min = 35.315 ft^3/min	
VOLUME FLOW RATE-LIQUIDS	liter per second	L/s	1 L/sec = 15.85 gpm	Allow time to vary to provide suitable numbers.
	liter per minute	L/min	1 L/min = .2642 gpm	
PRESSURE	bar	bar	1 bar = 14.5 psi (lb_f/lb^2 = 100 kPa)	bar more commonly used in industry. kPa used in Academia and technical publications.
	kilopascal	kPa	1 kPa = 1 kN/m^2 = 145 psi	
STRESS	megapascal	MPa	1 Mpa = MN/m^2 = 10^6 Pa = 145 psi (lb_f/in^2)	
POWER	watt	W	1 W = 1 J/s = 1 N · m/s = 44.25 ft-lb_f/min	
	kilowatt	kW	1 kW = 1 kJ/s = 1.34 hp (horsepower)	
VOLUME SPECIFIC ENERGY-GASES	kilojoule per cubic meter	kJ/m^3		1 kJ/m^3 = 1 J/L
VOLUME SPECIFIC ENERGY-LIQUIDS	joule per liter	J/L		
ENERGY, WORK, QUANTITY OF HEAT	joule	J	1 J = n · m = .7376 ft-lb_f 1 J = .948 x 10^{-3} Btu 1 kJ = .948 Btu	

METRICATION

TABLE 8.34 Preferred SI (Metric) Units (continued)

ROTATIONAL SPEED	revolution per second	r/s		
	revolution per minute	r/min		
VOLUME-GASES	cubic meter	m^3	1 m^3 = 35.315 ft^3	
VOLUME-LIQUIDS	liter	L	1 liter = .2642 gallon	
DENSITY	kilogram per cubic meter	kg/m^3	1 kg/m^3 = .0624 lb_m/ft^3	
VELOCITY	meter per second	m/s	1 m/s = 3.281 ft/sec	
VELOCITY-VEHICLE	kilometer per hour	km/h	1 km/h = .6214 miles/hr	
TEMPERATURE	degrees Celsius	°C	$t_{°C} = (t_{°F} - 32)\ 5/9$	Use °C for Celsius temperature and K for absolute temperature (thermodynamic)
	Kelvin	K	$T_k = T_{°C} + 273.15$	
SOUND PRESSURE LEVEL	decibel	dB		Reference level is 20 μ Pa = .0002 μ bar. Therefore unit remains the same.
VISCOSITY (DYNAMIC)	millipascal-second	mPa · s	1 mPa · s = 1 cP (centipoise)	
VISCOSITY (KINEMATIC)	square millimeter per second	mm^2/s	1 mm^2/s = 1 cSt (centistoke)	
MOISTURE CONTENT	kilogram per cubic feet	kg/m^3	1 kg/m^3 = .0624 lb_m/ft^3	
FORCE	Newton	N	1 N = .2248 lb_f	
	kilonewton	kN	1 kN = 224.8 lb_f	
MOMENT OF FORCE (TORQUE)	newton-meter	N · m	1 N · m = .7376 lb_f-ft	
MOMENT OF INERTIA	kilogram-meter squared	kg · m^2	1 kg · m^2 = 23.73 lb_m-ft^2	
FREQUENCY	hertz	Hz	1 Hz = 1 cycle per second	
GAS CONSTANT	joule per kilogram-kelvin	J(kg · K)	1 J/(kg · K) = .1859 ft-lb_f/lb_m-°R	
SPECIFIC HEAT	joule per kilogram-kelvin	J/(kg · K)	1 J/(kg · K) = .2389 x 10^{-3} Btu/lb_m-°R	

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Where there is a choice of SI units depending on quantity, the reference number has been put against the unit likely to be most frequently used.

1. The three units based on cm, dm and m, respectively, roughly correspond to use with fluidics, pneumatic controls, tools (consumption),up to medium-sized compressors, and large compressors. The alternatives of l/s and ml/s were rejected not only because the liter tends to be associated with liquids, but also because of the danger of confusion with l/min., widely used in Europe. One dm^3/s = approximately 2.1 cfm; that is, halving existing cfm tables is accurate within 5 percent and, in the case of consumption, cautious from the user's point of view.
2. This is the consistent unit but the long established use of rpm may call for the continued use of this alternative for some time, but this practice is not to be encouraged.
3. Weights of compressors, air tools, pneumatic equipment, and so on, will normally be described in these units.
4. Standard reference atmospheric conditions are as contained in ISO 1217 [i.e., 1 bar (14.5 psia); 20°C (68°F); 0 percent relative humidity (dry)].
5. The smaller unit (1 millibar = 100 N/m^2) will be used with fluidics and very low pressures. The high vacuum industry may use N/m^2 or rather the internationally and U-K preferred Pascal (Pa); 1 Pa = 1 N/m^2). As with pressure units hitherto in use, "absolute" or "gage" have to be stated where doubt could arise.

   At least one point in any document mentioning *bar*, the conversion 1 bar = 100 kPa should be stated as shown. Submultiples and multiples of Pa are used as with N/m (e.g., mPa, kPa, MPa).

   Designers of air receivers relating the pressure in bars to the MPa stress in the shell in one formula must not forget to include a factor of 10 (10 bars = 1 MPa).

   Users of low pressures and the fluidics industry have come across the use of inches water gage and mm of H_2O. 1 mm H_2O = 0.0985 m bar = 9.85 Pa approximately. Use of the w.g. will continue.
6. See also Note 5 for the explanation of MPa and the reason why this will replace the more cumbersome fraction, NM/m^2, preferable to N/mm^2. 1 ton/in.^{2} = 15.44 MPa.
7. J = N · m = W · s, for W = N · m/s. 746 W = 1 hp.
8. We are advised by BICEMA that the term *brake kilowatts* is likely to be used as standard practice in describing power outputs previously quoted in bhp (e.g., for prime moves such as diesel engines of portable compressors).

The following is a list of abbreviations of Metric SI Units in the order of their appearance in the last column of Table 8.34:

mm	millimeter (1 m = 1000 mm = 39.37 in. = 3.281 ft.)
m	meter
dm	decimeter (10 dm = 1 m)
cm	centimeter (100 cm = 1 m)
l	liter (originally 1 kg of water). In 1964 the liter was redefined as to b e equal to 10^{-3} m^3 = 1 dm^3.
km	kilometer (1000 m)
h	hour
s	second
ml	milliliter (1000 ml = 1 l) = 1 cm^3 (do not write ccm, cc, or ccs)
Hz	hertz (1 Hz = 1 cycle per second)
g	gram
kg	kilogram (= 1000 g)
t	ton (= 1000 kg). The abbreviation is not so widely used as, for instance g and kg, hence the unit is named full in the table.
N	newton. The force that will accelerate a freely movable mass of 1 kg by 1 m/s^2.
kN	kilonewton = N × 10^3
MN	meganewton = n × 10^6
J	Joule (see note 7)
W	watt
kW	kilowatt (= 1000 W)
C	Celsius = centigrade. The use of the word centigrade is deprecated.
K	Kelvin. Note that the ° sign is not used when quoting temperatures in kelvins.
cSt	centistokes.

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TABLE 8.35 Metric Conversion Factors

To convert:	Into:	Multiply by:
Atmospheres	Dynes per cm^2	1.0132×10^6
Atmospheres	Kilograms per square meter	1.0332×10^4
Amtospheres	Millimeters of mercury at 0°C	760
Atmospheres	Newtons per square meter	1.0133×10^5
British thermal units (Btu)	Kilogram-calories	0.2520
Centimeters	Feet	3.281×10^{-2}
Centimeters	Inches	0.3937
Centimeters	Mils (10^{-3} in.)	393.7
Centimeters per second	Feet per minute	1.969
Centimeters per second per second	Feet per second per second	3.281×10^{-2}
Circular mils	Square centimeters	5.067×10^{-6}
Cubic inches	Cubic centimeters	16.39
Cubic inches	Cubic meters	1.639×10^{-5}
Cubic inches	Liters	1.639×10^{-2}
Degrees Fahrenheit	Degrees centigrade	°C = 5/9 (°F – 32)
Dynes	Pounds	2.248×10^{-6}
Dyne-centimeters	Pounds-feet	7.376×10^{-8}
Grams	Ounces (avoir.)	3.527×10^{-2}
Grams per cm^3	Pounds per ft^3	62.43
Gram-cm^2	Pound-ft^2	2.37285×10^{-6}
Gram-cm^2	Slug-ft^2	7.37507×10^{-8}
Inches	Centimeters	2.540
Joules (int.)	Foot-pounds	0.7376
Kilograms	Pounds	2.205
Kilogram-calories	Foot-pounds	3.088
Kilometers	Feet	3.281
Liters	Gallons (U.S. liquid)	0.2642
Meters	Yards	1.094
Meters per second	Feet per second	3.28
Newton meters	Pound-feet	0.7376
Ounces (avoir.)	Grams	28.35
Pints (liquid)	Liters	0.4732
Pounds (avoir.)	Grams	453.6
Square Centimeters	Square feet	1.076×10^{-3}
Square Centimeters	Square inches	0.1550
Square feet	Square meters	0.09290

STANDARDS*

We suggest you reference the latest edition of the standards listed below.

Standards Organizations

AGMA = American Gear Manufacturers Association
ANSI = American National Standards Institute
API = American Petroleum Institute
ASME = American Society of Mechanical Engineers
CAGI = Compressed Air & Gas Institute
ISA = Instrument Society of America
ISO = International Standards Organization
NFPA = National Fire Protection Association
OSHA = Occupational Safety and Health Act
Note: ANSI and ISO Standards are available through:
ANSI, 25 West 43rd Street, 4th Floor, New York, NY 10036
Telephone: 1-212-642-4900 Fax: 1-212-398-0023 – www.ansi.org

Standards – Compressors

PN2CPTC1*	Acceptance Test Code for Bare Displacement Air Compressors
PN2CPTC2*	Acceptance Test Code for Electrically Driven Packaged Displacement Type Air Compressors
PN2CPTC3*	Acceptance Test Code for I.C. Engine Driven Packaged Displacement Type Air Compressors

* The standards have been incorporated in an Appendix to the latest edition of ISO 1217.

ISO 1217	Displacement Compressors - Acceptance Tests
ISO 5388	Stationary Air Compressors - Safety Rules and Code of Practice
ISO 5389	Turbocompressors - Performance Test Code
ISO 5390	Compressors - Classification
ISO 5941	Compressors, Pneumatic Tools and Machines-Preferred Pressures
ISO 6798	Reciprocating Internal Combustion Engines - Measurement of Airborne Noise
PN8NTC2.3	Measurement of Noise Emitted by Compressors. (Available from PNEUROP)
ISO 2151	Acoustics - Noise Test Code for Compressors and Vacuum Pumps Engineering Method (Grade 2)
API 617	Centrifugal Compressors for Petroleum, Chemical and Gas Industry Services
API 618	Reciprocating Compressors for Petroleum, Chemical and Gas Industry Services

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API 619	Rotary Compressors for Petroleum, Chemical and Gas Industry Services
API 672	Packaged Integral Geared Centrifugal Air Compressors
API 681	Liquid Ring Vacuum Pumps and Compressors
ASME B19.1	Safety Standard for Air Compressor Systems
ASME B19.3	Safety Standard for Compressors for Process Industries

Standards – Compressed Air Dryers

CAGI ADF100	Refrigerated Compressed Air Dryers - Methods for Testing and Rating
ANSI/CAGI ADF200	Dual Tower Regenerative Desiccant Compressed Air Dryers - Methods for Testing and Rating
ANSI/CAGI ADF300	Standard for Single Tower (Deliquescent) Compressed Air Dryers - Methods for Testing and Rating
ISO 7183-1	Compressed Air Dryers - Specifications and Testing
ISO 7183-2	Compressed Air Dryers - Part 2: Performance Ratings

Standards – Compressed Air Filters

ANSI/CAGI ADF400	Standards for Testing and Rating Coalescing Filters
ANSI/CAGI ADF500	Standard for Measuring the Adsorption Capacity of Oil Vapor Removal Adsorbent Filters
ANSI/CAGI ADF600	Standards for Particulate Filters
ANSI/CAGI ADF700	Standards for Membrane Compressed Air Dryers
ISO 8573-1	Compressed Air for General Use - Part 1: Contaminants and Quality Classes
ISO 8573-2	Compressed Air for General Use - Part 2: Test Methods for Aerosol Oil Content
ISO 8573-3	Compressed Air for General Use - Part 3: Test Methods for Humidity
ISO 8573-4	Compressed Air for General Use - Part 4: Test Methods for Solid Particle Content
ISO 8573-5	Compressed Air for General Use - Part 5: Test Methods for Oil Vapor and Organic Solvent Content
ISO 8573-6	Compressed Air for General Use - Part 6: Test Methods for Gaseous Contaminant Content
ISO 8573-7	Compressed Air for General Use - Part 7: Test Methods for Viable Microbiological Contaminant Content
ISO 8573-8	Compressed Air for General Use - Part 8: Contaminants and Purity Classes (by Mass Concentration of Solid Particles)
ISO 8573-9	Compressed Air for General Use - Part 9: Test Methods for Liquid Water Content
ISO 8573-10	Compressed Air for General Use - Part 10: Test Methods for Mass Concentration of Solid Particle Content

Standards – Pneumatic Tools

CAGI B186.1	Safety Code for Portable Air Tools
ISO 2787	Rotary and Percussive Tools - Performance Tests
ISO 5391	Pneumatic Tools and Machines - Vocabulary
ISO 5393	Rotary Pneumatic Tools for Threaded Fasteners - Performance Tests
ISO 6544	Hand-held Pneumatic Assembly Tools for Installing Threaded Fasteners - Reaction Torque Impulse Measurements
ISO 15744	Acoustics - Noise Test Code for Hand-held Non-Electric Power Tools - Engineering Method
PN8NTC1.2	Measurement of Noise Emitted by Hand-held Pneumatic Tools (Available from PNEUROP)
ANSI B7.1	The Use, Care, and Protection of Abrasive Wheels
ANSI B7.5	Safety Code for the Construction, Use, and Care of Gasoline-Powered, Hand-held, Portable, Abrasive Cutting off Machines
ANSI B30.16	Overhead Hoists (Underhung)
ANSI B74.18	American National Standard for Grading of Certain Abrasive Grain on Coated Abrasive Material
ISO 8662-1	Measurement of Vibration in Hand-held Power Tools - Part 1: General
ISO 8662-2	Measurement of Vibrations in Hand-held Power Tools - Part 2: Chipping Hammers, Riveting Hammers
ISO 8662-3	Measurement of Vibrations in Hand-held Power Tools - Part: 3: Rotary Hammers and Rock Drills
ISO 8662-4	Measurement of Vibrations in Hand-held Power Tools - Part 4: Grinders
ISO 8662-5	Measurement of Vibrations in Hand-held Power Tools - Part 5: Breakers and Hammers for Construction
ISO 8662-6	Measurement of Vibrations in Hand-held Power Tools - Part 6: Impact Drills
ISO 8662-7	Measurement of Vibrations in Hand-held Power Tools - Part 7: Wrenches, Screwdrivers and Nutrunners with Impact, Impulse or Ratcheting Action
ISO 8662-8	Measurement of Vibrations in Hand-held Power Tools - Part 8: Polishers and Rotary, Orbital and Random Orbital Sanders
ISO 8662-9	Measurement of Vibrations in Hand-held Power Tools - Part 9: Rammers
ISO 8662-10	Measurement of Vibrations in Hand-held Power Tools - Part 10: Nibblers and Shears
ISO 8662-11	Measurement of Vibrations in Hand-held Power Tools - Part 11: Fastener Driving Tools

ISO 8662-12	Measurement of Vibrations in Hand-held Power Tools - Part 12: Saws and Files with Reciprocating Action and Saws with Oscillating or Rotating Action
ISO 8662-13	Measurement of Vibrations in Hand-held Power Tools - Part 13: Die Grinders
ISO 8662-14	Measurement of Vibrations in Hand-held Power Tools - Part 14: Stone Working Tools and Needle Scalers
ISO 8662-16	Measurement of Vibrations in Hand-held Power Tools - Part 16: Screw Drivers (rapping)
ISO 5349-1	Mechanical Vibration - Measurement and Evaluation of Human Exposure to Hand-Transmitted Vibration - Part 1: General Requirements
ISO 5349-2	Measurement and Evaluation of Human Exposure to Hand-Transmitted Vibration - Part 2: Practical Guidance for Measurement at the Work Place

General

ANSI/ISA S7.0.01	Quality Standard for Instrument Air
ANSI S12.12	Engineering Method for Determination of Sound Power Levels of Noise Sources Using Sound Intensity
ASME B31.1	Power Piping
ASME BPVC Section VIII	Rules for Construction of Pressure Vessels
ISO 2398	Rubber Hose, Textile-Reinforced, for Compressed Air -Specification
ISO 3746	Acoustics – Determination of Sound Power Levels of Noise Sources Using Sound Pressure - Survey Method Using an Enveloping Measurement Surface Over a Reflecting Plane
ISO 3747	Acoustics – Determination of Sound Power Levels of Noise Sources Using Sound Pressure - Survey Method Using a Reference Sound Source
ISO 3857-1	Compressors, Pneumatic Tools and Machines - Vocabulary - Part 1: General
ISO 3857-2	Compressors, Pneumatic Tools and Machines - Vocabulary - Part 2: Compressors
ISO 3857-3	Compressors, Pneumatic Tools and Machines - Vocabulary - Part 3: Pneumatic Tools and Machines.
NFPA 99	Health Care Facilities
ANSI Z87.1	Occupational and Educational Eye and Face Protection
ANSI S12.6	Hearing Protectors

OSHA Regulations, 29 CFR, 1926.302 – Safety Equipment

OSHA Regulations, 29 CFR, 1910.133 – Eye and Face Protection

OSHA Regulations, 29 CFR, Section 1910.95 – Occupational Noise Exposure

OSHA Appendix F – Table 1 – Breathing Air Systems for use with Pressure-Demand Supplied Air Respirators in Asbestos Abatement