

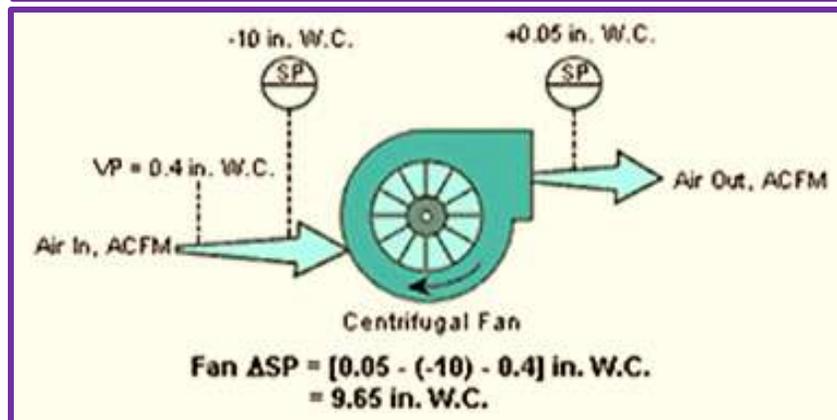
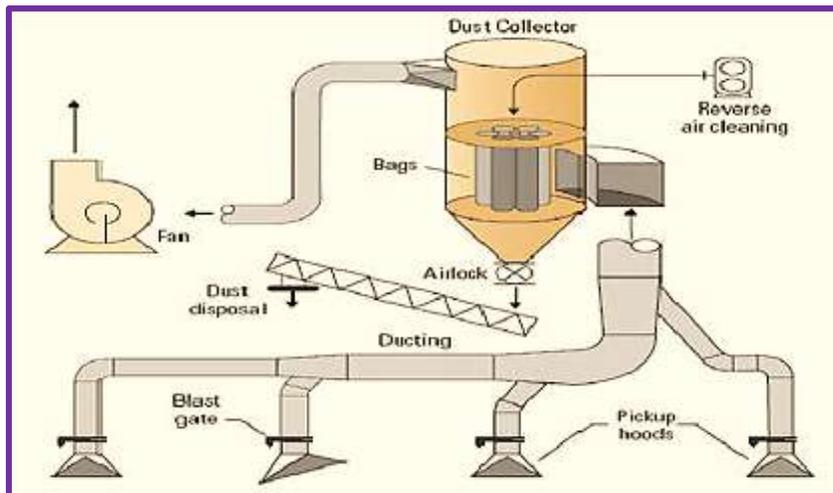


PDHonline Course M385 (3 PDH)

Industrial Ventilation

Practical Concepts and Calculations

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OBS.: This is a professional didactic manual. It's highly recommended to download and print the course content for your study, before answering the quiz questions.

I. INTRODUCTION:

According to Occupational Safety & Health Administration (OSHA), industrial ventilation is one of the most important engineering controls available for improving or maintaining the quality of the air in the occupational work environment.

Broadly defined, **ventilation is a method of controlling** the environment with air flow.

The quality of indoor air inside workplaces is important not only for workers' comfort but also for their health. Specific diseases have been linked to specific air contaminants due poor indoor air quality like asthma, lungs cancer, headaches, fatigue, trouble concentrating and irritation of the eyes, noses, throat and several other symptoms.

The **main scope** of this manual is the practical industrial ventilation concepts application and equipment calculation avoiding as possible, the tiresome extensive theory.

II. FANS AND BLOWERS:

There are two primary types of fans: **centrifugal and axial**.

1.) Centrifugal Fans:

- Centrifugal fans are the most used ventilation equipment, commonly applied in systems to control environment temperatures and in material handling applications to control room moisture or the air particulate content.
- Centrifugal fans use a rotating impeller to increase the velocity of an airstream. As the air is sucked in direction to the blades, it gains kinetic energy.
- The kinetic energy is then converted to a **static pressure increase** as the air slows before entering the discharge.

2.) Axial Fans:

- Axial fans move an airstream along the axis of the fan. The air is pressurized by aerodynamic fan blades, much like a propeller and an airplane wing frequently used in **exhaust applications** where air particulate sizes are small, such as dust streams, smoke and steam and **applications that require reverse** airflow.
- The reverse flow characteristic is useful when an environment may require contaminated **air to be exhausted or in reverse flow**, fresh air to be supplied.

3.) Fans Application:

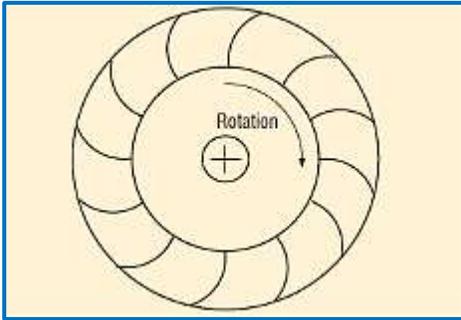
Fans are usually selected from a manufacturer range of models and sizes, rather than designed specifically for a particular application.

Fan selection is based on calculating the **airflow and pressure requirements of a system**, then finding a fan of the right design and materials to meet these requirements.

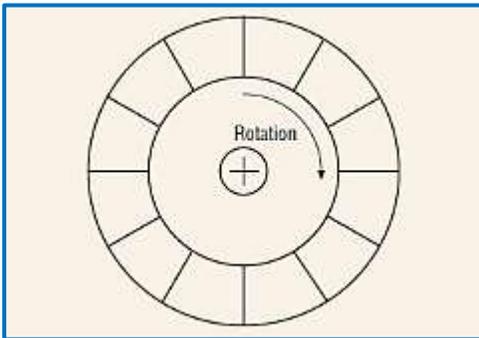
Noise reduction can be accomplished by insulating the duct, mounting the fan on rubber pads or spring isolators to limit the transmitted vibration energy, or installing sound damping material and baffles to absorb the noise energy.

4.) Centrifugal Fan Types:

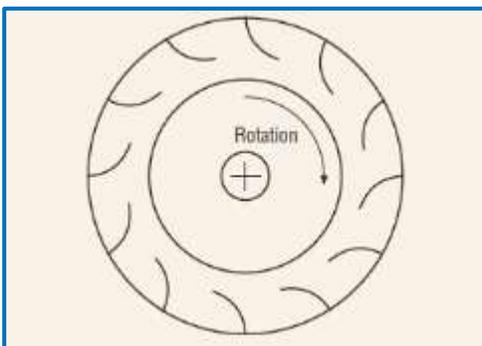
a) Forward-Curved Blades. This fan type, shown below, are typically used in applications that require low to medium air volumes at low pressure, well suited for HVAC applications, but, limited for cleaning systems. This type has low efficiency between 55% and 65% and can operate at relatively low speeds, with low levels of noise, commonly selected because of their small size comparing to other fan types.



b) Radial-Blade. This fan type, shown below, is commonly used in applications with low to medium airflow rates at high pressures. The flat blades, coated with protective compounds to improve erosion and corrosion resistance, are capable of handling high-particulate airstreams, including dust, wood chips, and metal scrap.

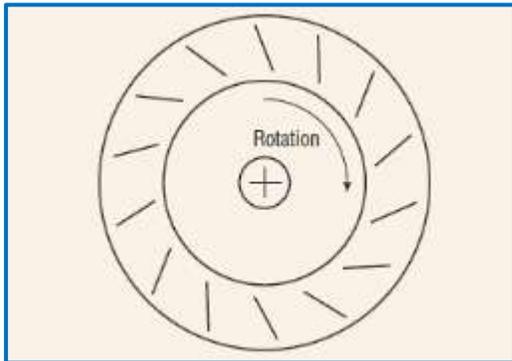


c) Radial-Tip. This fan type, shown below, is well-suited for use with airstreams that have small particulates at moderate concentrations and airstreams with high moisture contents. Radial-tip fans have low angle of attack between the blades which promotes low turbulence. Radial-tip fans can have efficiencies up to 75%.

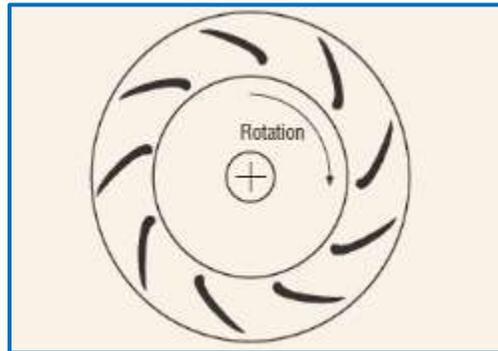


d) Backward-Inclined Fans. These fans types, shown below, have **three different blade shapes**: flat, curved, and airfoil.

Flat blade types are more robust, curved-blade fans are efficient but the airfoil-type blades are the most efficient of all and capable of achieving efficiencies exceeding 85%, commonly selected for forced-draft service to relatively clean airstream systems.



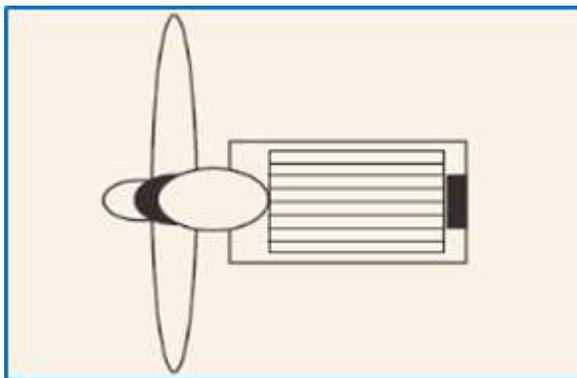
Flat blade type



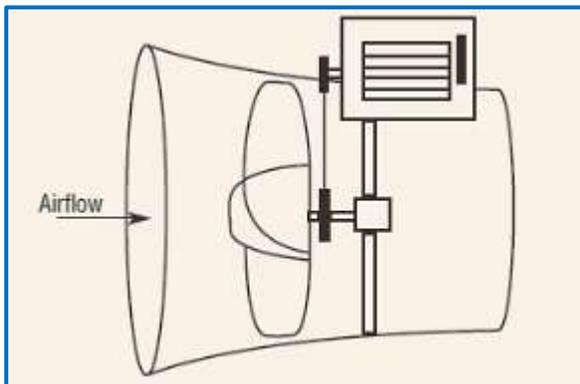
Curved and airfoil type

5.) Axial Fan Types:

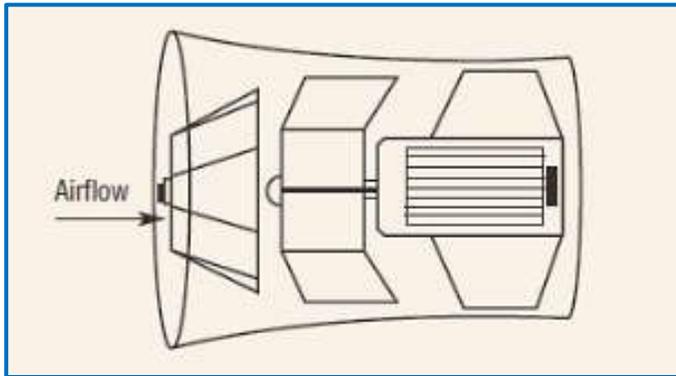
a) Propeller Fans. This fan type, shown below, generates high airflow rates but, creates low pressures so, is very limited to extensive ductwork application. Propeller fans tend to be comparatively noisy reflecting inefficient operation, then, are often used in **rooftop ventilation applications**.



b) Tube Axial Fans. This fan type, shown below, is used in medium-pressure and high airflow rate applications and is well-suited for ducted HVAC installations and also frequently used in exhaust applications because they create sufficient pressure to overcome duct losses accompanied by moderate airflow noise.

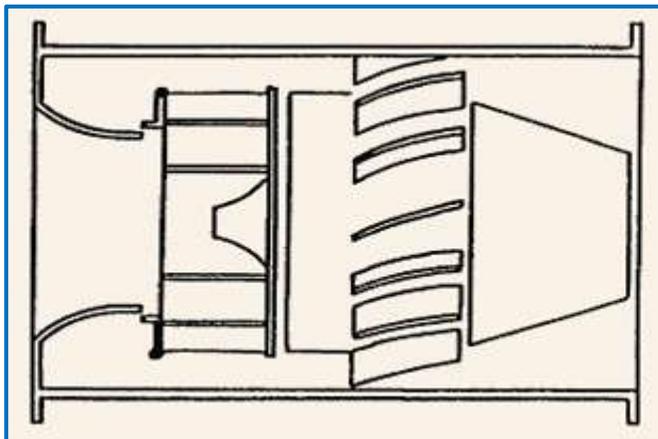


c) Vane Axial Fans. This fan type, shown below, is typically used in medium-to high-pressure applications, such as induced draft service for a boiler exhaust and ventilation applications where quick air removal or supply is required. Like other axial fans, vane axial fans can generate flow in reverse direction, which is also helpful in ventilation applications.

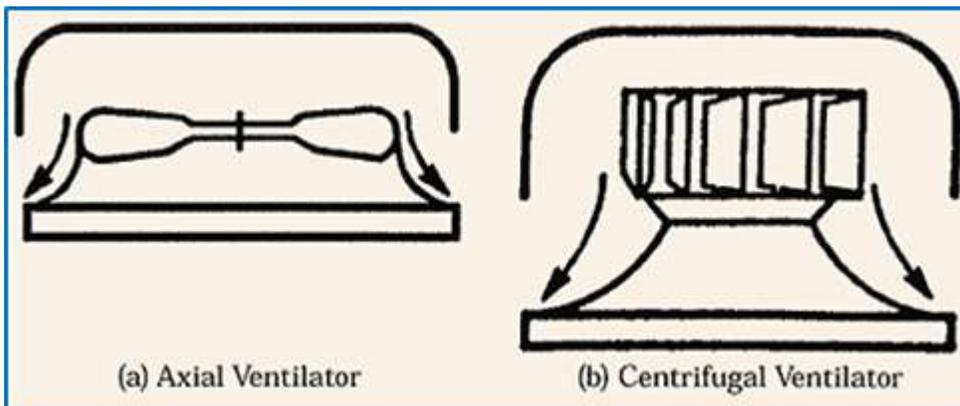


6.) Special Fan Types:

a) Inline Flow Centrifugal Fans. This type of fan, shown below, has backward-curved blades and a special housing to permit a straight line duct installation. The wheel is very similar to that of the airfoil, but space requirements are similar to a vane axial fan.



b) Power Roof Ventilators. This type of fan, shown below, has a special housing to permit a straight line roof installation to produce a high-volume flow at low pressure and can be centrifugal or axial type.



7.) Fan Power Capacity:

Horsepower (HP). The equations, below, to determine the required horsepower to drive fans or blowers do not compensate for temperature, density or airflow characteristics. HP is the power required to move a given volume against a given pressure.

$$\text{HP} = \frac{\text{Volume(cfm)} \times \text{Head (in. of water)}}{6356 \times \text{Fan Efficiency}} =$$

Or:

$$\text{HP} = \frac{\text{Volume(cfm)} \times \text{Pressure (lb/sq. ft.)}}{3300 \times \text{Fan Efficiency}} =$$

Or:

$$\text{HP} = \frac{\text{Volume(cfm)} \times \text{Pressure (psi)}}{229 \times \text{Fan Efficiency}} =$$

For purpose of estimating, the efficiency of a fan or blower may be assumed to be **0.65**.

Notes:

- Air capacity (**cfm**) **varies directly** with fan speed;
- Pressure **varies with square** of fan speed;
- HP **varies with cube** of fan speed.

Brake Horsepower (BHP): Is the actual horsepower required to drive the fan, includes other inefficiencies plus bearing losses. To calculate the BHP use the following formula:

1 phase motors

3 phase motors

$$\text{BHP} = \frac{V * I * E * \text{PF}}{746} = \quad \text{BHP} = \frac{V * I * E * \text{PF} * 1.73}{746} =$$

Where:

BHP = Brake Horsepower;

V = Line Voltage;

I = Line Current;

E = Motor Efficiency (Usually about 0.85 to 0.9);

PF = Motor Power Factor (Usually about 0.9).

Obs.: Once the BHP is known, the RPM of the fan can be measured. The motor BHP and fan RPM can then be matched on the fan performance curve to approximate airflow.

8.) Total Pressure:

The total pressure (TP) represents all energy requirements for moving air through the ventilation system.

TP = TP outlet - TP inlet;

TP = SPout - VP out - SPin - VP in;

TP = SPout - Spin.

8.1) Total Pressure, Air Flow and Velocity Pressure:

The operation of is often expressed in the terms of **Total Pressure, Air Flow and Velocity Pressure:**

$$TP = SP + DP = \text{constant} ;$$

$$TP = SP + VP.$$

$$VP = (V / 4005)^2$$

$$V = 4005 (VP)^{0.5}$$

$$Q = V \times A$$

$$V = Q/A$$

Where:

TP = Total Pressure (in. WG – Water Gauge);

SP = Static Pressure (in. WG);

DP = Dynamic Pressure (in. WG);

VP = Velocity Pressure (in. WG);

V = Air Velocity (FPM);

Q = Air Flow (CFM).

The **Dynamic Pressure (DP)**, measured **between the fan air inlet and the fan air outlet**, can also be expressed:

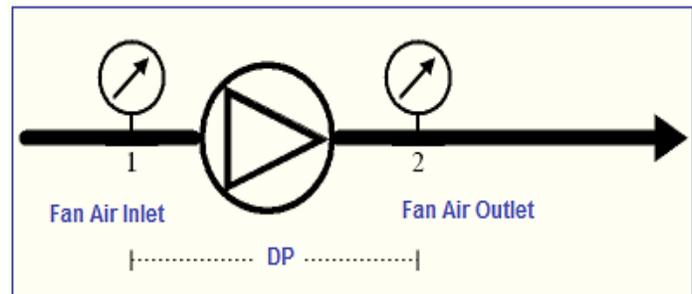
$$DP = \frac{\rho V^2}{2}$$

Where:

ρ = Density of air (0.075 lb/ft³); (1.204 kg/m³);

V = Flow velocity (fpm).

OBS.: WG is water gauge.

**Examples:**

1) Find the Fan **Total Pressure (TP)** given that the SP (inlet) = -5.0 in WG, SP (outlet) = +0.40 in. WG.

$$TP = SP_{out} - SP_{in} = 0.40 - (-5.0) = 5.4 \text{ in. WG (water gauge).}$$

2) Calculate de **Air Velocity (V)** and the **Velocity Pressure (VP)** for the below data:

Fan Air Flow Rate, *Q* = 4000 CFM; Duct Size: 10" x 36";

Outlet Duct Cross-Sectional Area, *A* = 10 x 36/144 = 2.5 ft².

Air Velocity, *V* = 4000/2.5 = 1600 ft/min (fpm)

Velocity Pressure, *VP* = (V/4005)² = (1600/4005)² = 0.16 in. WG

9.) Fan Inlet Pressure Drop:

The pressure drop occurs at the **fan inlet of a Cooling Tower fan stack** unless the shape of the fan inlet is elliptical and no obstruction, in case of induced draft fan arrangement.

The following **fan table** below could be **applied to a Cooling Tower fan** as a guide line in choosing the pressure drop coefficient or the ventilation constant (**K**).

Inlet Shape	K	Extra Factor	Total Factor
Elliptical (L/D = 1:1.5)	0.00	0.10	0.10
R/D = 0.15	0.00	0.10	0.10
R/D = 0.10	0.04	0.14	0.18
R/D = 0.05	0.13	0.15	0.28
R = 0	0.40	0.20	0.60

10.) Pressure Drop, Velocity and Velocity Pressure Relation:

The common formula for Pressure Drop is:

$$\text{Pressure Drop} = K \cdot \frac{\rho \cdot V^2}{2 \cdot g}$$

$$P_d = K \times (1 / 2) \times \text{Air Density} \times V^2 / 115,820 \text{ (lb/ft}^2\text{)} =$$

$$P_d = K \times 0.1922 \times (1 / 2) \times \text{Air Density} \times V^2 / 115,820 \text{ (inch WG = inch Water)} =$$

Or:

$$P_d = K \times 0.1922 \times (1 / 2) \times (\text{Density Ratio} \times 0.075) \times V^2 / 115,820 \text{ (inch WG = inch Water)} =$$

$$P_d = K \times 0.1922 \times (1 / 2) \times 0.075 / 115,820 \times V^2 \times \text{Density Ratio} \text{ (inch WG = inch Water)} =$$

$$P_d = K \times V^2 \times 1 / 16,069,372 \times \text{Density Ratio} =$$

$$P_d = K \times V^2 \times 1 / 4008.66^2 \times \text{Density Ratio} =$$

Then:

$$P_d = K \times (V / 4008.7)^2 \times \text{Density Ratio} =$$

Where:

G = Acceleration Gravity, ft/min² (**1g = 32.172 ft/sec² = 115,820 ft/min²**);

Density Ratio = Actual Air Density / 0.075 (**1 lb/ft³ = 0.1922 inch WG**).

And,

P_d = Pressure Drop ;

K = Pressure Drop Coefficient/Ventilation Constant ;

ρ = Air Density, lb/ft³;

V = Air Velocity, ft/min.

The **constant of 4008.7** was obtained to convert the **unit of pressure drop to inch WG**. Therefore **relation for Velocity and Velocity Pressure** became:

$$VP = (V / 4005)^2$$

and

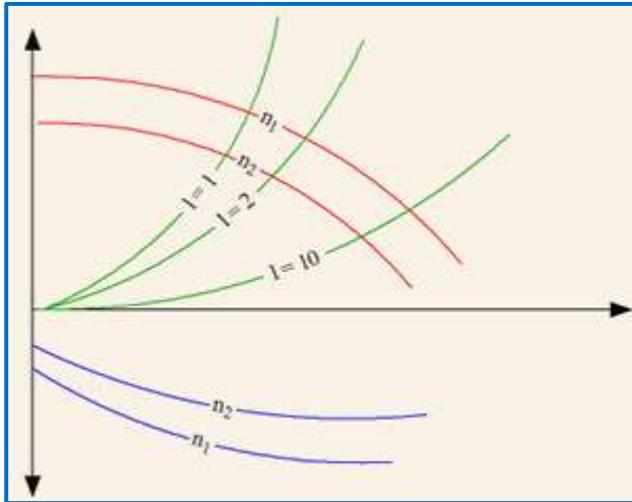
$$V = 4005 (VP)^{0.5}$$

11.) Pressure Head:

Term used in fluid mechanics to represent the internal energy of a fluid due to the pressure exerted on its container. It may also be called **static pressure head** or simply **static head**.

The head as a function of air flow volume is, in general, presented in diagrams as shown below:

- The red lines are the pressure head as a function of the air flow and the speed of the fan.
- The green lines marked - / - are the throttle lines.
- The blue lines are the power consumption of the fan.



12) Converting Head to Pressure:

Since pressure gauges often are calibrated in pressure - psi or bar, it may be necessary a conversion to head - feet or meter, commonly used in pump curves.

11.1) Converting head in feet to pressure in psi:

Pump curves in feet of head can be converted to pressure - psi - by the expression:

$$p = 0.434 h SG \quad (1)$$

Where:

p = Pressure (psi);

h = Head (ft);

SG = Specific Gravity.

11.2) Converting head in meter to pressure in bar:

Pump curves in meter of head can be converted to pressure - bar - by the expression:

$$p = 0.0981 h SG \quad (2)$$

Where:

h = Head (m);

p = Pressure (bar).

11.3) Converting pressure in psi to head in feet:

$$h = p \ 2.31 / SG \quad (3)$$

Where:

h = Head (ft);

p = Pressure (psi).

Converting pressure in bar to head in meter:

$$h = p \ 10.197 / SG \quad (4)$$

Where:

h = Head (m);

p = Pressure (bar).

Example:

1) Converting Pump Head - feet - to Pressure – psi. The pressure - psi - of a water pump operating with head **120 ft** can be expressed as:

$$p = (120 \text{ ft}) \ 1 / 2.31$$

$$p = \mathbf{52 \text{ psi}}$$

13.) Coefficient of Entry:

The Coefficient of Entry (**Ce**), a factor usually ranging **between 0.6 and 0.98**, indicates the relationship between the actual hood turbulence losses, and that of a "perfect" hood with no turbulence losses (coefficient of entry is the ratio of actual flow and theoretical flow).

Using the **Static Pressure (SP)** and **Velocity Pressure (VP)**, a fairly accurate estimation of flow rate of an exhaust opening can be determined, with **Coefficient of Entry (Ce)**.

Hood shapes can vary widely, but most have **hood openings of between 70% and 100%** of the corresponding duct area.

The Coefficient of Entry can also be estimated using the following formula:

$$Ce = (1 / 1+K)^{0.5}$$

Or:

$$Ce = (VP / SP)^{0.5}$$

Where:

K = Hood entry loss or ventilation constant (see table below)

$$Ce, \text{ for a plain hood opening} = (1 / 1+0.93)^{0.5} = \mathbf{0.72}$$

$$Ce, \text{ for a flanged hood opening} = (1 / 1+0.49)^{0.5} = \mathbf{0.82}$$

Some typical **coefficients of entry**, C_e , and **entry loss**, K , are as follows:

Hood Type	Coefficient of Entry, C_e	Entry Loss, K
Duct, round, square or rectangular, plain end	0.72	0.93
Duct, round, square or rectangular, flanged end	0.82	0.49
Bell Mouth Inlet (flared reducer)	0.98	0.04
Tapered Cone, 30° included angle	0.96	0.08
Tapered Cone, 90° included angle	0.93	0.15
Tapered Rect., 30° included angle	0.93	0.16
Tapered Rect., 90° included angle	0.89	0.25
Orifice plus flanged duct (duct/slot velocity)	0.55	2.3
Tapered hoods (face area > twice the duct area)	0.82 – 0.98, dependent on angle chosen, divide hood into simple shapes, sum factors for each shape.	

Note: In calculations using the **Velocity Pressure**, the hood entry loss for **chipping and grinding operations**, as an example, is **0.25 VP**.

13) Velocity and Velocity Pressure

For calculation of velocity pressures at velocities other than those listed below:

$$VP = (V / 4005)^2 =$$

For calculation of velocities when velocity pressures are known:

$$V = 4005 \sqrt{VP} =$$

Velocity (fpm)	Velocity Pressure (in wg)	Velocity (fpm)	Velocity Pressure (in wg)
300	0.0056	3500	0.7637
400	0.0097	3600	0.8079
500	0.0155	3700	0.8534
600	0.0224	3800	0.9002
700	0.0305	3900	0.9482
800	0.0399	4000	0.9975
900	0.0504	4100	1.0480
1000	0.0623	4200	1.0997
1100	0.0754	4300	1.1527
1200	0.0897	4400	1.2069
1300	0.1053	4500	1.2624
1400	0.1221	4600	1.3191
1500	0.1402	4700	1.3771
1600	0.1596	4800	1.4364
1700	0.1801	4900	1.4968
1800	0.2019	5000	1.5586
1900	0.2250	5100	1.6215
2000	0.2493	5200	1.6857

14.) Hood Static Pressure:

The volume of air can be determined by:

$$Q = 4005 C_e A \sqrt{SPh} =$$

Or:

$$Q = 4005 C_e A (SPh^{0.5}) =$$

Where:

Q = Rate of flow in cfm:

Ce = Coefficient of entry:

A = Cross-sectional area of duct in square feet:

SPh = Average static pressure reading in inches of water (in the hood).

Examples:

1) A fan will be designed for a **Cooling Tower** with an impeller **diameter 28 ft** and a **hub diameter 88 inch (7.33 ft)**. The fan inlet shape is rounded **R/D = 0.10**. The **airflow** volume of the Cooling Tower at the drift eliminator is **1,019,716.3 ft³/min**. Estimate the **pressure drop** at the fan inlet with the following data:

Air density and specific volume at **97°F and 100% RH** is **0.0696 lb/ft³**;

Air standard density is **0.075 lb/ft³**.

Let's calculate the net impeller area:

$$\text{Impeller Net Area} = 3.1416 / 4 \times (\text{Fan Dia}^2 - \text{Fan Hub}^2) =$$

$$\text{Impeller Net Area} = 3.1416 / 4 \times (28^2 - 7.33^2) = \mathbf{573.52 \text{ ft}^2}.$$

Air Velocity @ Fan:

$$Q = V \times A =$$

$$Q = \text{Airflow Volume} / \text{Net Fan Impeller Area} =$$

$$Q = 1,019,716.3 / 573.52 = \mathbf{1778.00 \text{ ft}^3/\text{min} \text{ (cfm)}}.$$

For a fan inlet shape **R/D = 0.10**, **K total factor** is = **0.18** (see fan table). Then the pressure drop is:

$$P_d = K (V / 4008.7)^2 \times \text{Density Ratio} =$$

$$P_d = 0.18 \times (1778.0 / 4008.7)^2 \times (0.0696 / 0.075) = \mathbf{0.0329 \text{ in. WG}}.$$

2) Calculate the hood **SP**, when the Velocity Pressure is **3.2 in. WG** and the Coefficient of Entry is **0.96**.

$$C_e = (VP / SP)^{0.5}$$

Where:

Ce = Coefficient of Entry;

VP = Velocity Pressure;

SP = Static Pressure.

$$\mathbf{SP = VP / C_e^2 = 3.2 / 0.96^2 = 3.47 \text{ in. WG}}$$

3) If the Static Pressure is **2.56 in. WG** and the Coefficient of Entry is **0.92**, calculate the VP.

Solution:

$$\mathbf{VP = SP \times Ce^2 = 2.56 \times 0.92^2 = 2.17 \text{ in. WG}}$$

4) The Velocity Pressure in the duct is **3.15 in. WG**. Calculate is the Velocity in the duct.

Solution:

$$\mathbf{V = 4005 (VP)^{0.5}}$$

Where:

V = Velocity in fpm;

VP = Velocity Pressure in inches of water (WG).

$$\mathbf{V = 4005 (3.15)^{0.5} = 7108.17 \text{ fpm.}}$$

5) The Velocity in a duct is **4000 fpm** and the area of the **duct is 1.5 m²**, what is the flow rate?

Solution:

$$\mathbf{Q = A * V =}$$

Where:

Q = Flow rate,;

A = Cross-sectional area (**1.5 m² = 16.145 ft²**);

V = Velocity.

$$\mathbf{Q = 16.145 \text{ ft}^2 \times 4000 \text{ fpm} = 64580.0 \text{ cfm.}}$$

6) The rate of flow in a pipe is **350 m³/s** and the area of the pipe is **25 ft²**, calculate the flow Velocity.

Solution:

$$\mathbf{A = 350 \text{ m}^3/\text{s} = 741608 \text{ ft}^3/\text{min.}}$$

$$\mathbf{V = Q / A = (741608 \text{ ft}^3/\text{min}) / (25 \text{ ft}^2) = 29664.3 \text{ fpm.}}$$

7) Calculate the **Velocity** in m/s, when the **Velocity Pressure** in a hood is **2.20 cm** of water.

Solution:

$$\mathbf{V = 4005 (VP)^{0.5} =}$$

Where:

V = Velocity in fpm;

VP = Velocity Pressure in inch of water (**2.20 cm WG = 0.866 in. WG**).

$$\mathbf{V = 4005 (0.866)^{0.5} = 3727.02 \text{ fpm} = 18.94 \text{ m/s.}}$$

8) Air flows in a **3 ft diameter circular duct**. At the end of a long straight run the centerline Velocity is measured to be **20m/s**. Estimate the average Velocity and mass flow rate of the air.

Solution:

Assuming Average velocity = **0.8 Centerline velocity**,

Vaverage:

$$0.8 \times V_{\text{centerline}} = 0.8 \times 20 \text{ m/s} = 16 \text{ m/s} = 3149.61 \text{ fpm.}$$

Mass flow rate:

$$V_{\text{average}} \times A \times \rho(\text{air}) = 3149.61 \text{ fpm} \times (\pi \times 3^2 / 4) \text{ ft}^2 \times 0.075 \text{ lbm/ft}^3 = 1669.75 \text{ lbm/min.}$$

9) The transport Velocity is **4500 fpm** and the duct is diameter **2 ft**. What air flow is needed?

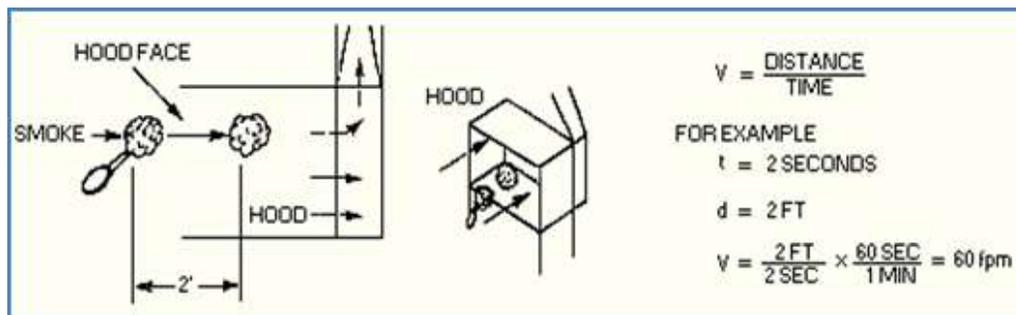
Solution:

$$\text{Area of duct, } A = \pi \times 2^2 / 4 = 3.14 \text{ ft}^2$$

$$\text{Required air flow} = V \times A = 4500 \text{ fpm} \times 3.14 \text{ ft}^2 = 14130 \text{ cfm.}$$

15.) Rate Velocity Measurements:

Smoke is useful for **measuring rate velocity** (see below) because it is visible. Nothing convinces more quickly that the ventilation is not functioning properly than to **show smoke drifting away** from the hood, escaping the hood, or traveling into the worker's breathing zone.



Velocity = Distance/Time =

$$V = \frac{D}{t}$$

16.) Air Density Correction Factors:

If the air stream temperature is **greater than 100°F** and/or if the altitude is **greater than 1,000 feet**, add a **density correction factor** to the above equation:

$$Q = 1096 C_e A (SPh/0.075 \rho)^{0.5} =$$

Where:

ρ = Relative Density of air.

AIR DENSITY CORRECTION FACTORS

Air Temp. °F	Elevation (Feet Above Sea Level)															
	0	1000	2000	3000	4000	5000	6000	7000	8000	9000	10000	11000	12000	13000	14000	15000
0	0.87	0.90	0.94	0.97	1.01	1.05	1.08	1.13	1.17	1.22	1.26	1.31	1.37	1.43	1.48	1.54
50	0.96	1.00	1.04	1.08	1.11	1.15	1.20	1.24	1.30	1.34	1.40	1.45	1.51	1.57	1.63	1.70
70	1.00	1.04	1.08	1.12	1.16	1.22	1.25	1.30	1.35	1.40	1.45	1.51	1.57	1.64	1.70	1.77
100	1.06	1.10	1.14	1.18	1.22	1.27	1.32	1.37	1.42	1.48	1.54	1.60	1.66	1.74	1.80	1.88
150	1.15	1.19	1.24	1.30	1.33	1.38	1.44	1.49	1.55	1.61	1.67	1.74	1.81	1.89	1.96	2.04
200	1.25	1.29	1.34	1.40	1.44	1.50	1.56	1.61	1.68	1.75	1.81	1.89	1.96	2.05	2.13	2.21
260	1.34	1.39	1.44	1.50	1.55	1.61	1.67	1.74	1.80	1.88	1.95	2.02	2.10	2.20	2.28	2.37
300	1.43	1.49	1.54	1.60	1.66	1.72	1.79	1.86	1.93	2.01	2.08	2.16	2.25	2.35	2.43	2.53
350	1.53	1.58	1.64	1.71	1.77	1.84	1.91	1.98	2.08	2.14	2.22	2.31	2.40	2.51	2.60	2.71
400	1.62	1.68	1.75	1.81	1.89	1.94	2.03	2.09	2.19	2.27	2.37	2.45	2.54	2.66	2.75	2.87
500	1.81	1.88	1.95	2.02	2.10	2.18	2.26	2.35	2.44	2.54	2.63	2.73	2.84	2.97	3.08	3.20
600	2.00	2.07	2.15	2.23	2.31	2.40	2.50	2.59	2.69	2.84	2.91	3.02	3.14	3.28	3.40	3.54
700	2.19	2.27	2.35	2.44	2.53	2.63	2.73	2.83	2.94	3.07	3.17	3.31	3.44	3.59	3.72	3.88
800	2.38	2.48	2.57	2.67	2.76	2.86	2.98	3.09	3.21	3.33	3.45	3.59	3.74	3.90	4.05	4.21
900	2.56	2.66	2.76	2.87	2.97	3.07	3.20	3.33	3.46	3.58	3.71	3.87	4.02	4.20	4.35	4.53
1000	2.76	2.87	2.99	3.09	3.20	3.31	3.45	3.59	3.73	3.86	4.00	4.17	4.33	4.53	4.69	4.89

Obs.: Commonly the density correction is **not needed** if the temperature is between **40°F to 100°F** and elevations are between **100 ft. to 1000 ft.**

Examples:

1) A fan is selected to deliver **7500 CFM at 1.5 inch SP** at an altitude of **6000 ft above sea level** and an operating **temperature of 200° F**. At **Air Standard** conditions and ambient temperature the fan is **3.53 BHP at 976 RPM**.

Solution:

From the table above, Air Density Factors for Altitude and Temperature, the **air density correction factor** is **1.56**. Using the fan’s operating altitude and temperature the correction is:

Corrected SP = 1.5” SP x 1.56 = 2.34” SP

At Air Standard conditions and temperature the **fan is required to be 3.53 BHP**. This should be **corrected** to new altitude condition that is 6000 ft above sea level:

Corrected BHP = 3.53 BHP / 1.56 = 2.26 BHP

2) A centrifugal fan with **20 inches** impeller diameter is required to deliver **5,000 CFM at 3.0 in. WG, Static Pressure (SP)**. Elevation is **0 (sea level)**. At Air Standard conditions and temperature between **40°F and 100°F**, the fan will require **6.76 BHP**. What is the operating **BHP at 600°F**?

Solution:

Using the chart above, the correction factor is **2.00**.

Corrected SP = 3.0 x 2.00 = 6.0 in. WG.

Operating BHP = 6.76 ÷ 2.00 = 3.38 BHP.

17.) Fan Average Capacities:

As a **rule of thumb**, the following table gives average capacities for various sizes of fans and the corresponding horse-power of electric motor for fans of the blower type having curved blades when run at a **speed corresponding to 1/2 ounce pressure** at the discharge outlet.

Fan Average Capacity – Curved Blades Type					
Housing Fan Height (Inches)	Fan Impeller Diameter (Inches)	Housing Width (Inches)	Impeller Speed (rpm)	Outlet Air Flow (cfm)	Minimum Horsepower
30	18	9	870	1000	1/2
40	24	12	580	1600	1
50	30	15	465	2600	1
60	36	18	390	4500	2
70	42	21	333	6000	2 1/2
80	48	24	293	8000	2 1/2
90	54	28	260	11000	4
100	60	32	233	12500	4
120	72	43	195	21500	7
140	84	48	167	28600	9
160	96	48	147	31800	10
180	108	54	130	40400	13
200	120	60	117	51000	16



Centrifugal Fan - flow within the fan is mainly **radial** to the shaft.



Axial Fan - flow within the fan is mainly **parallel** to the shaft.

18.) Fan Laws:

The fans operate under a predictable set of laws concerning speed, power and pressure. A change in speed (revolutions per minute or RPM) of any fan will predictably change the pressure rise and power necessary to operate it at the new RPM.

18.1) Affinity Laws for Fans or Blowers:

Flow ∞ Speed	Pressure ∞ (Speed) ²	Power ∞ (Speed) ³
$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$	$\frac{SP_1}{SP_2} = \left(\frac{N_1}{N_2}\right)^2$	$\frac{kW_1}{kW_2} = \left(\frac{N_1}{N_2}\right)^3$
<p><i>Varying the RPM by 10% decreases or increases air delivery by 10%.</i></p>	<p><i>Reducing the RPM by 10% decreases the static pressure by 19% and an increase in RPM by 10% increases the static pressure by 21%.</i></p>	<p><i>Reducing the RPM by 10% decreases the power requirement by 27% and an increase in RPM by 10% increases the power requirement by 33%.</i></p>
<p>Where Q – flow, SP – Static Pressure, kW – Power and N – speed (RPM)</p>		

18.2) Affinity Laws for Pumps

Impeller Diameter	Speed	Specific Gravity (SG)	To Correct for	Multiply by
Constant	Variable	Constant	Flow	$\left(\frac{\text{New Speed}}{\text{Old Speed}}\right)$
			Head	$\left(\frac{\text{New Speed}}{\text{Old Speed}}\right)^2$
			BHP (or kW)	$\left(\frac{\text{New Speed}}{\text{Old Speed}}\right)^3$
Variable	Constant		Flow	$\left(\frac{\text{New Diameter}}{\text{Old Diameter}}\right)$
			Head	$\left(\frac{\text{New Diameter}}{\text{Old Diameter}}\right)^2$
			BHP (or kW)	$\left(\frac{\text{New Diameter}}{\text{Old Diameter}}\right)^3$
	Constant	Variable	BHP (or kW)	$\frac{\text{New SG}}{\text{Old SG}}$

Adapted from ASHRAE "Pocket Handbook", 1987.

19.) Fan Basic Characteristics:

Fans basic calculations and their characteristics are summarized in the tables below:

Fan Type	Wheel Type	Static Pressure (inches WG)	Impeller Diameter (inches)	Air Flow (cubic feet per minute)	Break Horse Power (bhp)
Centrifugal	Backward Inclined	0 - 12	10 - 75	500 - 125, 000	1/3 - 200
	Backward Inclined Airfoil	0 - 14	20 - 120	1, 500 - 450, 000	1/3 - 1500
Propeller Axial	Direct Drive	0 - 1	10 - 50	50 - 50, 000	1/6 - 10
Tube Axial	Direct Drive	0 - 1	20 - 50	2, 500 - 50, 000	1/3 - 15
Vane Axial	Direct Drive	0 - 5	20 - 60	1, 200 - 150, 000	1/3 - 150

Fan Application Formulas

*d = 1.326 x $\frac{P_b}{\sqrt{F + 460}}$

Pt = Ps + Pv

*Pv = $\left(\frac{Q}{A \times 4005}\right)^2 = \left(\frac{V}{4005}\right)^2$

Fan BHP = $\frac{Q \times P_t}{6362 \times \eta_t} = \frac{Q \times P_s}{6362 \times \eta_s}$

For 3 phase motors: BHP output = $\frac{E \times I \times ME \times Pf \times 1.73}{746}$

For 3 phase motors: Kw input = $\frac{E \times I \times Pf \times 1.73}{1000}$

For 1 phase motors: BHP output = $\frac{E \times I \times ME \times Pf}{746}$

For 1 phase motors: Kw input = $\frac{E \times I \times Pf}{1000}$

To plot a System Curve where Ps1 and Q1 are known, use the following formula to find other curve points:

$$P_s 2 = P_s 1 \left(\frac{Q_2}{Q_1}\right)^2$$

To determine round duct equivalent of rectangular duct for same friction loss and volumetric capacity:

$$DR = 1.265 \sqrt[5]{\frac{(ab)^3}{a + b}}$$

Symbol	Definition
A	area (ft. ²)
a	side a of rectangular duct
b	side b of rectangular duct
BHP	brake horsepower
Q	air volume flow (ft. ³ /min.)
d	air density (lb/ft. ³)
DR	diameter of round duct
E	volts
°F	temperature (Fahrenheit)
I	amps
KW	kilowatts
ME	motor efficiency (dec.)
ηs	fan static efficiency (dec.)
ηt	fan total efficiency (dec.)
Pb	barometric pressure (inches mercury)
Pf	power factor
Ps	static pressure (inches WG)
Pt	total pressure (inches WG)
Pv	velocity pressure (inches WG)
V	velocity (ft./min)

* Formulas for d and Pv are applicable to dry air only.

20.) Pressure Correction Factors:

When the measured **inlet suction pressure exceeds 10 in WG (270 mm WG)** the change in air density should be compensated by increasing the suction pressures with a correction factor.

Suction Pressure (Water Gauge)		Pressure Correction Factor - f -
(mm)	(in)	
-270	-10	1.03
-380	-14	1.04
-490	-18	1.05
-600	-22	1.06

Corrected suction pressure can be expressed as:

$$\text{ScP} = P_s \times f =$$

Where:

ScP = Corrected suction pressure (in. WG);

f = Correction factor;

P_s = Design (and measured) suction pressure (in. WG).

Example:

1) For a measured **suction pressure $P_s = -14$ in. WG**. The correction factor from **table above is 1.04**. For proper fan selection the suction pressure **should be corrected to:**

$$\text{ScP} = -14 \text{ in WG} \times 1.04 = -14.56 \text{ in. WG}$$

21.) NTP - Normal Temperature and Pressure:

NTP - Normal Temperature and Pressure – is commonly used as a standard condition **for testing and documentation** of fan capacities. The standard conditions are:

- Air temperature at 68°F (293.15 K, 20°C)
- Air pressure at 29.92 in Hg (101.3 kPa, 14.7 psia, 0 psig, 1 atm, 760 torr)
- Air density at 0.075 lb/ft³ (1.204 kg/m³)
- 1 foot = 304,8 mm = 30.48 cm = 0.3048 m = 12 inches

22.) V-Belt Drives:

Most fan drive systems are based on the standard "V" drive belt which is relatively efficient and readily available and easily selected through a combination of AC motor RPM and drive pulley ratios. Calculate the approximate belt length using the following formula:

$$L = 2C + 1.57(D+d) + \frac{(D-d)^2}{4C}$$

L = Pitch Length of Belt

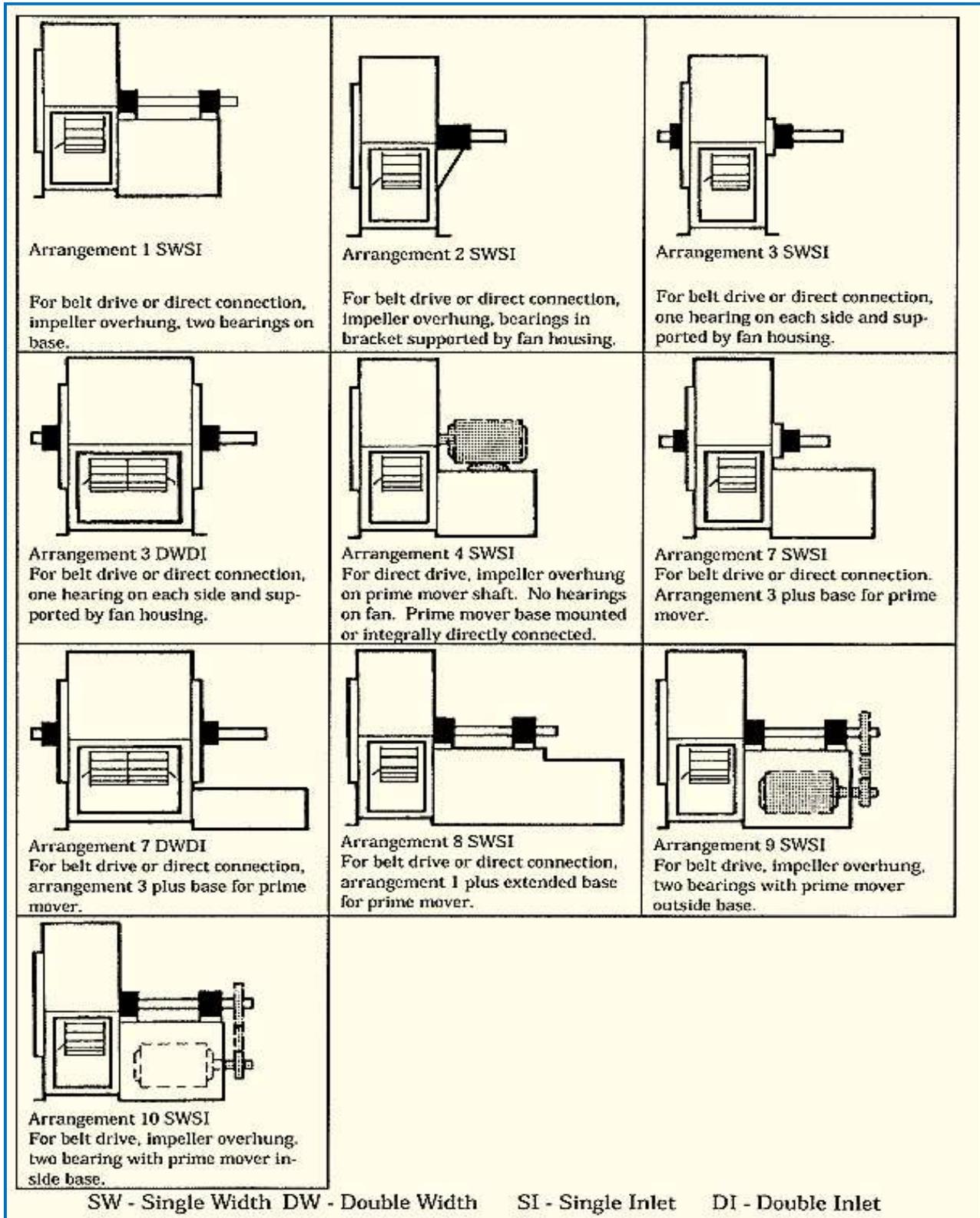
C = Center Distance of Sheaves

D = Pitch Diameter of Large Sheave

d = Pitch Diameter of Small Sheave

23.) Fan Drive Arrangements:

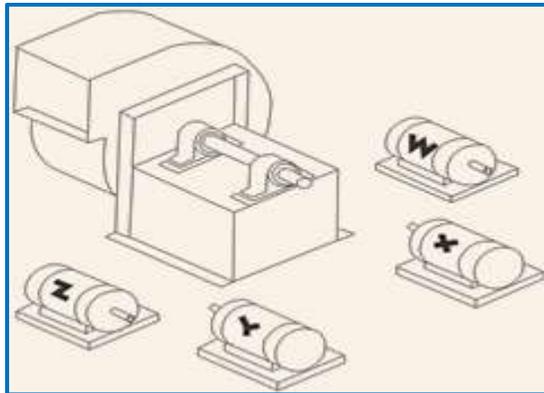
The fans and blowers drive arrangements are standardized by AMCA (Air Movement and Control Association) may be belt or direct electric motor driven:



21.1) Drive Design Conditions:

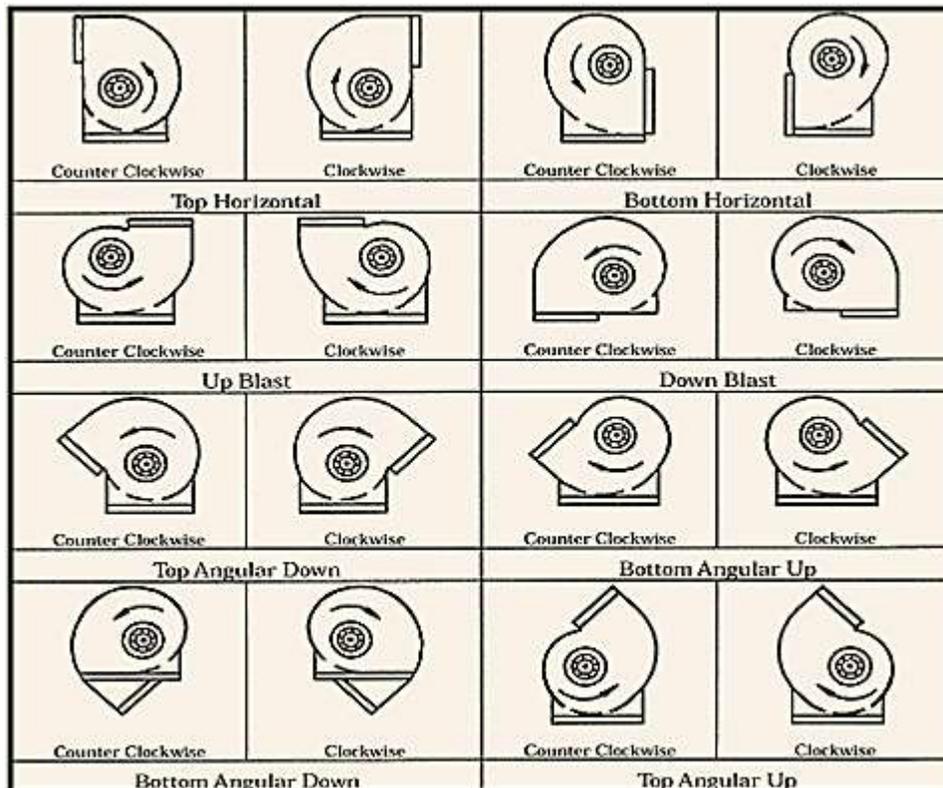
The conditions to consider in designing a satisfactory drive are:

- 1) Centers should not exceed 2.5 to 3 times the sum of the sheave diameters or be less than the diameter of the larger sheave.
- 2) The arc of contact on the smaller sheave should be not less than 120°.
- 3) Sheave diameter ratios should not exceed 8:1.
- 4) Belt speed should not exceed 5000 ft/min, or be less than 1000 ft/min, anyway, 4000 ft/min is the best practice.
- 5) Sheaves should be dynamically balanced for impeller speeds above 5000 ft/min.



24.) Fans or Blowers Standard Positions

The fan blowers positions depends entirely on application design. The standard positions are:



25.) Vibration:

The natural frequency of a system is the frequency at which the system prefers to vibrate. It can be calculated by the following equation:

$$f_n = 188 (1/d)^{0.5} \text{ (cycles per minute) =}$$

The static deflection corresponding to this natural frequency can be calculated by the following equation:

$$S_d = (188/f_n)^2 \text{ (inches) =}$$

By adding vibration isolation, the transmission of vibration can be minimized. A common **rule of thumb** for selection of vibration isolation is as follows:

Equipment RPM	Static Deflection of Isolation	
	Critical Installation	Non-critical Installation
1200+	1.0 in	0.5 in
600+	1.0 in	1.0 in
400+	2.0 in	1.0 in
300+	3.0 in	2.0 in

III. DUCTS AND DUCTWORK:

The term “**ductwork**” denotes all the equipment between capture devices of particulates and the control devices, including: straight ducts, fittings (elbows and tees), flow control devices (dampers) and duct supports. The design of the ductworks in ventilation systems are often done by using:

- Velocity Method Constant Pressure
- Loss Method (or Equal Friction Method)

1.) The Velocity Method

Sizes of ducts are then given by the continuity equation:

$$A = Q / V =$$

Where:

A = duct cross sectional area (ft²) (m²);

Q = air flow rate (ft³/min) (m³/s);

V = air speed (ft/min) (m/s).

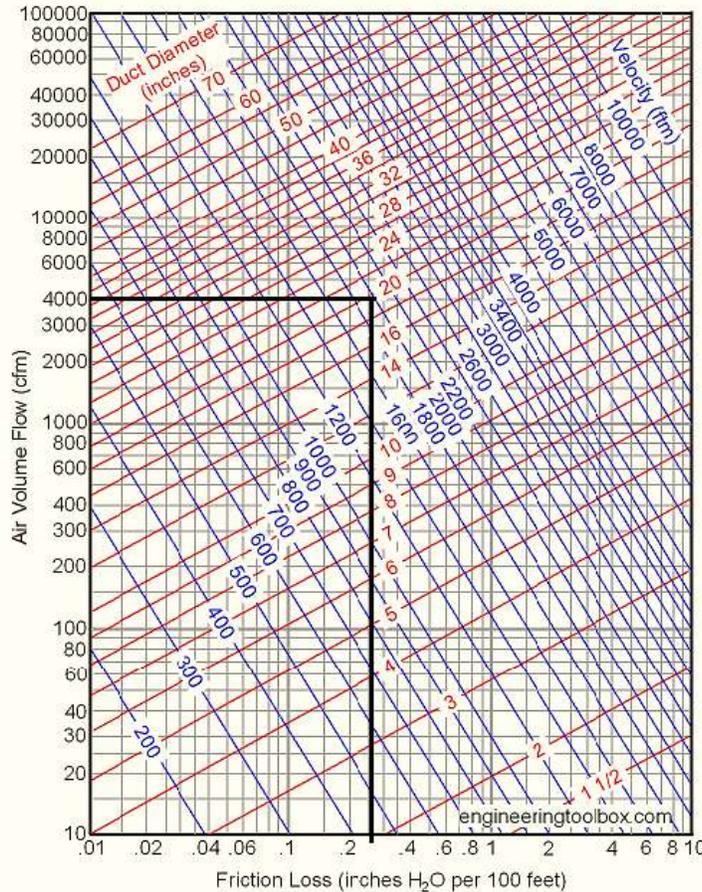
The **proper velocity** depends on the application. The table below indicates common velocity limits:

Type of Duct	Comfort Systems	Industrial Systems	High Speed Systems
Main ducts	800~1400 ft/min (4 ~ 7 m/s)	1600~1400 ft/min (8 ~ 12 m/s)	2000~3500 ft/min (10 ~ 18 m/s)
Main branch ducts	600~1000 ft/min (3 ~ 5 m/s)	1000~1600 ft/min (5 ~ 8 m/s)	1200~2400 ft/min (6 ~ 12 m/s)
Branch ducts	200~600 ft/min (1 ~ 3 m/s)	600~1000 ft/min (3 ~ 5 m/s)	1000~1600 ft/min (5 ~ 8 m/s)

Note.: Be aware that **high velocities** close to outlets and inlets may generate **unacceptable noise**.

2.) Chart Friction Loss Method:

Chart friction loss or head loss (k) in standard air ducts are indicated in the diagram below:



(Equation) $P_{fr} = (k_{loss} / 100 \text{ ft}) \times (L) =$

Example:

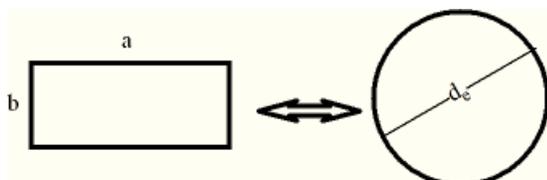
In a **20 inches duct** diameter with air flow **4000 cfm** and air velocity approximately **2000 fpm**, the **friction loss** is approximately **0.25 inches water per 100 feet duct** as shown in the diagram above.

The diagram is based on **standard air 0.075 lb/ft³** in **clean round galvanized metal ducts**.

- 1 inch water = 249 N/m² (Pa) = 0.036 lb/in² (psi) = 25.4 kg/m² = 0.0739 in mercury;
- 1 ft³/min (CFM) = 1.7 m³/h = 0.47 l/s;
- 1 ft/min (FPM) = 0.00508 m/s.

2.1) Equivalent diameters for rectangular and circular ducts - Air flows between 100 - 50,000 cfm:

The table below is based on the ducts friction loss formula, adapted to the equal friction loss method of sizing ventilation duct systems. An approximate **friction loss of 0.8 inches WG per 100 ft duct** is used.



Air flow - Q <i>(ft³/min, cfm)</i> <i>(m³/s)</i>	Rectangular Duct Sizes <i>(Inches)</i>	Equivalent Diameter Round Duct Sizes <i>(Inches)</i>	Velocity - V <i>(ft/min)</i> <i>(m/s)</i>	Friction Loss <i>(Inch WG</i> <i>per 100 ft duct)</i>
200 (0.09)	(3 x 7) – (4 x 5)	4.9	1527 (7.8)	0.88
300 (0.14)	(4 x 7) – (5 x 6)	5.7 ~ 6.0	1635 (8.3)	0.82
400 (0.19)	(4 x 9) – (5 x 7) – (6 x 6)	6.4 ~ 6.6	1736 (8.8)	0.80
500 (0.24)	(6 x 7)	7.1	1819 (9.2)	0.78
750 (0.35)	(5 x 12) – (6 x 10) – (7 x 8)	8.3 ~ 8.4	1996 (10.1)	0.77
1000 (0.47)	(7 x 10) – (8 x 9)	9.1 ~ 9.3	2166 (11)	0.79
1250 (0.59)	(8 x 10) – (9 x 9)	9.8	2386 (12.1)	0.88
1500 (0.71)	(8 x 12) – (10 x 10)	10.7 ~ 10.9	2358 (11.9)	0.77
2000 (0.94)	(8 x 15) – (10 x 12)	11.8 ~ 12.0	2589 (13.2)	0.81
2500 (1.2)	(10 x 14) – (12 x 12)	12.9 ~ 13.1	2712 (13.8)	0.8
3000 (1.4)	(12 x 14)	14.1	2767 (14.1)	0.75
3500 (1.7)	(12 x 15)	14.6	3010 (15.3)	0.84
4000 (1.9)	(10 x 22) – (14 x 15)	15.9 ~ 15.8	2938 (14.9)	0.73
4500 (2.1)	(12 x 19) – (14 x 16)	16.4	3068 (15.6)	0.76
5000 (2.4)	(10 x 25)-(12 x 20)-(15 x 16)	16.9	3248 (16.5)	0.82
6000 (2.8)	(14 x 20) – (15 x 18)	17.9 ~18.2	3358 (17.1)	0.8
7000 (3.3)	(12 x 26) – (16 x 20)	19.0 ~ 19.5	3482 (17.7)	0.8
8000 (3.8)	(12 x 30) – (14 x 25)	20.2	3595 (18.3)	0.8
9000 (4.3)	(12 x 34) – (15 x 25)	21.4 ~ 21.0	3671 (18.6)	0.78
10000 (4.7)	(12 x 36)–(16 x 25)-(20 x 20)	21.9 ~ 21.9	3858 (19.6)	0.83
15000 (7.1)	(16 x 36)-(18 x 30)-(23 x 25)	24.7 ~ 26.2	4331 (22)	0.87
17500 (8.3)	(16 x 40)-(20 x 32)-(25 x 25)	27.0 ~ 27.5	4337 (22)	0.79
20000 (9.4)	(20 x 35) – (25 x 28)	28.6 ~ 28.9	4483 (22.8)	0.79
25000 (11.8)	(16 x 55)-(20 x 43)-(25 x 38)	31.0 ~ 33.5	4709 (23.9)	0.78
30000 (14.2)	(20 x 50) – (30 x 32)	33.7 ~ 33.9	4815 (24.5)	0.74
35000 (16.5)	(20 x 55) – (30 x 35)	35.2 ~ 35.4	5179 (26.3)	0.81
40000 (18.9)	(25 x 48) – (30 x 40)	37.4 ~ 37.8	5243 (26.6)	0.77
45000 (21.2)	(32 x 40)	39.1	5397 (27.4)	0.77
50000 (23.6)	(32 x 45) – (35 x 40)	40.9 ~ 41.3	5222 (26.5)	0.66

Examples:

a) Air flows in a **0.6 m diameter** circular duct. At the end of a long straight run the centerline velocity is measured to be **9 m/sec**. Estimate the **average velocity and mass flow rate** of the air assuming standard conditions for the air.

Assuming a parabolic velocity distribution inside the duct, average velocity is 2/3 of centerline velocity.

Average Velocity, $V = 2/3 \times 9 = 6 \text{ m/s}$;

Mass Flow Rate, $Q = V \times \text{Area} = 6 \times (\pi/4 \times 0.6^2) = 1.696 \text{ m}^3/\text{s} = 3,593.62 \text{ cfm}$.

b) Fill in the blanks.

Duct. No	Shape of Duct	Area (ft ²)	Velocity (fpm)	Q (cfm)	Dimensions
a	Circle		1000		Diameter = 6"
b	Triangle			5000	Base=10",Height=20"
c	Square		4500		Side = 20"
d	Rectangle			1600	Side 1=10",Side 2=6"

Solution:

Duct. No	Shape of Duct	Area (ft ²)	Velocity (fpm)	Q (cfm)	Dimensions
a	Circle	0.196	1000	196	Diameter = 6"
b	Triangle	0.694	7204.611	5000	Base=10",Height=20"
c	Square	2.778	4500	12501	Side = 20"
d	Rectangle	0.417	3836.93	1600	Side 1=10",Side 2=6"

c) Assume duct diameter **54.33 inches, length 100 feet, velocity 4000 fpm** and that air is flowing through a black iron duct. Calculate the friction factor using **Wright's and Darcy-Weisbach equations**.

Using **Wright's** equation, friction loss h is given by:

$$h = 2.74 (V / 1000) \times 1.9 / D \times 1.22 =$$

Where:

V = Duct velocity in fpm;

D = Diameter in inches.

$$h = 2.74 (4000 / 1000) \times 1.9 / 54.33 \times 1.22 = \mathbf{0.29'' \text{ WG}}$$

Using **Darcy-Weisbach** equation, friction loss h is given by:

$$h = 12 \times (f / D) \times L \times VP =$$

From Moody Diagram, friction factor **f = 0.011**.

Where:

f = Friction factor, (f = 0.011);

D = Diameter in inches;

L = Length in ft;

VP = Velocity Pressure in inch WG.

$$VP = (V / 4005)^2 = 0.998'' \text{ WG.}$$

$$h = 12 \times (0.011 / 54.33) \times 100 \times 0.998 = 0.24'' \text{ WG.}$$

d) Determine the **velocity** of the following particles:

Saw dust:

D = 35 microns ;
Specific gravity = 0.80;

Glass particles:

D= 10 microns;
Specific gravity = 1.8.

Solution:

$$V = 0.0052 \times SG \times D^2 =$$

Where:

V = Velocity in fpm;
SG = Specific gravity;
D = Diameter in microns.

For saw dust, $V = 0.0052 \times 0.8 \times 35^2 = 5.1 \text{ fpm.}$

For glass particles, $V = 0.0052 \times 1.8 \times 10^2 = 0.94 \text{ fpm.}$

e) Determine the scfm of an air stream measured at **7000 cfm and 300°F.**

Solution:

$$\text{SCFM} = \text{ACFM} \times T_s / T_a =$$

Where:

SCFM = Standard Cubic Feet per Minute;
Ts = Standard temperature in R;
Ta = Actual temperature in R.

$$\text{SCFM} = 7000 \times 530 / (300 + 460) = 4882 \text{ cfm}$$

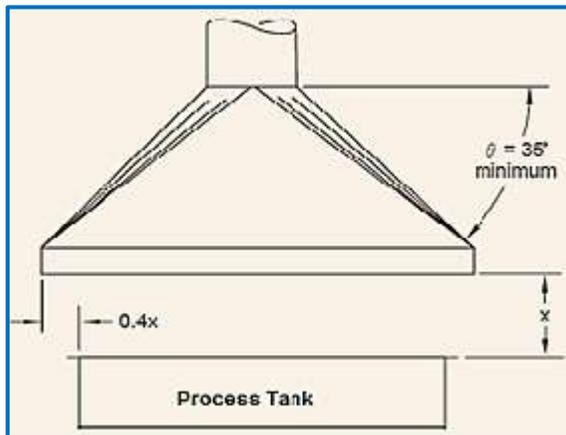
3.) Ducts and Ductwork Material:

Ductwork is fabricated from either metal or plastic, according to application, structural considerations, purchase and installation costs, aesthetics, and other factors. Metals used include carbon steel (bare or galvanized), stainless steel, and aluminum according to **SMACNA** (Sheet Metal and Air Conditioning Contractors' National Association) Standards.

3.1) Ductwork Components:

The ductwork system consists of straight ducts, fittings, flow control devices, and supports. The "fittings" change the direction of the ducted gas stream, modify the stream velocity, tie it to another duct(s), facilitate the connection of two or more components, or provide for expansion/contraction when thermal stresses arise using Expansion Joints.

a) Typical Canopy Hood:



b) Typical Fittings:

LONGITUDINAL SEAM DUCT
(Solid welded longitudinal seam)

DIMENSIONS:
8-inch minimum
90-inch maximum

GORED ELBOW

DIMENSIONS:
 $R = 1.5A$

θ	Number of gores
0 - 30°	2
36 - 71°	3
72 - 90°	5

For elbows where θ exceeds 90°, add one gore for each additional 10° of fraction thereof.

STRAIGHT TEE

DIMENSIONS:
 $V = C + 2$
Maximum $C = A$

STRAIGHT 90° CROSS

DIMENSIONS:
 $V = C + 2$
Maximum C or $D = A$

DAMPERS

Face View

CONCENTRIC REDUCER

$A - B$ (1-inch minimum or 12-inch maximum)

4.) Insulation:

Serves to reduce heat loss/gain, saves energy and prevents noise and condensation. The **insulation layer is typically 1-inch, although 2- and 3-inch thicknesses**, available for more extreme applications. The **thermal conductivities** of these thicknesses are **0.27, 0.13, and 0.09 Btu/hr-ft²-°F**, respectively.

5.) Expansion Joints:

The **bellows expansion joint**, consists of a piece of **flexible metal** welded to connect two duct ends, to allow the ductwork to **expand or contract** in response to temperature variation and thermal stresses.

For instance, **silicone fiberglass and aramid fiber cloth** can be used for duct temperatures of up to **500°F**, while **coated fiberglass cloth** is needed to accommodate temperatures of **1000°F**.

6.) Support Systems:

The last component to consider is the ductwork support system. The SMACNA - HVAC Duct Construction Standards manual recommends the selection of a hanging system **should not be taken lightly**, because the erection of an inadequate hanging system can be disastrous.

As a rule, a support should be provided for every **8 to 10 feet** of duct run. Ductwork can be suspended from a ceiling or other overhead structure via hangers or supported from below by girders, pillars, or other supports.

7.) Design Equations, Loss Factors and Coefficients of Entry

When air or gas enters a hood, there is mechanical energy loss due to friction. This friction loss is calculated using the equations described below:

7.1) Hood Loss Factor (unitless) - K

The **Static Gauge Pressure, SPh** is the **reference pressure** at one atmosphere. The hood is under a vacuum created by a fan downstream and should be always calculated considering the standard pressure and temperature. The empirical equations to estimate the pressure **loss factors (K)** are:

$$K = (SPh) / VP - 1 =$$
$$K = (Sph) - VP / VP =$$

7.2) Hood Coefficient of Entry – Ce:

The **hood loss factor (K)** varies according to the hood shape. It can range from **0.04 to 1.78** for various slotted hoods. The parameter related to the hood loss factor is the **Coefficient of Entry, Ce** defined as:

$$Ce = (1 / 1 + K)^{0.5}$$

(or) $Ce = (VP / SP)^{0.5}$

Example:

a) In some duct the transport velocity (**V**) is **3,500 ft/min** and the Static Gauge Pressure, (**SPh**) is **1.75 in. WG**. Calculate the **loss factor (K)** and **Coefficient of Entry (Ce)** considering standard temperature.

$$VP = (V / 4005)^2$$
$$VP = (3500 / 4005)^2 = 0.76'' \text{ in WG.}$$

$$K = (SPh / VP) - 1$$
$$K = (1.75) / 0.76 - 1 = 1.30 \text{ (unitless).}$$

$$Ce = (1 / 1 + K)^{0.5} =$$
$$Ce = (1 / 1 + 1.30)^{0.5} = 0.66 \text{ (unitless).}$$

7.3) Design Equations, Loss Factors, and Coefficients of Entry:

Hood Type	Design Equation	Hood Loss Factor (K)	Coefficient of Entry (Ce)
Duct End (round)	$Q = 4 \times X^2 \times Vs =$	0.93	0.72
Flanged Duct End	$Q = 2. X^2 \times Vs =$	0.50	0.8
Canopy Hood	$Q = 1.4 \times Px X \times Vs =$	1.78	N/A
Tapered Hood	$Q = 2 \times X \times Vs =$	0.06	0.9
Dip Tank Hood (slotted)	$Q = 125x A \text{ (tank)} =$	1.78	N/A

8.) Average or Slot Velocity – Vs:

The face velocity is the average velocity of the air passing through the hood inlet (face). A similar parameter is the **slot velocity**, which is the **average air velocity** through the hood slot openings, whose areas is only a fraction of the entire hood face area. The parameters are in the table below:

Operation/Hood Type	Velocity Type	Velocity Range(ft/min)
Tanks, degreasing	Capture	50-100
Drying oven	Face	75-125
Spray booth	Capture	100-200
Canopy hood	Capture	200-500
Grinding, abrasive blasting	Capture	500-2,000
Slot hood	Slot	2,000

9.) Hood Basic Sizing:

A simple sizing method for estimating purposes to determine the volumetric **flow rate (Q)** through the hood and ductwork, the following equation below can be used:

$Q = 1.4.P.X.Vs =$

Where:

- Q = Estimated Flow Rate (ft³/min);
- P = Perimeter of the Ventilation Source or Tank (ft);
- X = Distance of Hood Above Source or Tank (ft);
- Vs = Slot Velocity (ft/min).

Example:

a) A circular canopy hood is being used to capture emissions from a Chromium Electroplating Tank, **8-foot diameter**. The hood face is **6 feet above the tank, 40% bigger** than the tank diameter. The capture velocity is **200 ft/min**. At air standard conditions, calculate the required volumetric flow rate, the hood face area, and the hood face velocity.

Solution:

Because the tank is circular, $P = \pi \times 8 \text{ ft} = 25.1 \text{ ft}$.

$Q = 1.4.P.X.Vs =$

$$Q = 1.4 \times 25.1 \times 6 \times 200 = 42,168 \text{ ft}^3/\text{min}.$$

The hood diameter is 40% greater than the tank diameter. Then, the hood face area is:

$$A = [\pi \cdot (1.4 \cdot d)^2 / 4] = [\pi \cdot (1.4 \cdot 8)^2 / 4] = 98.5 \text{ ft}^2.$$

The hood slot velocity (**Vs**) would be:

$$V_s = Q / A =$$

$$V_s = 42,168 / 98.5 = 428 \text{ ft/min}.$$

10.) Stack Design:

Can be fabricated of steel, brick, or plastic (fiberglass-reinforced plastic, or FRP), installed after control devices to **disperse the exhaust gases** above ground level and surrounding buildings.

For study purposes, the only two stack design parameters that need to be determined are the **stack diameter** and the **stack height**. The other variables (wall thickness) are incorporated into the equipment.

The stack diameter (**Ds**) can be calculated via the duct diameter formula:

$$D_s = 1.128 (Q / V_s)^{0.5} =$$

Height should be dimensioned according to application. **Short stacks** are no taller than **120 feet**. **Tall stacks** can be up to **1,000 feet** high.

Example:

a) A fabric filter handling **16,500 ft³/min** of cocoa dust must be dispersed through a stack. Calculate the **diameter** of the stack knowing that the minimum transport velocity is **3000 ft/min**.

$$D_s = 1.128 (Q / V_s)^{0.5} =$$

$$V_s = 1.128 (16500 / 3000)^{0.5} = 2.64 \text{ inches}.$$

b) Assume a **4 inch duct diameter 0.087 ft²** using a bench grinder hood **Ce = 0.78** and velocity **4000 fpm**. Calculate **Velocity Pressure, Static Pressure and Flow Rate**.

The Velocity Pressure is:

$$V = 4005 (VP)^{0.5}$$

$$(VP)^{0.5} = (4000) / (4005) = 0.998 \text{ in. WG}$$

The Static Pressure is:

$$C_e = (VP/SP)^{0.5} = 0.78$$

$$(VP/SP) = (0.78)^2 = \text{thus, } SP = VP / (0.78)^2 =$$

$$SP = (0.998) / (0.608) = 1.64 \text{ in. WG}.$$

The Flow Rate is:

$$Q = V \times A =$$

$$Q = 4005 \times (A) \times (VP)^{0.5} =$$

$$Q = 4005 \times (0.087) \times (0.998)^{0.5} =$$

$$Q = 348 \text{ cfm.}$$

11.) Air Cooling Flow Rate:

If air is used for cooling, the needed **air flow rate** may be expressed as:

$$Q_c = W_c / \rho C_p (T_o - T_r) =$$

Where:

Q_c = Volume of air for cooling (m^3/s);

H_c = Cooling load (W);

T_o = Outlet temperature ($^{\circ}C$);

$T_i = T_r$ - If the air in the room is mixed.

12.) Air Heating Flow Rate:

If air is used for heating, the needed air flow rate may be expressed as:

$$Q_h = W_h / \rho C_p (T_i - T_o) =$$

Where:

Q_h = Volume of air for heating (m^3/s);

W_h = Heat load (W);

ρ = Density of air (kg/m^3).

C_p = Specific heat capacity of air (J/kg K)

T_i = Supply temperature ($^{\circ}C$)

T_o = Room temperature ($^{\circ}C$)

Example:

a) If the heat load is $W_h = 400 \text{ W}$, supply temperature $T_i = 30^{\circ}C$ and the room temperature $T_o = 22^{\circ}C$, the air flow rate can be calculated as:

$$Q_h = W_h / \rho C_p (T_i - T_o) =$$

$$Q_h = (400 \text{ W}) / (1.2 \text{ kg/m}^3) (1005 \text{ J/kg K}) ((30^{\circ}C) - (22^{\circ}C)) =$$

$$Q_h = 0.041 \text{ m}^3/\text{s} = 149 \text{ m}^3/\text{h.}$$

13.) Spray Booths and Tanks - Exhaust Hood Capture Velocities:

The Industrial Ventilation Manual gives ranges for various processes as shown in the table below:

Contaminant Dispersion Method	Examples	Capture Velocity, fpm
Released without velocity into quiet air	Evaporation from tanks, degreasing, etc.	50 – 100
Released at low velocity into moderately still air	Spray booths, intermittent container filling, plating, pickling, low speed conveyor transfers, welding	100 – 200
Active generation into a zone of rapid air motion.	Spray painting, barrel filling, conveyor loading, crushers	200 – 500
Released at high initial velocity into a zone of very rapid air motion	Grinding, abrasive blasting, tumbling	500 – 2000

Some additional guidelines for the use of slot hoods are as follows:

- If the width of a tank is 20" or less, a slot on one side is acceptable.
- If the width is 20" - 36", slots on both sides are desirable.
- If the width is 36" - 48", slots on both sides are necessary, unless all other conditions are optimal.
- If the length of the plenum is over 6 feet, multiple take-offs are recommended, and if over 10 feet considered necessary.

Based upon the above criteria, tables have been constructed to provide guidelines for tank ventilation with lateral (side slot) exhaust:

Required Min. Control Velocity, fpm	CFM per Square Foot Tank Area to Maintain the Required Minimum Control Velocities at the Following Tank Width to Length Ratios (W/L)				
	0.00-0.09	0.10-0.24	0.25-0.49	0.50-0.99	1.00-2.00
For a hood along one side or two parallel sides of a tank when one hood is against a wall or baffle. Also for a manifold along the tank centerline.					
50	50	60	75	90	100
75	75	90	110	130	150
100	100	125	150	175	200
150	150	190	225	250	250
Hood along one side or two parallel sides of a free standing tank not against a wall or baffle.					
50	75	90	100	110	125
75	110	130	150	170	190
100	150	175	200	225	250
150	225	250	250	250	250
Use W/2 as the tank width when the manifold is along the centerline or two parallel sides of the tank.					

14.) Hood Velocity Empirical Equation:

The equation for calculating flow for round or essentially square free hanging hoods is:

$$V(x) = Q / (10. X^2 + A) =$$

Where:

V(x) = Centerline velocity at distance x;

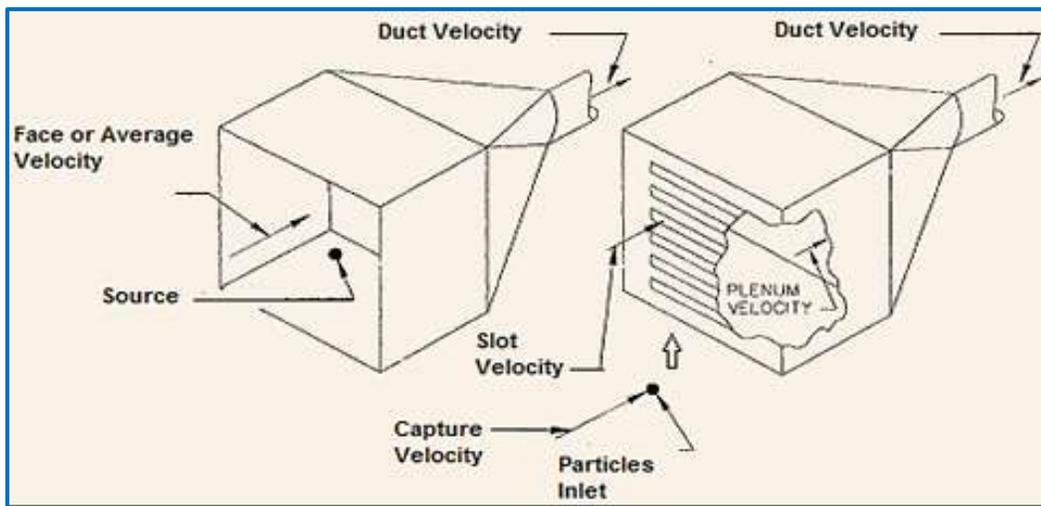
X = Distance along the axis in feet (and where X is less than or equal to 1.5 x D);

Q = Air flow, CFM;

A = Area of the hood opening, ft²;

D = Diameter of round hoods or the side of essentially square hoods, ft.

The velocity of the **air stream moving toward the hood** is (approximately) inversely proportional to the square of the distance from the slot or hood opening. This emphasizes how important hood location is.



Air volume and velocity ratios for some rectangular shapes may be represented as follows:

Hood Type	Aspect Ratio, W/L	Air Flow Rate, Q
Rectangular Slot, plain end	<0.2	$Q = 3.7 L * V * X$
Rectangular Slot, flanged end	<0.2	$Q = 2.8 L * V * X$
Rectangular Slot, plain end	>0.2	$Q = V * (10 * X^2 + A)$
Rectangular Slot, flanged end	>0.2	$Q = 0.75 * V * (10 * X^2 + A)$
Open Booth	As required	$Q = V * A = V * L * W$
Canopy Hoods	As required	$Q = 1.4 * P * D * V$

Where:

W = Width of slots or rectangular opening, ft

L = Length of slots or rectangular opening, ft

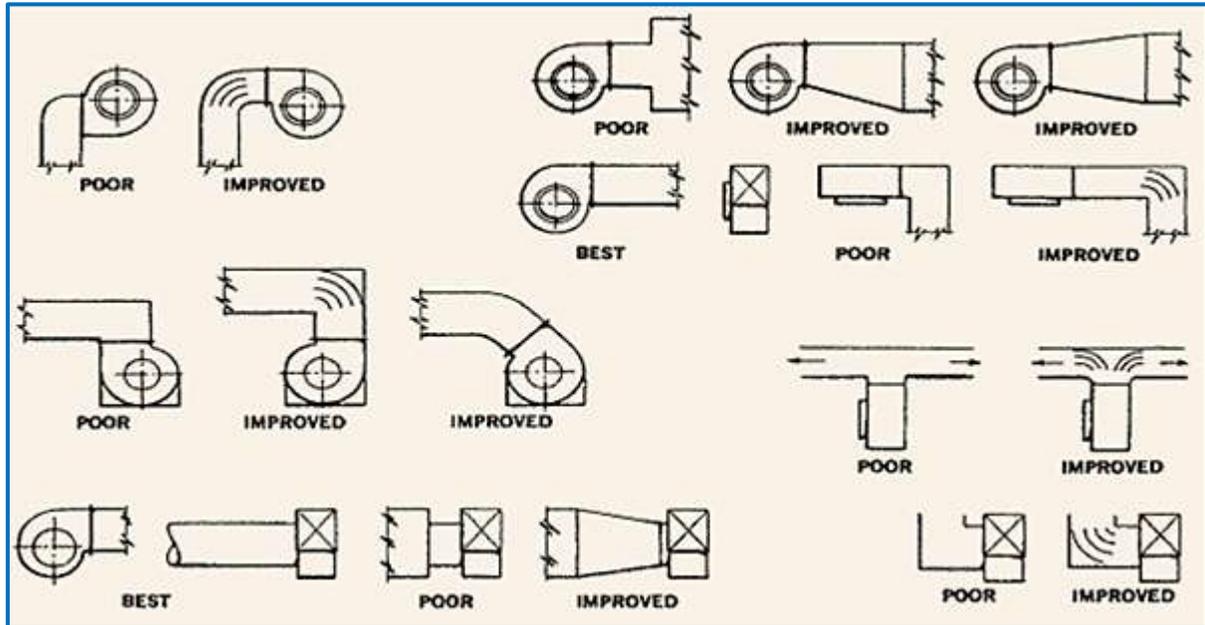
X = Distance from opening, ft

P = Perimeter of the hood, ft

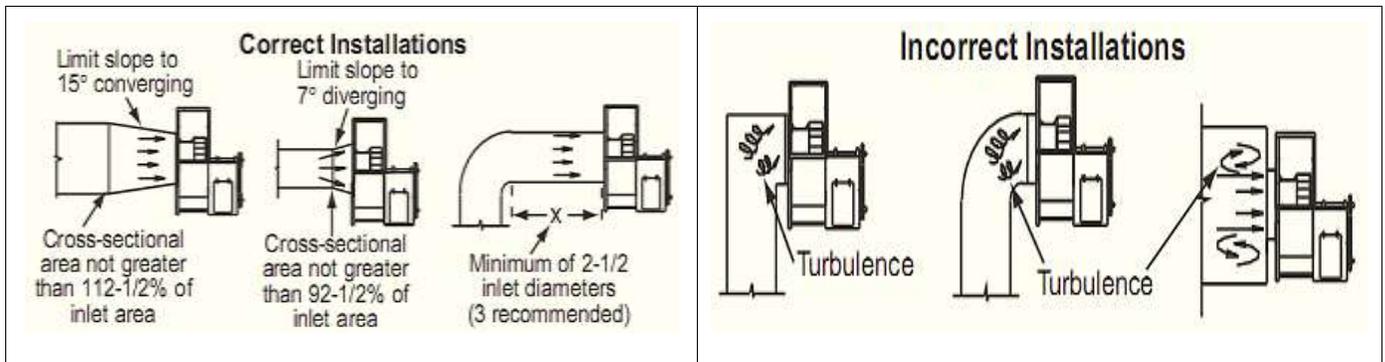
D = Distance of the canopy above the work, ft

15.) Air Flow Entrance and Exit:

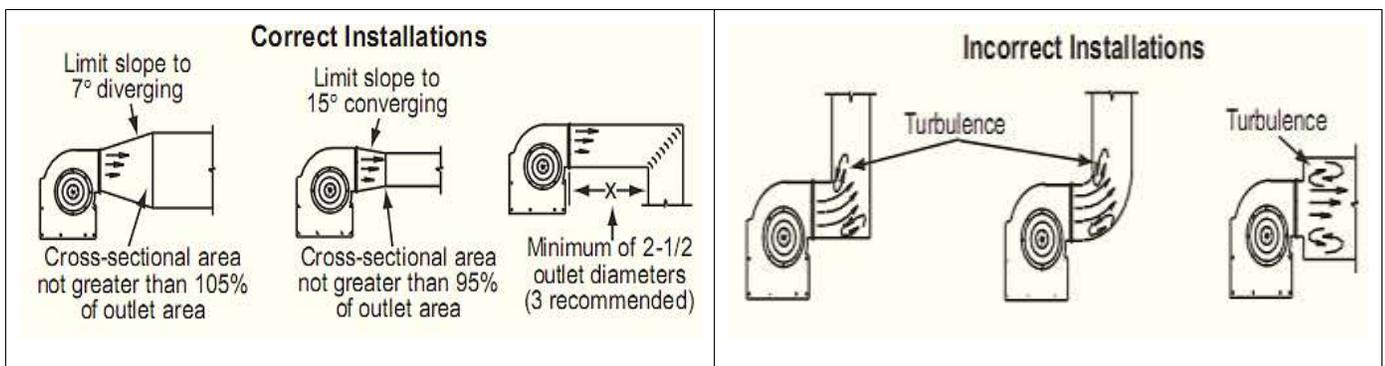
Air flow at the **entrance and exit of a fan should be as smooth** as possible to minimize the generation of turbulence. The conditions that produce turbulent air flow usually result in greater noise generation and increase Static Pressure Drop in the system. The design conditions are shown below:



15.1) Typical Inlet Conditions:



15.2) Typical Outlet Conditions:



16.) SCFM and ACFM Corrections:

a) Correcting for **Standard Cubic Feet per Minute (cfm):**

$$\text{SCFM} = \text{ACFM} [530 / (460 + T_{ct})] (P_{ct} / 29.92) =$$

Where:

SCFM = Standard flow rate, CFM;

ACFM = Actual flow rate = measured flow rate, CFM;

Tct = Measured dry bulb temperature of the actual airstream, °F;

Pct = Absolute pressure of the actual airstream, in. Hg.

b) Correcting for **Actual Cubic Feet per Minute (cfm):**

$$\text{ACFM} = \text{SCFM} [(460 + T_{ct}) / 530] (29.92 / P_{ct}) =$$

Where:

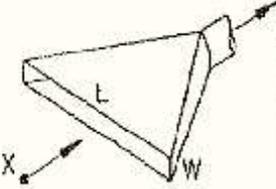
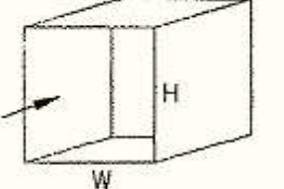
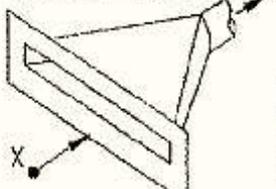
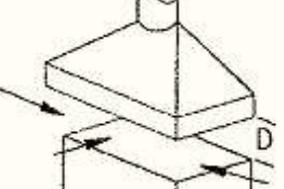
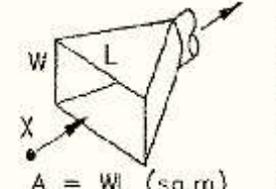
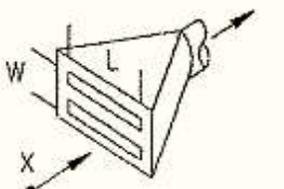
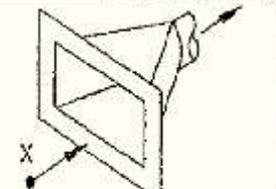
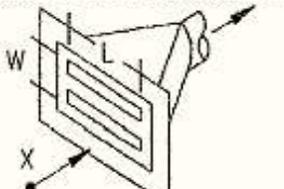
SCFM = Standard flow rate, CFM;

ACFM = Actual flow rate = measured flow rate, CFM;

Tct = measured dry bulb temperature of the actual airstream, °F;

Pct = absolute pressure of the actual airstream, in. Hg.

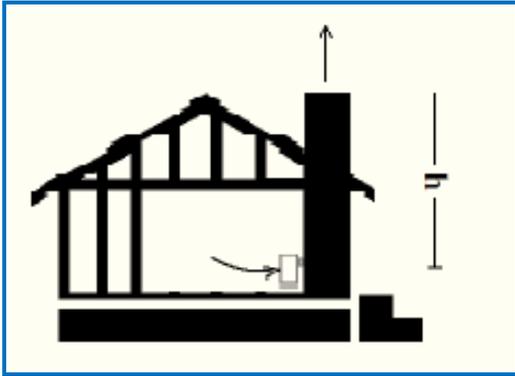
17.) Hood Types and Classification:

	SLOT		BOOTH
	FLANGED SLOT		CANOPY
	PLAIN OPENING		PLAIN MULTIPLE SLOT OPENING (2 OR MORE SLOTS)
	FLANGED OPENING		FLANGED MULTIPLE SLOT OPENING (2 OR MORE SLOTS)

18.) Chimney Sizing:

The capacity of a chimney depends on the potential chimney draft - the pressure difference created by the inside and outside air temperature difference and chimney height - and the chimney area.

The pressure difference in the chimney, pulls air and fuel gas from the boiler or fireplace through the chimney and out of the building.



The pressure difference or chimney draft can be expressed as:

$$Dp_c = h (\rho_o - \rho_i) g =$$

Where:

Dp_c = Pressure draft in the chimney, WG (Pa)

h = Height of chimney, ft (m)

ρ_o = Density of outside air, lb/ft³ (kg/m³)

ρ_i = Density of inside air, lb/ft³ (kg/m³)

g = Acceleration of gravity, 32.174 ft/s² (9.8 m/s²)

And can alternatively be expressed as:

$$Dp_c = 0.0465 \cdot dt \cdot h =$$

Where:

dt = temperature difference between inside and outside air, °F (°K) (°C).

18.1) Chimney Area:

The **velocity of air and flue gas** in a smaller furnace should not exceed 2 m/s. Design velocities of larger furnaces should not exceed 10 m/s. Required chimney area can be calculated as:

$$A = Q / V =$$

Where:

A = Cross-sectional area of chimney, ft² (m²);

Q = Flow rate at chimney temperature, ft³/min (m³/s);

V = Velocity, ft/min (m/s).

In general **1 kW** boiler heat requires **1100 mm²** of chimney area.

19.) HVAC, Air Flow:

- a) **Air Change Method:** Involves calculating the necessary number of air changes necessary to give the proper ventilation for a given space using the following equation:

$$\text{cfm} = \frac{\text{building volume (ft}^3\text{)}}{\text{Min/air change}} =$$

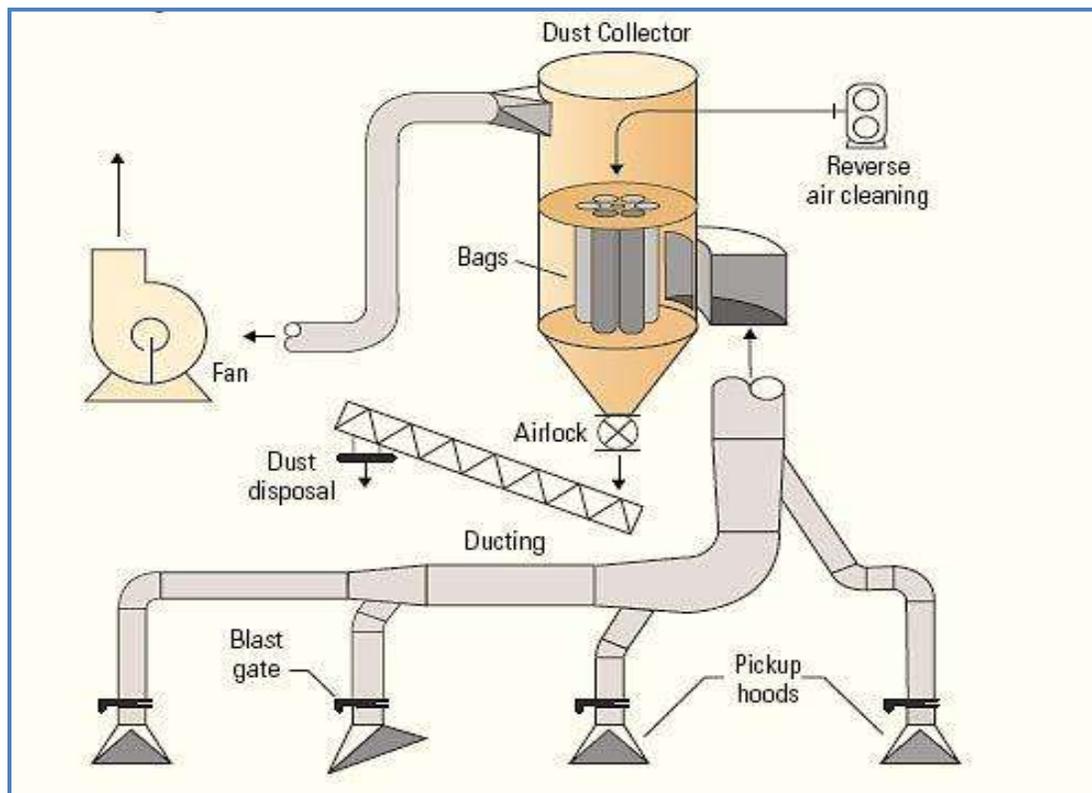
- b) **Heat Removal Method:** The average outside temperature, necessary inside temperature and Btu/min are required for this method. This formula gives the amount of air to be passed through the building to maintain desired inside temperature, using the following equation:

$$\text{cfm} = \frac{\text{total Btu/min}}{0.018 \times \text{temperature rise } ^\circ\text{F}} =$$

IV. BAG FILTERS:

Commonly known as **baghouses**, **fabric collectors** use filtration to separate dust particulates from dusty gases and can achieve a collection **efficiency of more than 99%** for very fine particulates.

The dusty-laden gases enter the baghouses and pass through fabric bags that act as filters. The bags can be of woven or felted cotton, synthetic, or glass-fiber material in either a tube or envelope shape.



1.) Selecting a Dust Collector:

Dust concentration and particle size: For minerals processing operations, the dust concentration can range from **0.1 to 5.0 grains per cubic feet of air (0.23 to 11.44 grams per standard cubic meter)**, and the particle size can vary from **0.5 to 100 μm**.

Characteristics of airstream: The characteristics of the airstream can have a significant impact on collector selection. For example, cotton fabric filters cannot be used where air temperatures exceed **180°F (82°C)**. Also, condensation of steam or water vapor can blind bags.

2) Sizing Bag Filters:

The **cleaning methods** for baghouses, using the **typical Air to Cloth ratio** are:

Shaker cleaning = **3 cfm/ft²**;
Pulse jet cleaning = **7 cfm/ft²**.

The following method will estimate the **fan size, air flow rate** (cfm), using the **surface area** of the bags and a typical **Air to Cloth** ratio.

The **Air to Cloth ratio** is the amount of air (ACFM) or process gas entering the baghouses.

Examples:

a) Calculate the main data for a bag diameter, 10 inches, length 8 ft, quantity 200 bag filters.

Calculate the bag circumference, using the bag diameter:

$$= 3.1416 \times \frac{10}{\text{Bag Diameter}} = \frac{31.42}{\text{"A"}} \text{ Inches}$$

Determine the **Filter Area** of all the bags, using the bag length:

$$\frac{8.0}{\text{Length}} \times \frac{31.42}{\text{"A"}} \times \frac{200}{\text{Number of Bags}} / 144 = \frac{349.07}{\text{"B"}} \text{ sq feet}$$

Calculate the **Flow Rate**, using the **Filter Area** and **Air to Cloth** ratio:

$$= \frac{349.07}{\text{"B"}} \times \frac{3}{\text{3 or 7 cfm/sq ft}} = \frac{1047}{\text{"C"}} \text{ CFM}$$

b) Calculate the Air to Cloth ratio for a 10' bag with a 6" tube sheet hole (**actual diameter 5.875"**) whose baghouses is handling **10,000 ACFM** with **144 bags**.

$$\text{Air to Cloth per bag} = 5.875'' / 12 \times \pi \times 10' = \mathbf{15.38 \text{ ft.}}$$

$$\text{Air to Cloth ratio} = 10,000 \text{ ACFM} / 144 \times 15.38 \text{ ft} = 10,000 / 2214 = \mathbf{4.52 : 1}$$

c) Calculate the number of bags required for an **8-compartment** pulse-jet baghouses with a process gas exhaust rate, **Q =100,000 ft³/min** and **Air to Cloth (A/C) ratio = 4 (ft³/min)/ft²**, **bag diameter = 6 inches (0.5 ft) x 12 ft height**:

Solution:

Calculate the total **cloth area**:

$$v_f = \frac{Q}{A_c} \text{ or } A_c = \frac{Q}{v_f}$$

Where:

A_c = Cloth area, ft²;

Q = Process exhaust rate, ft³/min.;

v_f = Air filtration velocity, ft/min.

$$\begin{aligned} A_c &= \frac{100,000 \text{ ft}^3 / \text{min}}{4 \text{ ft} / \text{min}} \\ &= 25,000 \text{ ft}^2 \end{aligned}$$

Determine the **area of each bag**:

$$A_b = \pi \cdot D \cdot h =$$

A_b = area of bag, ft²;

$$A_b = 3.14 \times 0.5 \text{ ft} \times 12 \text{ ft} = \mathbf{18.84 \text{ ft}^2}.$$

Calculate the **number of bags** required for the baghouses:

$$\text{Number of bags} = \frac{A_c}{A_b}$$

From step 1: $A_c = 25,000 \text{ ft}^2$

From step 2: $A_b = 18.84 \text{ ft}^2$

$$\begin{aligned} \text{Number of bags} &= \frac{25,000 \text{ ft}^2}{18.84 \text{ ft}^2} \\ &= 1,326.96 \text{ bags} \\ &\text{or } 1,328 \text{ bags} \end{aligned}$$

d) If the cloth area (**A_c**) is known to be **4,050 ft²**, how many bags would be used in a baghouses with the bag area of **78.5 ft²**?

$$\text{Number of Bags} = \frac{A_c}{A_b} = \frac{4050 \text{ ft}^2}{78.5 \text{ ft}^2} =$$

$$\text{Number of Bags} = 52$$

V. ELECTROSTATIC PRECIPITATORS (ESP):

Electrostatic precipitators, described here **only for reference** in ventilation systems, use electrostatic forces to separate dust particles from exhaust gases. A number of high-voltage, direct-current discharge electrodes are placed between grounded collecting electrodes.

There are **two main types** of precipitators:

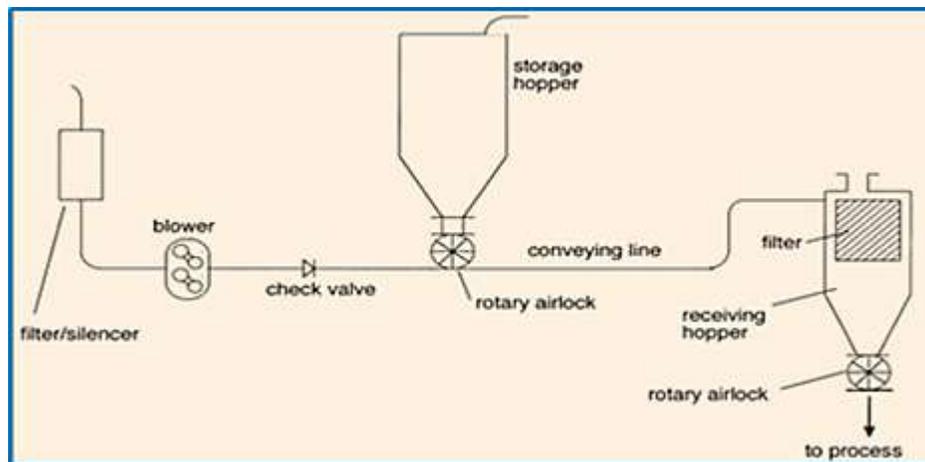
- **High-voltage, single-stage:** Single-stage precipitators combine ionization and a collection step. The high-voltage, single-stage precipitator is widely used in **minerals** processing operations.
- **Low-voltage, two-stage:** Two-stage precipitators use a similar principle; however, the ionizing section is followed by collection plates. The low-voltage, two-stage precipitator is generally used for filtration in air-conditioning systems.

VI. PNEUMATIC CONVEYING:

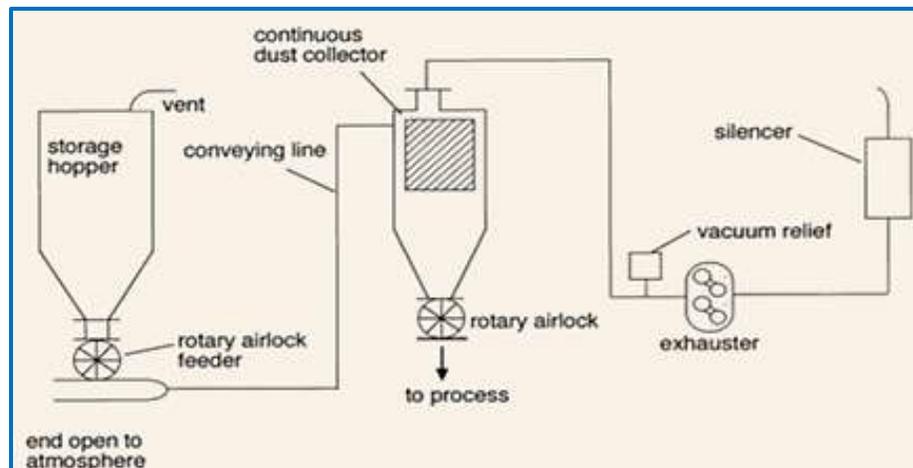
Pneumatic conveying can be used for particles ranging from fine powders to pellets and bulk densities of **1 to 200 lb/ft³** - (16 to 3200 kg/m³). Most millwork plants deal with dry wood in the form of sawdust and shavings. The bulk densities of these materials are in the range of **24 lb/ft³**.

The pneumatic systems are defined as low pressure systems, up to **18 inches WG static pressure**. A **15,000 cfm** fan system could transport **22.5 lb material per minute**. With this value of material loading, the mixture of air and material can be treated as if the load were **exclusively air**.

a) Dilute phase transport positive pressure system:



b) Dilute phase transport negative pressure system:



Examples:

a) Conveying sawdust at a rate of **1800 lb/h through a 6" pipe** with a material loading ratio of **1:2** will result in an air velocity of:

Solution:

Conveying material = $1800 \text{ lb/h} / 60 = 30 \text{ lb/min}$;
Standard air flow = $60 \text{ (lb/min) air} / 0.075 \text{ (lb/ft.}^3\text{) std. density} = 800 \text{ CFM}$;
Pipe area = 6" pipe = **0.1964 ft²**.

Velocity = $800 \text{ CFM} / 0.1964 \text{ ft}^2 = 4073 \text{ FPM}$.

b) Giving the following data:

ACFM = 26,000;
Material Loading = 19,500 lb/h;
Air Inlet Density = 0.05 lb/ft².

Solution:

Air flow rate = $26,000 \times 0.05 = 1,300 \text{ lb/min.}$;
Material flow rate = $19,500/60 = 325 \text{ lb/min}$,

Fan or Blower Selection:

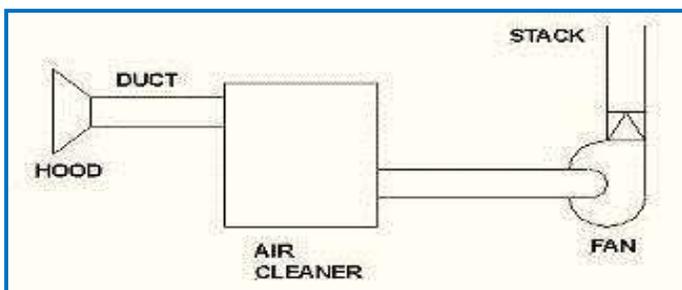
In general, materials up to approximately **50 lb/ft³** can be conveyed with an air velocity of **5000 fpm**. Once the calculations for the system have been performed and the appropriate pipe size has been selected based on material being conveyed, the required flow rate of the fan is:

Air flow rate = velocity x pipe area =

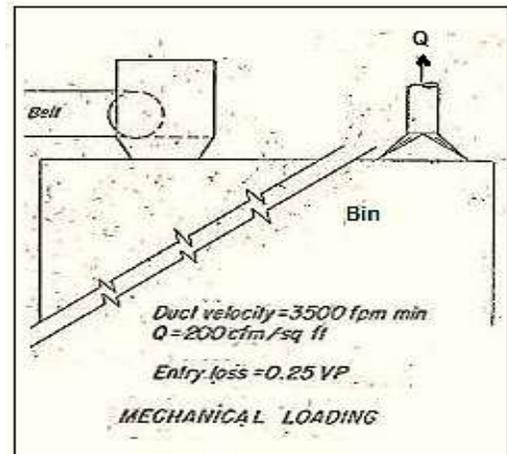
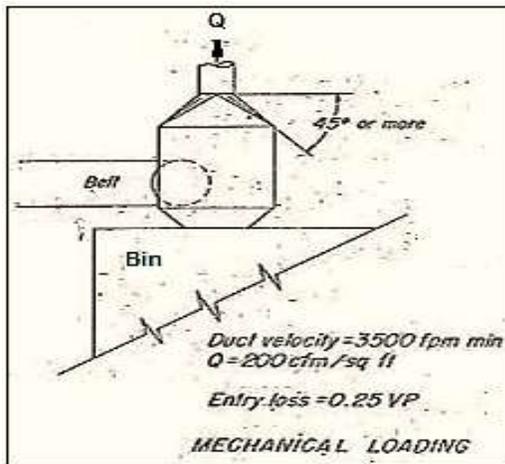
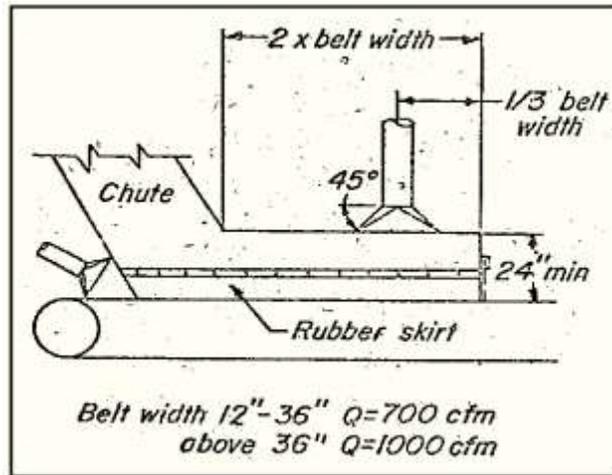
Note: SCFM (Standard Cubic Feet per Minute): a volume of air at **0.075 lbs/ft.³** standard density is used at sea level, temperature at **70 °F**.

VII. PARTICLES EXHAUST CONTROL:

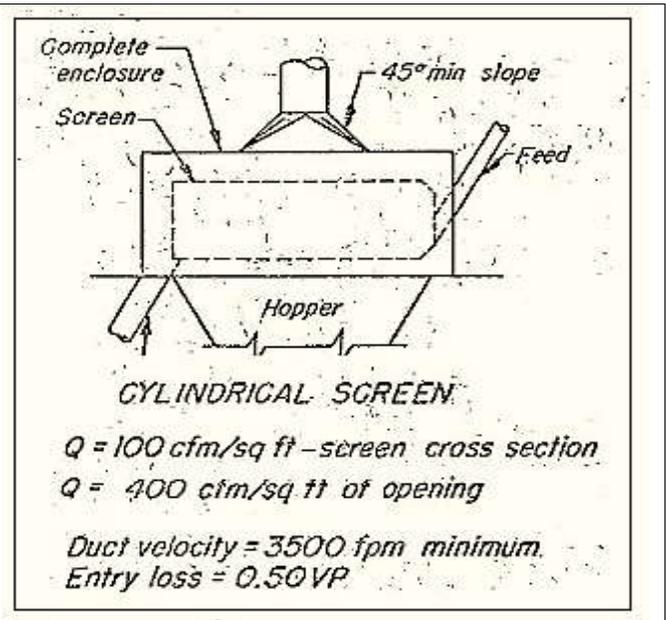
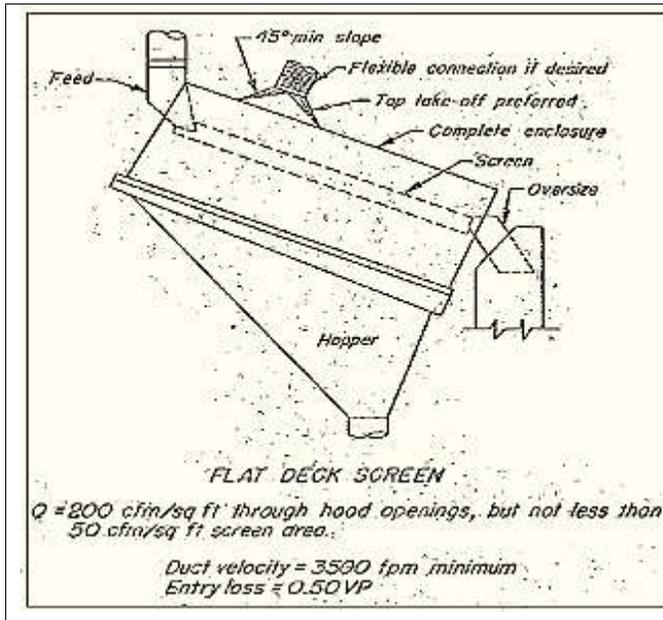
For displacement ventilation systems, designers normally assume that all pollutants follow the buoyant air flow into an upper zone. The ventilation systems are designed with **exhaust fans or blowers** in line with ducts and air cleaners exiting the clean air to atmosphere. A variety pollutant powder of organic or inorganic bulky materials like stones, chalk, fertilizers, saw dust, wood chips, fumes and particles can be exhausted to filter bags or any other pollution control equipment using canopies or plain openings, as shown in the sketch below:



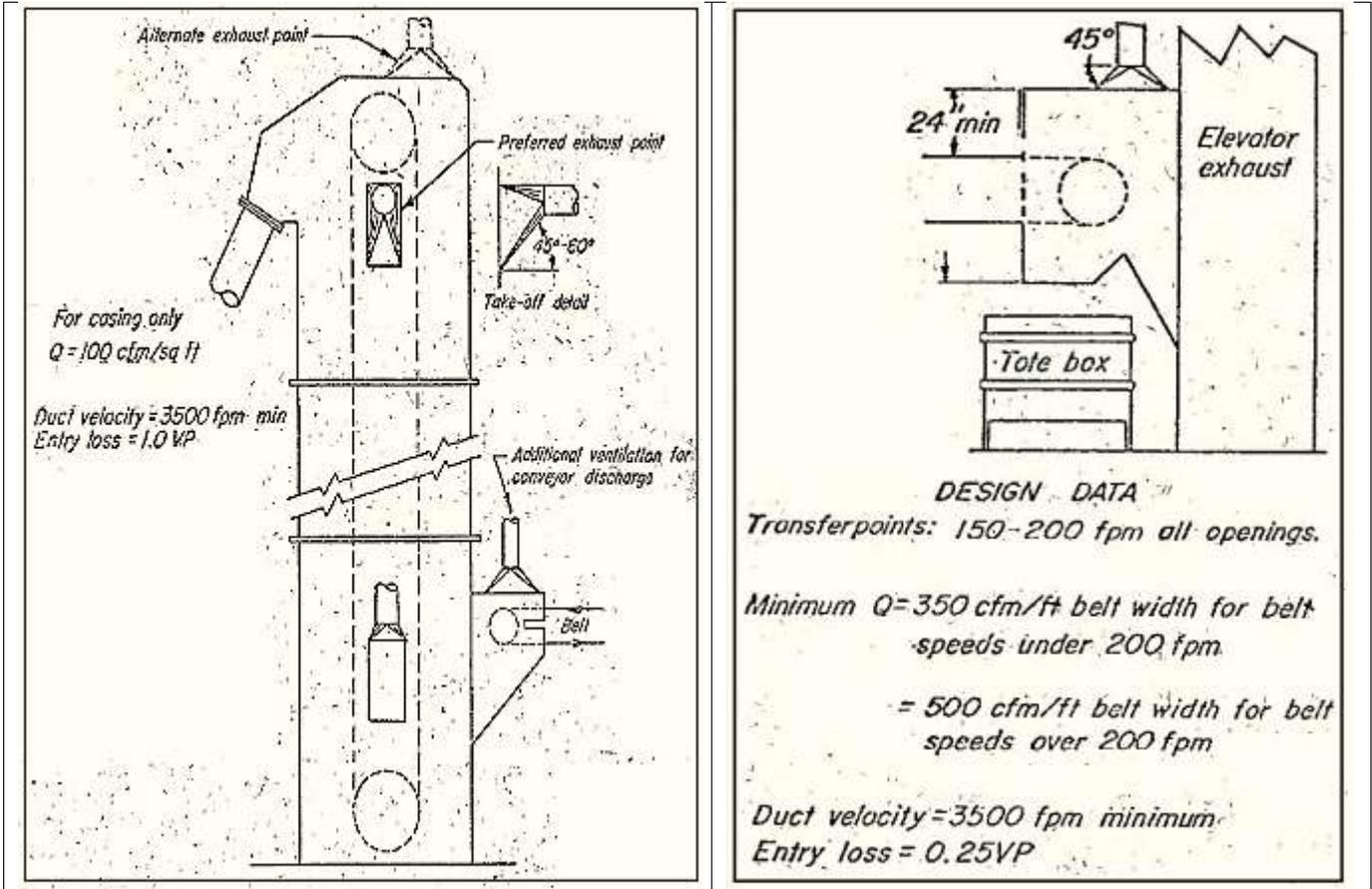
a) Chute to belt transfer and conveyor transfer for dusty materials as follows:



b) Exhaust system recommended for vibrating screens:



c) Exhaust system recommended for bucket elevators:



Note: From American Conference of Governmental Industrial Hygienists (ACGIH®) *Industrial Ventilation: A Manual of Recommended Practice*, 23rd Edition.

1) Ducts and Fittings Losses:

Duct and fitting losses for some common shapes are given below as a fraction of the VP. These are often converted to equivalent lengths of straight run duct for summing losses:

Fitting Style	Loss as Fraction of VP
Standard Long Radius 45° Elbow (1.5:1)	0.20.VP
Standard Long Radius 90° Elbow (1.5:1)	0.39.VP
Branch Entry Loss @ 45° Entry, as Fraction VP in Branch	0.28.VP
Branch Entry Loss @ 90° Entry, as Fraction VP in Branch	1.00.VP
Reducer @ 10° Taper	0.06.VP
Reducer @ 20° Taper	0.10.VP

1.2) Chart Frictional Loss:

Use the **Chart Friction Loss (see chapter III)** to look up the pressure loss per 100 ft of duct.

$$P_{fr} = (k_{loss} / 100 \text{ ft}) \times (L) =$$

2) Ventilation Systems Constant:

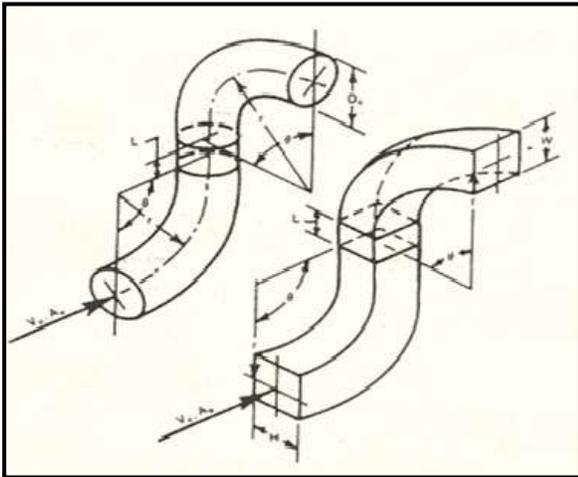
Every air system or sub-system has a system constant (K). This constant (K) can be calculated as long as you know one (CFM, Static Pressure) point. To calculate the system constant:

$$K_{\text{system}} = \text{S.P.} / (\text{CFM})^2 =$$

Once you have the system constant you can calculate the Static Pressure for any flow rate.

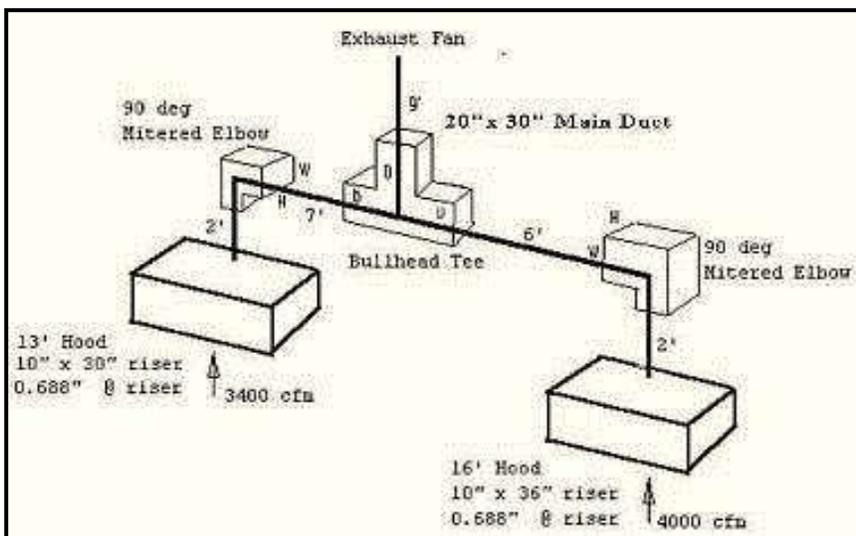
$$\text{S.P.} = (\text{CFM})^2 \times K_{\text{system}} =$$

Constant “K” for some exhausting ducts:



K													
L/D													
Deg°	0	1	2	3	4	6	8	10	12	14	20	25	40
60	2.0	1.90	1.50	1.35	1.30	1.20	1.25	1.50	1.63	1.73	1.85	1.95	2.0
90	2.0	1.80	1.60	1.55	1.55	1.65	1.80	1.90	1.93	1.98	2.0	2.0	2.0

Example – An Exhaust System:



The **first step** is to break the system into sections:

Section 1 - runs from the 16' Hood to the Bullhead Tee;

Section 2 - runs from the 13' Hood to the Bullhead Tee;

Section 3 - runs from the Bullhead Tee to the Exhaust Fan.

Now calculate Velocity, Velocity Pressure and Frictional Loss for each section.

Section 1:

Air Flow Rate, $Q = 4000 \text{ cfm}$

Cross-Sectional Area, $A = 10 \times 36/144 = 2.5 \text{ ft}^2$

Center Line Distance, $L = 2' + 6' = 8'$

Velocity $V = 4000/2.5 = 1600 \text{ ft/min}$

Velocity Pressure = $VP_1 = (V/4005)^2 = (1600/4005)^2 = 0.16''$

Frictional Loss = $P_{fr} = (k_{loss} / 100 \text{ ft}) \times (L) = (0.15 / 100 \text{ ft}) \times 8' = 0.012''$

Section 2:

Air Flow Rate, $Q = 3400 \text{ cfm}$

Cross-Sectional Area, $A = 10 \times 30/144 = 2.1 \text{ ft}^2$

Center Line Distance, $L = 2' + 7' = 9'$

Velocity $V = 3400/2.1 = 1619 \text{ ft/min}$

Velocity Pressure = $VP_2 = (V/4005)^2 = (1619/4005)^2 = 0.16''$

Frictional Loss = $P_{fr} = (k_{loss} / 100 \text{ ft}) \times (L) = (0.20''/100 \text{ ft}) \times (9') = 0.018''$

Section 3:

Air Flow Rate, $Q = 3400 \text{ cfm} + 4000 \text{ cfm} = 7000 \text{ cfm}$

Cross-Sectional Area, $A = 20 \times 30/144 = 4.17 \text{ ft}^2$

Center Line Distance, $L = 9'$

Velocity $V = 7000/4.17 = 1678 \text{ ft/min}$

Velocity Pressure = $VP_3 = (V/4005)^2 = (1785/4005)^2 = 0.20''$

Frictional Loss = $P_{fr} = (k_{loss} / 100 \text{ ft}) \times (L) = (0.15''/100 \text{ ft}) \times (9') = 0.014''$

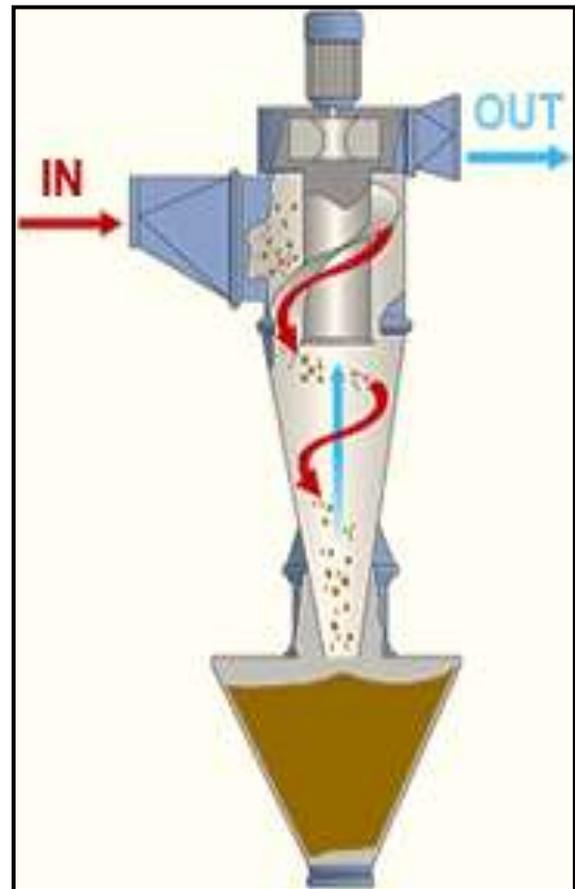
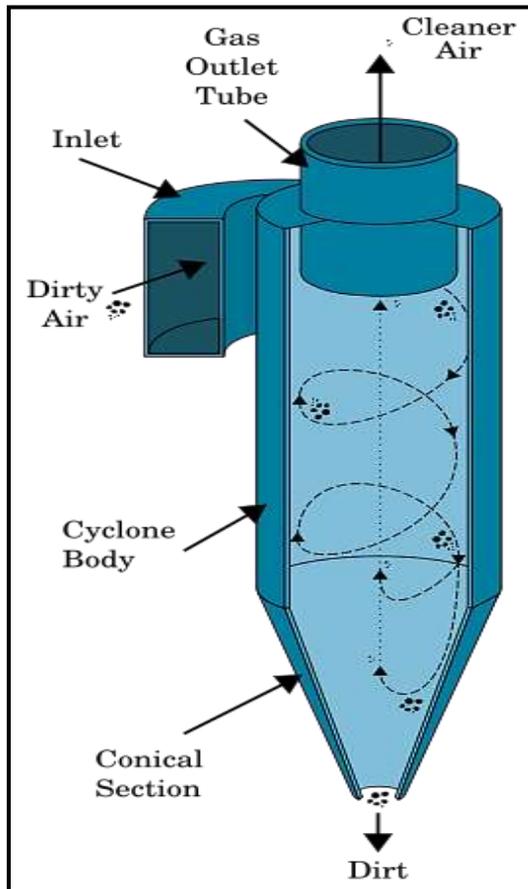
VIII. CYCLONES – CYCLONIC SEPARATION:

Cyclonic separation is a method of removing particulates from air, gas or water stream, using cylindrical or conical vessels called cyclones. **Fans or blowers** commonly suck the air in line after the cyclones. Air flows in a spiral pattern, beginning at the wide top of the cyclone and ending at the narrow bottom end before exiting the cyclone in a straight stream through the center of the cyclone and out the top.

Obs.: Larger or denser particles entering the rotating stream have much inertia to follow the tight curve of the top outlet nozzle and when strike the outside wall, fall to the bottom of the cyclone to be removed.

In a conical system, as the rotating flow moves towards the narrow end of the cyclone the rotational radius of the stream is reduced, separating smaller and smaller particles. The cyclone geometry, together with flow rate, defines the cut point of the cyclone. This is the size of particle that will be removed from the stream with 50% efficiency. Large scale cyclones are used in sawmills to remove sawdust, in oil refineries to separate oils and gases and in the cement industry as components of kiln pre-heaters. Analogous devices for separating particles or solids from liquids are called **hydrocyclones**, used to separate solid waste from water in wastewater and sewage treatment.

1) Basic Cyclone Types:



Small diameter cyclones:

Dust loads = 0.1 to 6 grains/ft³;

High pressure drops = 6 to 10 inches WG.

Large diameter cyclones:

Dust loads = 50 to 100 grains/ft³;

Low pressure drops = 1.5 to 3 inches WG.

Average flow rates = 1000 to 25000 ft³/min - standard (scfm).

IX. REFERENCES:

1. (ACGIH®) *Industrial Ventilation: A Manual of Recommended Practice*, 23rd Edition.
2. ANSI/ASHRAE Standards.
3. Engineering Cookbook, Loren Cook Company, 2nd Edition, 1999.
4. Pneumatic Grain Conveyors, 1985.
5. Wikipedia - The Free Encyclopedia.
6. The Engineering ToolBox.
7. Occupational Safety & Health Administration (OSHA).