Building Design and Engineerin
Approaches to Airborne Infection

Basic Concepts of Ventilation Design

Jack Price

General Principles of Ventilation

Introduction

Need for ventilation:

- **Comfort**
- **Contamination Control**

both maintain healthy work environment

General Principles of Ventilation

- Office buildings ----- In-door air quality
- Occupational exposure ---- OSHA
- Environmental releases ---- EPA

General Principles of Ventilation

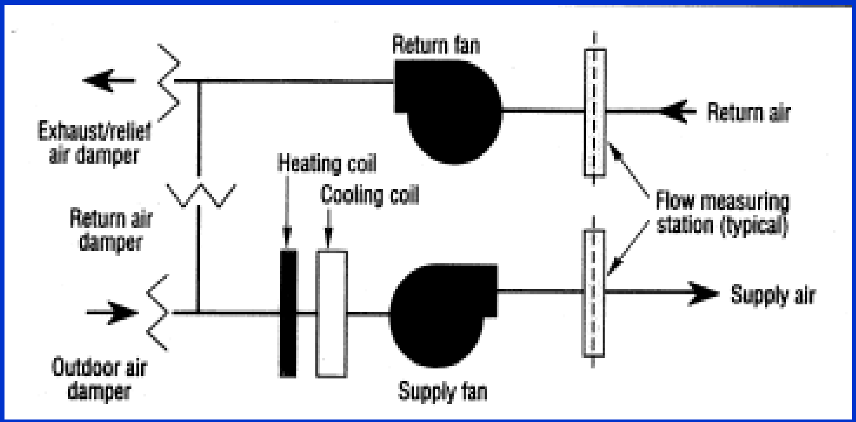
- Regulatory Agencies (compliance concerns)
  - Federal
  - State
  - Local
- Good Practice
  - Standard of care (industry standards ANSI, ASME, etc.)
  - Work productivity
  - Process control

Types of Systems

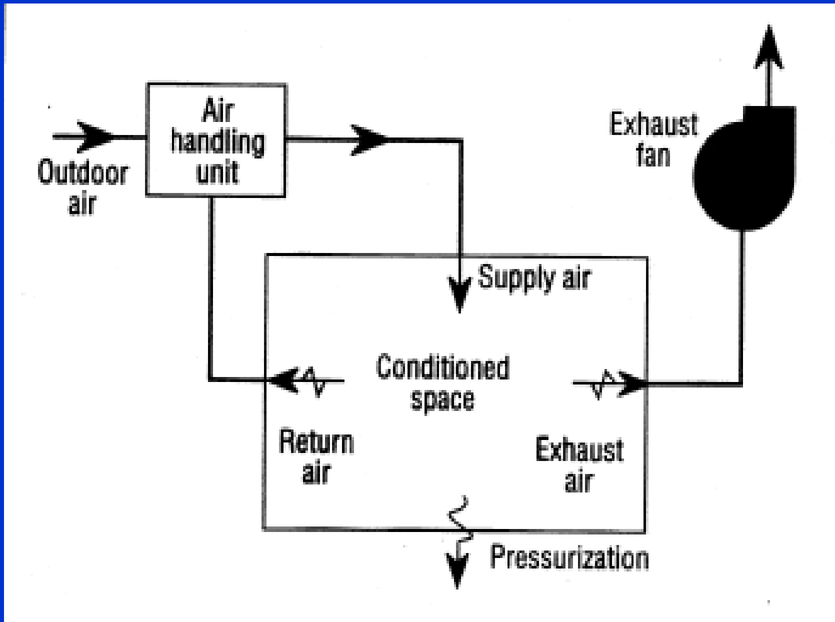
- **Supply**
  - Temperature & Humidity
  - Replacement (make-up air)
  - Return (recirculated air)

- **Exhaust**
  - General (dilution)
  - Local Control (hoods)

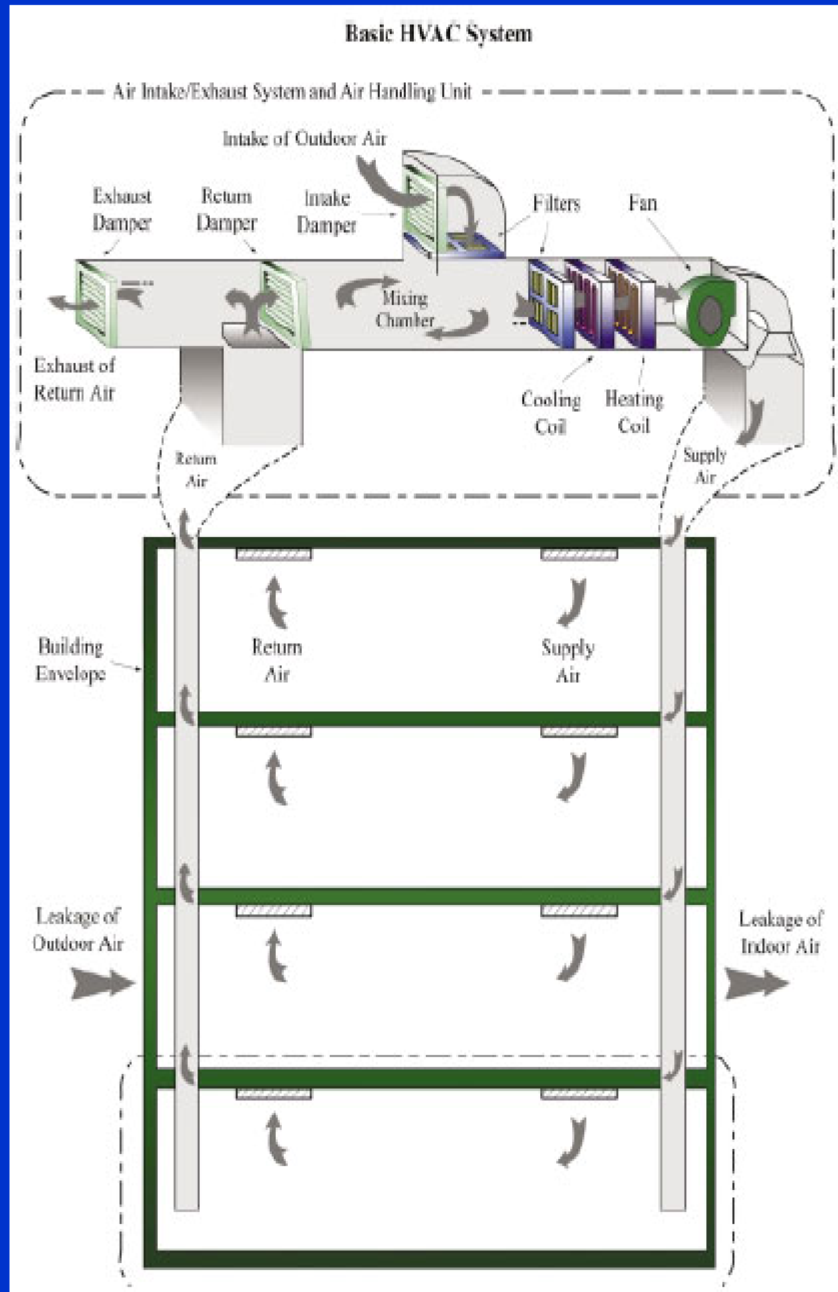
HVAC Systems



Air Handling System with Economizer



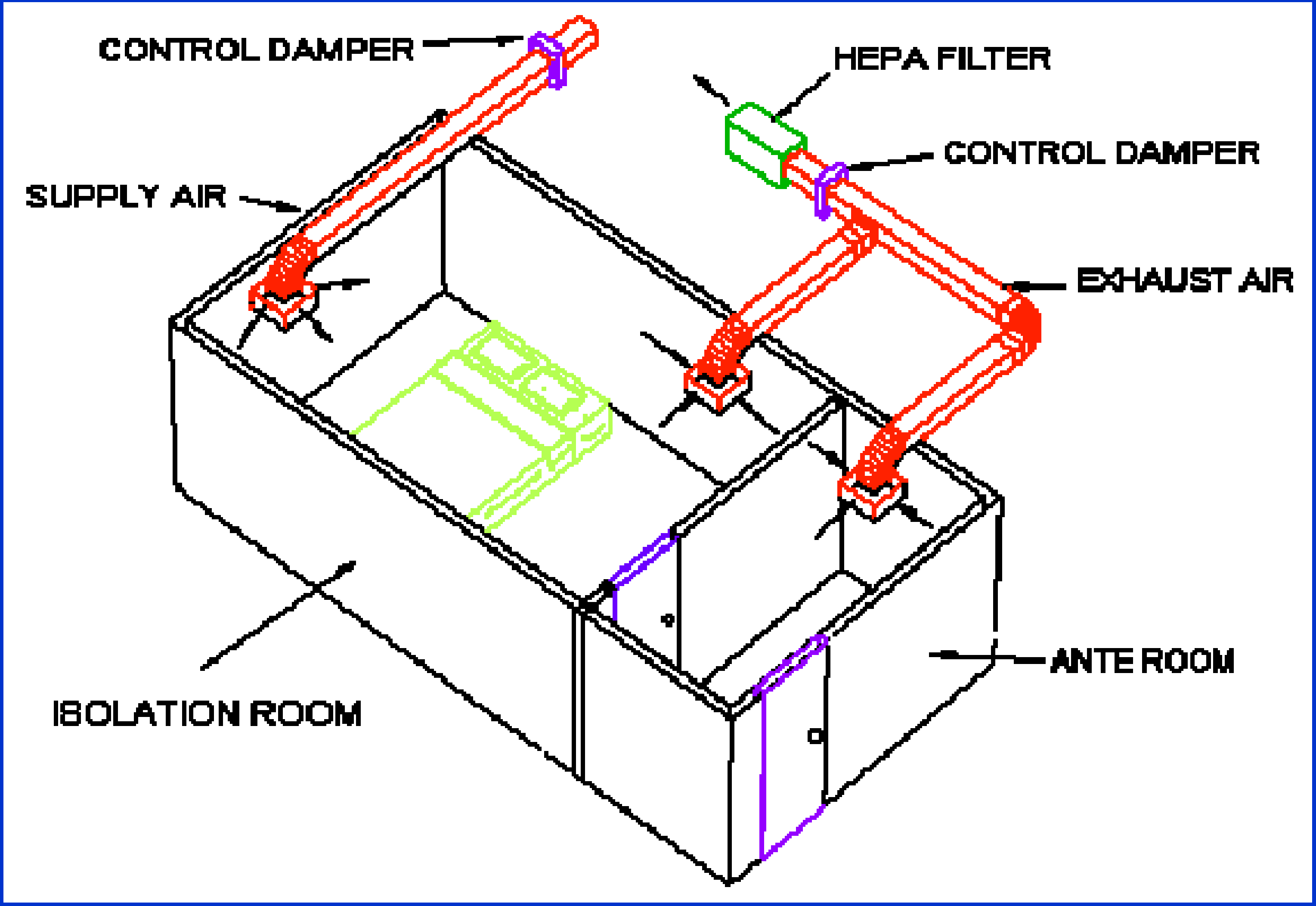
Air Balance in a Conditioned Space


Basic HVAC System
Air Intake/Exhaust System and Air Handling Unit
Intake of Outdoor Air
Exhaust Damper
Return Damper
Intake Damper
Filters
Fan
Mixing Chamber
Exhaust of Return Air
Cooling Coil
Heating Coil
Return Air
Supply Air
Building Envelope
Return Air
Supply Air
Leakage of Outdoor Air
Leakage of Indoor Air

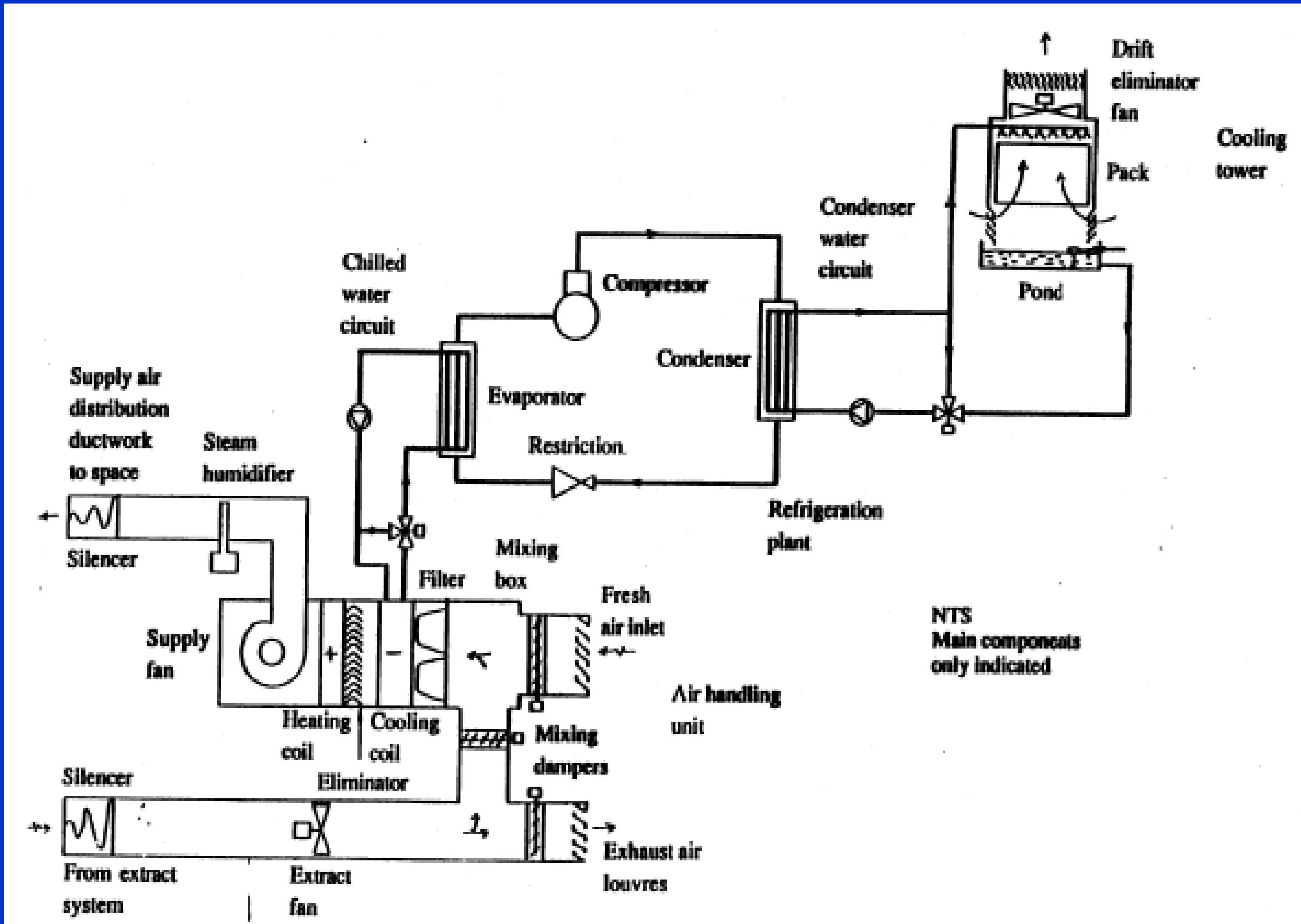
Design Concerns

- Temperature
- Pressure
- Air Contaminants
- Work Practices
- Product Protection
- Worker Protection
- Building Codes

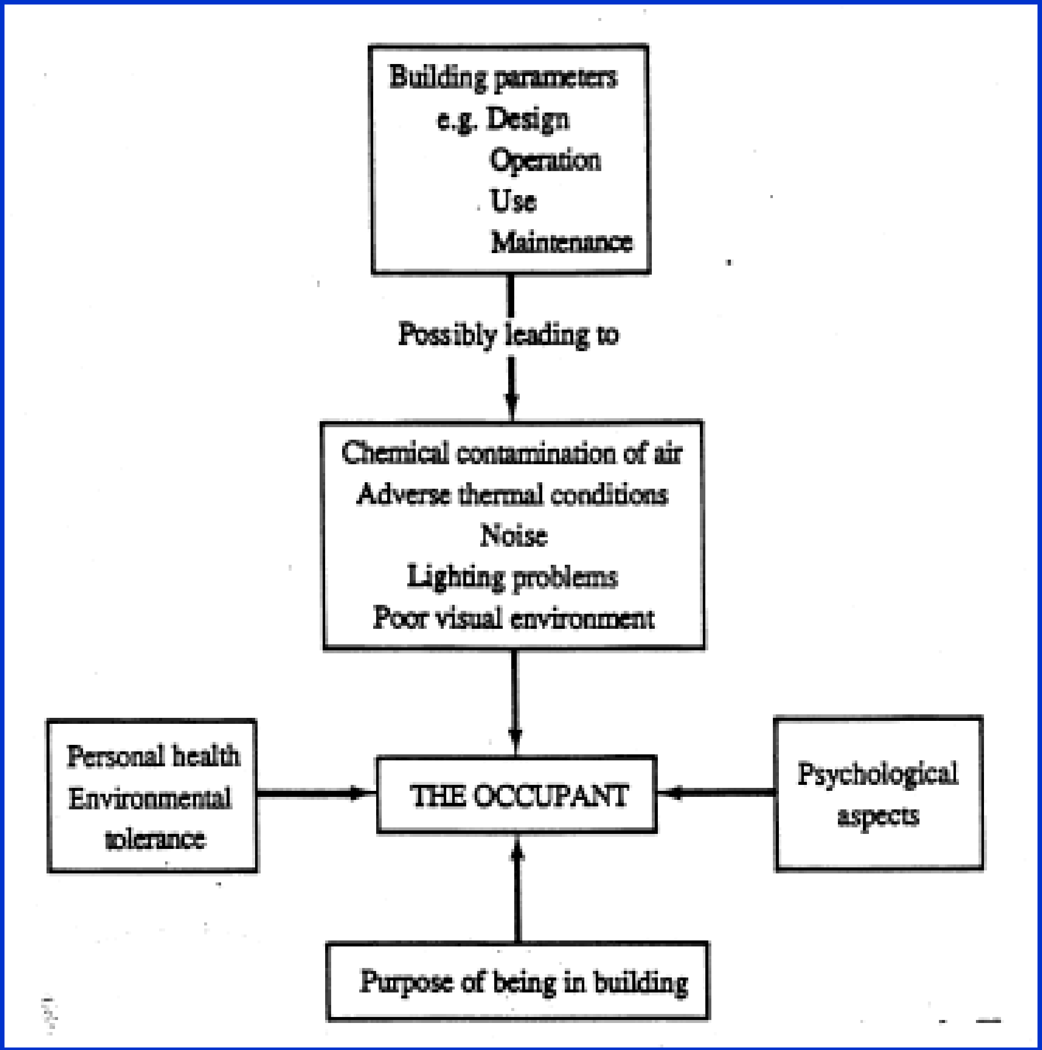
- Equipment Selection
- Energy Conservation
- Maintenance
- Security
- Expansion



Patient Isolation Room with HEPA Exhaust Filtration



Air Conditioning System Water and Refrigeration Circuits

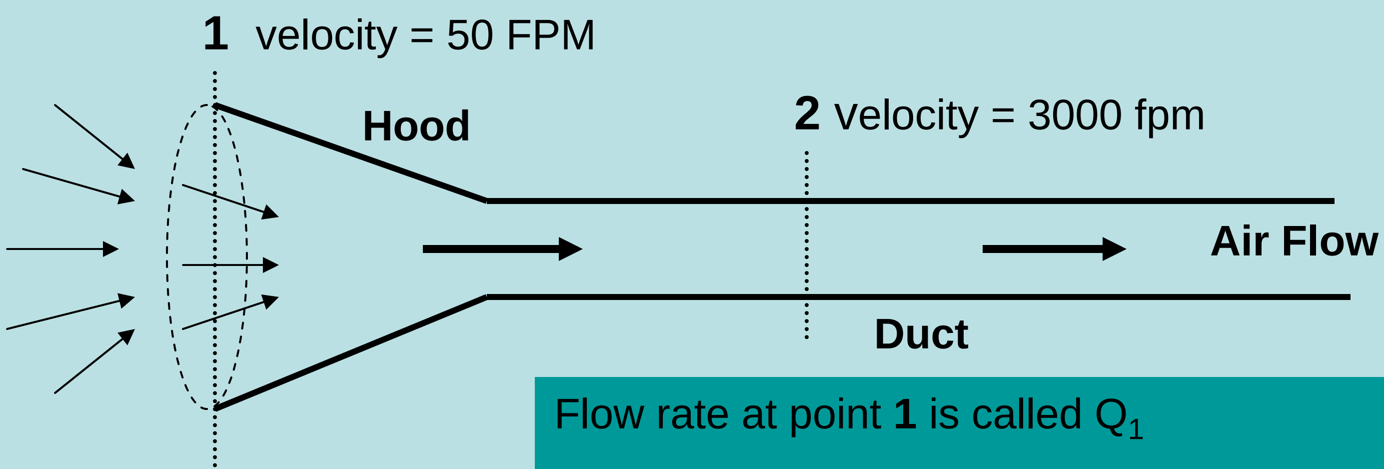


Factors in the Perception of Air Quality

Conversion Factors

Quantity	To Convert	Into	Multiply By:
Volumetric Flow	cubic feet/minute (ft^3/min)	cubic meters/second (m^3/sec)	4.719×10^{-4}
Velocity	feet/minute (fpm)	meters/second (m/s)	0.00508
Pressure	inches water (in w.g.)	Pascals (Pa)	249.1

Conservation of Mass



Flow rate at point **1** is called Q_1

and is equal to

flow rate at point **2** which is called Q_2

$$Q = V \cdot A$$

Where

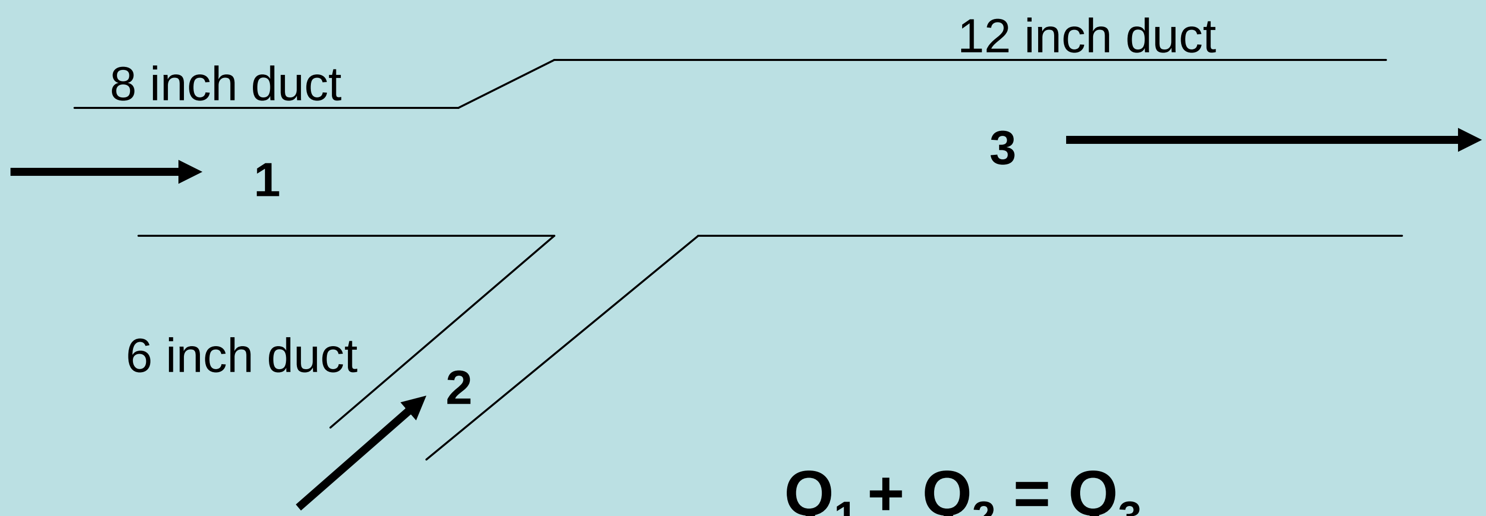
Q = Volumetric Flow Rate, ft^3/min

V = Air Velocity, ft/min or fpm

A = Cross Sectional Area, ft^2 or SF

Conservation of Mass

$$Q = V \cdot A$$



$$Q_1 + Q_2 = Q_3$$

$$V_1A_1 + V_2A_2 = V_3A_3$$

AIR FLOW

- At standard temperature and pressure (STP):

  * 1 atmosphere & 70^{o} F *

  The density of air is 0.075 lb_m/ft^3

- Air will flow from a higher pressure region to a lower pressure region

- Three Different Types of Pressure Measurements

  * Static * Velocity * Total *

Types of Pressure Measurements

- Static Pressure (S_P)

  potential energy

  can be + or –

  bursting or collapsing

  measured perpendicular to flow

- Velocity Pressure (V_P)

  kinetic energy

  Exerted in direction of flow

  accelerates from 0 to some velocity

  always +

- Total Pressure (T_P)

  combined static & velocity components

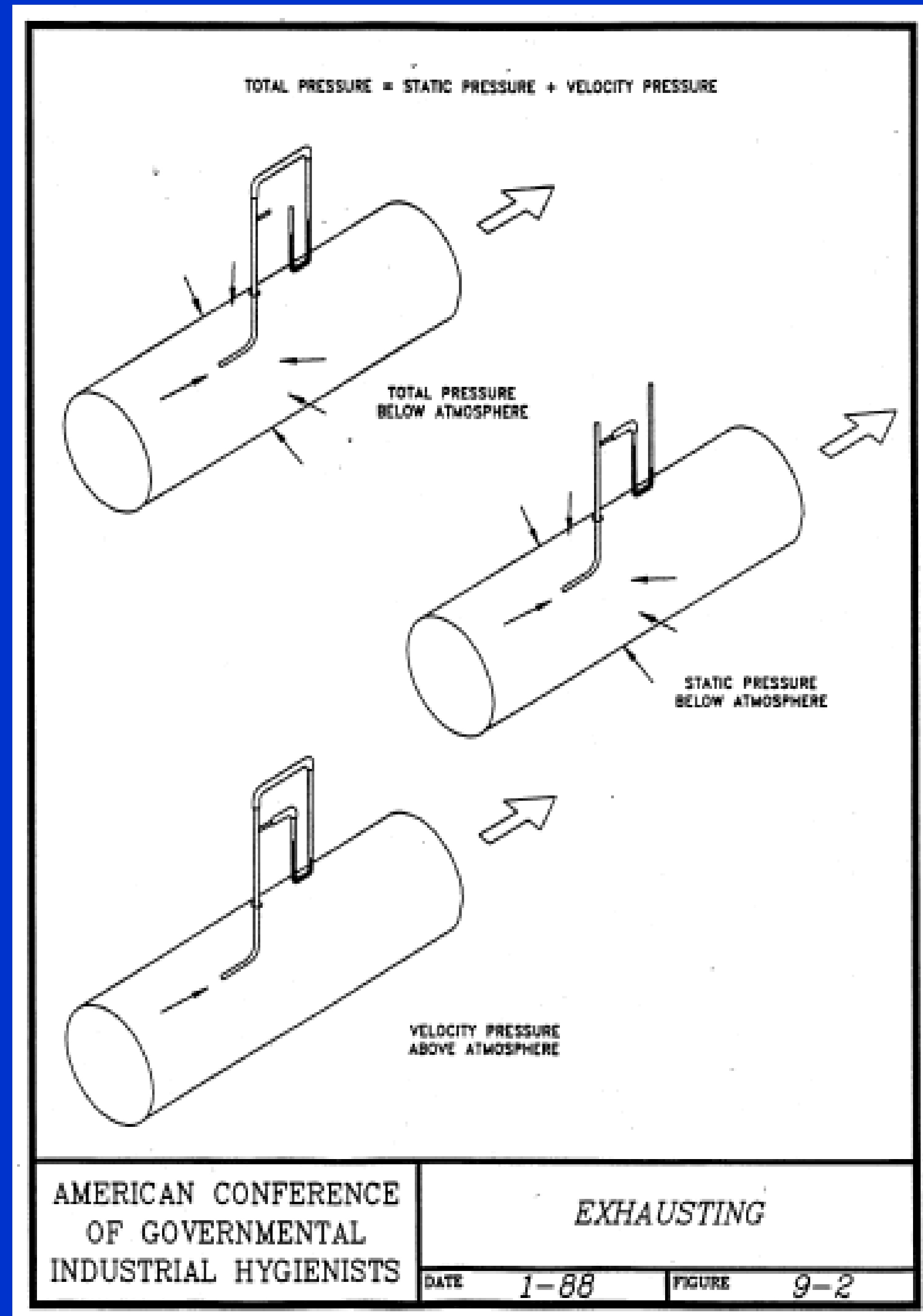
  can be + or -

  measure of energy content of air stream

  Always decreasing as flow travels downstream thru a system only rising when going across a fan

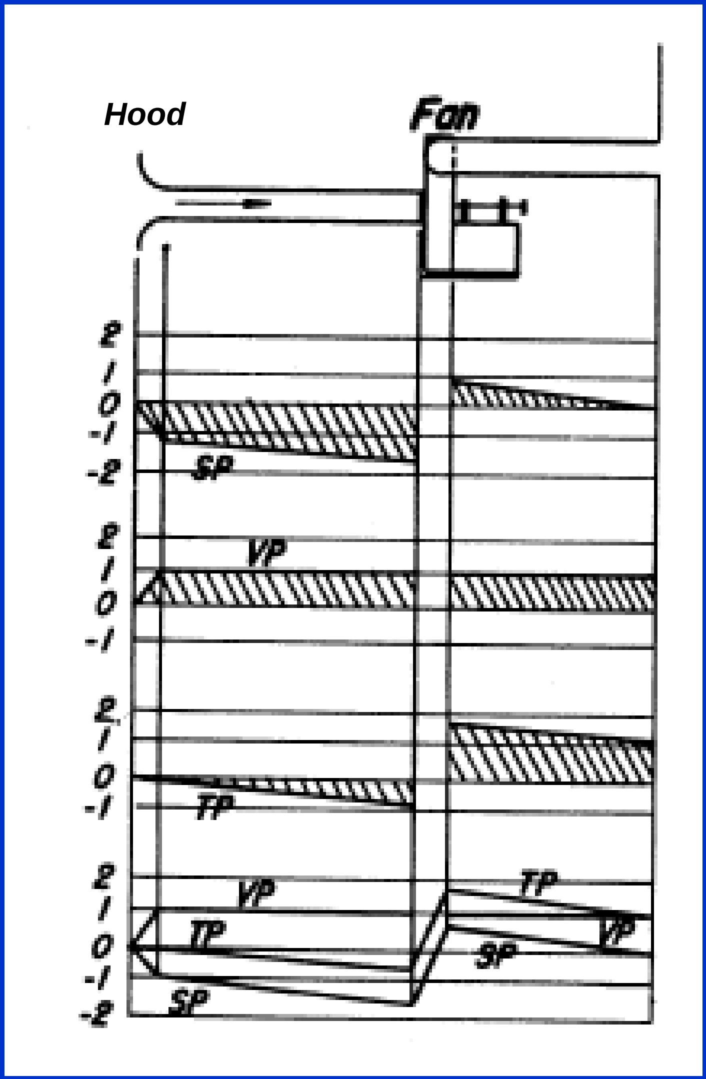
TP = SP + VP

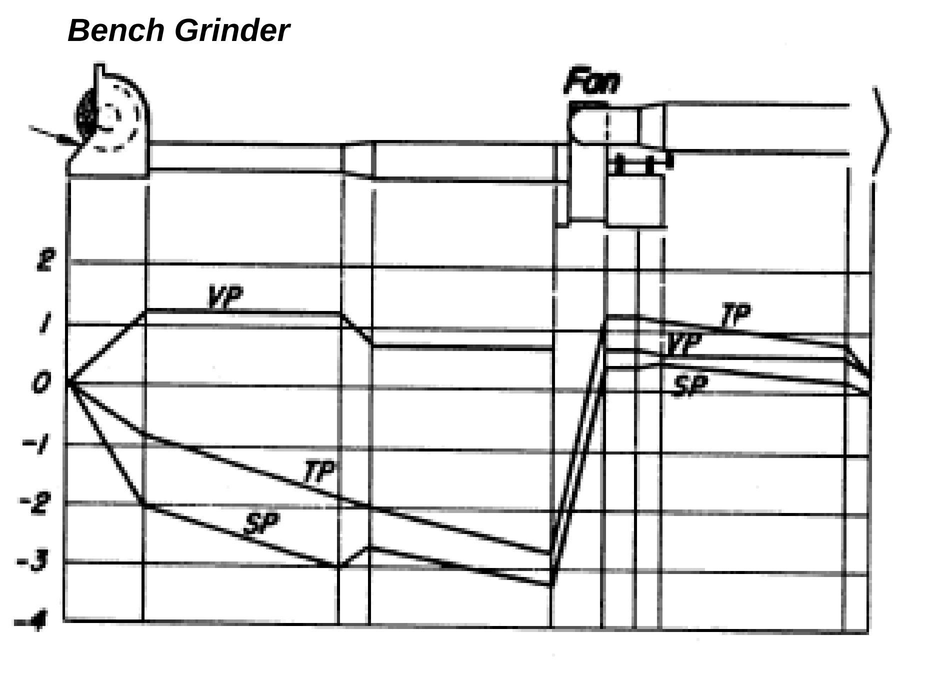
TP	— **+**
SP	— **+**
VP	**+**



Conservation of Energy

- $TP = SP + VP$ or $T_P = S_P + V_P$
- Energy losses:
  - Acceleration of air
  - Hood entry
  - Duct losses: friction (function of system materials & design)
  - Fitting losses: contractions & expansions
- $T_{P1} = T_{P2} + h_L$ now substitute $T_P = S_P + V_P$
- $S_{P1} + V_{P1} = S_{P2} + V_{P2} + h_L$





Pressure Graphs for TP, SP, and VP

Velocity Pressure & Velocity

- $V = 1096\ (V_P/p)^{0.5}$

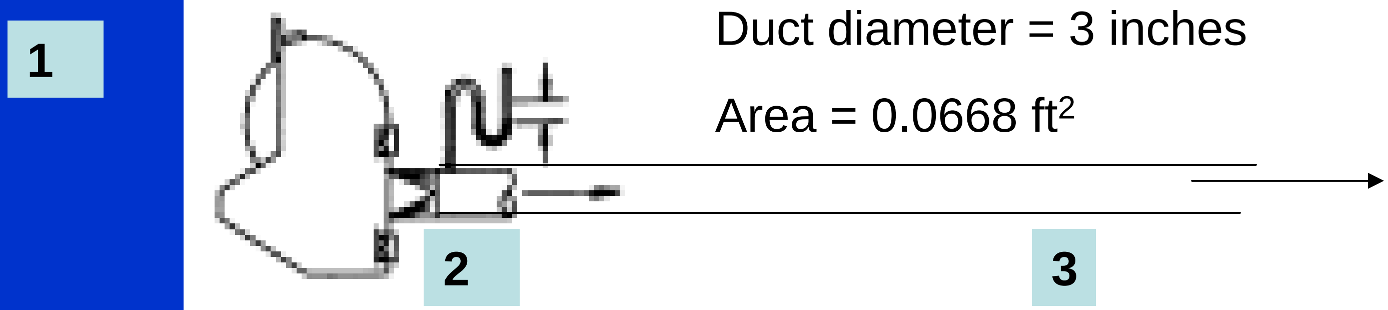
where p = air density

@ STP $p = 0.075\ lb_m/ft^3$

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

- **Velocity pressure is a function of the velocity and fluid density.**
- **Velocity pressure will only be exerted in the direction of air flow and is always positive.**

Bench Grinder Exhaust Ventilation



Duct diameter = 3 inches

Area = 0.0668 ft^2

- $Q_1 = Q_2$
- If Q desired is 300 cfm
- Then Q = V A

  V = Q A

  V = (300) / (0.0068)

  V = 4490 fpm

- If there are no losses from the grinder hood entry then:

$$SP_1 + VP_1 = SP_2 + VP_2$$

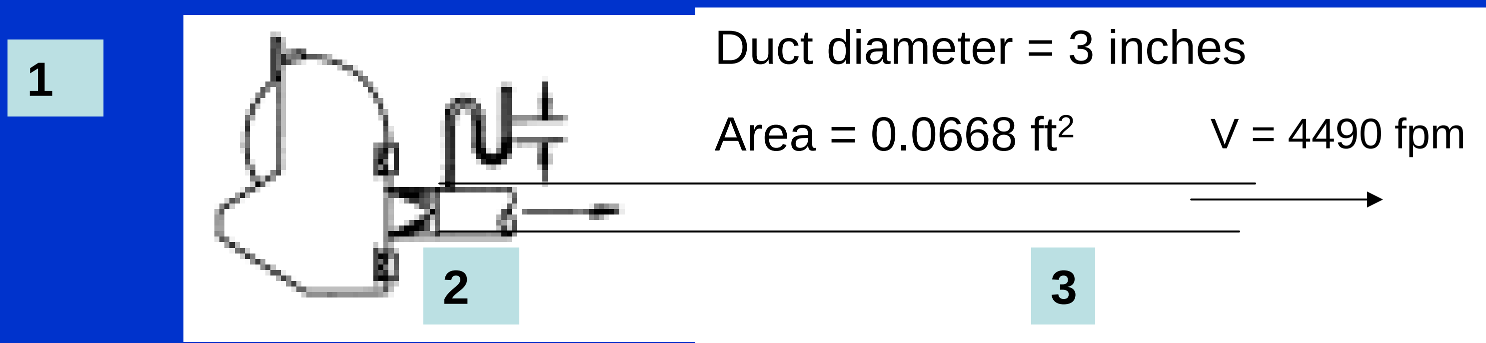
but: $SP_1 = 0$ and $VP_1 \longrightarrow 0$

we then have:

$$0 = SP_2 + VP_2$$

or $$-VP_2 = SP_2$$

Bench Grinder Exhaust Ventilation



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but: $SP_1 = 0$ and $VP_1 \longrightarrow 0$

we then have:

$$0 = SP_2 + VP_2$$

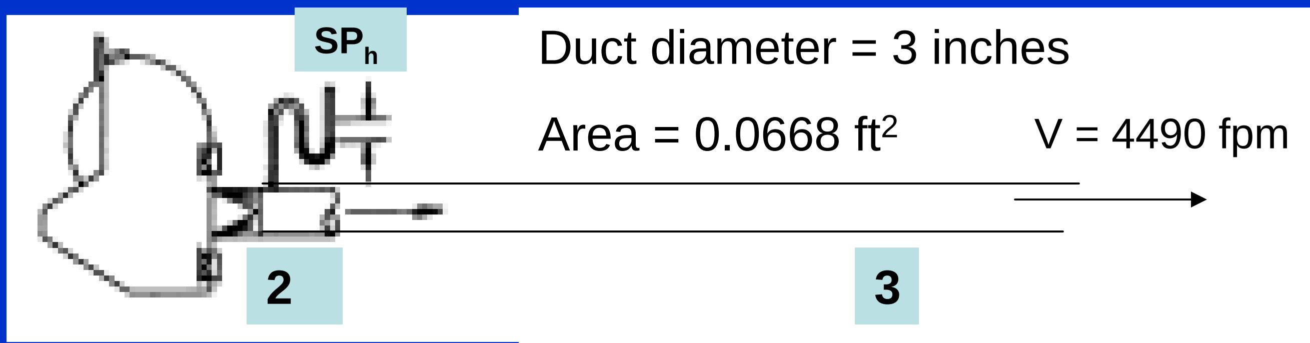
or $\quad SP_2 = (-VP_2)$

- **from $V = 4005\,(VP)^{0.5}$**
- **$VP_2 = (4490/4005)^2$**
- **$VP_2 = 1.26$ in w.g.**
- **then: $SP_2 = (-VP_2)$**

$SP_2 = -1.26$ in w.g.

Bench Grinder Exhaust Ventilation

1



- However there are losses thru the grinder hood entry

  $\mathbf{SP_2 = -(VP_2 + h_e)}$ where h_e is the energy loss of the hood entry

- Static pressure **(SP)** must decrease due to acceleration of air up to the duct velocity

- $\mathbf{F_h}$ is defined as the energy loss factor (for that hood design)

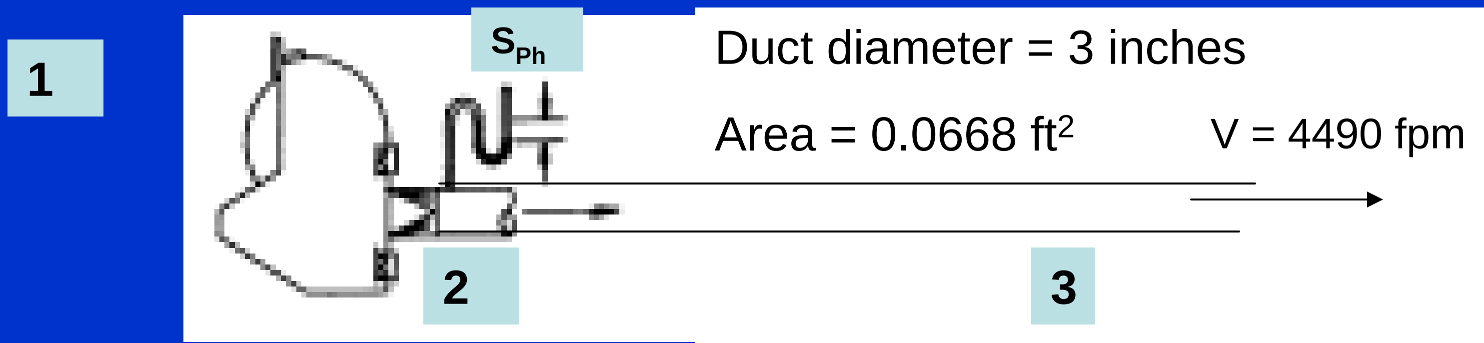
- Energy losses will be measured as a function of the velocity pressure in the system

  $\mathbf{h_e = (F_h)(VP)}$

- Now we define the static pressure at the hood as $\mathbf{SP_h}$

- $\mathbf{SP_h}$ is also called the hood static suction and is the absolute value of $\mathbf{SP_2}$

Bench Grinder Exhaust Ventilation



- Now add the hood entry loss:

$$SP_h = VP_2 + h_e = VP_{2} + (F_h)\ (VP_2)$$

Assume that the hood energy loss factor for this hood is 0.40

- **$SP_h = 1.26 + (0.40)\ (1.26) = 1.76$ in w.g.**

Figure 1. Relationship Between Hood Static Pressure and Flow Rate Entering Hood

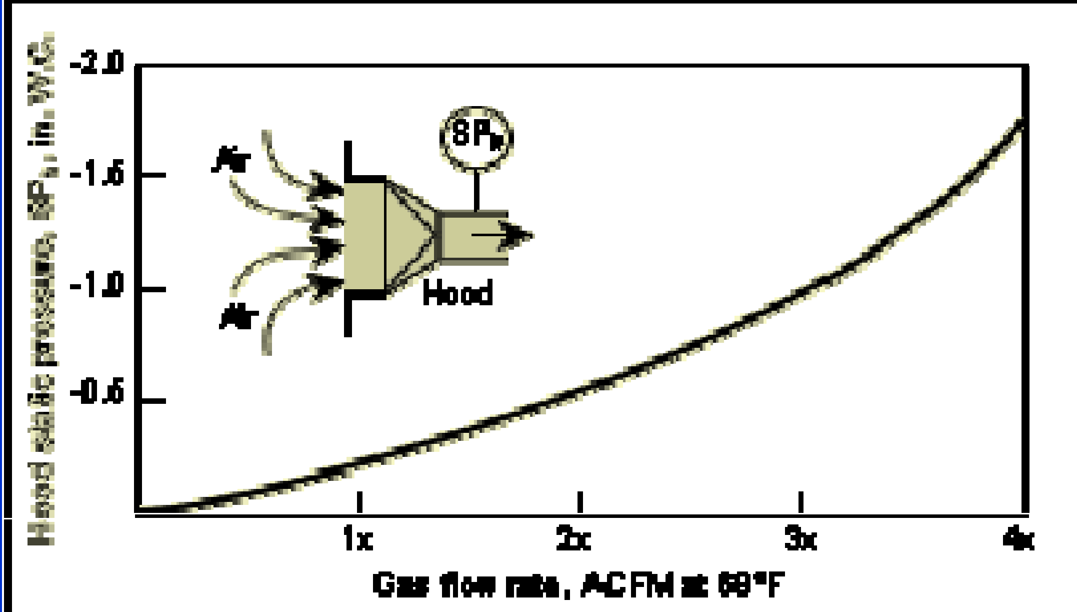


Figure 2. Air Flow Convergence in a Duct

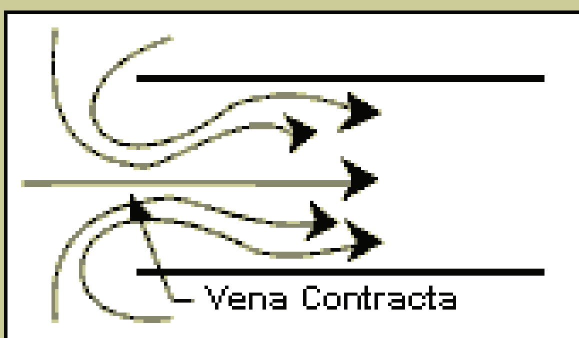


Figure 3. Hood Entry Loss Coefficients (F_d) for Various Duct Designs

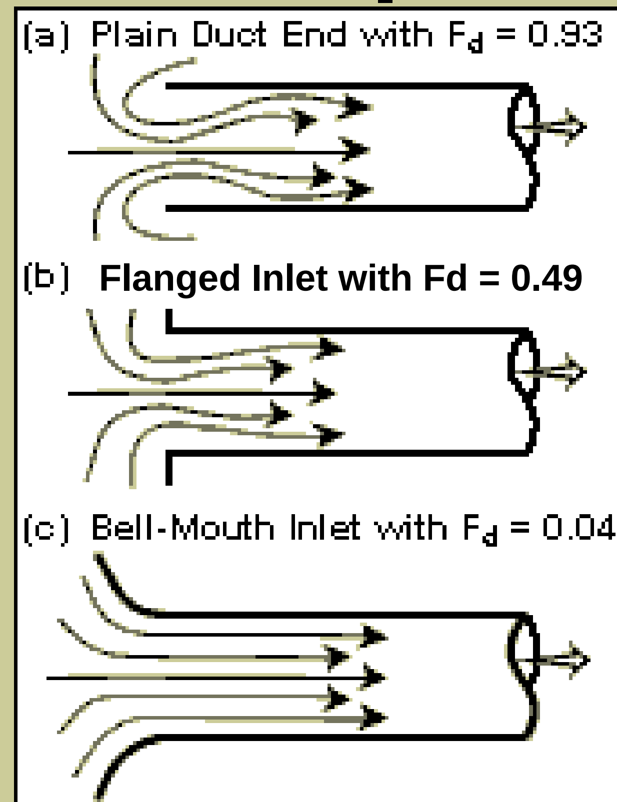
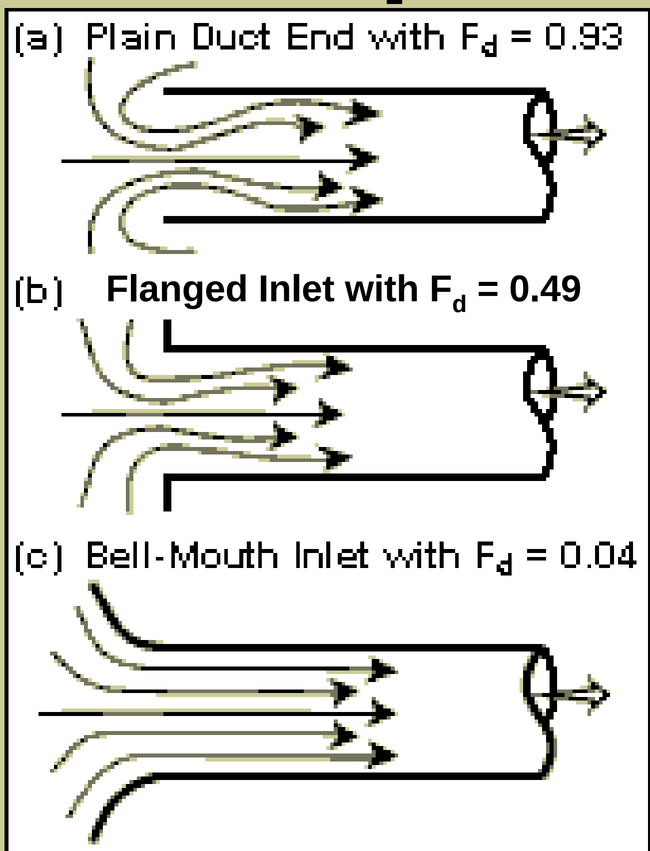
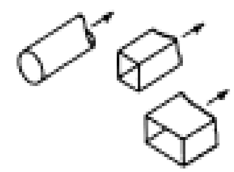
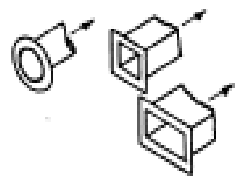
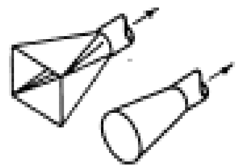
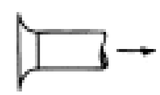
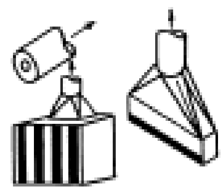
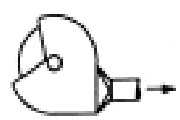


Figure 3. Hood Entry Loss Coefficients (F_d) for Various Duct Designs



HOOD TYPE	DESCRIPTION	COEFFICIENT OF ENTRY, Ce	ENTRY LOSS
	PLAIN OPENING	0.72	0.93 VP
	FLANGED OPENING	0.82	0.49 VP
	TAPER or CONE HOOD	Varies with angle of taper or cone. See Fig. 6-10	
	BELL MOUTH INLET	0.98	0.04 VP
	ORIFICE	See Fig. 6-10	
	TYPICAL GRINDING HOOD	STRAIGHT TAKE-OFF	
		0.78	0.65 VP
		TAPERED TAKE-OFF	
		0.85	0.40 VP

Hood Entry Coefficients

$$C_e = \frac{\textbf{Actual Flow}}{\textbf{Hypothetical Flow}\ \textit{no losses}}$$

$$C_e = \frac{(4005)\ (VP)^{0.5}\ (A)}{(4005)\ (SP_h)^{0.5}(A)} = \frac{(VP)^{0.5}}{(SP_h)^{0.5}}$$

$$\mathbf{C_e = (VP/SP_h)^{0.5}}$$

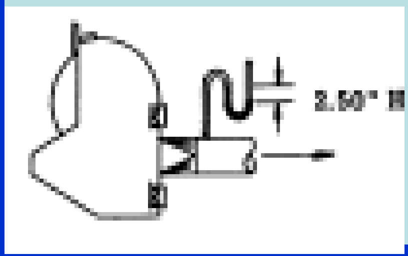
Hood Entry Coefficients

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

Typical values for C_e are known for some hoods.

For the bench grinder hood with a straight take-off :

$$C_e = 0.78$$



Example Problem

- What static pressure (SP_h) should be set at the bench grinder hood to maintain a duct velocity of 4000 fpm if the take-off duct size is 4 inch diameter ?
- What is the volumetric flow rate ?

Example Problem

- V = 4000 fpm   Q = VA = 4005(A)(VP)$^{0.5}$ Q = VA = 348 cfm
- A for 4 inch duct diameter = 0.087 ft^2
- C_e bench grinder hood = 0.78

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

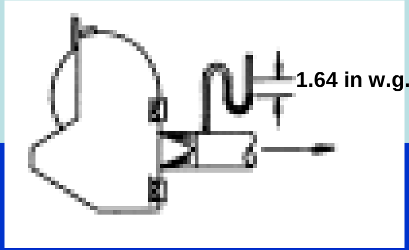
$$(VP/SP_h) = (0.78)^2$$

$$SP_h = VP/(0.78)^2 = (0.998)/(0.608) = 1.64 \text{ in w.g.}$$

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

(VP)$^{0.5}$ = (4000)/(4005)

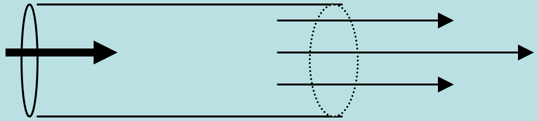
VP = 0.998 in w.g.

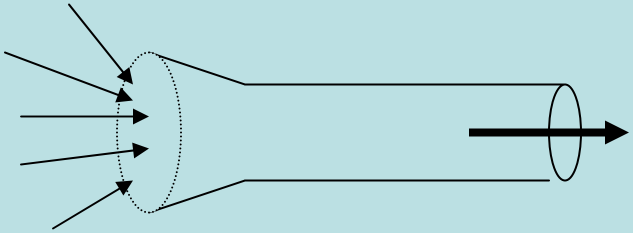


Air Flow Characteristics

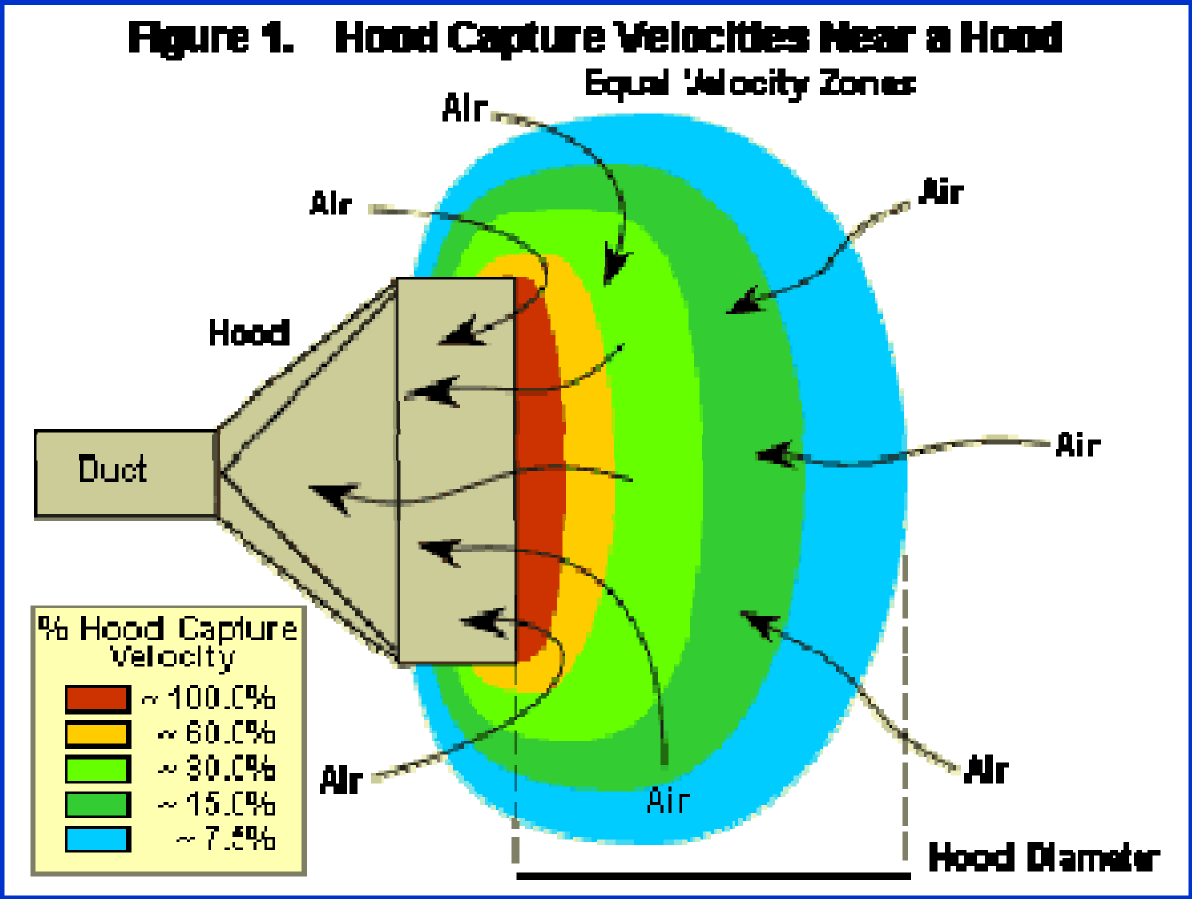
- See Industrial Ventilation Manual notes

Blowing vs. Exhausting





Air Flow Characteristics



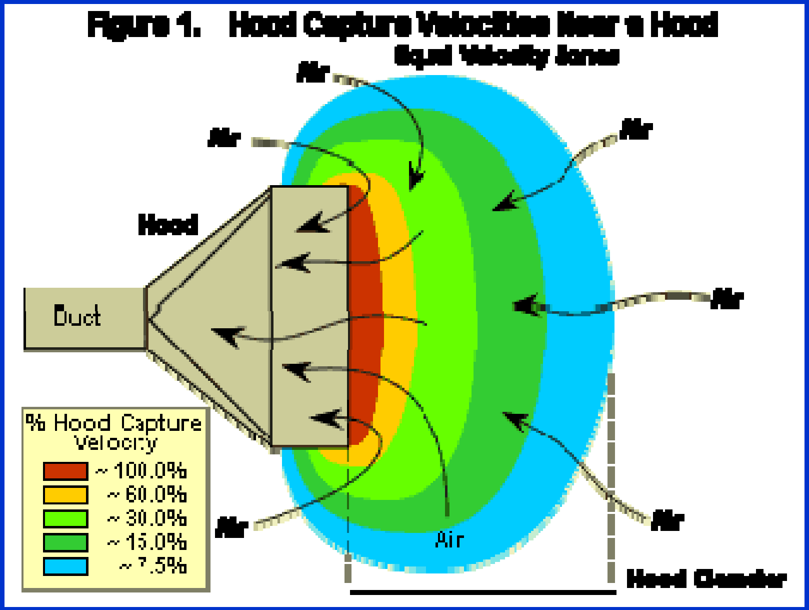
Exhaust Hoods

Capture Velocity

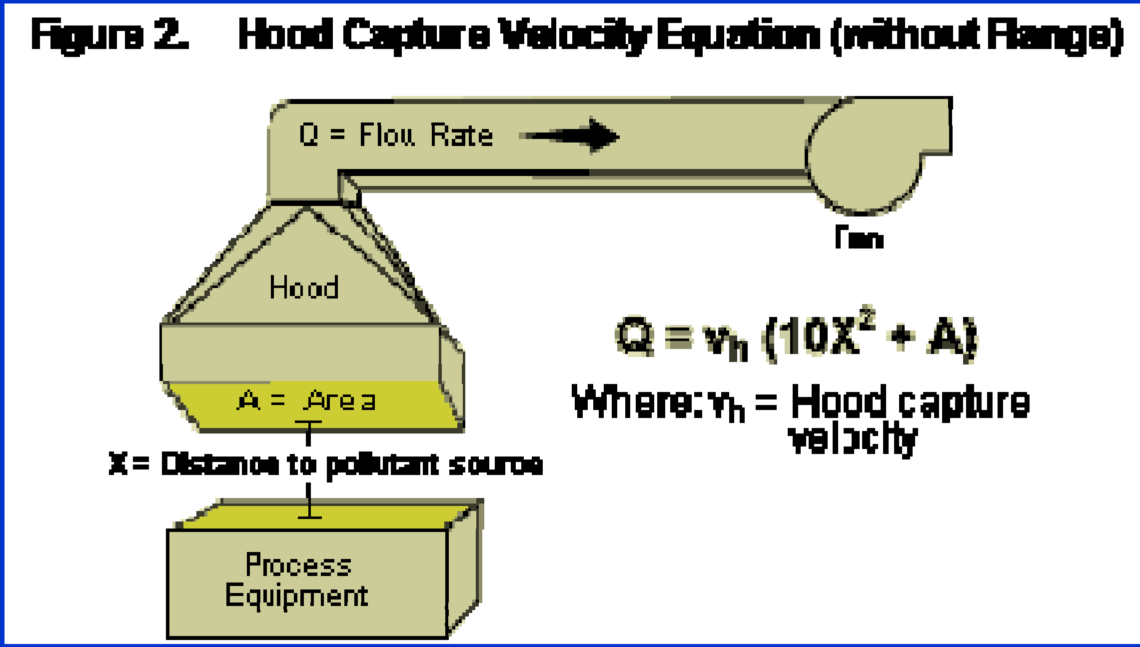
From Dalla Valle's empirical work

$$V_{(x)} = Q/\,(10\,x^2 + A)$$

Capture Velocity

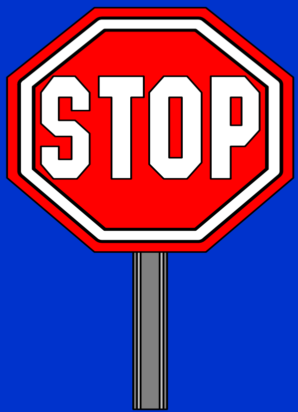


$$V_{(x)} = Q / (10\,X^2 + A)$$

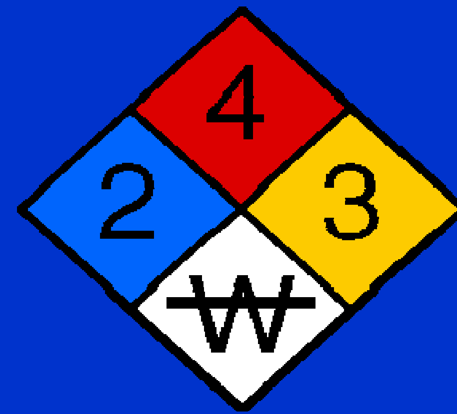


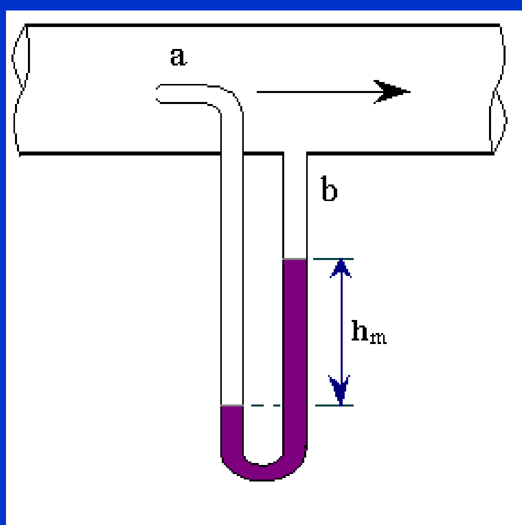
Capture velocity is only effective in the immediate vicinity of the hood

Room supply air (make-up air) discharge can influence effectiveness of hood capture



Questions ?





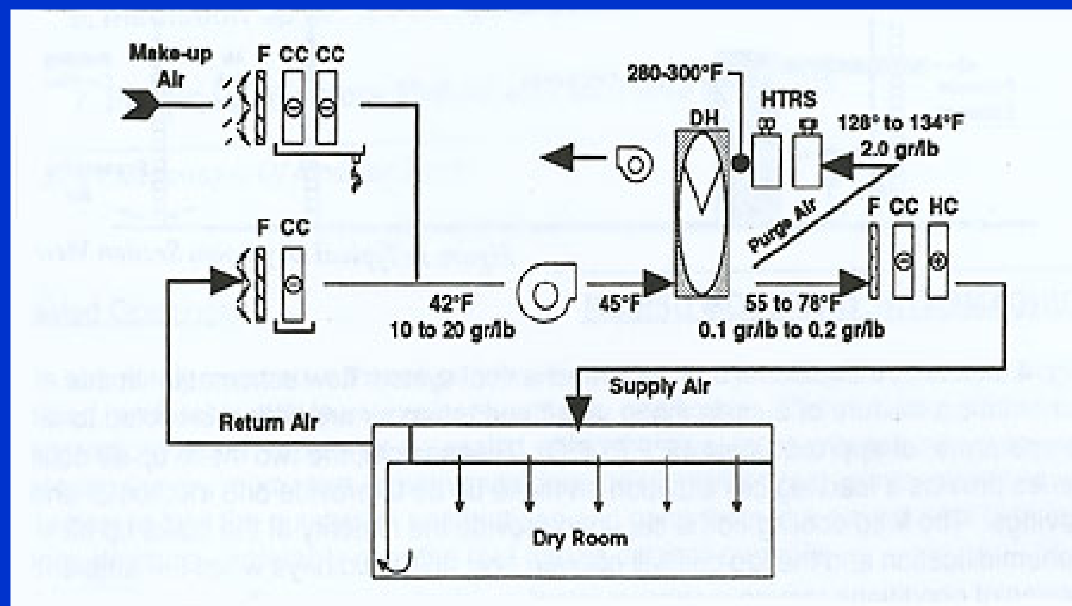
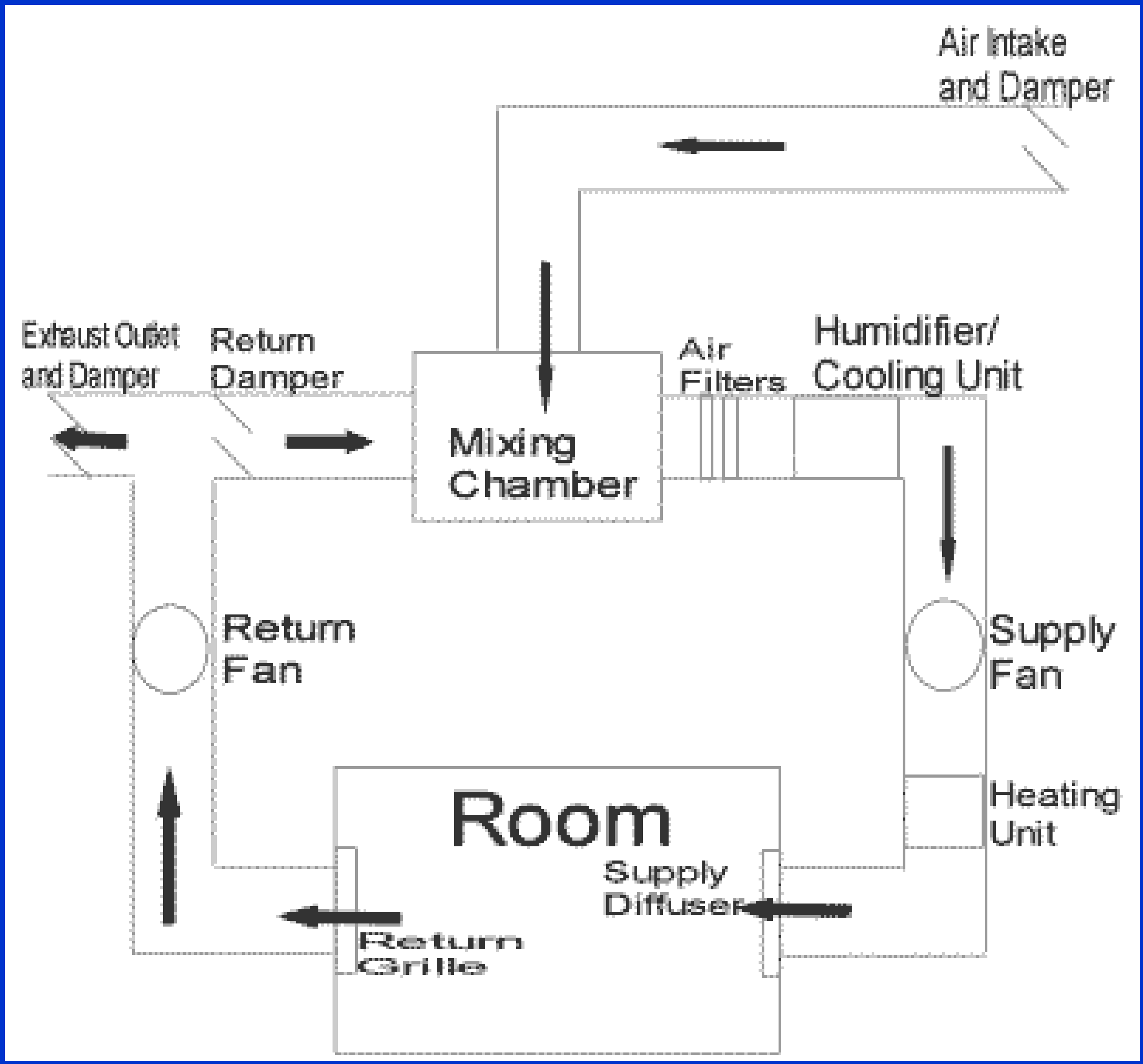


Figure 4- Typical Dry Room Mechanical System

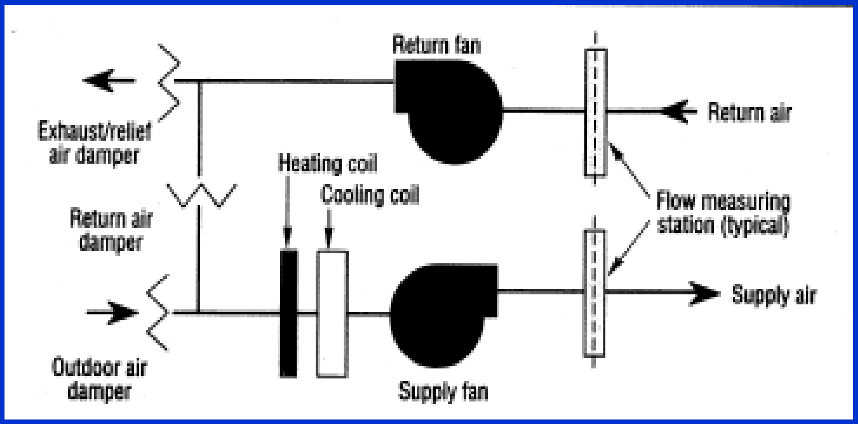
In-Place Filter Testing Workshop

Ventilation Systems: Operation and Testing

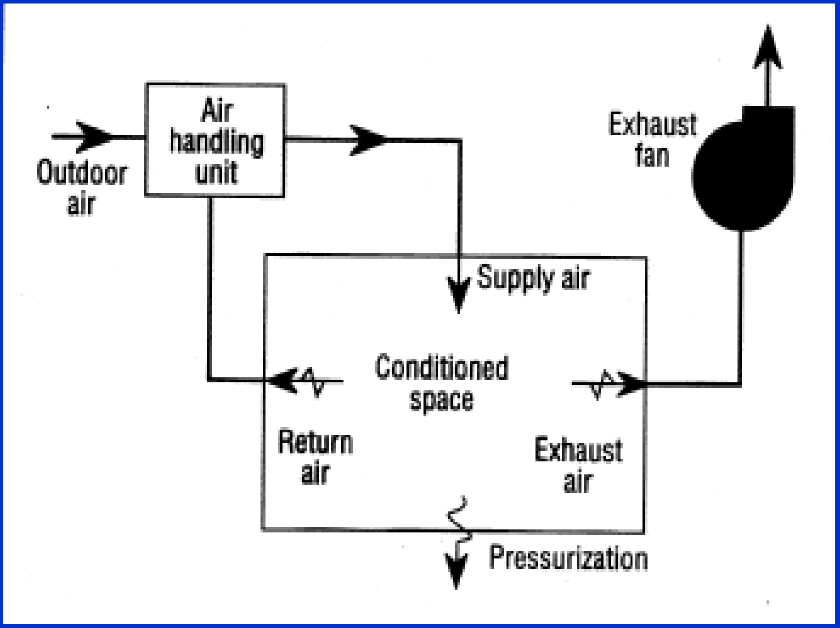
HVAC Systems



HVAC Systems



Air Handling System with Economizer



Air Balance in a Conditioned Space

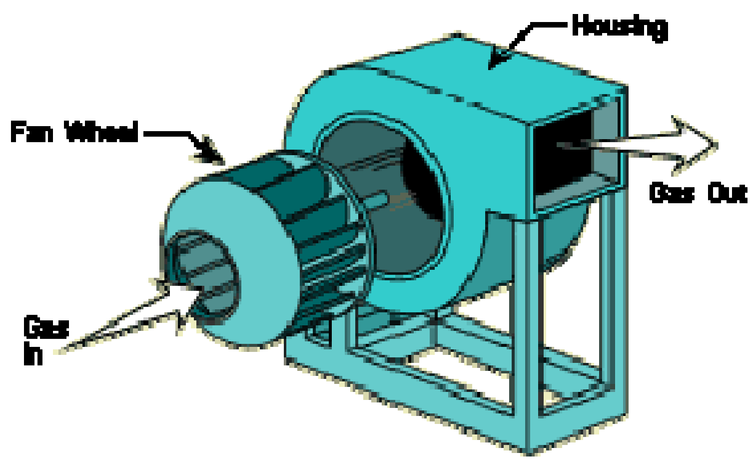


Figure 2. Centrifugal Fan Components

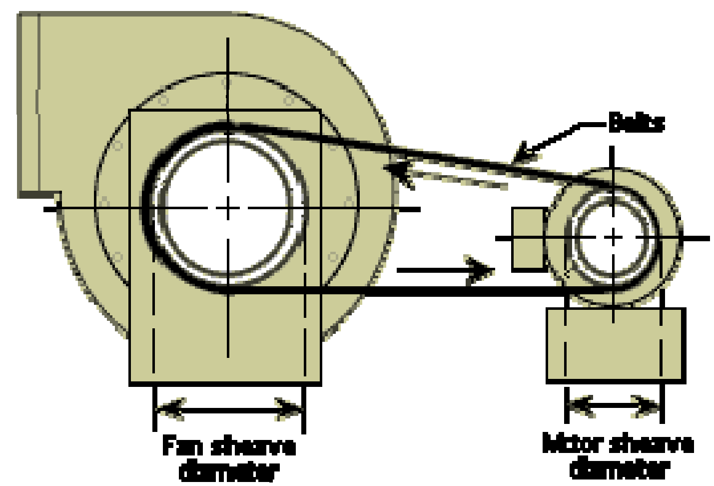


Figure 3. Centrifugal Fan and Motor Sheaves

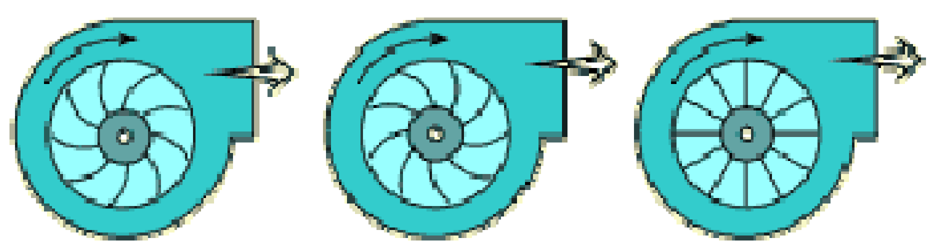

(a) Forward Curved (b) Backward Curved (c) Radial

Figure 4 . Types of Fan Wheels

Figure 3. Total System Static Pressure Drop

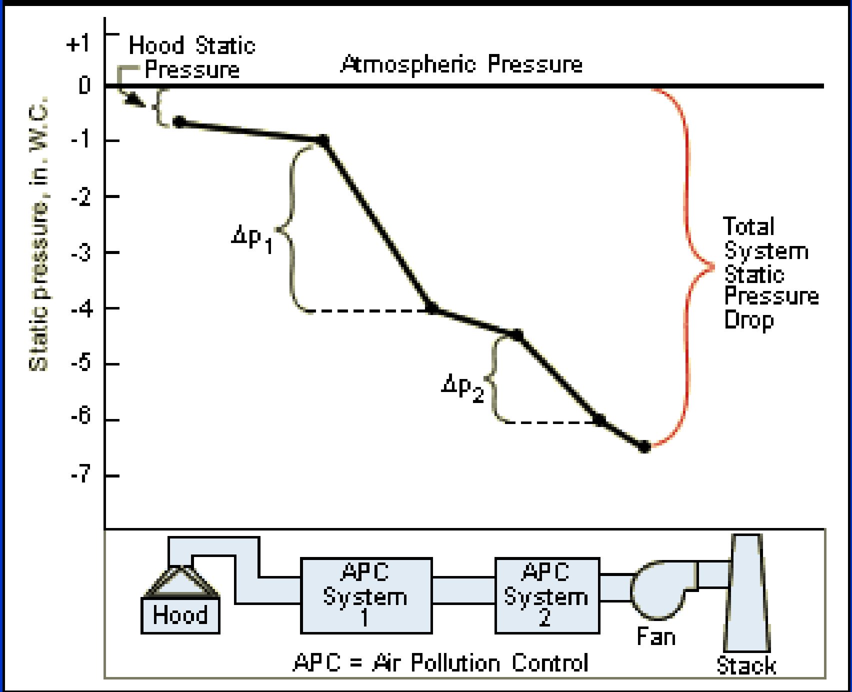


Figure 4. System Characteristic Curve

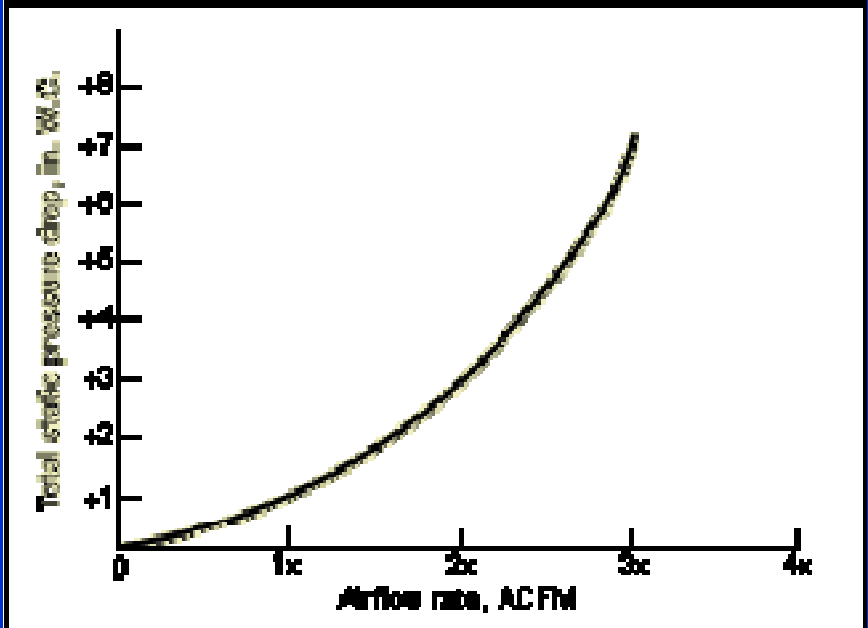


Figure 7. Fan Characteristic Curve

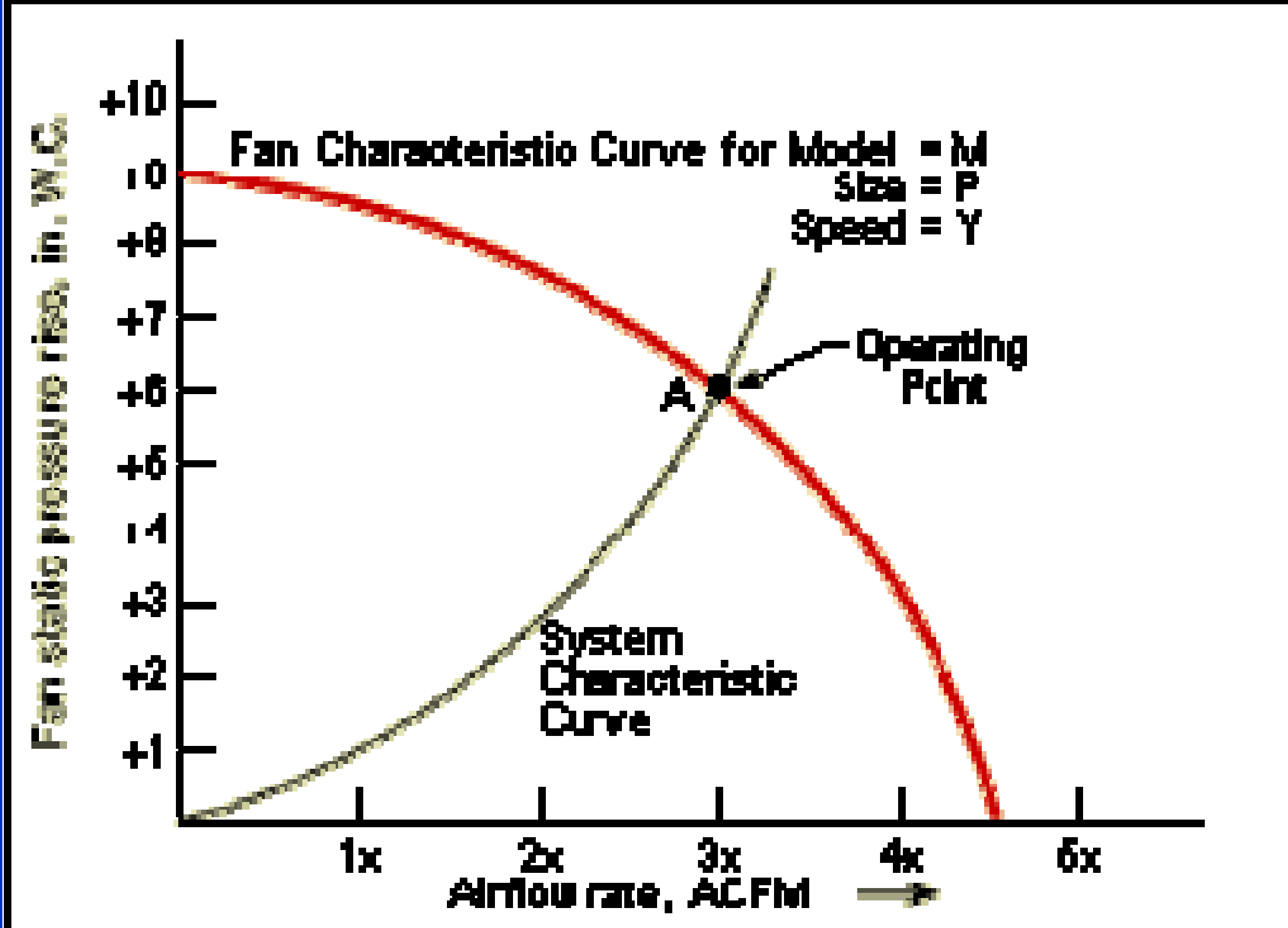


Figure 8. Effect of a Change in the System Characteristic on the Operating Point

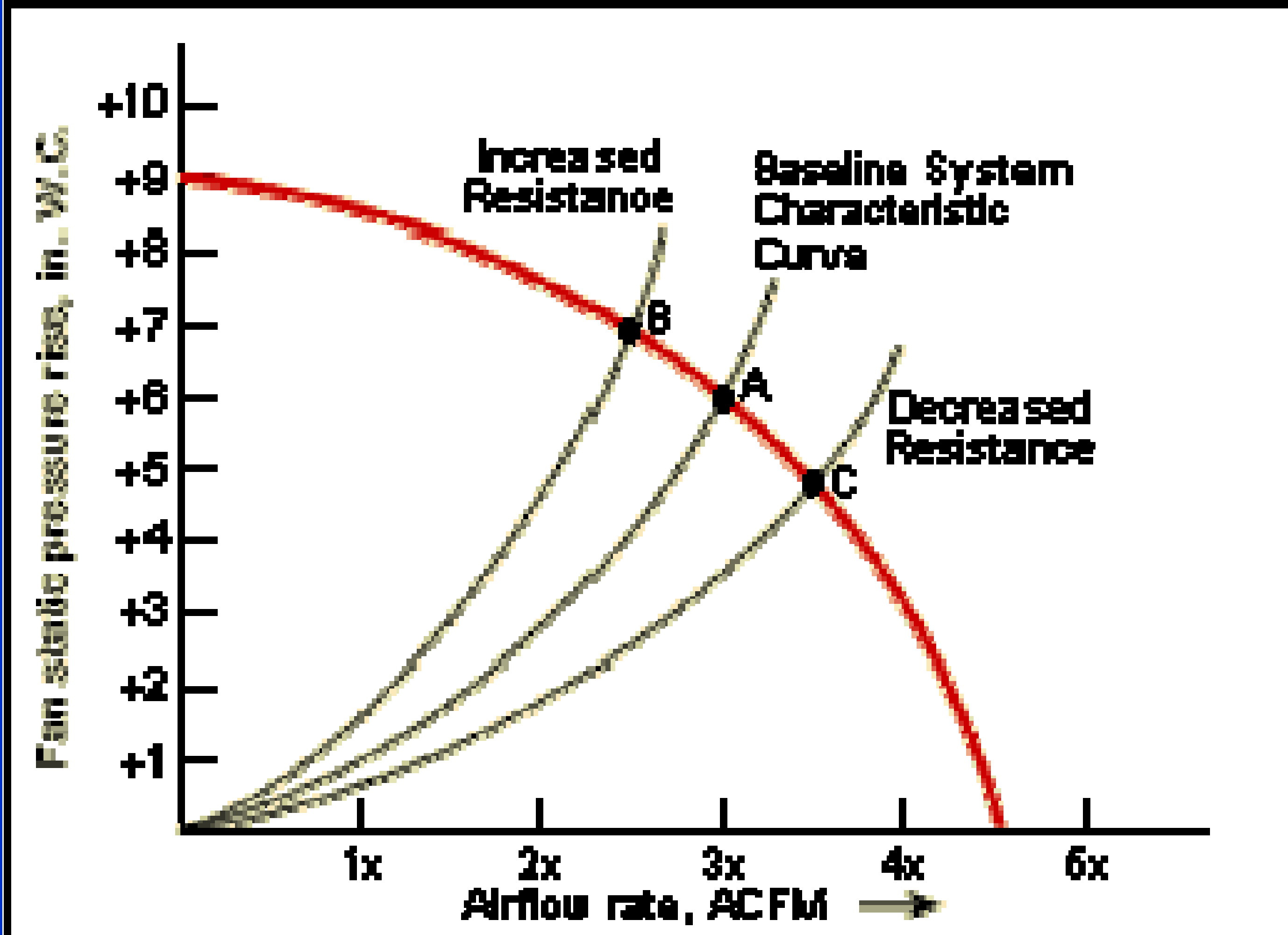
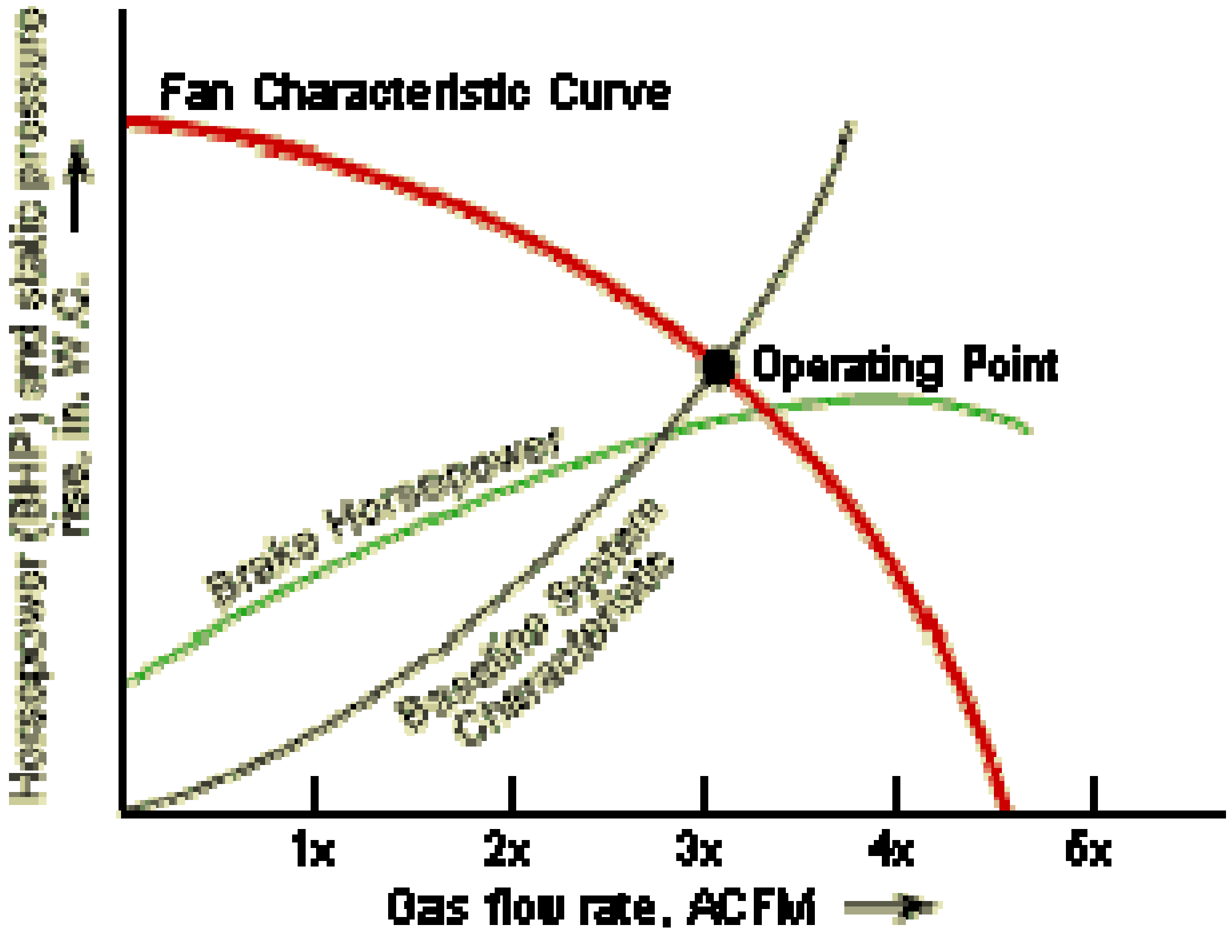
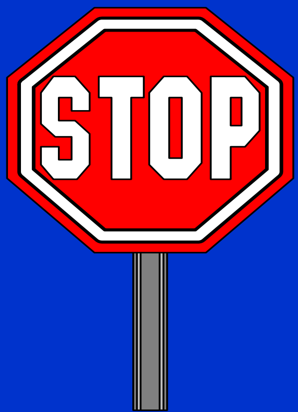
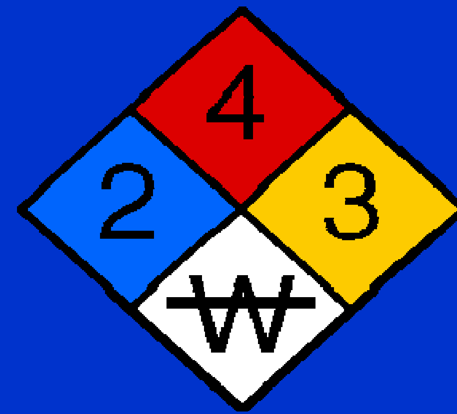


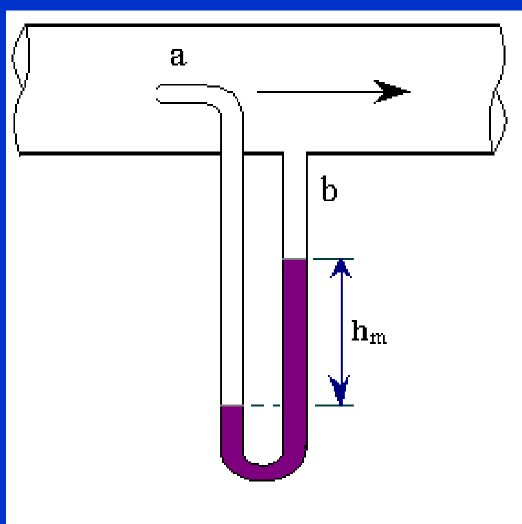
Figure 1. Example of a Brake Horsepower Curve





Questions ?





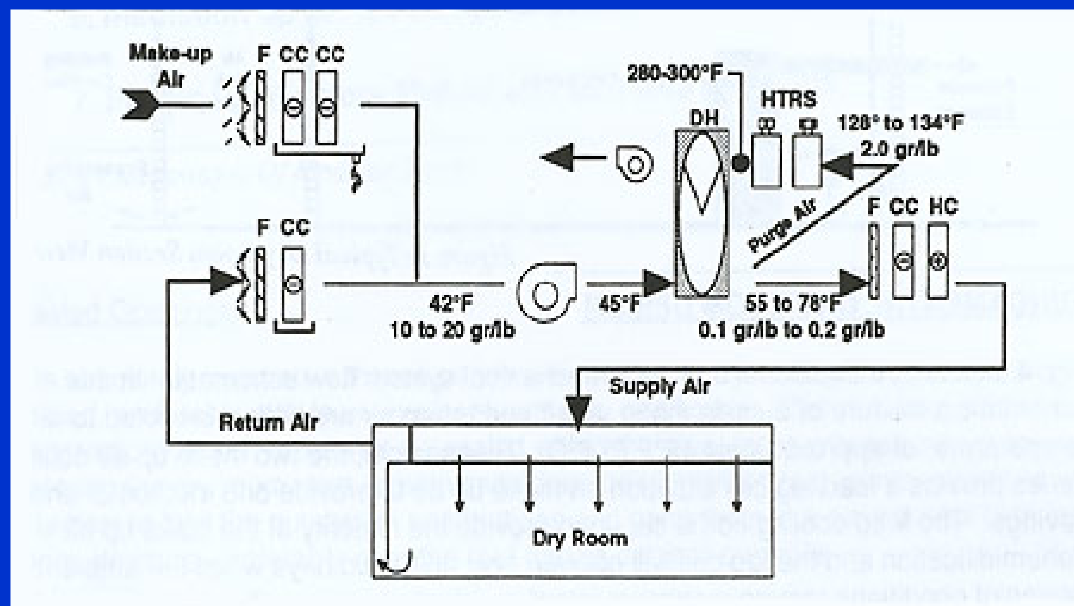


Figure 4- Typical Dry Room Mechanical System