Figure 1. Water Vapor Content

Also % Saturation at Sea Level Pressure

Pounds of Water per 1,000 Standard Cubic Feet of Air

Temperature –

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Determining Water Content in Compressed Air Systems . . 2
Determining Pressure Drop in Compressed Air Systems . . 4
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Useful Formulas . . 22
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 0.15 vapor content line and the 100 psig line — see Figure 1).

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, 6,000 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 = .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

Also % Saturation at Sea Level Pressure

1 lb/1,000 ft³ = 16 gr/m³
1 lb. water = 15.3 fl. ozs.
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:

\[ \Delta P = \frac{KQ^2}{1000} \left[ \frac{14.7}{14.7 + P} \right] \left[ \frac{460 + t}{520} \right] \]

\( \Delta P \) = 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 3/4" 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 \( 4.74 \times 150 = 7.11 \) psig since pressure drop is proportional to length.

Example 2:

Determine the pressure drop in a system containing 100 feet of 3/4" schedule 40 pipe, two 90° standard elbows, one globe valve and one 3/4" 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 left corresponding to this intersection: \( P = 0.8 \).
3. Select from Table 1 the \( K \) value for 3/4" 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: \( 0.119 \times 2 = 0.238 \).
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 3/4" 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 \( Kt \): \( 0.8 \times 9.308 = 7.446 \). 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 1/2" 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 1/8" 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 (\( \Delta P \)) 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, \( \Delta P \), by \( \frac{460 + \text{°F}}{520} \).

### Table 1. Values of K for 100 Feet of Schedule 40 Pipe

<table>
<thead>
<tr>
<th>Pipe Size</th>
<th>K</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/8&quot;</td>
<td>2300.</td>
</tr>
<tr>
<td>1/4&quot;</td>
<td>450.0</td>
</tr>
<tr>
<td>3/8&quot;</td>
<td>91.0</td>
</tr>
<tr>
<td>1/2&quot;</td>
<td>26.4</td>
</tr>
<tr>
<td>3/4&quot;</td>
<td>5.93</td>
</tr>
<tr>
<td>1&quot;</td>
<td>1.66</td>
</tr>
<tr>
<td>1-1/4&quot;</td>
<td>0.400</td>
</tr>
<tr>
<td>1-1/2&quot;</td>
<td>0.174</td>
</tr>
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<td>2&quot;</td>
<td>0.0467</td>
</tr>
<tr>
<td>2-1/2&quot;</td>
<td>0.0186</td>
</tr>
<tr>
<td>3&quot;</td>
<td>0.0060</td>
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</table>

### Table 2. Values of K for Commonly Used Fittings

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<tr>
<th>Fitting</th>
<th>Pipe Size</th>
<th>1/8&quot;</th>
<th>1/4&quot;</th>
<th>3/8&quot;</th>
<th>1/2&quot;</th>
<th>3/4&quot;</th>
<th>1&quot;</th>
<th>1-1/4&quot;</th>
<th>1-1/2&quot;</th>
<th>2&quot;</th>
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<tbody>
<tr>
<td>90° Standard Elbow</td>
<td></td>
<td>15.4</td>
<td>4.09</td>
<td>1.09</td>
<td>0.422</td>
<td>0.119</td>
<td>0.0432</td>
<td>0.01400</td>
<td>0.00711</td>
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<td>45° Standard Elbow</td>
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<td>8.3</td>
<td>2.20</td>
<td>0.53</td>
<td>0.216</td>
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<td>0.0216</td>
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<td>90° Street Elbow</td>
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<td>0.80</td>
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<td>0.0282</td>
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### Table 3. Values of K for Norgren Filters

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<th>Filter Type</th>
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</table>

### Table 4. Maximum Recommended Air Flow (scfm) thru A.N.S.I. Standard Weight Schedule 40 Pipe

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.

<table>
<thead>
<tr>
<th>Pressure Drop per 100 ft of Pipe</th>
<th>Pipe Size</th>
</tr>
</thead>
<tbody>
<tr>
<td>10% of Applied Pressure</td>
<td>1/8&quot;, 1/4&quot;, 3/8&quot;, 1/2&quot;</td>
</tr>
<tr>
<td>5% of Applied Pressure</td>
<td>3/4&quot;, 1&quot;, 1-1/4&quot;, 1-1/2&quot;, 2&quot;, 2-1/2&quot;, 3&quot;</td>
</tr>
</tbody>
</table>

### Table 4. Maximum Recommended Air Flow (scfm) thru A.N.S.I. Standard Weight Schedule 40 Pipe

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

Table 4. Maximum Recommended Air Flow (scfm) thru A.N.S.I. Standard Weight Schedule 40 Pipe

Helpful Engineering Information

Littleton, CO USA Phone 303-794-2611 www.norgren.com
FIGURE 2. Air Flow – Pressure Drop Graph For K = 1 In Equation \( \Delta P = KQ^2 \).
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)

∆P – Pressure Drop per 100 Feet of Pipe – psig

AIR FLOW – scfm

Applied Pressure – psig

DO NOT USE DOTTED LINE PORTION OF GRAPH

A: Maximum Flow 5%
B: Maximum Flow 10%
C: Maximum Flow 20%
D: Maximum Flow 40% Pressure Drop
Figure 5. Air Flow – Pressure Drop Graph (3/8", 1", 1-1/2" Pipe)

ΔP – Pressure Drop per 100 Feet of Pipe – psig

AIR FLOW – scfm – 1-1/2" Pipe

AIR FLOW – scfm – 3/8” & 1" Pipe

ΔP – Pressure Drop per 100 Feet of Pipe – psig

DO NOT USE DOTTED LINE PORTION OF GRAPH

A: Maximum Flow 5%
B: Maximum Flow 10%
C: Maximum Flow 20%
D: Maximum Flow 40%
Pressure Drop

Applied Pressure – psig

3/8" Pipe

1" Pipe

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

\[ \Delta P \text{ – Pressure Drop per 100 Feet of Pipe – psig} \]

- DO NOT USE DOTTED LINE PORTION OF GRAPH
- A: Maximum Flow 5%
- B: Maximum Flow 10%
- C: Maximum Flow 20%
- D: Maximum Flow 40%

\[ \text{AIR FLOW — scfm — 3" Pipe} \]

\[ \text{AIR FLOW — scfm — 2-1/2" Pipe} \]

Applied Pressure – psig

2-1/2" Pipe

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 (\(\Delta P\)) or flow (gpm) for a variety of operating conditions.

Figure 7 gives the relationship between pressure drop (\(\Delta P\)) and flow (gpm) for pipe sizes 1/8” to 3”. Two auxiliary scales on Figure 7 provide the applied pressure corresponding to a (\(\Delta P\)) of 5% and 10%.

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

Example 1:
Determine the flow in 1/2” pipe (gpm) that will produce a pressure drop (\(\Delta P\)) 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 1/2” pipe and the 10 psig \(\Delta P\) line: Read flow = 4.6 gpm.

Example 2:
Determine the flow in 1/2” pipe (gpm) that will produce a pressure drop (\(\Delta P\)) of 12 psig in 150 feet of pipe when operating at an applied pressure of 100 psig:

First—Determine the \(\Delta P\) 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 1/2” pipe and the 8 psig \(\Delta P\) line: Read flow = 4.2 gpm.

Example 3:
Determine the pressure drop (\(\Delta P\)) 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 \(\Delta P\) for 100 feet of 3/4” pipe by locating the intersection of the diagonal line for 3/4” pipe and the 10 gpm line: Read \(\Delta P\) = 7.5 psig.

Second—For 75 feet of pipe:

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

<table>
<thead>
<tr>
<th>Applied Pressure</th>
<th>Nominal Standard Pipe Size</th>
</tr>
</thead>
<tbody>
<tr>
<td>PSIG</td>
<td>1/8”</td>
</tr>
<tr>
<td>5</td>
<td>0.10</td>
</tr>
<tr>
<td>10</td>
<td>0.14</td>
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<tr>
<td>20</td>
<td>0.21</td>
</tr>
<tr>
<td>40</td>
<td>0.30</td>
</tr>
<tr>
<td>60</td>
<td>0.37</td>
</tr>
<tr>
<td>80</td>
<td>0.43</td>
</tr>
<tr>
<td>100</td>
<td>0.48</td>
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<tr>
<td>150</td>
<td>0.60</td>
</tr>
<tr>
<td>200</td>
<td>0.71</td>
</tr>
<tr>
<td>250</td>
<td>0.80</td>
</tr>
</tbody>
</table>

Table 5. Maximum Recommended Water Flow (gpm) Through A.N.S.I. Standard Weight Schedule 40 Pipe.
Figure 7. Water Flow - Pressure Drop Graph

Appired Pressure - psig — Corresponding to 5% ∆P

Appired Pressure - psig — Corresponding to 10% ∆P

Pipe Size

WATER FLOW — gpm

∆P — Pressure Drop per 100 Feet of Schedule 40 Steel Pipe For Water at 60°F — psig
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 may become sonic and the \( Cv \) formula does not apply. To calculate air flow to atmosphere, enter outlet pressure \( p_2 \) as 53% of absolute \( p_2 \). Pressure drop \( \Delta P \) 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)
- **p_1**: Inlet or Primary Pressure (PSIG)
- **p_2**: Outlet or Secondary Pressure (PSIG)
- **\( \Delta P \)**: Pressure Differential (\( p_1 - p_2 \)) (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 Pressure. Deg R = Deg F + 460

### Valve Sizing For Cylinder Actuation — Direct Formula

\[
Cv = \frac{\text{cylinder area} \times \text{cylinder stroke} \times \text{compression factor}}{\text{pressure drop factor} \times \text{time to complete} \times 29}
\]

**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 .7 or higher. In most cases a 1/4" valve would be sufficient.

It is considered good engineering practice to limit the pressure drop \( \Delta P \) to approximately 10% of primary pressure \( p_1 \). 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.
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. \[
Q = 0.273 \times \frac{D^2 L}{t} x \frac{p_2 + 14.7}{14.7} \quad \text{(SCFM)}
\]

Conversion of CFM to SCFM

2. \[
Q = q \times \frac{p_2 + 14.7}{14.7} x \frac{528}{T} \quad \text{(SCFM)}
\]

Flow Factor \( Cv \) (standard conditions)

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

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

<table>
<thead>
<tr>
<th>Inlet Pressure (psig)</th>
<th>Air Flow Q (SCFM) for Various Pressure Drops ( \Delta P ) at ( Cv = 1 )</th>
<th>Air Flow Q (SCFM) to Atmosphere</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>6.7</td>
<td>12.0</td>
</tr>
<tr>
<td>20</td>
<td>7.9</td>
<td>16.9</td>
</tr>
<tr>
<td>30</td>
<td>9.2</td>
<td>21.8</td>
</tr>
<tr>
<td>40</td>
<td>9.9</td>
<td>26.6</td>
</tr>
<tr>
<td>50</td>
<td>10.8</td>
<td>31.5</td>
</tr>
<tr>
<td>60</td>
<td>11.6</td>
<td>36.4</td>
</tr>
<tr>
<td>70</td>
<td>12.3</td>
<td>41.2</td>
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<tr>
<td>80</td>
<td>13.0</td>
<td>46.1</td>
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<tr>
<td>90</td>
<td>13.7</td>
<td>51.0</td>
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<tr>
<td>100</td>
<td>14.4</td>
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<td>110</td>
<td>15.0</td>
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<td>130</td>
<td>16.1</td>
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<td>140</td>
<td>16.7</td>
<td>75.3</td>
</tr>
<tr>
<td>150</td>
<td>17.2</td>
<td>80.2</td>
</tr>
<tr>
<td>160</td>
<td>17.7</td>
<td>85.1</td>
</tr>
<tr>
<td>170</td>
<td>18.2</td>
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<tr>
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<td>18.7</td>
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<td>190</td>
<td>19.2</td>
<td>99.7</td>
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<td>109.4</td>
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<td>220</td>
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<td>114.3</td>
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<tr>
<td>230</td>
<td>21.0</td>
<td>119.2</td>
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<tr>
<td>240</td>
<td>21.4</td>
<td>124.0</td>
</tr>
<tr>
<td>250</td>
<td>21.8</td>
<td>128.9</td>
</tr>
</tbody>
</table>

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 \( p_1 = 90 \) PSIG
Pressure drop across valve \( \Delta P = 5 \) PSID

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

\[
Q = 1.8 \times 21.8 = 39.2 \text{ SCFM}
\]

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

Primary pressure \( p_1 = 90 \) PSIG
Pressure drop \( \Delta P = 10 \) PSID
Air Flow - \( Q = 60 \) SCFM

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

\[
Cv = \frac{60 \text{ SCFM}}{30} = 2.0
\]

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 \( p_1 = 90 \) PSIG
Air flow to atmosphere \( Q = 100 \) SCFM

Flow to atmosphere through valve from Table 3 for \( Cv = 1: 51 \) 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.

\[
Q = 0.0273 = \frac{4^2 \times 10}{2} \times \frac{85 + 14.7}{14.7} = 14.81 \text{ SCFM}
\]

\[
Cv = \frac{1.024 \times 14.81}{\sqrt{5 \times (85 + 14.7)}} = .7
\]
“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.

\[
K = \frac{D}{560,000} \times S \times (2xP_1 - p_2 - p_3) \times Z \times N \text{ ($HR$)}
\]

\[
N = \frac{60 \text{ Sec}}{t_1 + t_2} \text{ (CPM)}
\]

### Nomenclature
- **D** = Piston Diameter of Cylinder (IN)
- **K** = Cost Savings per Hour ($HR$)
- **p_1** = Plant Air Pressure (PSIG)
- **p_2** = Work Stroke Pressure (Reduced) (PSIG)
- **p_3** = Return Stroke Pressure (Reduced) (PSIG)
- **t_1** = Work Stroke (SEC)
- **t_2** = Return Stroke (SEC)
- **S** = Cylinder Stroke (IN)
- **N** = Cycles Per Minute (CPM)
- **Z** = Cost to compress 1,000 SCF of air to 150 psig ($/1000 SCF$)


### 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 \(Z = .24\)

\[
N = \frac{60}{2 + 2} = 15
\]

Calculate Savings per 8 Hour Shift

\[
K = \frac{2^2 \times 12 \times (150 \times 2 - 100 - 30) \times .24 \times 15}{5.6 \times 10^5} = .053/HR
\]

**Savings are $ .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 - $C_v$

$C_v = \frac{d}{2}$

$C_v$ = AIR CYLINDER PORT FLOW COEFFICIENT = $2\sqrt{d}$ WHERE 'd' IS THE INSIDE DIAMETER OF SUPPLY TUBE

$C_v$ GIVEN IS BASED ON FITTING HAVING SAME ID AS TUBING

FLOW LENGTH IN FEET - $L$

$C_v = \frac{d}{2}$

$C_V = \frac{32d^2}{\pi}$

$C_V$ = FLOW COEFFICIENT - $C_v$

TUBE SIZE (INCHES)

<table>
<thead>
<tr>
<th>1/4 Nylon</th>
<th>3/8 Nylon</th>
<th>1/2</th>
<th>3/4</th>
<th>1</th>
<th>1-1/4</th>
</tr>
</thead>
<tbody>
<tr>
<td>.52</td>
<td>.57</td>
<td>1</td>
<td>1.14</td>
<td>2.14</td>
<td>3.18</td>
</tr>
<tr>
<td>.52</td>
<td>.57</td>
<td>1</td>
<td>1.14</td>
<td>2.14</td>
<td>3.18</td>
</tr>
<tr>
<td>.52</td>
<td>.57</td>
<td>1</td>
<td>1.14</td>
<td>2.14</td>
<td>3.18</td>
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<tr>
<td>.52</td>
<td>.57</td>
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<td>1.14</td>
<td>2.14</td>
<td>3.18</td>
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<tr>
<td>.52</td>
<td>.57</td>
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<td>2.14</td>
<td>3.18</td>
</tr>
<tr>
<td>.52</td>
<td>.57</td>
<td>1</td>
<td>1.14</td>
<td>2.14</td>
<td>3.18</td>
</tr>
</tbody>
</table>

TUBE LENGTH IN FEET

<table>
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<tr>
<th>100</th>
<th>80</th>
<th>60</th>
<th>40</th>
<th>30</th>
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<th>20</th>
<th>15</th>
<th>10</th>
<th>5</th>
<th>2.5</th>
<th>1.5</th>
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</thead>
<tbody>
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<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
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<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
</tbody>
</table>
FLOW COEFFICIENT FOR SCHEDULE 40 STEEL PIPE - $C_{vp}$

**TUBE FRICTION FACTOR** $f = 0.03$

$d =$ INSIDE DIAMETER OF PIPE (INCHES), $l =$ LENGTH OF PIPE (INCHES)

**Supply Pipe Size**

<table>
<thead>
<tr>
<th>Schedule 40</th>
<th>$C_{vc}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/8</td>
<td>1.66</td>
</tr>
<tr>
<td>1/4</td>
<td>3.05</td>
</tr>
<tr>
<td>3/8</td>
<td>5.59</td>
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<td>1/2</td>
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<tr>
<td>3/4</td>
<td>15.6</td>
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<tr>
<td>1</td>
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<td>1-1/4</td>
<td>43.8</td>
</tr>
<tr>
<td>1-1/2</td>
<td>59.6</td>
</tr>
</tbody>
</table>

$C_{vc} =$ AIR CYLINDER PORT FLOW

**COEFFICIENT = $23d^2$ WHERE 'd' IS THE INSIDE DIAMETER OF SUPPLY PIPE**

**CURVES BASED ON FOLLOWING FORMULA -**

$$C_{vp} = 33.2d^2 \sqrt{\frac{d}{f}}$$
## Selected SI Units for Fluid Power Usage

Extracted from ISO 1000 with National Fluid Power Association permission.

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Symbol</th>
<th>Customary U.S. Unit</th>
<th>Abbreviation</th>
<th>Preferred SI Unit</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Angular Velocity</td>
<td>( \omega )</td>
<td>radian per second</td>
<td>rad/s</td>
<td>rad/s</td>
<td>1, 7</td>
</tr>
<tr>
<td>Area</td>
<td>A or S</td>
<td>square inch</td>
<td>( \text{in}^2 )</td>
<td>( \text{cm}^2 )</td>
<td>( \text{m}^2 )</td>
</tr>
<tr>
<td>Bulk Modulus Liquids</td>
<td>K</td>
<td>pounds per square inch</td>
<td>psi</td>
<td>bar</td>
<td>N/m(^2)</td>
</tr>
<tr>
<td>Capacity (Displacement)</td>
<td>V</td>
<td>cubic inches per revolution</td>
<td>cipr</td>
<td>( \text{m}^3/r )</td>
<td>( \ell/r )</td>
</tr>
<tr>
<td>Coefficient of Thermal Expansion (cubic)</td>
<td>( \alpha )</td>
<td>( ^\circ\text{F}-1 )</td>
<td>1/( ^\circ\text{F} )</td>
<td>1/K</td>
<td></td>
</tr>
<tr>
<td>Dynamic Viscosity</td>
<td>( \mu )</td>
<td>centipoise</td>
<td>cP</td>
<td>cP</td>
<td>P</td>
</tr>
<tr>
<td>Efficiency</td>
<td>( \eta )</td>
<td>percent</td>
<td>percent</td>
<td>percent</td>
<td>3</td>
</tr>
<tr>
<td>Force</td>
<td>F</td>
<td>pound (f)</td>
<td>(lb) f</td>
<td>N</td>
<td>kN</td>
</tr>
<tr>
<td>Frequency</td>
<td>f</td>
<td>cycles per second</td>
<td>cps</td>
<td>Hz</td>
<td>kHz</td>
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<td>Kinematic Viscosity</td>
<td>( \nu )</td>
<td>Saybolt Universal Seconds</td>
<td>SUS</td>
<td>cSt</td>
<td>m(^2/s)</td>
</tr>
<tr>
<td>Length</td>
<td>l</td>
<td>inch</td>
<td>in.</td>
<td>mm</td>
<td>m</td>
</tr>
<tr>
<td>Linear Velocity</td>
<td>v</td>
<td>feet per second</td>
<td>ft/s</td>
<td>m/s</td>
<td></td>
</tr>
<tr>
<td>Mass</td>
<td>m</td>
<td>pound (m)</td>
<td>lb (m)</td>
<td>kg</td>
<td>Mg</td>
</tr>
<tr>
<td>Mass Density</td>
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<td>pound (m) per cubic foot</td>
<td>lb (m)/ft(^3)</td>
<td>kg/m(^3)</td>
<td>kg/dm(^3)</td>
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<tr>
<td>Mass Flow</td>
<td>M</td>
<td>pound (m) per second</td>
<td>lb (m)/s</td>
<td>kg/s</td>
<td>g/s</td>
</tr>
<tr>
<td>Power</td>
<td>P</td>
<td>horsepower</td>
<td>HP</td>
<td>kW</td>
<td>W</td>
</tr>
<tr>
<td>Pressure (Above Atmospheric)</td>
<td>p</td>
<td>pounds per square inch</td>
<td>psi</td>
<td>bar</td>
<td>mbar</td>
</tr>
<tr>
<td>Pressure (Below Atmospheric)</td>
<td>p</td>
<td>inches of mercury, absolute</td>
<td>in. Hg</td>
<td>bar, abs</td>
<td>Pa</td>
</tr>
<tr>
<td>Quantity of Heat</td>
<td>( Q_c )</td>
<td>British Thermal Unit</td>
<td>BTU</td>
<td>J</td>
<td>kJ</td>
</tr>
<tr>
<td>Rotational Frequency</td>
<td>n</td>
<td>revolutions per minute</td>
<td>HPM</td>
<td>r/min</td>
<td>r/s</td>
</tr>
<tr>
<td>Specific Heat Capacity</td>
<td>c</td>
<td>British Thermal Unit per pounds mass degree Fahrenheit</td>
<td>BTU/lb(m)(^\circ\text{F})</td>
<td>J(kgK)</td>
<td></td>
</tr>
<tr>
<td>Stress (Materials)</td>
<td>( \sigma )</td>
<td>pounds per square inch</td>
<td>psi</td>
<td>daN/mm(^2)</td>
<td>MPa</td>
</tr>
<tr>
<td>Surface Roughness</td>
<td>( \mu )</td>
<td>microinch</td>
<td>( \mu ) in</td>
<td>grade N.</td>
<td>( \mu ) m</td>
</tr>
<tr>
<td>Temperature (Customary)</td>
<td>( \theta )</td>
<td>degree Fahrenheit</td>
<td>( ^\circ\text{F} )</td>
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### 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/\( \mu \) 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.
10. For conversion from U.S. to SI units, see ISO/R 1302-1971.
## Conversion Tables

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<td>Valve Operations</td>
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<td>Direction of Flow in Pneumatic System</td>
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</tr>
<tr>
<td>5</td>
<td>5.0000</td>
<td>5.0000</td>
<td>19.64</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>6.0000</td>
<td>6.0000</td>
<td>28.27</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>7.0000</td>
<td>7.0000</td>
<td>38.49</td>
<td></td>
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<tr>
<td>8</td>
<td>8.0000</td>
<td>8.0000</td>
<td>50.27</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>10.0000</td>
<td>10.0000</td>
<td>78.54</td>
<td></td>
</tr>
</tbody>
</table>
Summary of Formulas and Equivalents

### Area & Volume

\[ A = D^2 \times 0.7854 \text{ (or } A = \pi R^2) \]

\[ V = D^2 \times 0.7854 \times L \]

\[ \sqrt{\frac{A}{0.7854}} \]

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

### Temperature

Absolute temperature \( ^\circ R = ^\circ F + 460 \)

### Pressure

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

Compression Ratio (standard conditions) = \( \frac{psig + 14.7}{14.7} \)

Compression Ratio (corrected for elevation) = \( \frac{psig + psia}{psia} \)

Pascal's Law – \( F = P \times A \)

\( F = \) Force in lbs./sq. in.

\( P = F/A \)

\( P = \) Pounds (lbs)

\( A = \) Area in sq. in.

### Flow

\( scfm = \) (area in sq. inches \( \times \) stroke inches \( \times \) CPM\(^*\)) / 1728

\( cfm = \) area in sq. inches \( \times \) velocity in ft./min.

\( 144 \text{ in.}^2/\text{ft.}^2 \)

\( scfm = cfm \times \) compression ratio

\( \text{*CPM} = \) Cycles per minute

### Pressure Drop (\( \Delta P \))

\( psid = P_1 - P_2 \)

\( \Delta P \) Averaged for distance = \( \frac{psig \text{ rcvr.} - psig \text{ tool}}{\text{distance ft.}} \)

### Pressure / Volume

Boyles Law – \( P_1 V_1 = P_2 V_2 \)

General Gas Law – \( P_1 V_1 = P_2 V_2 \)

\( T_1 \)

\( T_2 \)

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

### Coefficient of Flow

\( Cv = \frac{Q \text{ (scfm)}}{22.67} \sqrt{\frac{^\circ F + 460}{\Delta P x K}} \)

\( K = P_2 \text{ absolute...if } \Delta P \text{ is less than 10\%} \)

\( K = \frac{(P_1 \text{ abs.} + P_2 \text{ abs.})}{2} \text{...if } \Delta P \text{ is 10\% to 25\%} \)

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

### Line Drop

\( \text{drop/inches} = \text{run/ft} \times \% \text{ grade} \times 0.12 \)

\( \% \text{ grade} = \frac{\text{drop/inches}/0.12}{\text{run/ft}} \)

1% to 2% grade recommended

### Compressed Air Cost

\( \text{Cost} = cfm \times 60 \times \# \text{ hrs.} \times \text{kWh/} \text{cfm} \times \$/\text{kWh} \)
### Vacuum
- Negative psig = inches Hg x 0.49
- Inches Hg = psi/0.49
- Inches Hg x 1.133 = ft. H₂O
- Inches H₂O x 0.036 = psi
- 1 foot H₂O x 0.8826 = 1 inch Hg
- Force = -P x A
- Lifting force = inches Hg x 0.4912 x sq. in. area

### Receiver Sizing

<table>
<thead>
<tr>
<th>Volume (gallons)</th>
<th>Volume (gallons)</th>
</tr>
</thead>
<tbody>
<tr>
<td>K x cfm x (\frac{14.7}{\text{psig} + 14.7}) x 7.48</td>
<td>K x cfm x (\frac{14.7}{\text{psig} + 14.7}) x (\frac{1728}{231})</td>
</tr>
</tbody>
</table>

(V = volume/gal. K = 1 continuous, K = 3 intermittent)
(7.48 converts cu. ft. to gal.)

Time = cu. ft. volume x \((P_{max} - P_{min})\)

Cfm rcvr. consumption x 14.7

### Cylinder Velocity

<table>
<thead>
<tr>
<th>Velocity (ft./sec. extend)</th>
<th>Velocity (ft./sec. retract)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(\frac{\text{inches stroke}}{\text{extend time seconds}}) + (\frac{\text{extended dwell sec. x 60}}{12})</td>
<td>(\frac{\text{inches stroke}}{\text{extend time seconds}}) + (\frac{\text{extended dwell sec. x 60}}{12})</td>
</tr>
</tbody>
</table>

### Electrical

\[
\begin{align*}
E & = I \times R \\
I & = E / R \\
R & = E / I
\end{align*}
\]

\[
\begin{align*}
P & = I \times E \\
P & = I^2 \times R \\
E & = P / I
\end{align*}
\]

(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 – \(R_t = R_1 + R_2 + R_3\)

\(R_t\) = total resistance

### Moisture Content of Air

- Dewpoint = Temperature at which moisture will condense
- Relative Humidity = (Absolute humidity / humidity at saturation) x 100
**Electrical**

Sin θ = opposite / hypotenuse
Cos θ = adjacent / hypotenuse
Secant θ = hypotenuse / adjacent
Cosecant θ = hypotenuse / opposite
Tangent θ = opposite / adjacent
Cotangent θ = adjacent / opposite
Hypotenuse = \( \sqrt{\text{adjacent squared} + \text{opposite squared}} \)

<table>
<thead>
<tr>
<th>Angle</th>
<th>Sin</th>
<th>Cos</th>
<th>Secant</th>
<th>Cosecant</th>
<th>Tangent</th>
<th>Cotangent</th>
</tr>
</thead>
<tbody>
<tr>
<td>30°</td>
<td>.500</td>
<td>.866</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>45°</td>
<td>.707</td>
<td>.707</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>60°</td>
<td>.866</td>
<td>.500</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Mechanical**

Speed Ratio = driven shaft or gear / 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
Horsepower = hp = rpm x ft. lb. torque / 5252

First class lever = \( F_1 \times L_1 = F_2 \times L_2 \) (F = force, L = Length)
Third class lever = \( F_1 \times L_2 = F_2 \times L_1 \) (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 θ
Incline force = load x sin θ
Force along an incline = \( F_1 \times D_1 = F_2 \times D_2 \) (F = force, D = distance)
Rotary actuator torque = Torque = psig x area x pitch radius
**Terminal Velocity** = $2 \times \frac{\text{distance}}{\text{time in seconds}}$

**Kinetic Energy (KE)** = $\frac{\text{weight} \times \text{terminal velocity}^2}{2 \times \text{acceleration of gravity}}$

(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 (H₂O) = 1
- 1 mm Hg = 0.0446 ft. water
- Nm to Hp constant = 7124

**Common Friction Factors**

<table>
<thead>
<tr>
<th>Valves</th>
<th>Friction Factors</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gate Valves</td>
<td></td>
</tr>
<tr>
<td>full-open</td>
<td>0.19</td>
</tr>
<tr>
<td>1/4 closed</td>
<td>1.15</td>
</tr>
<tr>
<td>1/2 closed</td>
<td>5.60</td>
</tr>
<tr>
<td>3/4 closed</td>
<td>24.00</td>
</tr>
<tr>
<td>Globe valve</td>
<td>10.00</td>
</tr>
<tr>
<td>Plug cock</td>
<td>0.26</td>
</tr>
<tr>
<td>Swing check</td>
<td>2.50</td>
</tr>
<tr>
<td>45° elbow</td>
<td>0.42</td>
</tr>
<tr>
<td>90° elbow</td>
<td>0.90</td>
</tr>
<tr>
<td>Close return bend</td>
<td>2.20</td>
</tr>
<tr>
<td>Standard tee</td>
<td>1.80</td>
</tr>
</tbody>
</table>