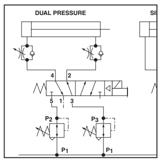
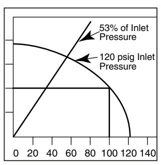
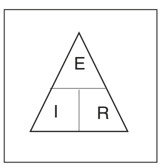
# HELPFUL ENGINEERING INFORMATION

NORGREN




## Contents



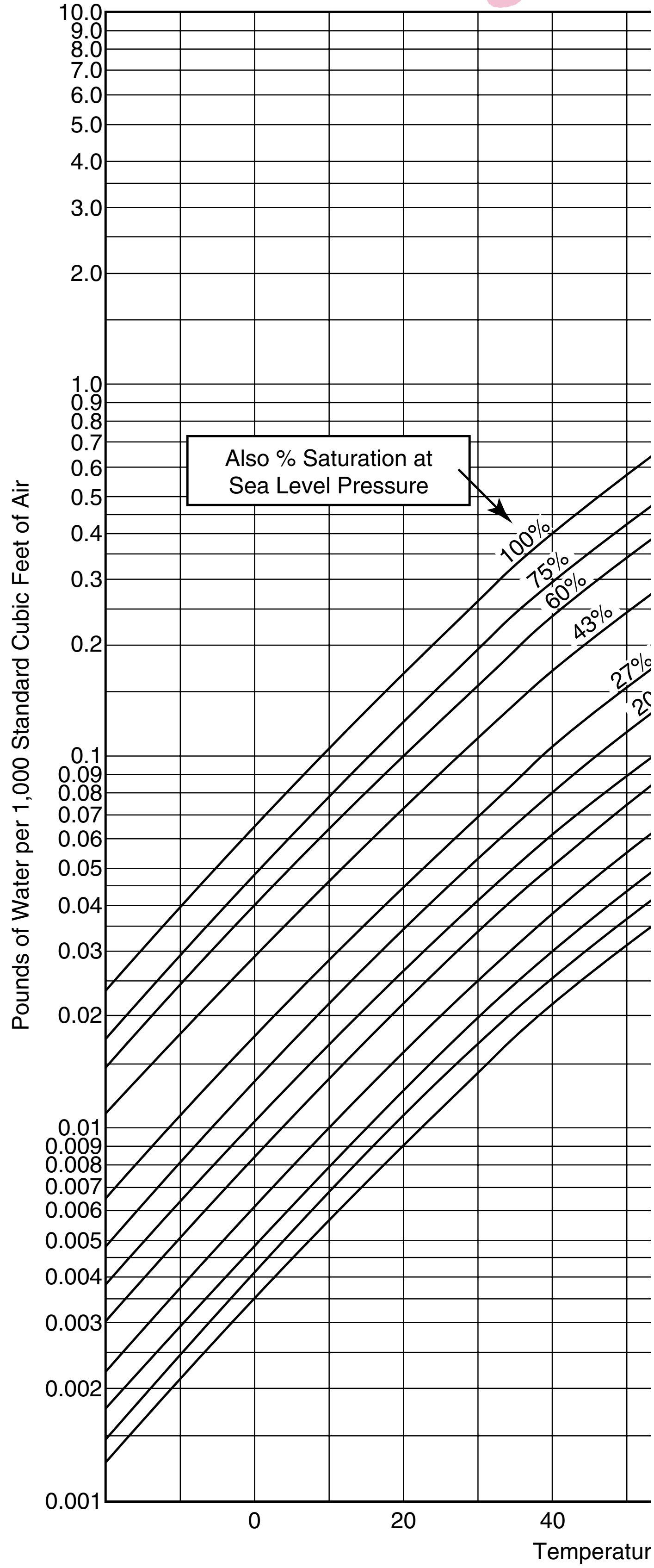



Figure 1. Water Vapor Co

# HOW TO DETERMINE WATER CONTENT IN COMPRESSED AIR SYSTEMS

The more sophisticated pneumatic equipment and instrumentation being used throughout the industry today requires greater attention to the purity of the compressed air which supplies this equipment. Compressed air, free of condensate, has become increasingly important for many industrial applications.

The question, "How much water or condensate must be removed from the system?" Today, more frequently requires an answer.

The data presented in Figure 1 permits simple determination of the amount of condensate to be found in a compressed air system under a variety of operating conditions—pressure, temperature, and humidity.

Figure 1 gives this information in pounds of water per 1,000 cubic feet of air at different operating temperatures (°F) and pressures (psig). The data presented, water vapor content of saturated air at various temperatures and pressures, represent the worst possible condition. There is no guarantee that the water vapor content of compressed air will be any less than saturation at any given operating pressure and temperature; therefore, the saturated content should be used in all calculations.

## The Following Examples Illustrate the Use of Figure 1

### Example 1:

How much condensate will there be in a compressed air system operating at 100 scfm and 100 psig if the air at the compressor intake is at a temperature of 80°F and 75% saturation (relative humidity)?

The water vapor content of air at 80°F, 75% saturation, and 0 psig (atmospheric pressure) is 1.12 pounds of water per 1,000 cubic feet of air (intersection of the 75% saturation line and the 80°F line — see Figure 1).

If this air is compressed to 100 psig and then cooled to 70°F, either in an after cooler or as it flows through the distribution piping, the maximum water vapor content that this air can carry is 0.15 pounds of water per 1,000 cubic feet of air (intersection of the 100 psig operating pressure line and the 70°F line).

The difference, 1.12 - 0.15 = 0.97 pounds of water per 1,000 cubic feet of air. This quantity of water appears in the system as condensate.

At an air consumption of 100 scfm, 6000 cubic feet of air will be compressed each hour. 6 x 0.97 = 5.82 pounds of water or 0.698 gallons of water must be removed from the system each hour.

In an eight-hour operating day, 8 x 0.698 = 5.584 gallons of water must be removed from the system.

### Example 2:

Assume, as in Example 1, that air is compressed at the rate of 100 scfm to an operating pressure of 100 psig and cooled to 70°F. The water vapor content equals 0.15 pounds of water per 1,000 cubic feet of air (intersection of the 100 psig line and the 70°F line - see Figure 1).

If this air is then used in an environment at 0°F, or if it is desired to maintain a 0°F dewpoint to protect delicate pneumatic equipment or instruments, additional condensate or ice will form.

At 100 psig and 0°F, the saturated water vapor content of air is 0.0085 pounds of water per 1,000 cubic feet of air (intersection of the 100 psig line and the 0°F line). The difference, 0.1500 - 0.0085 = 0.1415 pounds of water per 1,000 cubic feet of air, must be removed from the system.

Each hour of operation, 6 x 0.1415 = 0.849 pounds or 0.1018 gallons of water will appear as condensate.

In an eight-hour operating day, 8 x 0.1018 = 0.814 gallons of condensate.

Adding the results of Example 1 and 2, the total condensate to be removed from the system when air is compressed to 100 psig at the rate of 100 scfm and cooled to 0°F from a source at 80°F and 75% saturation is 5.584 plus 0.814 = 6.40 gallons per eight-hour day. If the air at the compressor intake was more than 75% saturation, the amount of condensate forming in the system would be even greater and could be as high as 8.86 gallons of water per eight-hour day.

### Example 3:

If compressed air at 100 psig is saturated at 70°F (70°F dewpoint): What is the dewpoint at 40 psig? What is the dewpoint at 0 psig?

The water vapor content at 100 psig and 70°F is 0.15 pounds of water per 1,000 cubic feet of air (intersection of 100 psig line and 70°F line - see Figure 1). Move horizontally along the 0.15 vapor content line to the intersection with the 40 psig line - read temperature: 50°F. The dewpoint at 40 psig is 50°F.

Continue along the 0.15 vapor content line to the intersection with the 0 psig line - read temperature: 17°F. The dewpoint at 0 psig (atmospheric pressure) is 17°F.

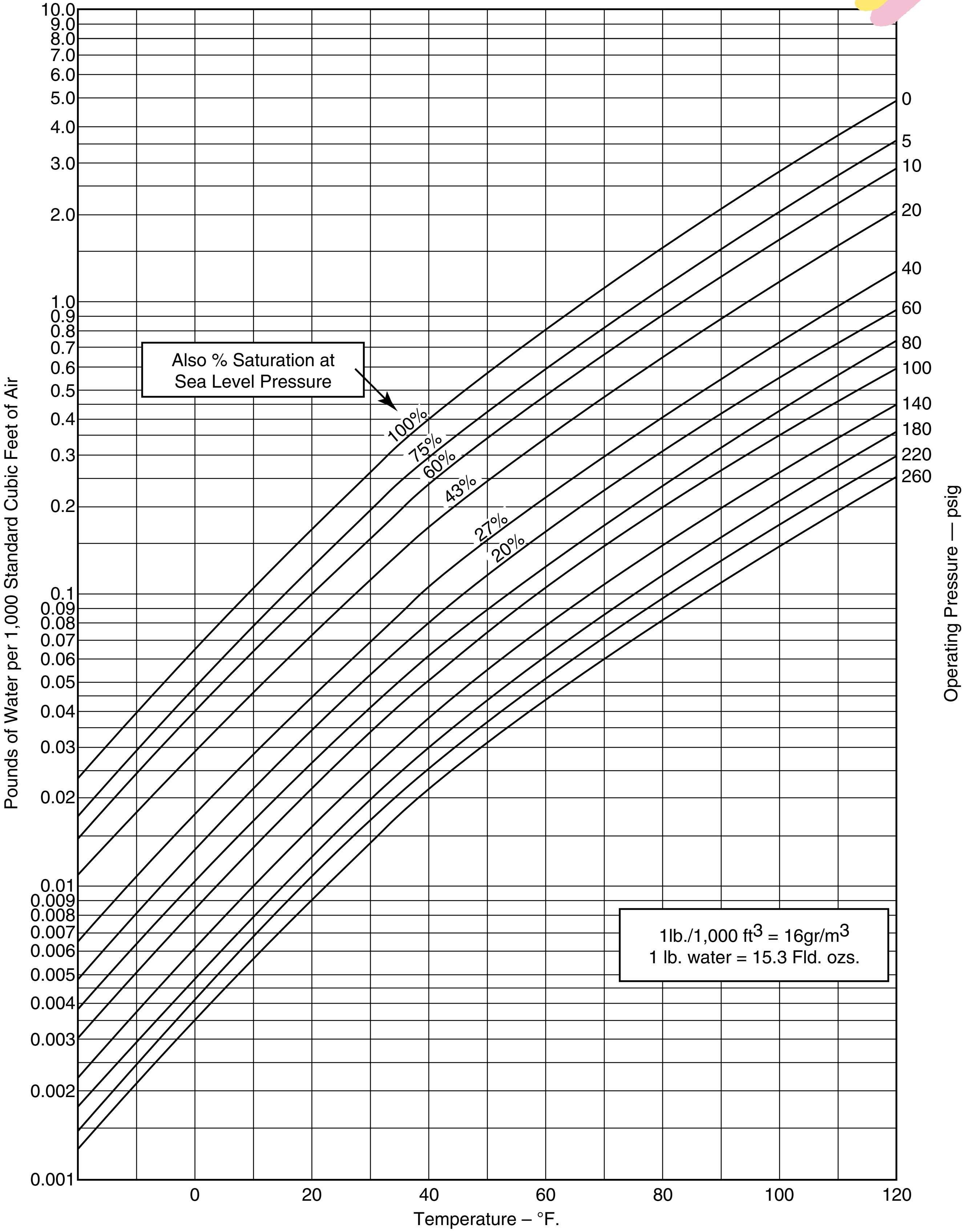



Figure 1. Water Vapor Content of Saturated Air

# HOW TO DETERMINE PRESSURE DROP IN COMPRESSED AIR SYSTEMS

## Distribution Piping, Fittings, and Filters

The method used in this section represents a simplified approach to the determination of pressure drop in compressed air systems. It permits easy determination of the pressure-drop across any component installed in the system as well as determination of the pressure drop for the complete system or any segment of the system.

This method is based upon the recognized Darcy formula presented here in a somewhat different form:

$$\Delta P = \frac{KQ2}{1000}\left[\frac{14.71}{14.7=P}\right]\left[\frac{460+t}{520}\right]$$

$\Delta P$ = Pressure drop (psig)
K = Constant for pipe or unit
Q = Constant for flow (scfm)
P = Working pressure (psig)
t = Compressed air temperature (°F)

Figure 2 presents the relationship between air flow (scfm) and pressure drop (psig) for K = 1. Figure 2, when used in conjunction with the values of K presented in Tables 1, 2 and 3, readily permits the determination of pressure drop ($\Delta$ P) across any component installed in a compressed air system, the pressure drop of the entire system, or any segment of the system.

### Example 1:

Determine the pressure drop ($\Delta$P) in 150 feet of 3/4" schedule 40 pipe, at a flow of 80 scfm and an operating pressure of 100 psig:

1. Refer to Figure 2: Follow vertically the 80 scfm line to its intersection with the 100 psig operating pressure line.
2. Read the pressure drop ($\Delta$P) at left corresponding to this intersection: P = 0.8.
3. Select from Table 1 the K value for ¾" pipe: K = 5.93.
4. Multiply 5.93 x 0.8 = 4.74 psig per 100 feet of pipe.
5. $\Delta$P for 150 feet of pipe equals $\frac{4.74 \times 150}{100}$ = 7.11 psig since pressure drop is proportional to length.

### Example 2:

Determine the pressure drop in a system containing 100 feet of ¾" schedule 40 pipe, two 90° standard elbows, one globe valve and one ¾" 40-micron filter (F74). The system pressure is 100 psig, and the flow requirement is 80 scfm:

1. Refer to Figure 2: Follow vertically the 80 scfm line to its intersection with the 100 psig operating pressure line.
2. Read the pressure drop ($\Delta$P) at the left of the graph, corresponding to this intersection: $\Delta$P = 0.8 psig.
3. From Table 1, select the K value for ¾" pipe: K = 5.93
4. From Table 2, select the K value for 3/4" standard 90° elbow: K = 0.119. There are two elbows; therefore, multiply by 2: 0.119 x 2 = 0.238.
5. From Table 2, select the K value for a fully open globe valve: K = 1.36.
6. From Table 3, select the K value for a ¾" 40-micron filter (F74); K = 1.78.
7. Add the K values from steps 3, 4, 5 and 6 (5.930 + 0.238 + 1.360 + 1.78 = 9.308=Kt).
8. Multiply the $\Delta$P value determined from step 2 by Kt: 0.8 x 9.308 = 7.446. The pressure drop under the foregoing conditions will be approximately 7.5 psig.
9. If a higher pressure drop is permissible, make a similar computation for ½" pipe and fittings; if a lower pressure drop is desirable, consider 1" pipe and fittings.

## Distribution Piping

Figures 3, 4, 5 and 6 present the relationship between air flow (scfm) and pressure drop ($\Delta$P = psig) for pipe sizes ⅛" through 3" inclusive at operating pressures of 5 to 250 psig. Lines "A", "B", "C" and "D" represent the maximum flow for pressure drops equal to 5%, 10%, 20% and 40% of the supply pressure respectively over the operating range of 5 to 250 psig.

These figures are a convenience in that they permit direct reading of the pressure drop through 100 feet of schedule 40 pipe. The pressure drop read from these charts will not always agree exactly with the pressure drop calculated from the information contained on Figure 2. The differences, however, are minor and result primarily from limiting the computations to three significant figures. The results obtained using either method are well within the accuracy capabilities of the flow computations.

### Example 1:

Determine the pressure drop in 100 feet of 3/4" schedule 40 pipe at a flow rate of 150 scfm and an operating pressure of 100 psig:

1. Refer to Figure 4—follow the vertical 150 scfm line until it intersects the diagonal 100 psig applied pressure line.
2. Read the pressure drop on the scale at the left: 17 psig.
3. At an applied pressure of 100 psig, this represents a pressure drop of 17%. You will note that this point falls between lines "B" and "C" representing 10% and 20% pressure drop.
4. If the operating pressure was 80 psig, a flow of 150 scfm would produce a pressure drop of 20 psig or 25% of the applied pressure. You will note that this point falls between the lines "C" and "D" indicating pressure drops of 20% and 40% respectively.

The information on the following tables and figures is based on a compressed air temperature of 60°F. For temperatures other than 60°F, multiply the final result, ΔP by $\frac{460 + °F}{520}$

| Fitting | Pipe Size | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | 1/8" | 1/4" | 3/8" | 1/2" | 3/4" | 1" | 1-1/4" | 1-1/2" | 2" |
| 90° Standard Elbow | 15.4 | 4.09 | 1.09 | 0.422 | 0.119 | .0432 | .01400 | .00711 | .00219 |
| 45° Standard Elbow | 8.3 | 2.20 | 0.53 | 0.216 | 0.059 | .0216 | .00720 | .00382 | .00131 |
| 90° Street Elbow | 25.8 | 6.80 | 1.91 | 0.686 | 0.196 | .0714 | .02320 | .01180 | .00406 |
| -45° Street Elbow | 13.3 | 3.56 | 0.91 | 0.343 | 0.107 | 0.365 | .01200 | .00607 | .00205 |
| 90° Long Radius Elbow | 10.4 | 2.74 | 0.80 | 0.264 | 0.083 | .0282 | .00920 | .00468 | .00163 |
| Standard Tee – Run | 10.4 | 2.74 | 0.80 | 0.264 | 0.083 | .0282 | .00920 | .00468 | .00163 |
| Standard Tee – Side | 31.0 | 8.14 | 2.37 | 0.818 | 0.243 | .0845 | .02760 | .01390 | .00490 |
| Globe Valve – Full Open | 175.3 | 46.40 | 12.70 | 4.750 | 1.360 | .4820 | .15600 | .08150 | .02750 |
| Gate Valve – Full Open | 6.7 | 1.76 | 0.47 | 0.180 | 0.053 | .0183 | .00600 | .00295 | .00107 |
| Angle Valve – Full Open | 74.8 | 19.80 | 5.46 | 1.800 | 0.593 | .1990 | .06800 | .03470 | .01210 |

Table 2. Values of K for Commonly Used Fittings

| Pipe Size | K |
|---|---|
| 1/8" | 2300. |
| 1/4" | 450.0 |
| 3/8" | 91.0 |
| 1/2" | 26.4 |
| 3/4" | 5.93 |
| 1" | 1.66 |
| 1-1/4" | 0.400 |
| 1-1/2" | 0.174 |
| 2" | 0.0467 |
| 2-1/2" | 0.0186 |
| 3" | 0.0060 |

Table 1. Values of K for 100 Feet of Schedule 40 pipe

| Filter Type | Micron Size | Pipe Size | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | 1/8" | 1/4" | 3/8" | 1/2" | 3/4" | 1" | 1-1/4" | 1-1/2" | 2" |
| F07 | 5 | 115 | 55.0 | | | | | | | |
| | 25 | 112 | 49.0 | | | | | | | |
| | 100 | 92 | 41.0 | | | | | | | |
| F72 | 5 | | 22.62 | 18.18 | | | | | | |
| | 25 | | 29.99 | 23.95 | | | | | | |
| | 40 | | 15.71 | 11.03 | | | | | | |
| F73 | 5 | | 14.93 | 10.83 | 9.75 | | | | | |
| | 25 | | 14.93 | 11.48 | 10.54 | | | | | |
| | 40 | | 12.86 | 8.99 | 8.02 | | | | | |
| F74 | 5 | | | 5.15 | 3.72 | 2.92 | | | | |
| | 25 | | | 4.17 | 3.01 | 2.25 | | | | |
| | 40 | | | 3.67 | 2.52 | 1.78 | | | | |
| F17 | 5 | | | | | .47 | .34 | .34 | .340 | |
| | 25 | | | | | .34 | .23 | .20 | .200 | |
| | 50 | | | | | .32 | .20 | .19 | .190 | |
| | 75 | | | | | .32 | .20 | .19 | .190 | |
| F18 | 25 | | | | | | | | .050 | .028 |
| | 50 | | | | | | | | .036 | .020 |
| | 75 | | | | | | | | .032 | .018 |

Table 3. Values of K for Norgren Filters

| Applied Pressure PSIG | Nominal Standard Pipe Size | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1/8" | 1/4" | 3/8" | 1/2" | 3/4" | 1" | 1-1/4" | 1-1/2" | 2" | 2-1/2" | 3" |
| 5 | 0.5 | 1.2 | 2.7 | 4.9 | 6.6 | 13 | 27 | 40 | 80 | 135 | 240 |
| 10 | 0.8 | 1.7 | 3.9 | 7.7 | 11.0 | 21 | 44 | 64 | 125 | 200 | 370 |
| 20 | 1.3 | 3.0 | 6.6 | 13.0 | 18.5 | 35 | 75 | 110 | 215 | 350 | 600 |
| 40 | 2.5 | 5.5 | 12.0 | 23.0 | 34.0 | 62 | 135 | 200 | 385 | 640 | 1100 |
| 60 | 3.5 | 8.0 | 18.0 | 34.0 | 50.0 | 93 | 195 | 290 | 560 | 900 | 1600 |
| 80 | 4.7 | 10.5 | 23.0 | 44.0 | 65.0 | 120 | 255 | 380 | 720 | 1200 | 2100 |
| 100 | 5.8 | 13.0 | 29.0 | 54.0 | 80.0 | 150 | 315 | 470 | 900 | 1450 | 2600 |
| 150 | 8.6 | 20.0 | 41.0 | 80.0 | 115.0 | 220 | 460 | 680 | 1350 | 2200 | 3900 |
| 200 | 11.5 | 26.0 | 58.0 | 108.0 | 155.0 | 290 | 620 | 910 | 1750 | 2800 | 5000 |
| 250 | 14.5 | 33.0 | 73.0 | 135.0 | 200.0 | 370 | 770 | 1150 | 2200 | 3500 | 6100 |

Use Table 4 as a guide in sizing piping and equipment in compressed air systems.

The flow values in Table 4 are based on a pressure drop as shown below.

| Pressure Drop per 100 ft of Pipe | Pipe Size (Inches) |
|---|---|
| 10% of Applied Pressure | ⅛, ¼, ⅜, ½ |
| 5% of Applied Pressure | ¾, 1, 1¼, 1½, 2, 2½, 3 |

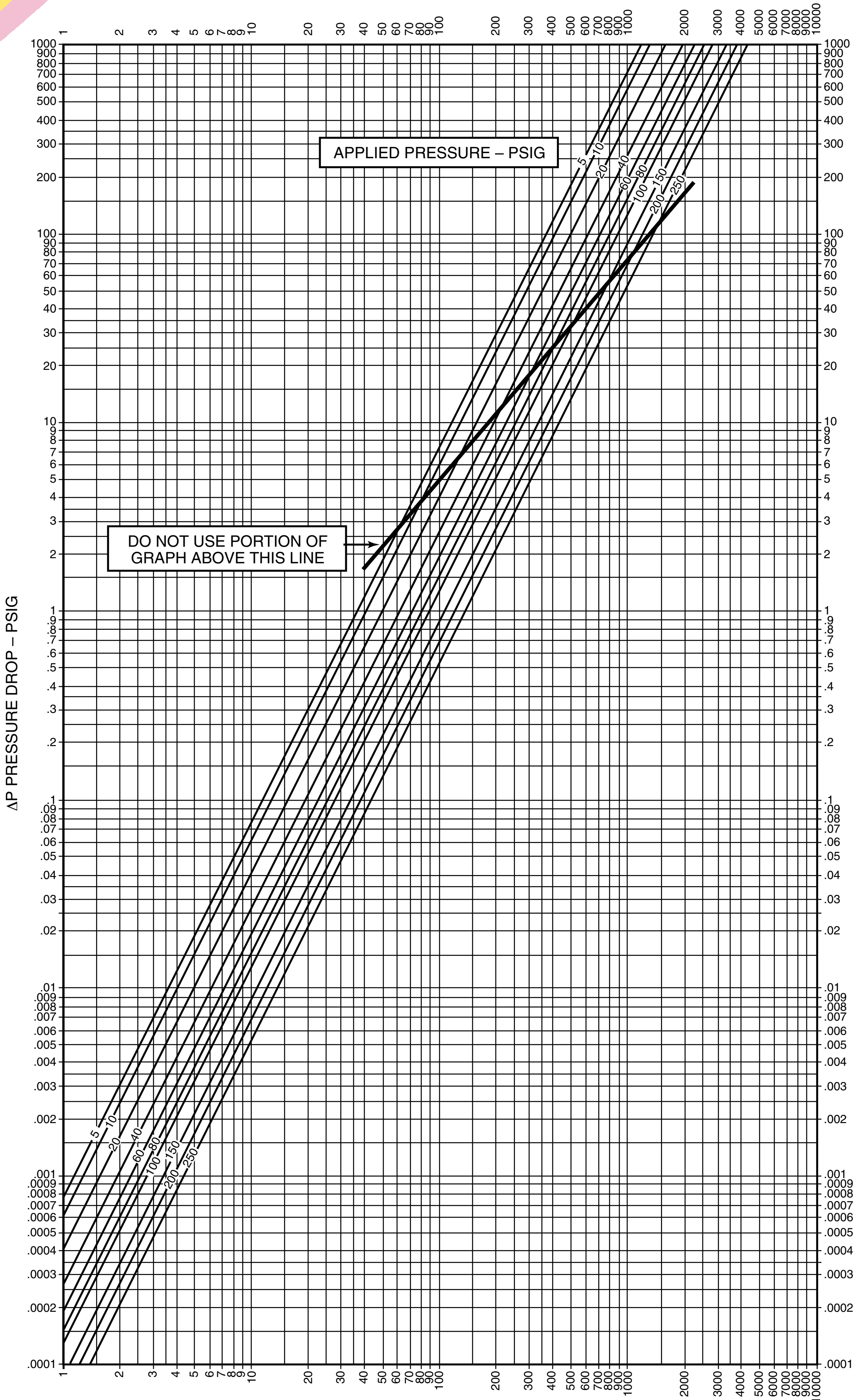



FIGURE 2. Air Flow – Pressure Drop Graph For K = 1 In Equation $\Delta P = \frac{KQ^2}{1000}\left[\frac{14.7}{14.7 + P}\right]\left[\frac{460 + t}{520}\right]$

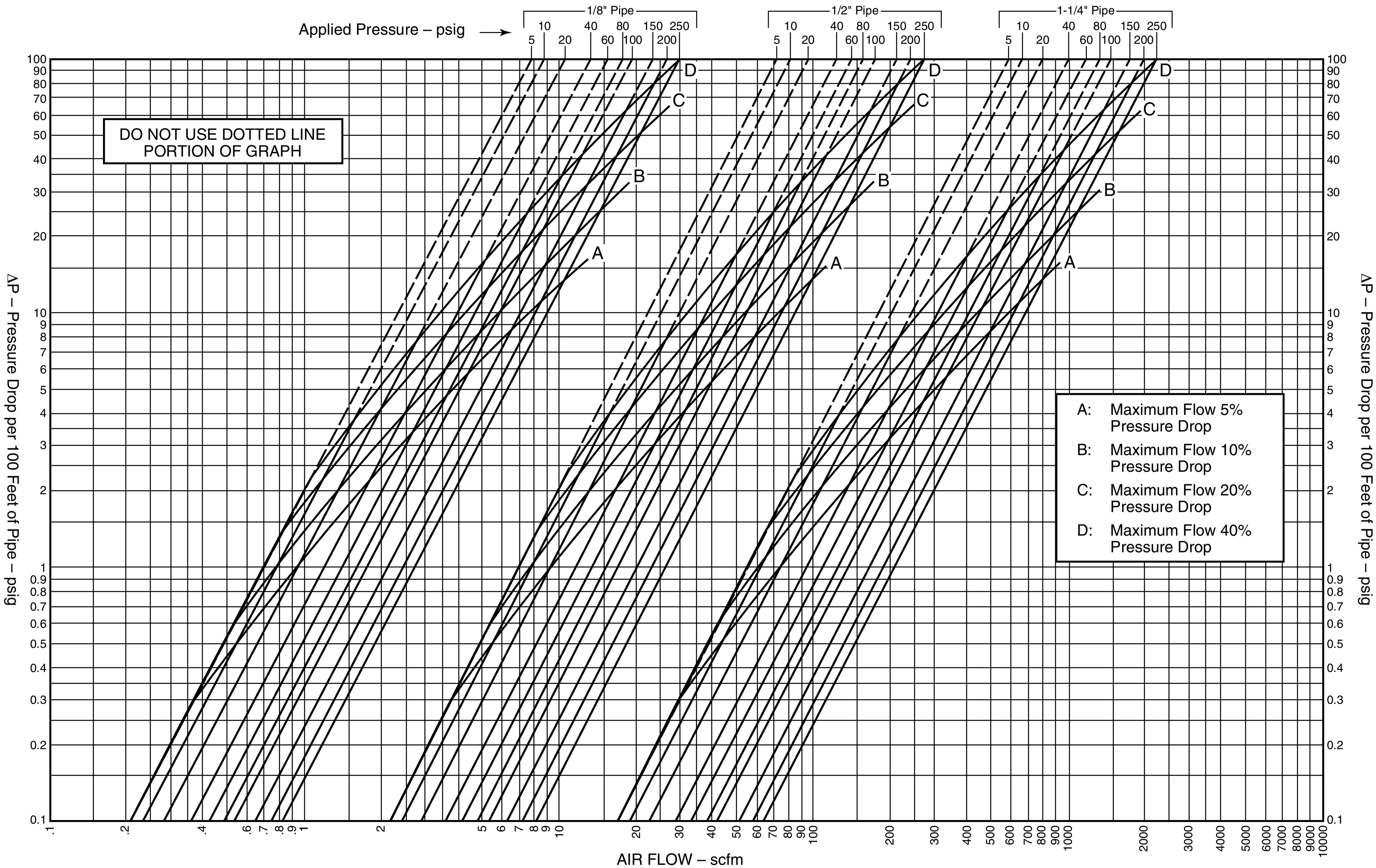



Figure 3. Air Flow – Pressure Drop Graph (1/8", 1/2", 1-1/4" Pipe)

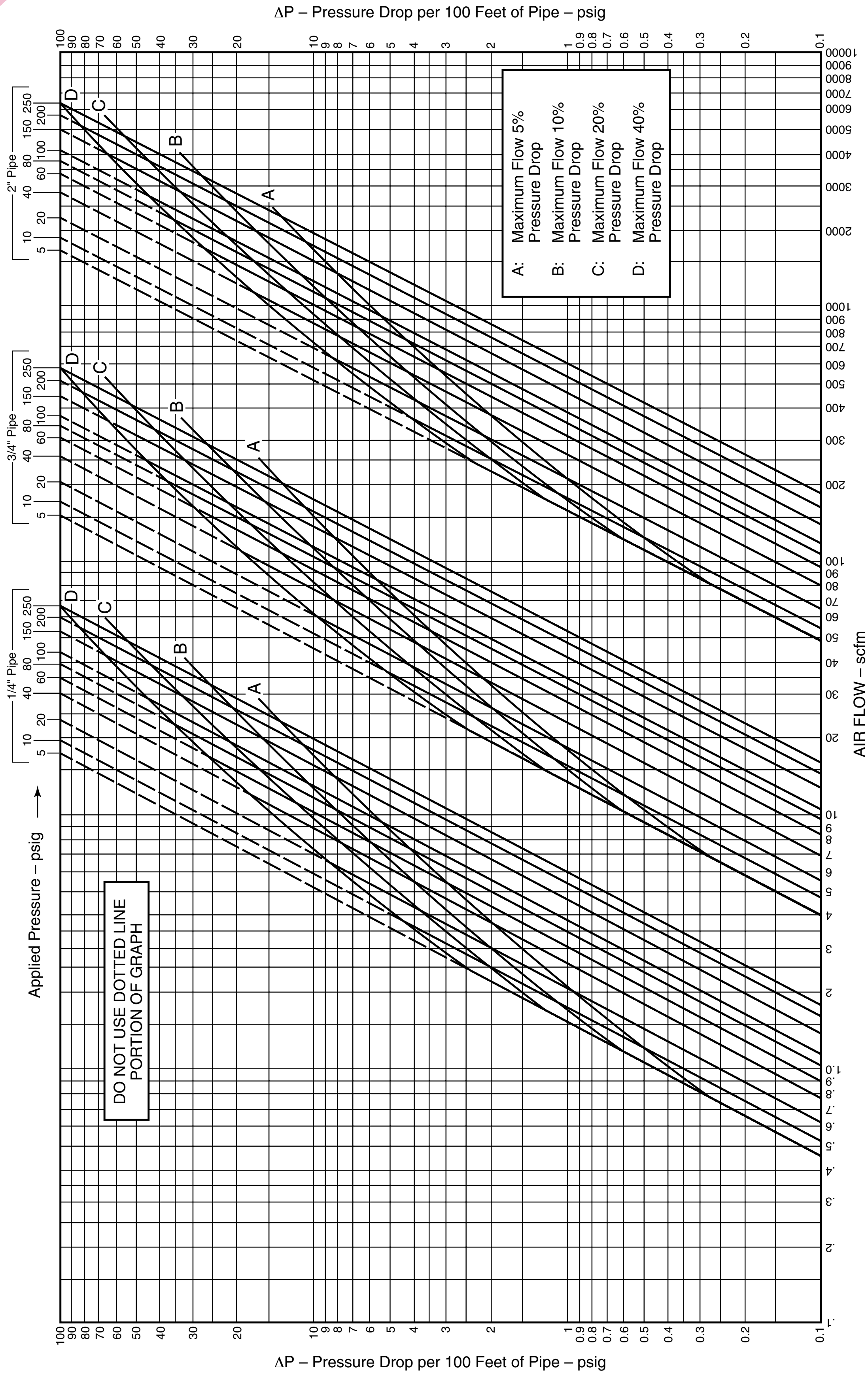



Figure 4. Air Flow – Pressure Drop Graph (1/4", 3/4", 2" Pipe)

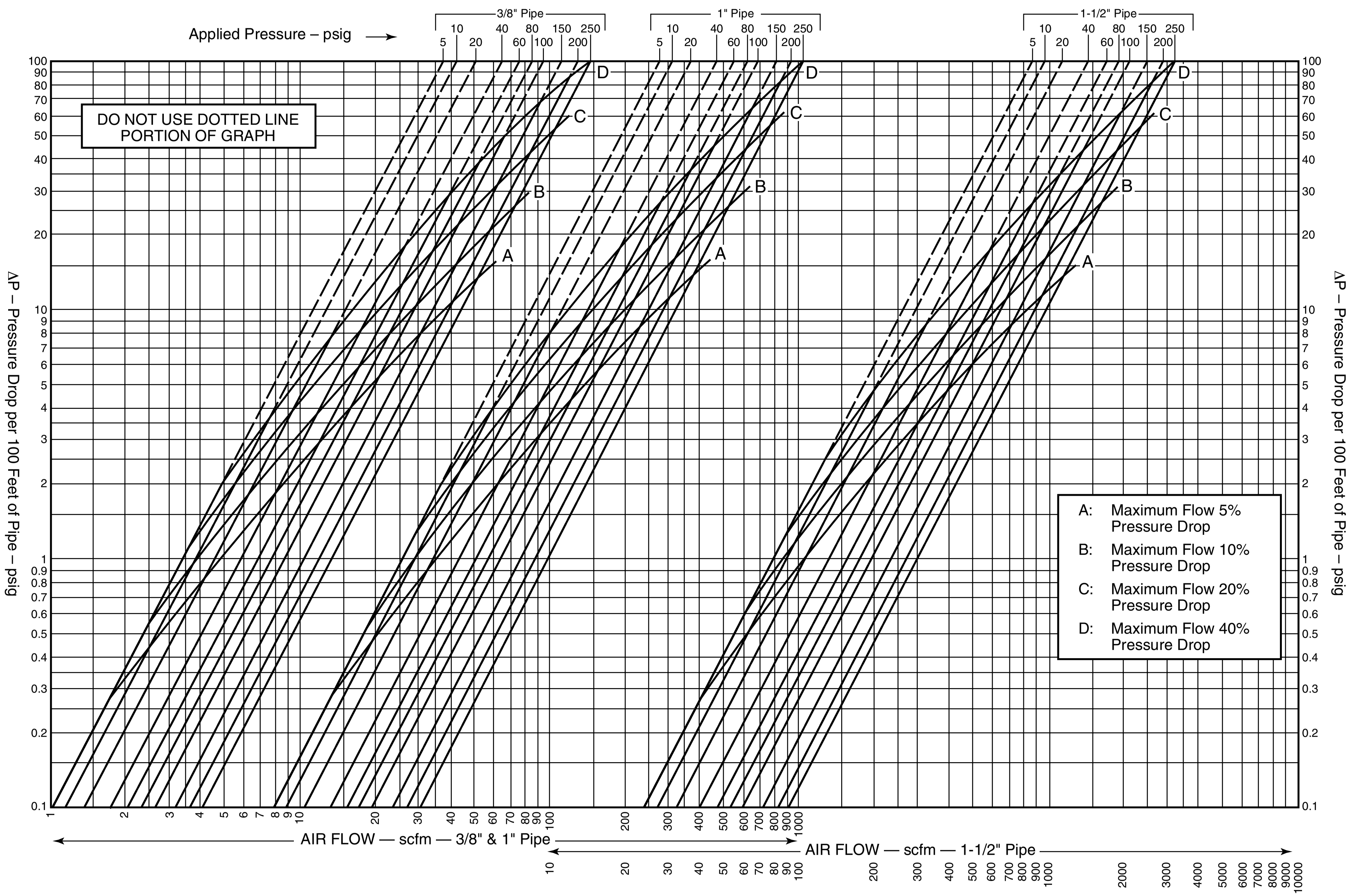



Figure 5. Air Flow – Pressure Drop Graph (3/8", 1", 1-1/2" Pipe)

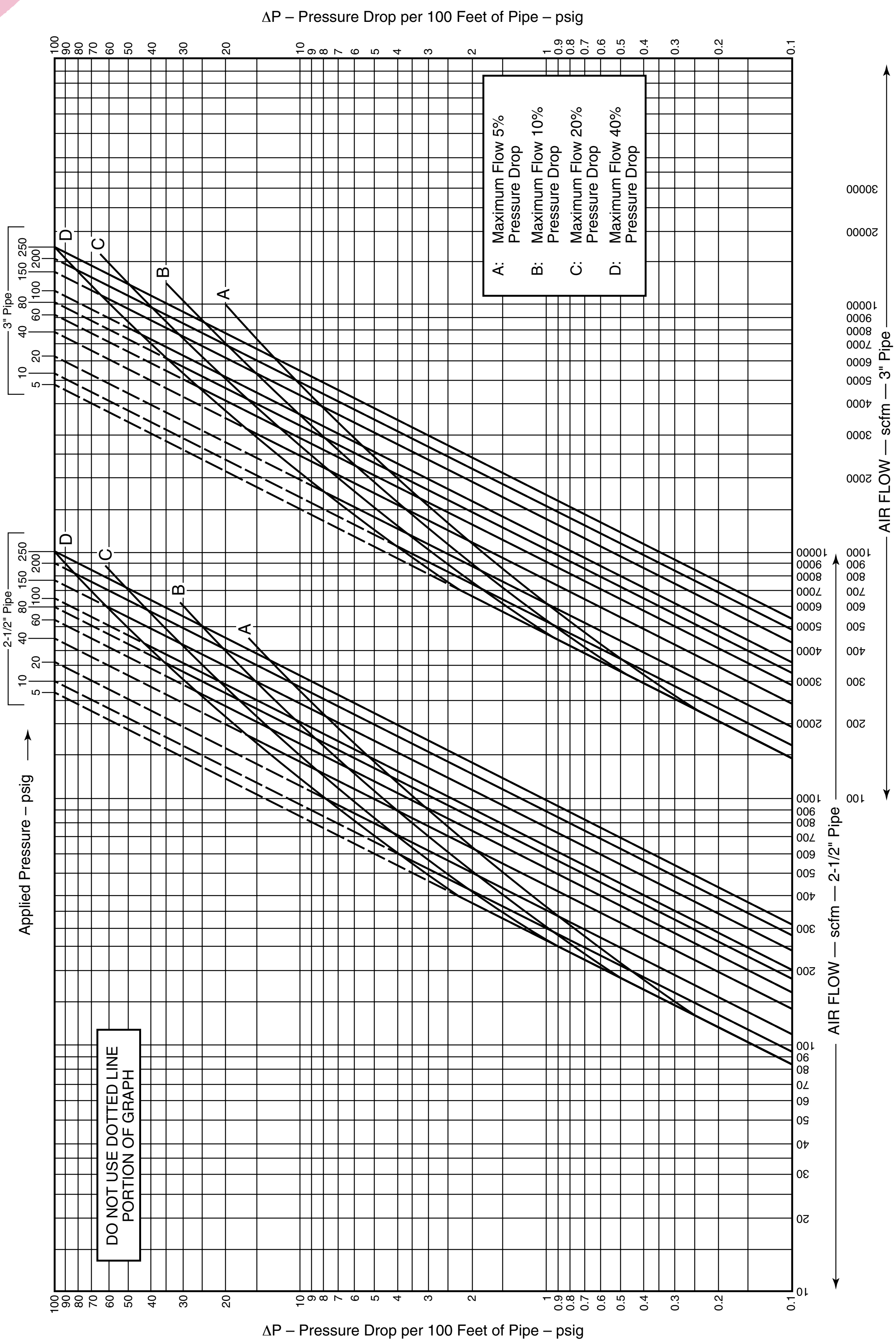



Figure 6. Air Flow – Pressure Drop Graph (2-1/2" & 3" Pipe)

# HOW TO DETERMINE FLOW AND PRESSURE DROP IN WATER SYSTEMS

Table 5 is self-explanatory. For the conditions given, flow values can be read directly from the chart

Figure 7 is more versatile - it provides the means for determining pressure drop (ΔP) or flow (gp) for a variety of operating conditions.

Figure 7 gives the relationship between pressure drop (ΔP) and flow (gpm) for pipe sizes ⅛" to 3". Two auxiliary scales on Figure 7 provide the applied pressure corresponding to a (ΔP) of 5% and 10%.

## The Following Examples Illustrate the Use of Table 5 and Figure 7

### Example 1:

Determine the flow in ½" pipe (gpm) that will produce a pressure drop (ΔP) of 10 psig per 100 feet of pipe when operating at an applied pressure of 100 psig:

From Table 5, the flow can be read directly = 4.6 gpm or from Figure 7, locate the intersection of the diagonal line for ½" pipe and the 10 psig ΔP line: Read flow = 4.6 gpm.

### Example 2:

Determine the flow in ½" pipe (gpm) that will produce a pressure drop (ΔP) of 12 psig in 150 feet of pipe when operating at an applied pressure of 100 psig:

First—Determine the ΔP for 100 feet of pipe:

$$\Delta P = \frac{12 \times 100}{150} = 8 \text{ psig}$$

Second—From Figure 7, locate the intersection of the diagonal line for ½" pipe and the 8 psig ΔP line: Read flow = 4.2 gpm.

### Example 3:

Determine the pressure drop (ΔP) in 75 feet of 3/4" pipe when operating at a flow of 10 gpm and an applied pressure of 150 psig:

First—From Figure 7, determine the ΔP for 100 feet of ¾" pipe by locating the intersection of the diagonal line for ¾" pipe and the 10 gpm line: Read ΔP = 10 psig.

Second—For 75 feet of pipe:

$$\Delta P = \frac{75 \times 10}{100} = 7.5 \text{ psig}$$

| Applied Pressure | Nominal Standard Pipe Size | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| PSIG | 1/8" | 1/4" | 3/8" | 1/2" | 3/4" | 1" | 1-1/4" | 1-1/2" | 2" | 2-1/2" | 3" |
| 5 | 0.10 | 0.24 | 0.50 | 0.92 | 1.4 | 2.6 | 5.3 | 8.0 | 16 | 25 | 47 |
| 10 | 0.14 | 0.34 | 0.73 | 1.3 | 2.0 | 3.7 | 7.8 | 12 | 23 | 37 | 68 |
| 20 | 0.21 | 0.50 | 1.1 | 1.9 | 2.9 | 5.4 | 11 | 17 | 33 | 53 | 100 |
| 40 | 0.30 | 0.73 | 1.5 | 2.8 | 4.2 | 8.0 | 16 | 25 | 48 | 78 | 145 |
| 60 | 0.37 | 0.90 | 1.9 | 3.5 | 5.2 | 10 | 21 | 31 | 60 | 96 | 180 |
| 80 | 0.43 | 1.1 | 2.2 | 4.1 | 6.1 | 12 | 24 | 36 | 70 | 112 | 210 |
| 100 | 0.48 | 1.2 | 2.5 | 4.6 | 6.8 | 13 | 27 | 41 | 80 | 128 | 240 |
| 150 | 0.60 | 1.5 | 3.1 | 5.8 | 8.5 | 16 | 33 | 51 | 99 | 155 | 290 |
| 200 | 0.71 | 1.7 | 3.7 | 6.8 | 10 | 19 | 39 | 60 | 115 | 185 | 350 |
| 250 | 0.80 | 2.0 | 4.2 | 7.6 | 11 | 21 | 44 | 67 | 130 | 210 | 390 |

Table 5. Maximum Recommended Water Flow (gpm) Through A.N.S.I. Standard Weight Schedule 40 Pipe.

Use Table 5 as a guide in sizing piping in water systems.

The flow values in Table 5 are based on a pressure drop as shown below.

| Pressure Drop per 100 ft of Pipe | Pipe Size (Inches) |
|---|---|
| 10% of Applied Pressure | ⅛, ¼, ⅜, ½ |
| 5% of Applied Pressure | ¾, 1, 1¼, 1½, 2, 2½, 3 |

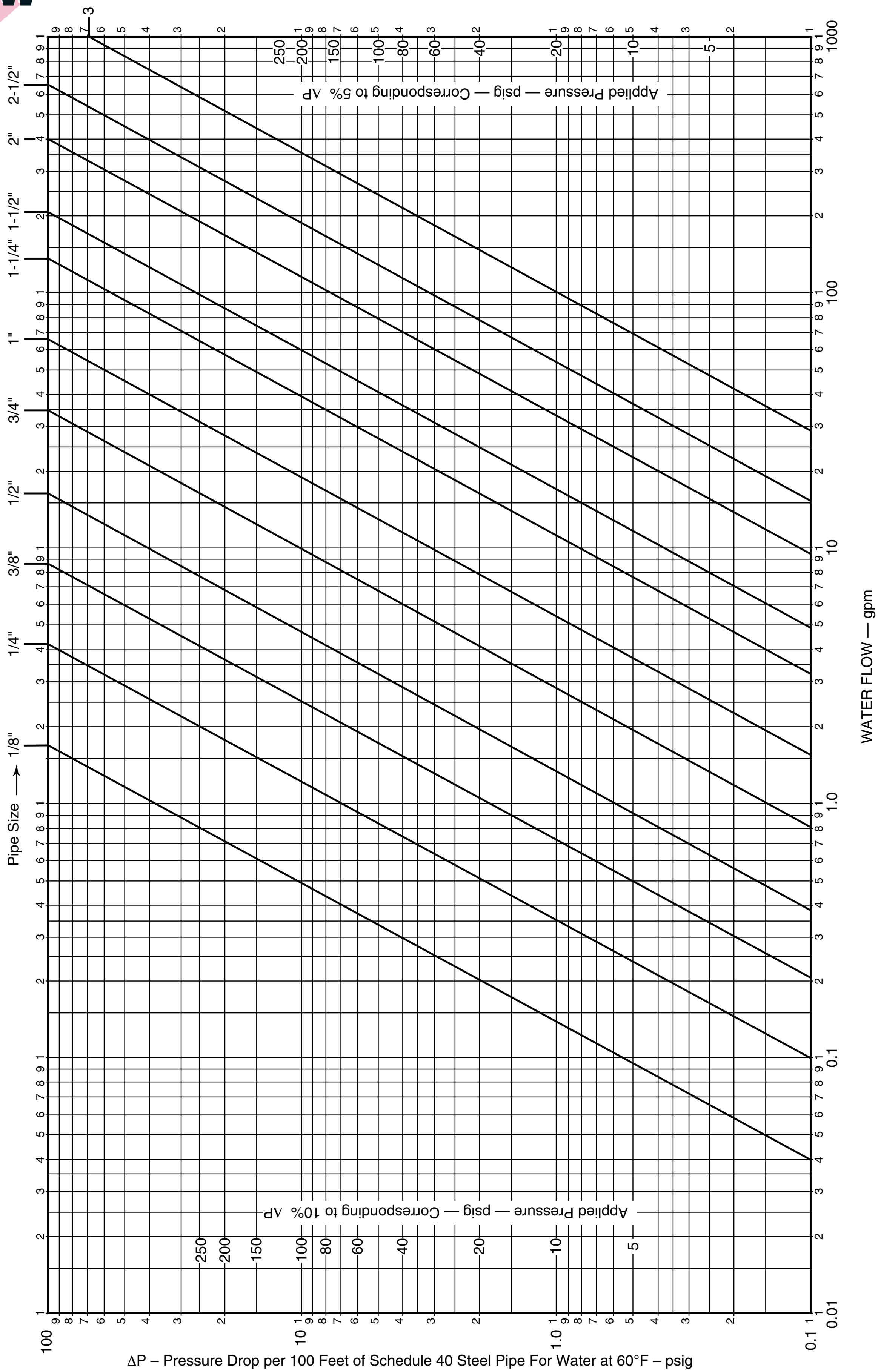



Figure 7. Water Flow – Pressure Drop Graph

# HOW TO DETERMINE PROPER AIR VALVE SIZE

Most manufacturers catalogs give flow rating Cv for the valve, which was established using proposed National Fluid Power Association (NFPA) standard T3.21.3. The following tables and formulas will enable you to quickly size a valve properly. The traditional, often used, approach of using the valve size equivalent to the port in the cylinder can be very costly. Cylinder speed, not port size, should be the determining factor.

The following Cv calculations are based upon simplified formulas which yield results with acceptable accuracy under the following standard conditions: **Air at a temperature of 68°F (20°C)**

Absolute downstream or secondary pressure must be 53% of absolute inlet or primary pressure or greater. Below 53%, the air velocity my become sonic and the Cv formula does not apply. To calculate air flow to atmosphere, enter outlet pressure p2 as 53% of absolute p2. Pressure drop ∆P would be 47% of absolute inlet pressure. These valves have been calculated for a Cv = 1 in Table 3.

## Nomenclature

- B Pressure Drop Factor
- C Compression Factor
- Cv Flow Factor
- D Cylinder Diameter (IN)
- F Cylinder Area (SQ IN)
- L Cylinder Stroke (IN)
- p1 Inlet or Primary Pressure (PSIG)
- p2 Outlet or Secondary Pressure (PSIG)
- ∆P Pressure Differential (p1 - p2) (PSID)
- q Air Flow at Actual Condition (CFM)
- Q Air Flow of Free Air (SCFM)
- t Time to Complete One Cylinder Stroke (SEC)
- T Absolute Temperature at Operating (°R) Pressure. Deg R = Deg F + 460

| Bore Size D (in.) | Push Bore F (sq. in.) | Bore Size D (in.) | Push Bore F (sq. in.) |
|---|---|---|---|
| 3/4" | .44 | 4" | 12.57 |
| 1" | .79 | 4-1/2" | 15.90 |
| 1-1/8" | .99 | 5" | 19.64 |
| 1-1/4" | 1.23 | 6" | 28.27 |
| 1-1/2" | 1.77 | 7" | 34.48 |
| 1-3/4" | 2.41 | 8" | 50.27 |
| 2" | 3.14 | 10" | 78.54 |
| 2-1/2" | 4.91 | 12" | 113.10 |
| 3-1/4" | 8.30 | 14" | 153.94 |

Table 1: Cylinder Push Bore Area F for Standard Size Cylinders

## Valve Sizing For Cylinder Actuation—Direct Formula

cylinder area (SQ IN) (see table 1) **F x** cylinder stroke (IN) **L x** compression factor (see table 2) **C**

**Cv =** ————————————————

pressure drop factor **B x** (see table 2) time to complete cylinder stroke **t x 29** (SEC)

$$Cv = \frac{F \times L \times C}{B \times t \times 29}$$

## Example:

Cylinder size 4" Dia. x 10" stroke. Time to extend: 2 seconds. Inlet pressure 90 PSIG. Allowable pressure drop 5 PSID. Determine Cv.

Solution: Table 1 F = 12.57 SQ IN
Table 2 C = 7.1
B = 21.6

$$Cv = \frac{12.57 \times 10 \times 7.1}{21.6 \times 2 \times 29} = 0.7$$

Select a valve that has a Cv factor of 0.7 or higher. In most cases a ¼" valve would be sufficient

It is considered good engineering practice to limit the pressure drop ∆P to approximately 10% of primary pressure p1. The smaller the allowable pressure drop, the larger the required valve will become.

After the minimum required Cv has been calculated, the proper size valve can be selected from the catalog.

| Inlet Pressure (psig) | Com-pression Factor C | Pressure Drop Factor B For Various Pressure Drops ∆P | | | | |
|---|---|---|---|---|---|---|
| | | 2 PSID | 5 PSID | 10 PSID | 15 PSID | 20 PSID |
| 10 | 1.7 | 6.5 | | | | |
| 20 | 2.4 | 7.8 | 11.8 | | | |
| 30 | 3.0 | 8.9 | 13.6 | 18.0 | | |
| 40 | 3.7 | 9.9 | 15.3 | 20.5 | 23.6 | |
| 50 | 4.4 | 10.8 | 16.7 | 22.6 | 26.4 | 29.0 |
| 60 | 5.1 | 11.7 | 18.1 | 24.6 | 29.0 | 32.0 |
| 70 | 5.8 | 12.5 | 19.3 | 26.5 | 31.3 | 34.8 |
| 80 | 6.4 | 13.2 | 20.5 | 28.2 | 33.5 | 37.4 |
| 90 | 7.1 | 13.9 | 21.6 | 29.8 | 35.5 | 39.9 |
| 100 | 7.8 | 14.5 | 22.7 | 31.3 | 37.4 | 42.1 |
| 110 | 8.5 | 15.2 | 23.7 | 32.8 | 39.3 | 44.3 |
| 120 | 9.2 | 15.8 | 24.7 | 34.2 | 41.0 | 46.4 |
| 130 | 9.8 | 16.4 | 25.6 | 35.5 | 42.7 | 48.4 |
| 140 | 10.5 | 16.9 | 26.5 | 36.8 | 44.3 | 50.3 |
| 150 | 11.2 | 17.5 | 27.4 | 38.1 | 45.9 | 52.1 |
| 160 | 11.9 | 18.0 | 28.2 | 39.3 | 47.4 | 53.9 |
| 170 | 12.6 | 18.5 | 29.0 | 40.5 | 48.9 | 55.6 |
| 180 | 13.2 | 19.0 | 29.8 | 41.6 | 50.3 | 57.2 |
| 190 | 13.9 | 19.5 | 30.6 | 42.7 | 51.7 | 58.9 |
| 200 | 14.6 | 20.0 | 31.4 | 43.8 | 53.0 | 60.4 |
| 210 | 15.3 | 20.4 | 32.1 | 44.9 | 54.3 | 62.0 |
| 220 | 16.0 | 20.9 | 32.8 | 45.9 | 55.6 | 63.5 |
| 230 | 16.7 | 21.3 | 33.5 | 46.9 | 56.8 | 64.9 |
| 240 | 17.3 | 21.8 | 34.2 | 47.9 | 58.1 | 66.3 |
| 250 | 18.0 | 22.2 | 34.9 | 48.9 | 59.3 | 67.7 |

Table 2: Compression Factor C and Pressure Drop Factor B.

## Valve Sizing with Cv = 1 Table

(For nomenclature see previous page)

This method can be used if the required are flow is known or has been calculated with the formulas as shown below:

1. $$Q = 0.0273 \frac{D^2L}{t} \times \frac{p_2 + 14.7}{14.7} \text{ (SCFM)}$$

Conversion of CFM to SCFM

2. $$Q = q \times \frac{p_2 + 14.7}{14.7} \times \frac{528}{T} \text{ (SCFM)}$$

Flow Factor Cv (standard conditions)

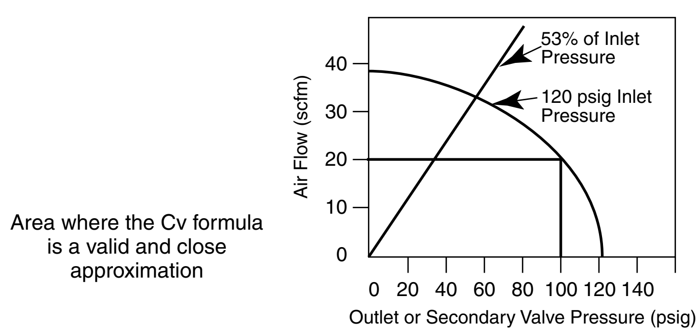
3. $$Cv = \frac{1.024 \times Q}{\sqrt{\Delta P \times (p_2 + 14.7)}}$$ Proposed NFPA Standard T3.21.3

Maximum pressure drop $\Delta p$ across the valve should be less than 10% of inlet pressure $p_1$.

| Inlet Pressure (psig) | Air Flow Q (SCFM) for Variou Pressure Drops ΔP at Cv = | | | | | ir Flow (SCFM) to Atmosphere |
|---|---|---|---|---|---|---|
| | 2 PSID | 5 PSID | 10 PSID | 15 PSID | 20 PSID | |
| 10 | 6.7 | | | | | 12.0 |
| 20 | 7.9 | 11.9 | | | | 16.9 |
| 30 | 9.0 | 13.8 | 18.2 | | | 21.8 |
| 40 | 9.9 | 15.4 | 20.6 | 23.8 | | 26.6 |
| 50 | 10.8 | 16.9 | 22.8 | 26.7 | 29.2 | 31.5 |
| 60 | 11.6 | 18.2 | 24.8 | 29.2 | 32.3 | 36.4 |
| 70 | 12.3 | 19.5 | 26.7 | 31.6 | 35.1 | 41.2 |
| 80 | 13.0 | 20.7 | 28.4 | 33.8 | 37.7 | 46.1 |
| 90 | 13.7 | 21.8 | 30.0 | 35.8 | 40.2 | 51.0 |
| 100 | 14.4 | 22.9 | 31.6 | 37.8 | 42.5 | 55.9 |
| 110 | 15.0 | 23.9 | 33.1 | 39.6 | 44.7 | 60.7 |
| 120 | 15.6 | 24.9 | 34.5 | 41.4 | 46.8 | 65.6 |
| 130 | 16.1 | 25.8 | 35.8 | 43.1 | 48.8 | 70.5 |
| 140 | 16.7 | 26.7 | 37.1 | 44.7 | 50.7 | 75.3 |
| 150 | 17.2 | 27.6 | 38.4 | 46.3 | 52.5 | 80.2 |
| 160 | 17.7 | 28.4 | 39.6 | 47.8 | 54.3 | 85.1 |
| 170 | 18.2 | 29.3 | 40.8 | 49.3 | 56.0 | 90.0 |
| 180 | 18.7 | 30.1 | 42.0 | 50.7 | 52.7 | 94.8 |
| 190 | 19.2 | 30.9 | 43.1 | 52.1 | 59.4 | 99.7 |
| 200 | 19.6 | 31.6 | 44.2 | 53.4 | 60.9 | 104.6 |
| 210 | 20.1 | 32.4 | 45.2 | 54.8 | 62.5 | 109.4 |
| 220 | 20.5 | 33.1 | 46.3 | 56.1 | 64.0 | 114.3 |
| 230 | 21.0 | 33.8 | 47.3 | 57.3 | 65.5 | 119.2 |
| 240 | 21.4 | 34.5 | 48.3 | 58.6 | 66.9 | 124.0 |
| 250 | 21.8 | 35.2 | 49.3 | 59.8 | 68.3 | 128.9 |

Table 3: Air Flow Q (SCFM) For Cv = 1

**Flow Curves — How to Read Them**




## Example 1: Find air flow Q (SCFM) if Cv is known. Cv (from valve catalog) = 1.8

Primary pressure p1 = 90 PSIG

Pressure drop across valve $\Delta P$ = 5 PSID

Flow through valve from Table 3 for Cv = 1: 21.8 SCFM

Q = Cv of valve x air flow at Cv = 1 (SCFM)

Q = 1.8 x 21.8 = 39.2 SCFM

## Example 2: Find Cv if air flow Q (SCFM) is given.

Primary pressure p1 = 90 PSIG

Pressure drop $\Delta P$ = 10 PSID

Air Flow-Q = 60 SCFM

Flow through valve from Table 3 for Cv = 1: 30 SCFM

$$Cv = \frac{\text{Air Flow Q (SCFM)}}{\text{Air Flow at Cv = 1 (SCFM)}}$$

$$Cv = \frac{60 \text{ SCFM}}{30} = 2.0$$

A valve with a Cv of minimum 2 should be selected.

## Example 3: Find Cv if air flow Q (SCFM) to atmosphere is given (from catalog).

Primary pressure p1 = 90 PSIG

Air flow to atmosphere Q = 100 SCFM

Flow to atmosphere through valve from Table 3 for Cv = 1: 51 SCFM

$$Cv = \frac{\text{Air Flow to atmosphere Q (SCFM)}}{\text{Air Flow to atmosphere at Cv = 1 (SCFM)}}$$

$$Cv = \frac{100}{51} = 2.0$$

Flow given in catalog is equivalent to a valve with Cv = 2. This conversion is often necessary to size a valve properly, since some manufacturers do not show the standard Cv to allow a comparison.

## Example 4: Find Cv if cylinder size and stroke speed is known, using the formulas 1 and 3

Primary pressure = 90 PSIG

Pressure drop across valve 5 PSID

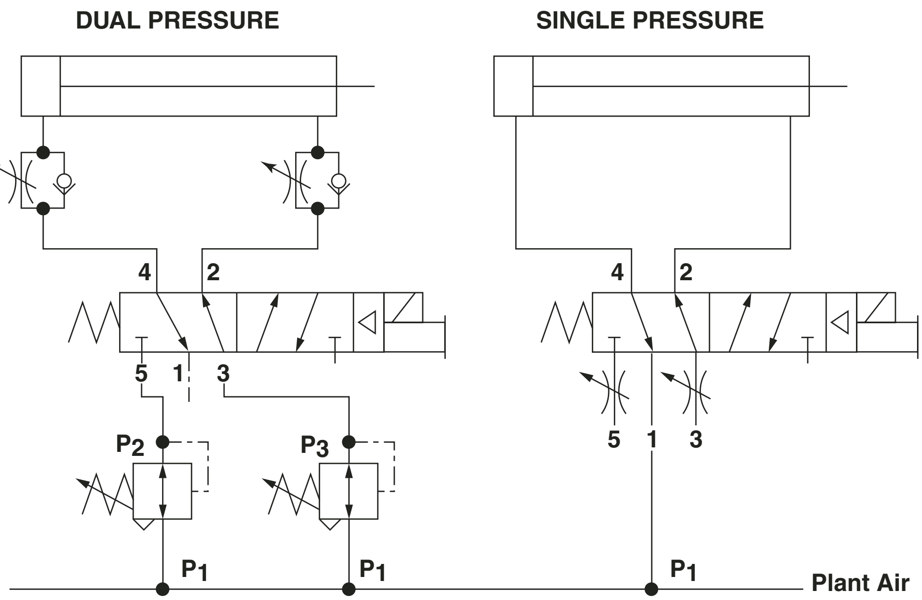
Cylinder size 4" dia. x 10" stroke

Time to complete stroke 2 sec.

$$Q = 0.0273 = \frac{42 \times 10}{2} \times \frac{85 + 14.7}{14.7} = 14.81 \text{ SCFM}$$

$$Cv = \frac{1.024 \times 14.81}{\sqrt{5 \times (85 + 14.7)}} = 0.7$$

# SAVINGS WITH DUAL PRESSURES VALVES




"Dual pressure" means using two different supply pressures to the valve. One supply acts to extend the cylinder, and the other supply acts to retract the cylinder when the valve is shifted.

Justification of a dual pressure versus a single pressure valve can be done quickly, using this simple formula. Savings in air consumption is the most important consideration of the use of dual pressure valves.

$$K = \frac{D2 \times S \times (2xp_1 - p_2 - p_3) \times Z \times N}{560{,}000} \text{ (\$HR)} \qquad N = \frac{60 \text{ Sec}}{t_1 + t_2} \text{ (CPM)}$$

## Nomenclature

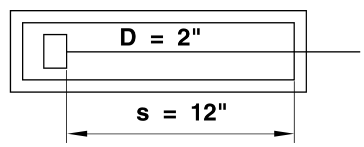
| | | |
|---|---|---|
| D = | Piston Diameter of Cylinder | (IN) |
| K = | Cost Savings per Hour | ($HR) |
| p1 = | Plant Air Pressure | (PSIG) |
| p2 = | Work Stroke Pressure (Reduced) | (PSIG) |
| p3 = | Return Stroke Pressure (Reduced) | (PSIG) |
| t1 = | Work Stroke | (SEC) |
| t2 = | Return Stroke | (SEC) |
| S = | Cylinder Stroke | (IN) |
| N = | Cycles Per Minute | (CPM) |
| Z = | Cost to compress 1000 SCF of air to 150 psig | ($/1000 SCF) |

(1976 estimate: $0.24/1000 SCF at 150 psig. Source: Assembly Engineering, page 50, May 1976)

## Assumptions:

1. Rod diameter of cylinder is partially accounted for in the constant (560,000). Except for very small cylinders, where the use of dual pressure is questionable anyway, the formula is sufficiently accurate for most practical applications.
2. Atmospheric Pressure = 14.7 psia
3. Standard Temperature = 68°F

## Example:




Work Stroke $t_1$ = 2 sec
Return Stroke $t_2$ = 2 sec
Plant Air Pressure $p_1$ = 150 psig
Work Stroke Pressure $p_2$ = 100 psig
Return Stroke Pressure $p_3$ = 30 psig
Cost of 1000 SCF Compressed Air Z = $0.24

$$N = \frac{60}{2 + 2} = 15$$

Calculate Savings per 8 Hour Shift

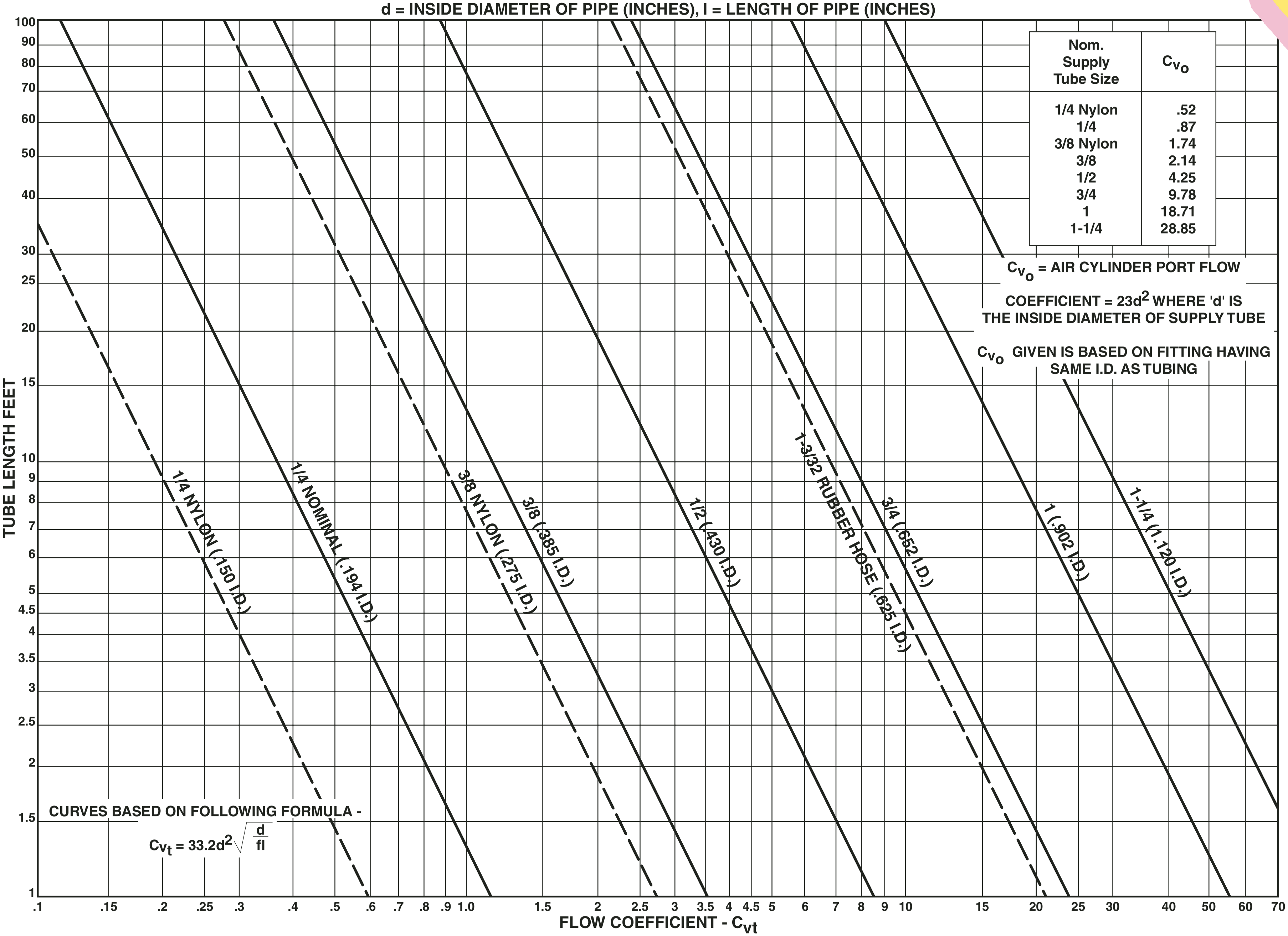
$$K = \frac{2^2 \times 12 \times (150 \times 2 - 100 - 30) \times 0.24 \times 15}{5.6 \times 10^5} = \$0.053/HR$$

## Savings are $0.42 for 8 hours

Conclusion:

As demonstrated in this example, savings for just one small cylinder result in a very short pay back period for the required additional one or two regulators. It should be kept in mind that a pressure reduction will result in a cylinder speed reduction. It is also important that relieving regulators be used.

## FLOW COEFFICIENT FOR SMOOTH WALL TUBING - $C_{vt}$
TUBE FRICTION FACTOR f - .02
d = INSIDE DIAMETER OF PIPE (INCHES), l = LENGTH OF PIPE (INCHES)




| Nom. Supply Tube Size | $C_{v_o}$ |
|---|---|
| 1/4 Nylon | .52 |
| 1/4 | .87 |
| 3/8 Nylon | 1.74 |
| 3/8 | 2.14 |
| 1/2 | 4.25 |
| 3/4 | 9.78 |
| 1 | 18.71 |
| 1-1/4 | 28.85 |

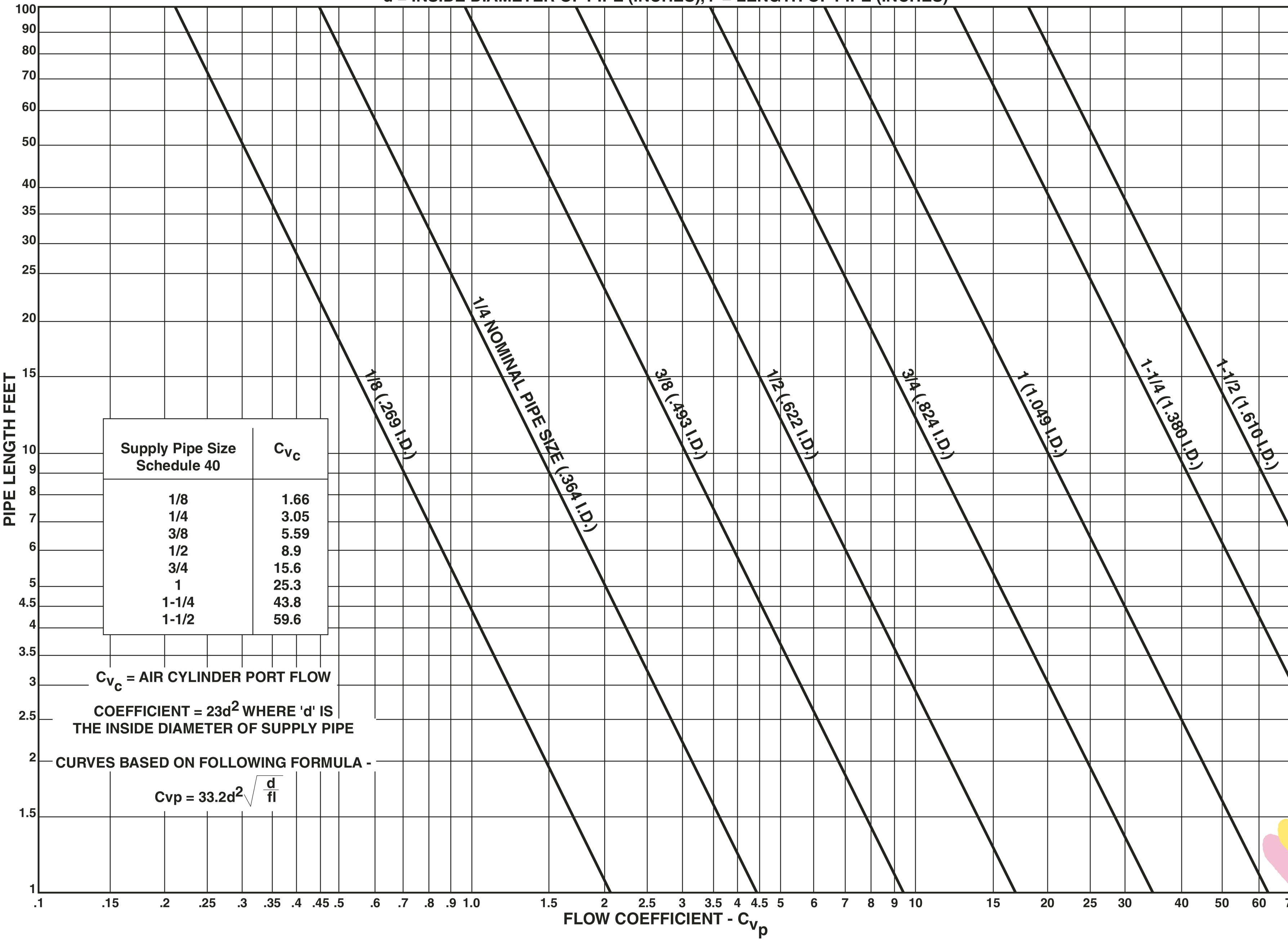
$C_{v_o}$ = AIR CYLINDER PORT FLOW COEFFICIENT = $23d^2$ WHERE 'd' IS THE INSIDE DIAMETER OF SUPPLY TUBE

$C_{v_o}$ GIVEN IS BASED ON FITTING HAVING SAME I.D. AS TUBING

CURVES BASED ON FOLLOWING FORMULA -

$$C_{v_t} = 33.2d^2\sqrt{\frac{d}{fl}}$$

## FLOW COEFFICIENT FOR SCHEDULE 40 STEEL PIPE - $C_{v_p}$
TUBE FRICTION FACTOR f - .03
d = INSIDE DIAMETER OF PIPE (INCHES), l = LENGTH OF PIPE (INCHES)




| Supply Pipe Size Schedule 40 | $C_{v_c}$ |
|---|---|
| 1/8 | 1.66 |
| 1/4 | 3.05 |
| 3/8 | 5.59 |
| 1/2 | 8.9 |
| 3/4 | 15.6 |
| 1 | 25.3 |
| 1-1/4 | 43.8 |
| 1-1/2 | 59.6 |

$C_{v_c}$ = AIR CYLINDER PORT FLOW COEFFICIENT = $23d^2$ WHERE 'd' IS THE INSIDE DIAMETER OF SUPPLY PIPE

CURVES BASED ON FOLLOWING FORMULA -

$$Cvp = 33.2d^2\sqrt{\frac{d}{fl}}$$

# SELECTED SI UNITS FOR FLUID POWER USAGE

## Extracted from ISO 1000 with National Fluid Power Association Permission

| Quantity | Symbol | Customary U.S. Unit | | SI Units | | | | Notes |
|---|---|---|---|---|---|---|---|---|
| | | | Abbreviation | Preferred Unit | | | | |
| Angular Velocity | $\omega$ | radian per second | rad/s | rad/s | | | | |
| Area | A or S | square inch | in$^2$ | cm$^2$ | m$^2$ | mm$^2$ | | |
| Bulk Modulus Liquids) | K | pounds per square inch | psi | bar | N/m$^2$ | | | |
| Capacity (Displacement) | V | cubic inches per revolution | cipr | ml /r | l /r | | | 1, 7 |
| Coefficient of Thermal | $\alpha$ | °F-1 | 1/°F | 1/K | | | | |
| Expansion (cubic) | | | | | | | | |
| Dynamic Viscosity | $\mu$ | centipoise | cP | cP | P | Pa s | | 2 |
| Efficiency | $\eta$ | percent | | percent | | | | 3 |
| Force | F | pound (f) | (lb) f | N | kN | | | |
| Frequency | f | cycles per second | cps | Hz | kHz | | | |
| Kinematic Viscosity | $\nu$ | Saybolt Universal Seconds | SUS | cSt | m$^2$/s | | | 4, 9 |
| Length | l | inch | in. | mm | m | µm | | |
| Linear Velocity | v | feet per second | ft/s | m/s | | | | |
| Mass | m | pound (m) | lb (m) | kg | Mg | g | | |
| Mass Density | $\rho$ | pound (m) per cubic foot | lb (m)/ft$^3$ | kg/m$^3$ | kg/dm$^3$ | kg/l | | 5 |
| Mass Flow | M | pound (m) per second | lb (m)/s | kg/s | g/s | | | |
| Power | P | horsepower | HP | kW | W | | | |
| Pressure (Above Atmospheric) | p | pounds per square inch | psi | bar | mbar | Pa | kPa | 6 |
| Pressure (Below Atmospheric) | p | inches of mercury, absolute | in. Hg | bar, abs | Pa | kPa | | 6 |
| Quantity of Heat | $Q_c$ | British Thermal Unit | BTU | J | kJ | MJ | | |
| Rotational Frequency (Shaft Speed) | n | revolutions per minute | RPM | r/min | r/s | | | |
| Specific Heat Capacity | c | British Thermal Unit per pounds mass degree Fahrenheit | BTU/lb(m)°F | J(kgK) | | | | |
| Stress (Materials) | $\sigma$ | pounds per square inch | psi | daN/mm$^2$ | MPa | | | |
| Surface Roughness | | microinch | µ in | grade N_ | µ m | | | 10 |
| Temperature (Customary) | $\theta$ | degree Fahrenheit | °F | °C | | | | |
| Temperature (Interval) | | degree Fahrenheit | °F | °C | | | | |
| Temperature (Thermodynamic) | T | Rankine | °R | K | | | | |
| Time | t | second | s | min | s | µ | | |
| Torque (Moment of Force) | T | pounds (F) - inch | lb (f) - in. | Nm | kNm | mNm | | |
| Volume | V | gallon | U.S. gal | l | m$^3$ | cm$^3$ | | 7 |
| Volumetric Flow (Gases) | Q (ANR) | standard cubic feet per minute | scfm | dm$^3$/s$_n$ | m$^3$/s$_n$ | cm$^3$/s$_n$ | | 8 |
| Volumetric Flow (Liquids) | Q | gallons per minute | USGPM | l /min | l /sec | ml /s | | 7 |
| Work | W | foot-pound (f) | ft-lb (f) | J | | | | |

## Notes to the Table of Selected SI Units for Fluid Power Usage

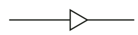
1. The capacity (displacement) of a rotary device is given as "per revolution" Non-rotary devices are expressed as "per cycle".
2. The centipoise, cP, is a non-SI unit, use of which is permitted by ISO 1000. The centipoise is equal to $10^{-3}$ N s/m$^2$.
3. Efficiencies are normally stated as "percent" but the use of a ratio is also permitted.
4. The centistokes, cSt, is a non-SI unit, use of which is permitted by ISO 1000. The centistokes is equal to $10^{-6}$ m$^2$/s.
5. Subject to change to kg/_ to correspond to recent action by ISO/TC 28 (Petroleum Fluids).
6. The bar is a non-SI unit, use of which is permitted by ISO 1000. The bar is a special name for a unit of pressure and is assumed to be "gage" unless otherwise specified. 1 bar = 100 kPa; 1 bar = $10^5$ N/m$^2$.
7. The litre is a non-SI unit use of which is permitted by ISO 1000. The litre is a special name for a unit of liquid measure and is exactly equal to the cubic decimetre.
8. The abbreviation "ANR" means that the result of the measurement has been referred to the Standard Reference Atmosphere (Atmosphere Normale de Reference) as defined in clause 2.2 of ISO/R 554, "Standard atmospheres for conditioning and/or testing - Standard reference atmosphere - Specifications." This abbreviation should immediately follow the unit used or the expression of the quantity.
9. For conversion from U.S. to Si units, see ANSI/Z11.129-1972 (ASTM/D2161-1971).
10. For conversion from U.S. to SI units, see ISO/R 1302-1971.

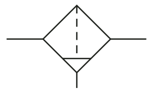
# CONVERSION TABLES

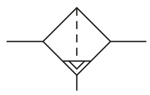
| To Convert | Into | Multiply By |
|---|---|---|
| atmospheres | bar | 1.0135 |
| atmospheres | mm of mercury | 760.0 |
| atmospheres | pounds/sq. in. | 14.696 |
| bars | atmospheres | 0.9869 |
| bars | kilopascal | 100.0 |
| bars | Newton/sq. meters | 100,000.0 |
| bars | pounds/sq. in. | 14.5 |
| Btu | foot-lbs. | 778.3 |
| Btu | horsepower/hrs | $3.927 \times 10^{-4}$ |
| Btu | joules | 1054.8 |
| Btu | kilogram-calories | 0.252 |
| Btu | kilowatts-hrs. | $2.928 \times 10^{-4}$ |
| Btu/pound °F | kilogram-calories/kg °C | 1.0 |
| Centigrade | Fahrenheit | 9/5C° + 32° |
| Centigrade | Kelvin | C° + 273° |
| centimeters | feet | 0.0328 |
| centimeters | inches | 0.3937 |
| centipoise | gram/cm. sec. | 0.01 |
| centipoise | pound mass/ft. sec. | 0.000672 |
| centistokes | sq. feet/sec. | $1.076 \times 10^{-6}$ |
| cubic centimeters | cu inches | 0.06102 |
| cubic feet | cu cms | 28,317.0 |
| cubic feet | cu meters | 0.028317 |
| cubic feet | liters | 28.317 |
| cubic feet/min. | cu dms.sec. | 0.472 |
| cubic feet/min. | pounds of air/hr. | 4.5 |
| cubic feet/min. | cu Newton meters/hr. | 1.7 |
| cubic inches | cu cms | 16.39 |
| cubic inches | cu mm | 16,387.0 |
| cubic inches | liters | 0.01639 |
| cubic meters | cu feet | 35.31 |
| Fahrenheit | Centigrade | 5/9 (F° -32°) |
| Fahrenheit | Rankine | F° + 460° |
| feet | centimeters | 30.48 |
| feet | meters | 0.3048 |
| feet | millimeters | 304.8 |
| foot-pounds | Newton-meters | 1.356 |
| foot-pounds/sec. | Newton meters/sec. | 1.356 |
| gallons (US) | liters | 3.785 |
| gallons/min. | cu in./min. | 231.0 |
| gallons/min. | liters/min. | 3.785 |
| gallons/min. | pounds of water/hr. | 500.0 |
| grams | ounces (avdp) | 0.3527 |
| grams/cu cm | pounds/cu ft | 62.43 |
| grams/cu cm | pounds/cu in. | 0.03613 |
| horsepower | foot-lbs/min. | 33,000.00 |
| horsepower | foot-lbs/sec. | 550.0 |
| horsepower (metric) | horsepower | 0.9863 |
| horsepower | horsepower (metric) | 1.014 |
| horsepower | watts | 745.7 |
| inches | centimeters | 2.54 |
| inches | meters | 0.0254 |
| inches | millimeters | 25.4 |
| inches of mercury | pounds/sq. in. | 0.4912 |
| inches of water (4°C) | pounds/sq. in. | 0.03613 |
| kilograms | pounds | 2.205 |
| kilograms/cu meter | pounds/cu ft. | 0.06243 |
| kilograms-calories | Btu | 3.968 |
| kilopascal | bar | 0.01 |
| kilopascal | psi | 0.145 |
| kilowatt-hrs. | Btu | 3415.0 |

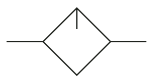
| To Convert | Into | Multiply By |
|---|---|---|
| liters | cu dm | 1.0 |
| liters | cu feet | 0.0351 |
| liters | cu inches | 61.02 |
| liters | cu meters | 0.001 |
| liters | gallons (US) | 0.2642 |
| liters/min | gals/min | 0.2642 |
| meters | feet | 3.281 |
| meters | inches | 39.37 |
| meters | yards | 1.094 |
| millimeters | inches | 0.03937 |
| millimeters of mercury | psi | 0.0194 |
| Newton/sq. meter | pascal | 1.0 |
| Newton-meter | foot-pounds | 0.7375 |
| Newton-meter | joule | 1.0 |
| Newtonmeter/sec. | foot-pounds/sec. | 0.7375 |
| Newton-meter/sec. | watts | 1.0 |
| ounces | grams | 28.35 |
| pounds | kilograms | 0.4536 |
| pounds/cu ft. | grams/cu cm | 0.01602 |
| pounds/cu ft. | kgs/cu meter | 16.02 |
| pounds/cu in. | gms/cu cm | 27.68 |
| pounds/hr. | kilograms/hr. | 0.454 |
| pounds/sec. | kilograms/hr. | 1,633.0 |
| pounds-sec./sq. ft. | pounds mass/ft. sec. | 32.2 |
| pounds/sq. in. | atmospheres | 0.06804 |
| pounds/sq. in. | bar | 0.069 |
| pounds/sq. in. | inches of mercury | 2.036 |
| pounds/sq. in. | inches of water | 27.7 |
| pounds/sq. in. | kilopascal | 6.895 |
| pounds/sq. in. | mm of mercury | 51.6 |
| square centimeters | sq. feet | 0.001076 |
| square centimeters | sq. inches | 0.155 |
| square feet | sq. cms | 929.0 |
| square feet | sq. meters | 0.0929 |
| square feet/sec. | centistokes | 92,903.0 |
| square inches | sq. cms | 6.452 |
| square inches | sq. millimeters | 645.2 |
| square meters | sq. feet | 10.76 |
| square meters | sq. yards | 1.196 |
| square millimeters | sq. inches | 0.00155 |
| square yards | sq. meters | 0.8361 |
| tons (metric) | kilograms | 1000.0 |
| tons (metric) | pounds | 2205.0 |
| tons (short) | pounds | 2000.0 |
| tons (short) | tons (metric) | 0.9072 |
| yards | meter | 0.9144 |

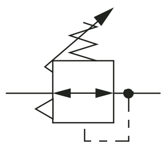
# CIRCUIT SYMBOLS

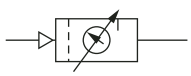
Direction of Flow in Pneumatic System


Manual Drain Filter


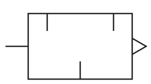
Automatic Drain Filter


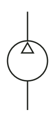
Lubricator


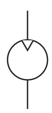
Airline Pressure Regulator (Adjustable, Relieving)


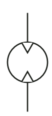
Airline Pressure Regulator and Lubricator With Gauge


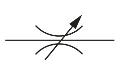
Pressure Gauge

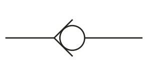

Muffler


Fixed Displacement Compressor


Undirectional Motor


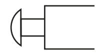
Bidirectional Motor


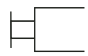
Adjustable Flow Control Valve


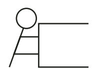
Check Valve


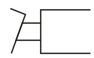
## Valve Operations

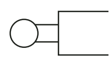
Spring

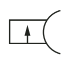

Push Button


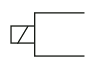
Manual Actuator


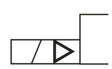
Push-Pull Lever


Pedal or Treadle


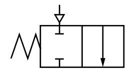
Mechanical Actuator


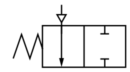
Pressure Compensated Actuator


Solenoid Actuator


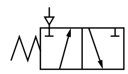
Solenoid and Pilot Actuator


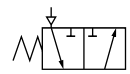
## 2-Way Valves

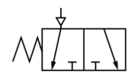
Normally Closed


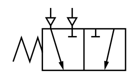
Normally Open


## 3-Way 2-Position Valves

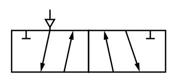
Normally Closed


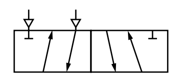
Normally Open


Distributor


Selector


## 4-Way 2-Position Valves

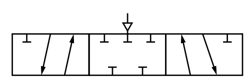
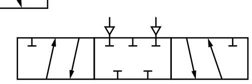
Single Pressure


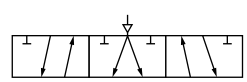
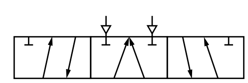
Dual Pressure


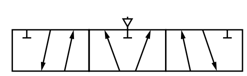
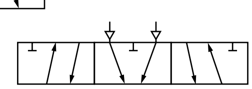
## 4-Way 3-Position Valves

Single Pressure

Dual Pressure

All Ports Blocked Center



Inlet To Cylinder Center



Cylinder To Exhaust Center



# USEFUL DIMENSIONAL DATA

| Diameter | | Internal Area Sq. In. | | | |
|---|---|---|---|---|---|
| | | Circle Area | Hose | Std. Pipe | .032 Wall Copper Tubing |
| 1/32 | (.0312) | .00077 | | | |
| 1/16 | (.0625) | .00307 | | | |
| 3/32 | (.0938) | .0069 | | | |
| 1/8 | (.1250) | .01227 | .01227 | .057 | .0029 |
| 5/32 | (.1562) | .01917 | | | |
| 3/16 | (.1875) | .02761 | | | .012 |
| 7/32 | (.2188) | .03758 | | | |
| 1/4 | (.2500) | .04909 | | | .0271 |
| 9/32 | (.2812) | .06213 | | | |
| 5/16 | (.3125) | .0767 | | | .0485 |
| 11/32 | (.3438) | .09281 | | | |
| 3/8 | (.3750) | .1104 | .11 | .191 | .076 |
| 13/32 | (.4062) | .1296 | | | |
| 7/16 | (.4375) | .1503 | | | .1095 |
| 15/32 | (.4688) | .1726 | | | |
| 1/2 | (.5000) | .1963 | .196 | .304 | .149 |
| 17/32 | (.2217) | .2217 | | | |
| 9/16 | (.2485) | .2485 | | | |
| 19/32 | (.2769) | .2769 | | | |
| 5/8 | (.3068) | .3068 | .307 | | .247 |
| 21/32 | (.5312) | .3382 | | | |
| 11/16 | (.5625) | .3712 | | | |
| 23/32 | (.5938) | .4057 | | | |
| 3/4 | (.7500) | .4418 | .442 | .533 | .370 |
| 13/16 | (.8125) | .5185 | | | |
| 7/8 | (.8750) | .6013 | | | |
| 15/16 | (.9375) | .6903 | | | |
| 1 | (1.000) | .7854 | .785 | .864 | .594 |
| 1-1/4 | (1.250) | 1.2272 | 1.227 | 1.496 | .922 |
| 1-1/2 | (1.500) | 1.767 | | 2.036 | |
| 2 | (2.000) | 3.1416 | 3.14 | 3.356 | |
| 2-1/2 | (2.500) | 4.9088 | | 4.788 | |
| 3 | (3.000) | 7.07 | 7.07 | 7.39 | |
| 3-1/2 | (3.500) | 9.62 | | | |
| 4 | (4.000) | 12.57 | 12.57 | | |
| 5 | (5.000) | 19.64 | | | |
| 6 | (6.000) | 28.27 | | | |
| 7 | (7.000) | 38.49 | | | |
| 8 | (8.000) | 50.27 | | | |
| 10 | (10.000) | 78.54 | | | |

# SUMMARY OF FORMULAS AND EQUIVALENTS

## Area and Volume

$A = D^2 \times 0.7854$ (or $A = \pi R^2$)

$V = D^2 \times 0.7854 \times L$

$\sqrt{\text{Area} / 0.7854}$

(A = area in sq. in., diameter in inches, V = volume in cu. in., L = length

## Temperature

Absolute temperature °R = °F + 460

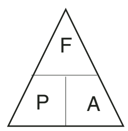
## Pressure

Standard conditions = 14.7 psia @ sea level (68°F, 36% Relative Humidity

Compression Ratio (standard conditions) – $\dfrac{\text{psig} + 14.7}{14.7}$

Compression Ratio (corrected for elevation) – $\dfrac{\text{psig} + \text{psia}}{\text{psia}}$

Pascal's Law –




F = P x A  F = Force in lbs./sq. in.

P = F/A  P = Pounds (lbs)

A = F/P  A = Area in sq. in.

psig (standard conditions) = psia -14.7
psia (standard conditions) = psig +14.7

## Flow

scfm = (area in sq. inches x stroke inches x CPM*) / 1728

cfm = $\dfrac{\text{area in sq. inches} \times \text{velocity in ft./min.}}{144\ \text{in}^2/\text{ft}^2}$

scfm = cfm x compression ratio

** CPM = Cycles per minute*

## Pressure Drop (ΔP)

psid = P1 - P2

$\Delta P$ Averaged for distance = $\dfrac{\text{psig rcvr.} - \text{psig tool}}{\text{distance ft.}}$

## Pressure / Volume

Boyles Law – $P_1V_1 = P_2V_2$

General Gas Law – $\dfrac{P_1V_1}{T_1} = \dfrac{P_2V_2}{T_2}$

Charles Law (variation) – $P_1 \times V_1 \times T_1 = P_2 \times V_2 \times T_2$

## Coefficient of Flow

$Cv = \dfrac{Q\ (\text{scfm})}{22.67} \sqrt{\dfrac{°F + 460}{\Delta P \times K}}$

K = $P_2$ absolute...if $\Delta P$ is less than 10%

K = ($P_1$ abs. + $P_2$ abs.) /2...if $\Delta P$ is 10% to 25%

K = $P_1$ absolute...if $\Delta P$ is greater than 25% (critical velocity)

## Line Drop

drop/inches = run/ft x % grade x 0.12
% grade = (drop/inches/0.12) / run/ft
1% to 2% grade recommended

## Compressed Air Cost

Cost = cfm x 60 x # hrs. x kWh/cfm x $/kWh

## Vacuum

negative psig = inches Hg x 0.49

inches Hg = psi/0.49

inches Hg x 1.133 = ft. $H_2O$

inches $H_2O$ x 0.036 = psi

1 foot $H_2O$ x 0.8826 = 1 inch Hg

Force = -P x A

Lifting force = inches Hg x 0.4912 x sq. in.area

## Receiver Sizing

$$\text{Volume (gallons)} = K \times \text{cfm} \times \frac{14.7}{\text{psig} + 14.7} \times 7.48$$

$$\text{Volume (gallons)} = K \times \text{cfm} \times \frac{14.7}{\text{psig} + 14.7} \times \frac{1728}{231}$$

(V = volume/gal. K = 1 continuous, K = 3 intermittent)

(7.48 converts cu. ft. to gal.)

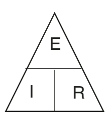
$$\text{Time} = \frac{\text{cu. ft. volume} \times (\text{Pmax-Pmin.})}{\text{cfm rcvr. consumption} \times 14.7}$$

## Cylinder Velocity

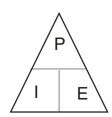
$$\text{Velocity (ft./sec. extend)} = \frac{\text{inches stroke}}{\text{extend time seconds}} + \frac{\text{extended dwell sec.} \times 60}{12}$$

$$\text{Velocity (ft./sec. retract)} = \frac{\text{inches stroke}}{\text{extend time seconds}} + \frac{\text{extended dwell sec.} \times 60}{12}$$

## Electrical




("the eagle flies over the indian at the river")




("pie")

E = I x R

I = E / R

R = E / I

P = I x E

I = P / E

E = P / I

P = I2 R

P = E2/R

(E = volts, I = amperes (current), R = Ohms (resistance), P = (Watts power)

8 bit = 256 increments of resolution

signal ratio (I/P) = amperes output / pressure input

volts per inch = stroke / reference potential

Kirchoffs Law – Rt = R1 + R2 + R3

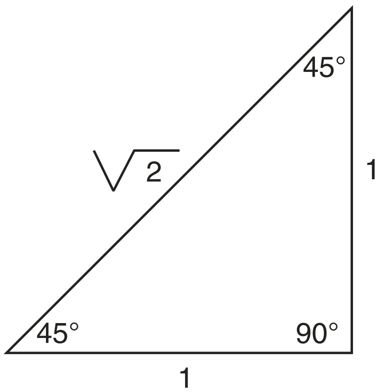
(Rt = total resistance)

## Moisture Content of Air

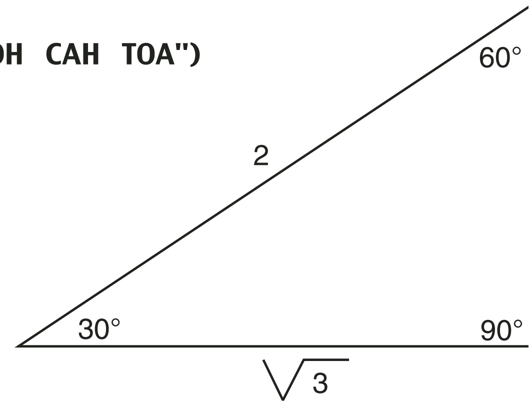
Dewpoint = Temperature at which moisture will condense

Relative Humidity = (Absolute humidity / humidity at saturation x 100

## Electrical




("SOH CAH TOA")




Sin 30° = 0.500 Sin 45° = 0.707 Sin 60° = 0.866

Cos 30° = 0.866 Cos 45° = 0.707 Cos 60° = 0.500

sin ø = opposite / hypotenuse cos ø = adjacent / hypotenuse

secant ø = hypotenuse / adjacent cosecant ø = hypotenuse / opposite

tan ø = opposite / adjacent cotan ø = adjacent / opposite

hypotenuse = $\sqrt{\text{(adjacent squared + opposite squared)}}$

## Mechanical

Speed Ratio = $\frac{\text{driven shaft or gear}}{\text{drive shaft or gear}}$

Torque = force x radius

Force = torque / radius

Motor Torque lb. - ft. = 5252 x hp / rpm

Motor Torque lb.- in. = 63025 x hp / rpm

Motor hp = lb. - in. torque x rpm / 5252

Motor hp = lb. - in. torque x rpm / 63025

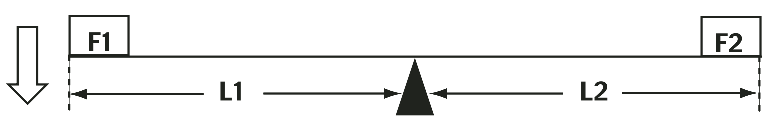
Work = force x distance

Power = force x distance / time

Horesepower – hp = rpm x ft. lb. torque / 5252

First class lever = $F_1$ x $L_1$ = $F_2$ x $L_2$ (F = force, L = Length)

Third class lever = $F_1$ x $L_2$ = $F_2$ x $L_1$ (F = force, L = length)




Mechanical advantage = total rod length / supported rod length

Bending moment = mechanical advantage x side force

Total Force = coefficient of friction x load

Up incline force = surface force + incline force

Down incline force = surface force - incline force

Surface force = coefficient of friction x load x Cos $\theta$

Incline force = load x sin $\theta$

Force along an incline = $F_1$ x $D_1$ = $F_2$ x $D_2$ (F = force, D = distance)

Rotary actuator torque - Torque = psig x area x pitch radius

## Mechanical Cont.

Gripper – $F_1 \times L_1 = F_2 \times 2$ or $F_2 = F_1 \times L_1 / L_2$ (F = force, L = load)

Jib Crane force = L x ($D_1 + D_2$) / sin x $D_1$

Jib Crane load = F x Sin X $D_1$ / ($D_1 + D_2$)

(L = load lbs., D1 = distance (in.) pivot to rod clevis. D2 = distance (in.) rod clevis to load)

Feet per minute = 0.2618 x dia. inches x rpm

Inches Hg = inches $H_2O$ / specific gravity Hg

Intensifier sizing – Pressure air x area air = pressure oil x area oil

Max. flow throgh an orifice (critical backpressure ratio) = > 53% P1 abs.

GPM = Area in. x Stroke in. x cycles per mn. x 0.004329

## Terminal Velocity

= 2 x distance / time in seconds

## Kinetic Energy (KE)

$$= \frac{\text{weight x terminal velocity squared}}{\text{2 x acceleration of gravity}}$$

(Acceleration of Gravity = 32.2 ft./sec./sec. OR 9.81 Meters/sec./sec.)

## Conversions and Equivalents:

29.92 in. Hg = 14.7 psia

760 mm Hg = 29.92 in. Hg = 33.899 ft-water = 10.34 Meters-water

1 micron = 0.000001 meter = 0.000039 inch

1 in. = 25,400 micron

231 cu.in. = 1 gallon

1728 cu. in. = 1 cu.ft.

7.48 gallons = 1 cu. ft.

1 micron Hg. = .0000193 psia

Newton = 0.1022 Kilograms = .2248 lbs.

Pounds = 4.448 Newtons

Specific gravity of mercury (Hg) = 13.5951

Specific gravity of water ($H_2O$) = 1

1 mm Hg = 0.0446 ft. water

Nm to Hp constant = 7124

## Common Friction Factors

| | Valves | Friction Factors |
|---|---|---|
| Gate Valves | full-open | 0.19 |
| | ¼ closed | 1.15 |
| | ½ closed | 5.60 |
| | ¾ closed | 24.00 |
| | Globe valve | 10.00 |
| | Plug cock | 0.26 |
| | Swing check | 2.50 |
| | 45° elbow | 0.42 |
| | 90° elbow | 0.90 |
| | Close return bend | 2.20 |
| | Standard tee | 1.80 |