

***Building Design and Engineering  
Approaches to Airborne Infection***

# **Basic Concepts of Ventilation Design**

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**Ketabton.com**

# 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)

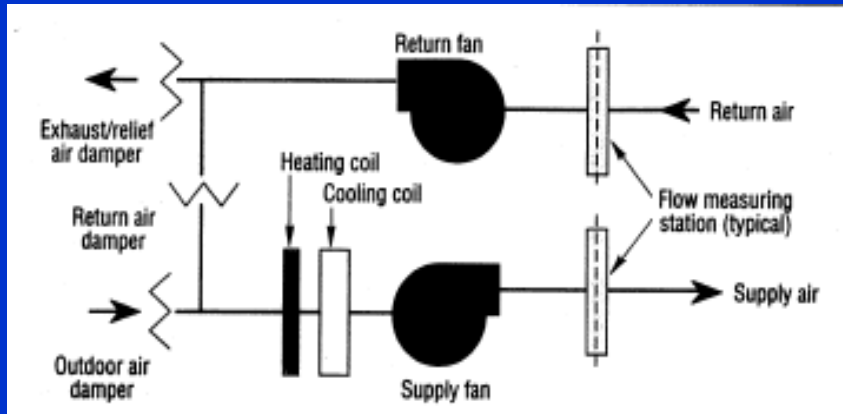
Return (recirculated air)

- **Exhaust**

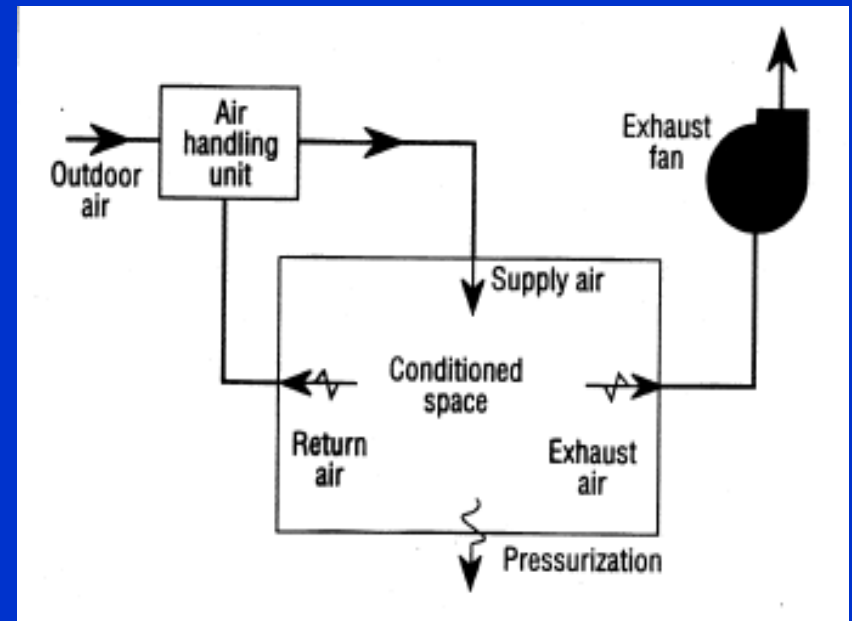
General (dilution)

Local Control (hoods)

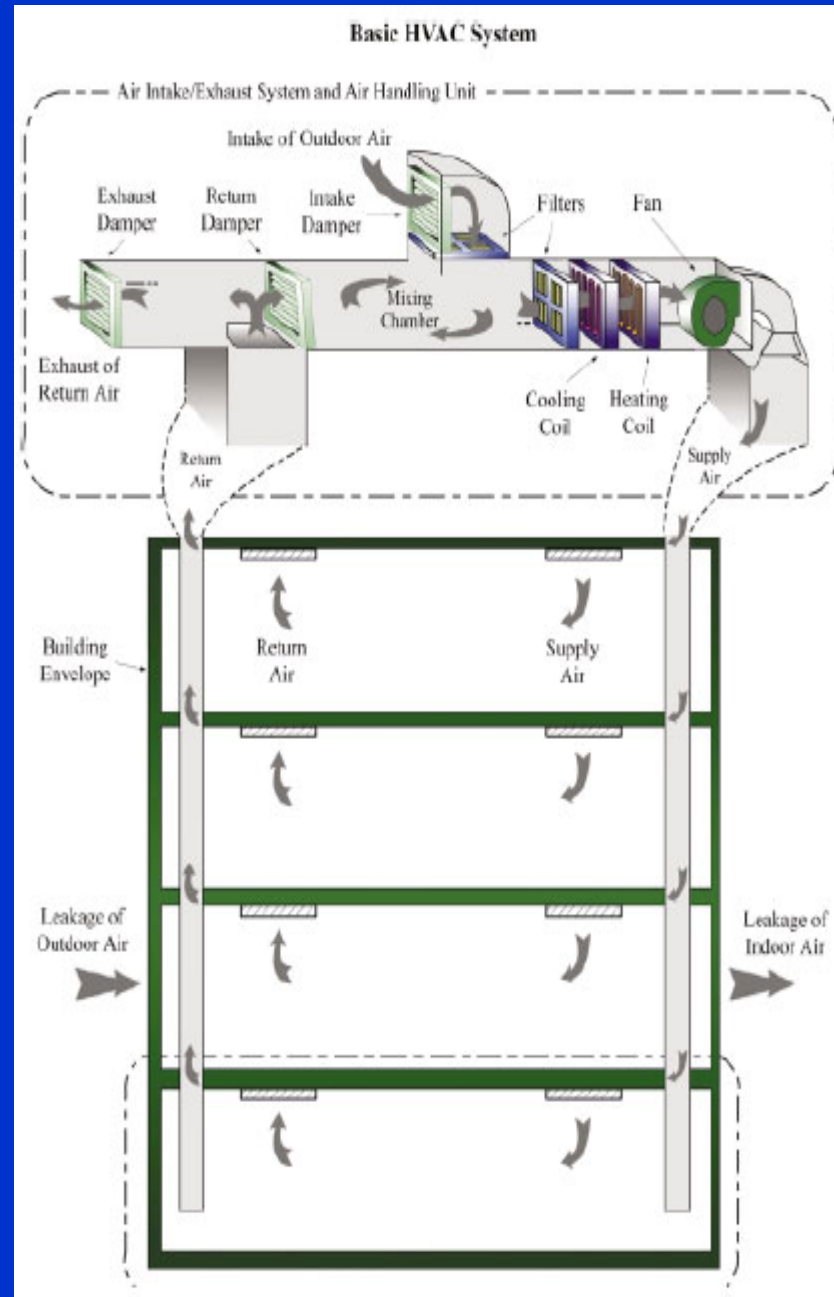
# HVAC Systems



**Air Handling System with Economizer**



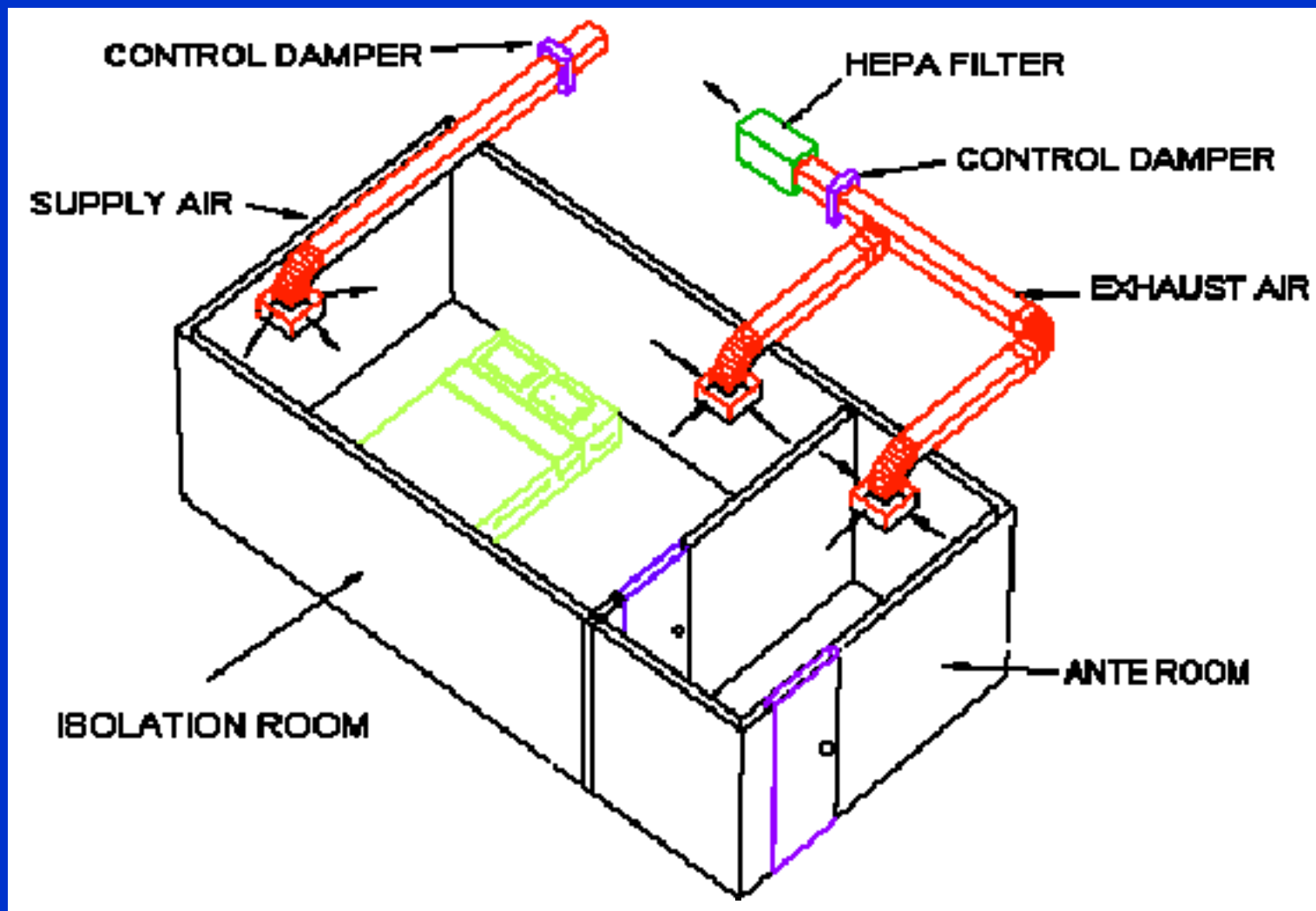
**Air Balance in a Conditioned Space**



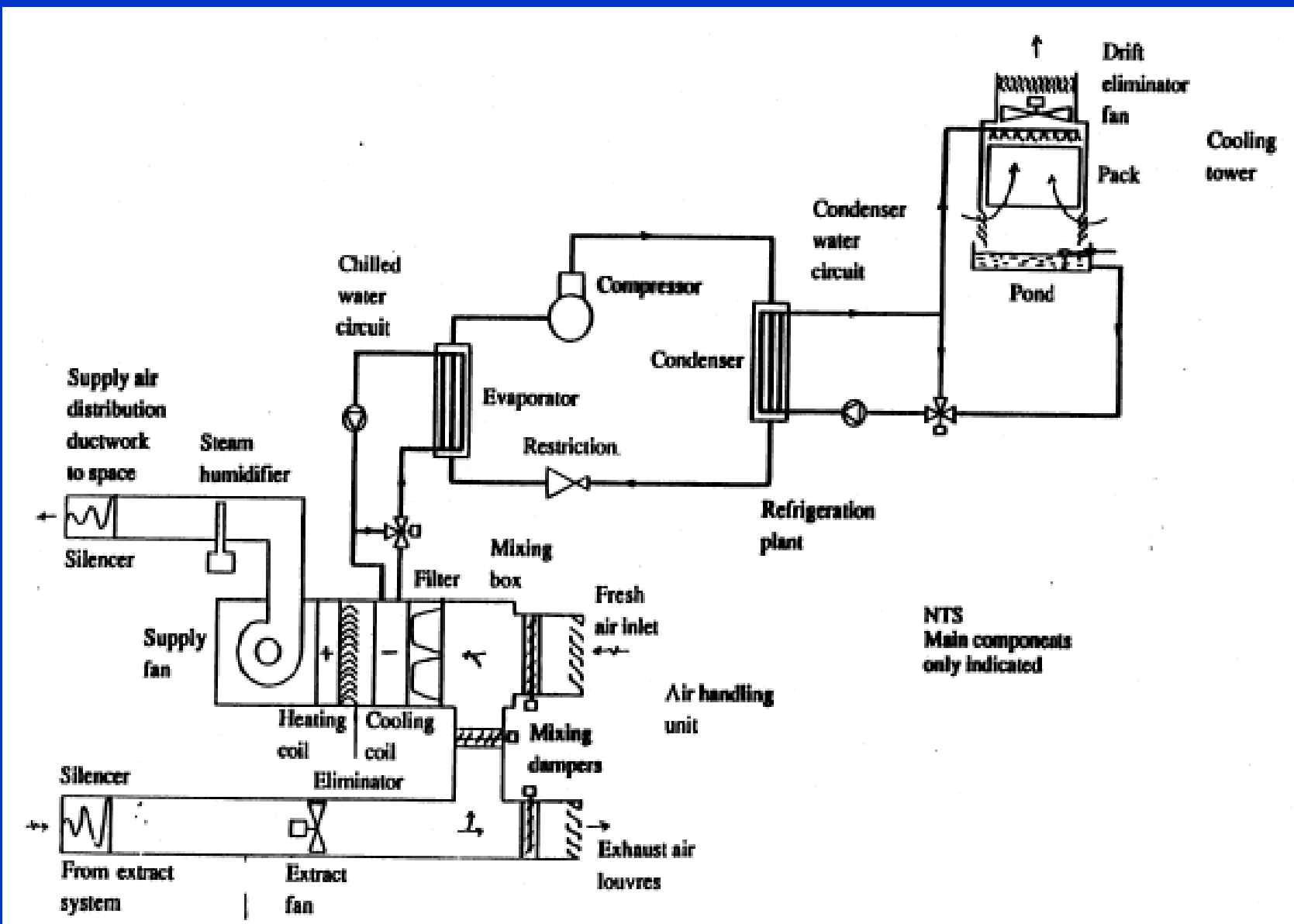
# Design Concerns

- Temperature
- Pressure
- Air Contaminants
- Work Practices
- Product Protection
- Worker Protection
- Building Codes
- Equipment Selection
- Energy Conservation
- Maintenance
- Security
- Expansion

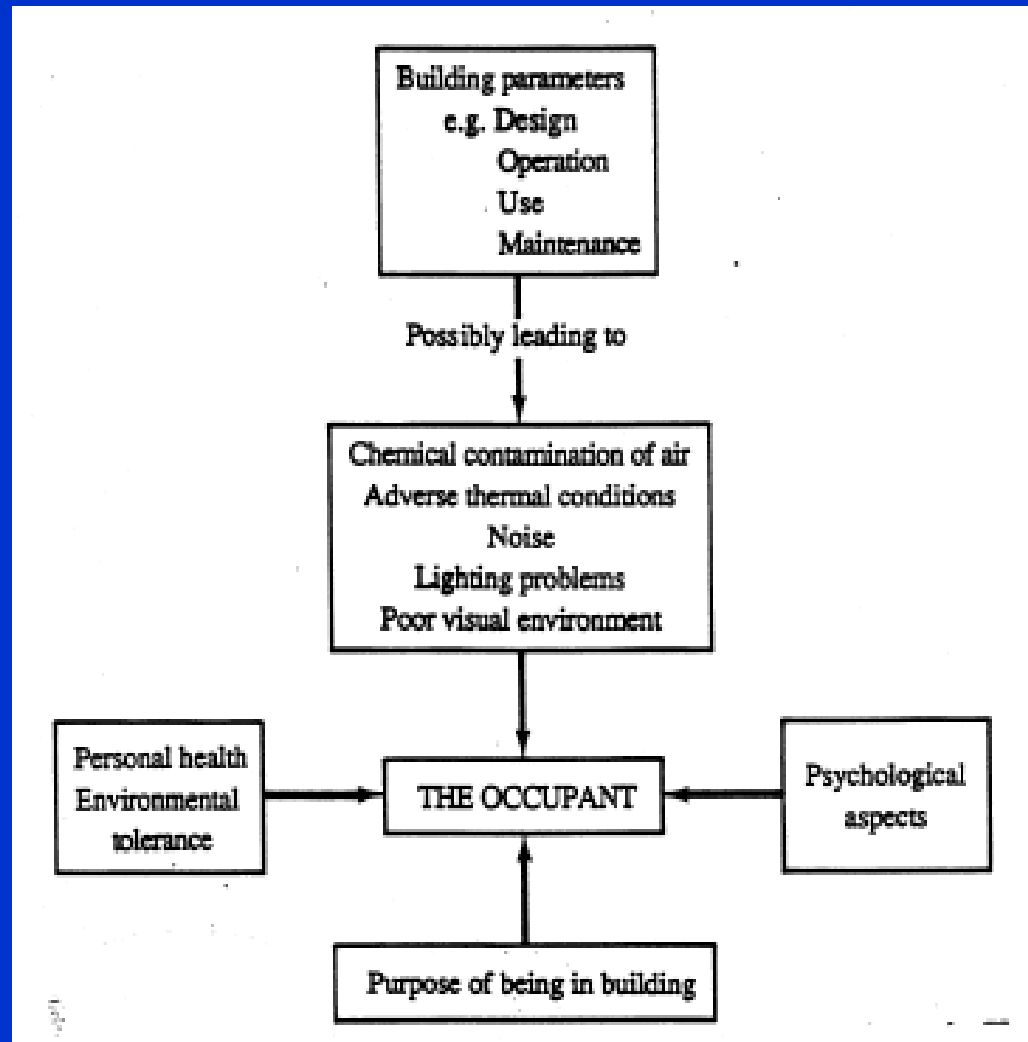




**Patient Isolation Room with HEPA Exhaust Filtration**



**Air Conditioning System Water and Refrigeration Circuits**

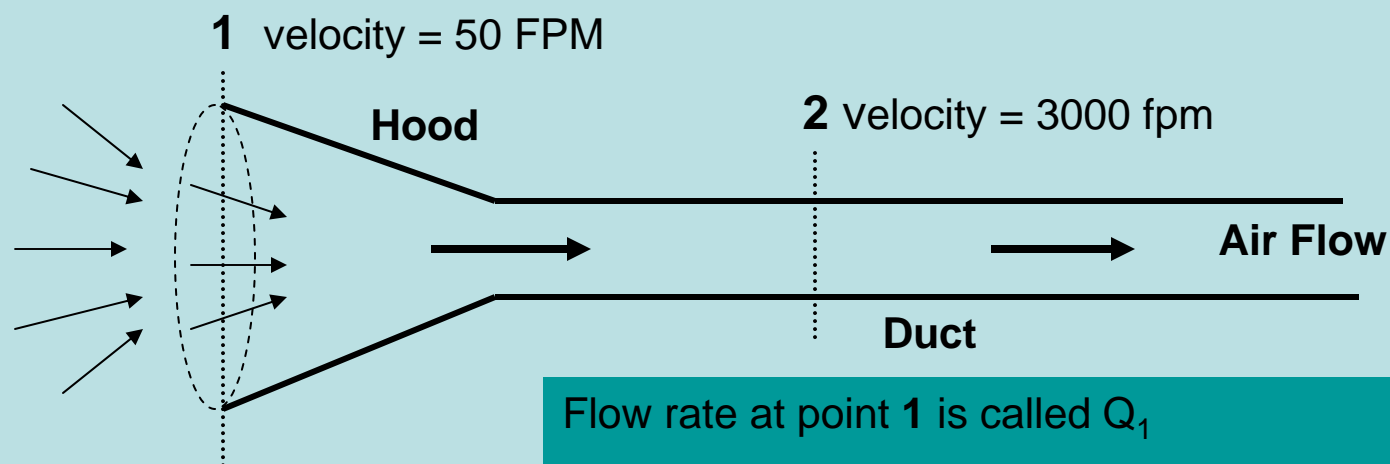


## Factors in the Perception of Air Quality

# Conversion Factors

Quantity	To Convert	Into	Multiply By:
Volumetric Flow	cubic feet/minute (ft <sup>3</sup> /min)	cubic meters/second (m <sup>3</sup> /sec)	$4.719 \times 10^{-4}$
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

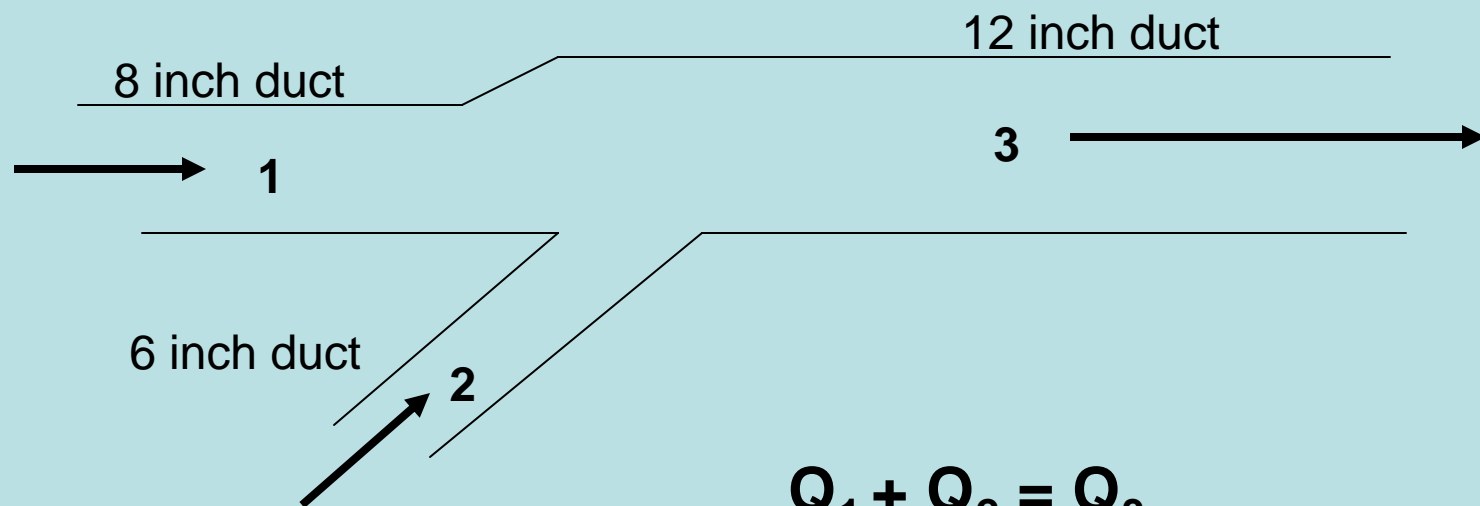
$Q$  = Volumetric Flow Rate,  $\text{ft}^3/\text{min}$

$V$  = Air Velocity,  $\text{ft}/\text{min}$  or  $\text{fpm}$

$A$  = Cross Sectional Area,  $\text{ft}^2$  or  $\text{SF}$

# Conservation of Mass

$$Q = V \cdot A$$



$$Q_1 + Q_2 = Q_3$$

$$V_1 A_1 + V_2 A_2 = V_3 A_3$$

# AIR FLOW

- At standard temperature and pressure (STP):
  - \* 1 atmosphere & 70° F \*

The density of air is  $0.075 \text{ lb}_m/\text{ft}^3$
- 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 ( $T_P$ )**

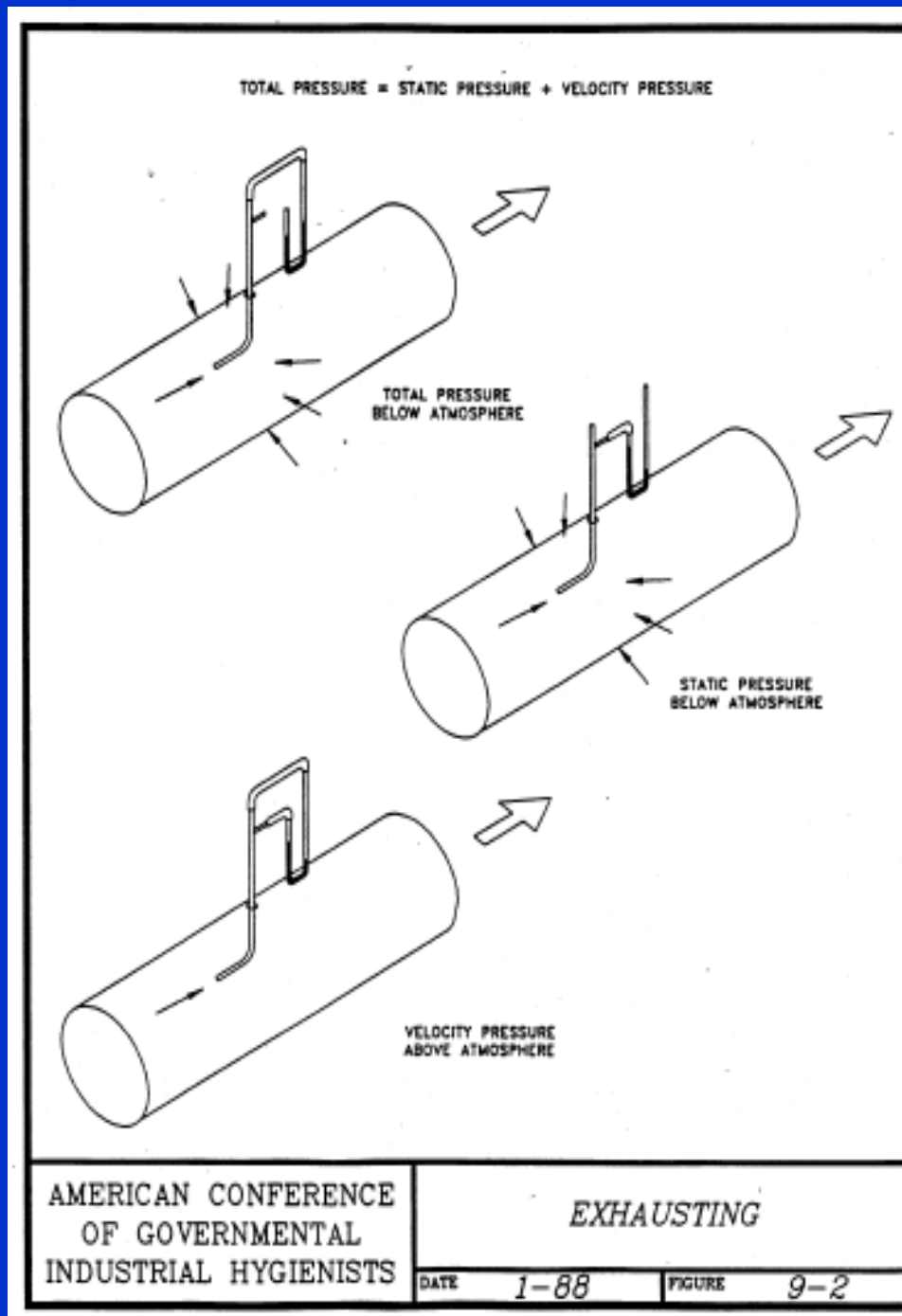
combined static & velocity components  
can be + or -

measure of energy content of air stream  
Always decreasing as flow travels  
downstream thru a system only rising when  
going across a fan



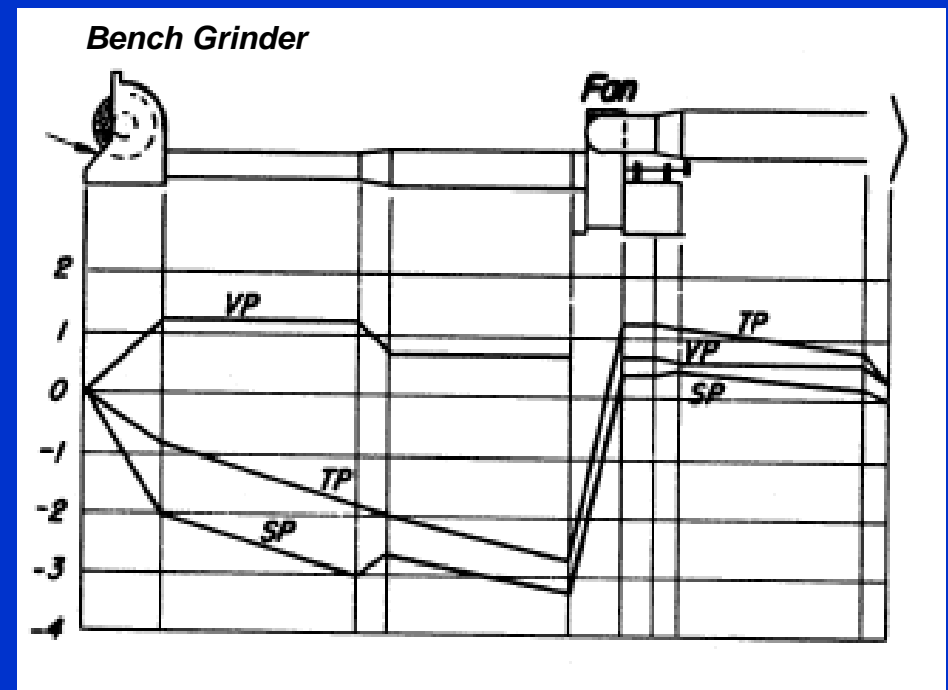
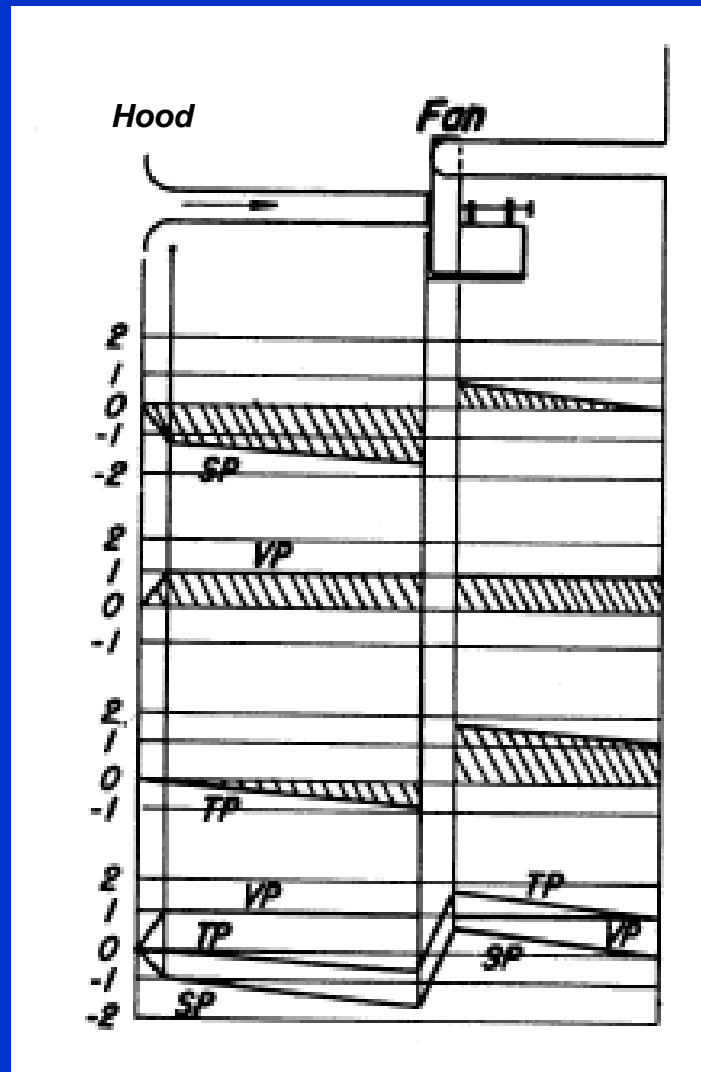
$$TP = SP + VP$$

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

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

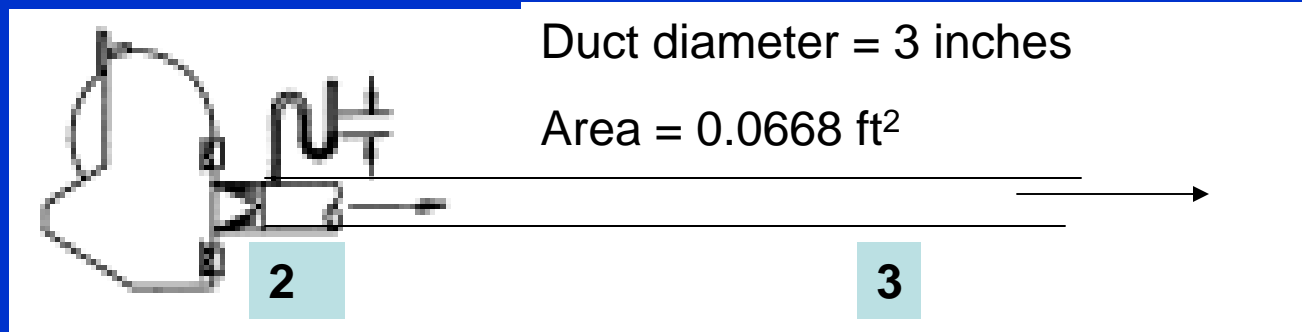
where  $\rho$  = air density  
@ STP  $\rho = 0.075 \text{ lb}_m/\text{ft}^3$

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

- 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

1



- $Q_1 = Q_2$
- If Q desired is 300 cfm
- Then  $Q = V A$   
 $V = Q / A$   
 $V = (300) / (0.0068)$   
 $V = 4490 \text{ fpm}$

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

$$SP_1 + VP_1 = SP_2 + VP_2$$

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

we then have:

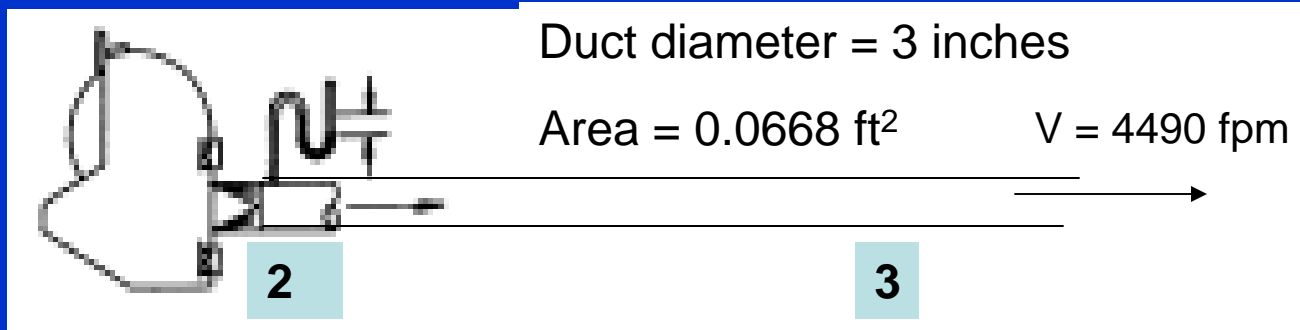
$$0 = SP_2 + VP_2$$

or

$$-VP_2 = SP_2$$

# Bench Grinder Exhaust Ventilation

1



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

$$SP_1 + VP_1 = SP_2 + VP_2$$

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

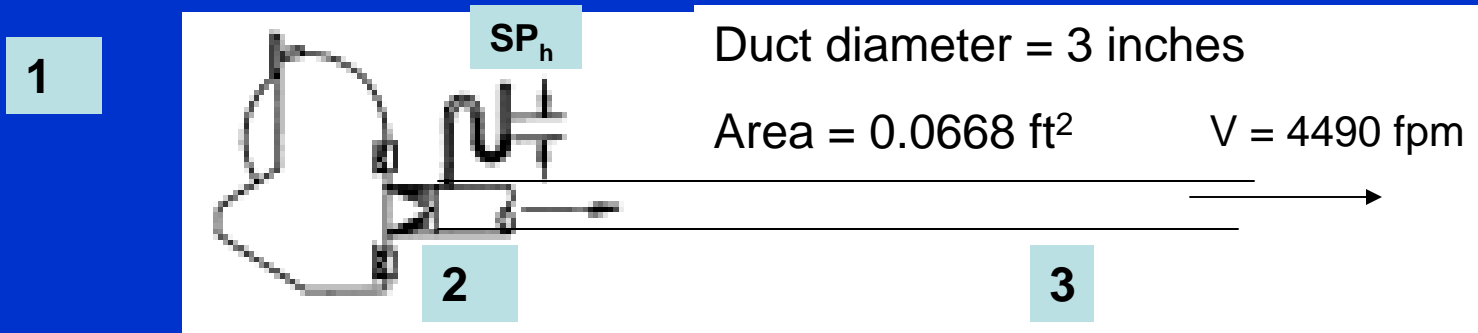
we then have:

$$0 = SP_2 + VP_2$$

or  $SP_2 = (-VP_2)$

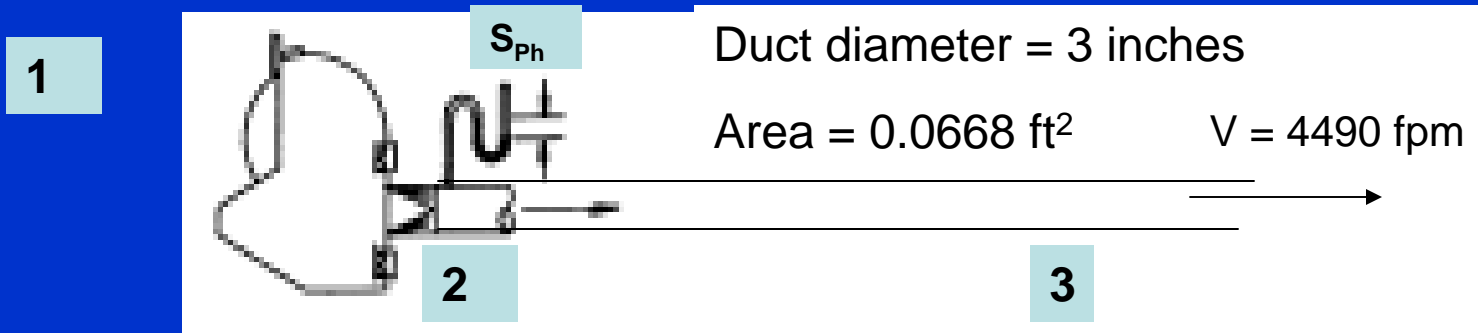
- from  $V = 4005 (VP)^{0.5}$
- $VP_2 = (4490/4005)^2$
- $VP_2 = 1.26$  in w.g.
- then:  $SP_2 = (-VP_2)$   
 $SP_2 = -1.26$  in w.g.

# Bench Grinder Exhaust Ventilation



- 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
- $F_h$  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 **SP<sub>h</sub>**
- **SP<sub>h</sub>** is also called the hood static suction and is the absolute value of **SP<sub>2</sub>**

# Bench Grinder Exhaust Ventilation



- Now add the hood entry loss:

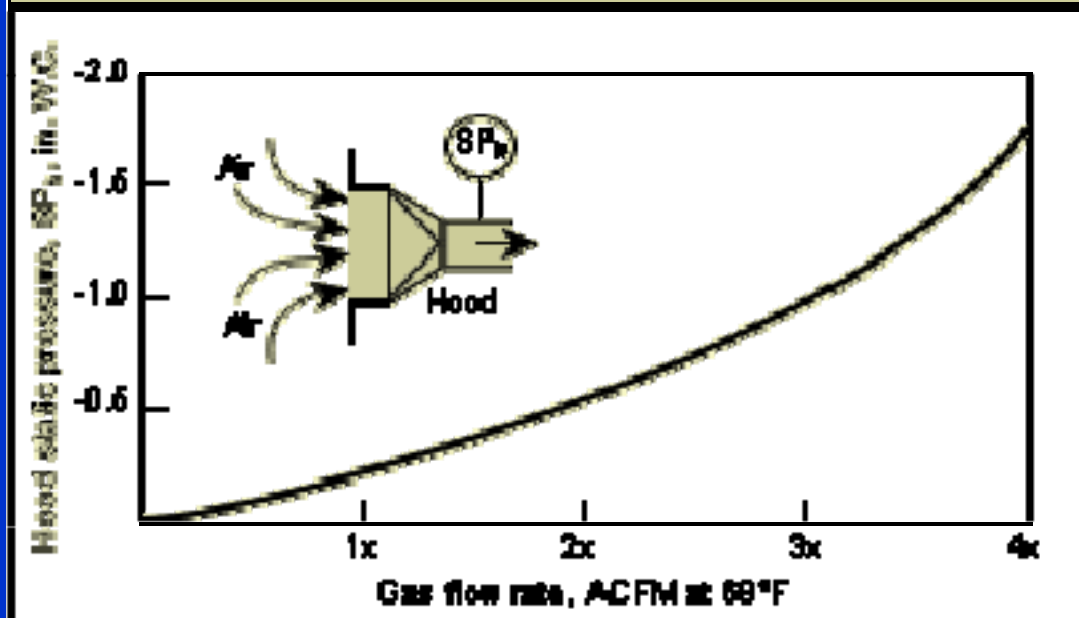
$$SP_h = VP_2 + h_e = VP_2 + (F_h) (VP_2)$$

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

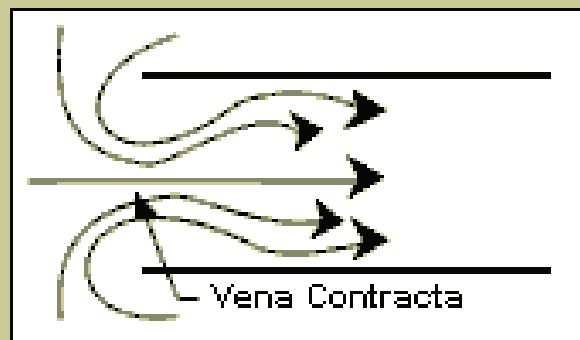
- **$SP_h = 1.26 + (0.40) (1.26) = 1.76$  in w.g.**



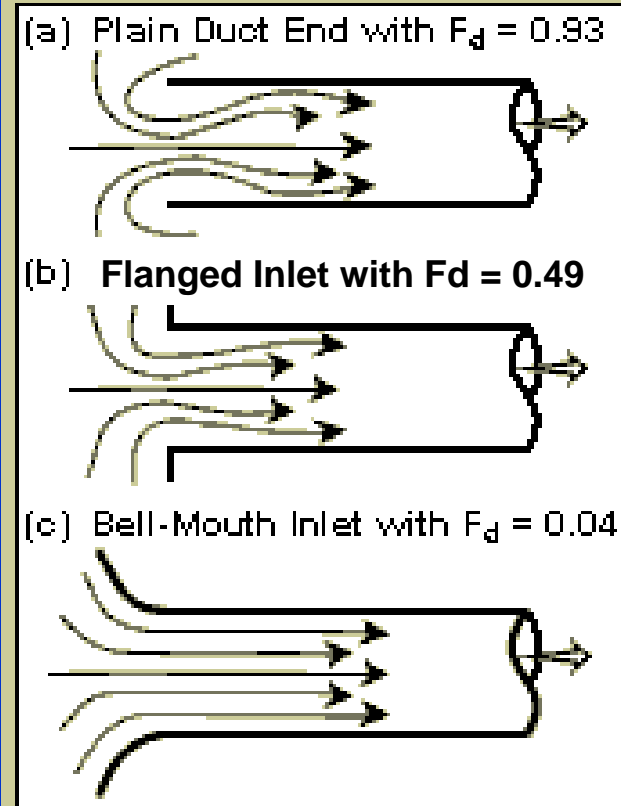
**Figure 1. Relationship Between Hood Static Pressure and Flow Rate Entering Hood**



**Figure 2. Air Flow Convergence in a Duct**

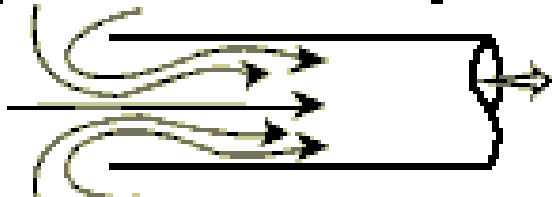


**Figure 3. Hood Entry Loss Coefficients ( $F_d$ ) for Various Duct Designs**

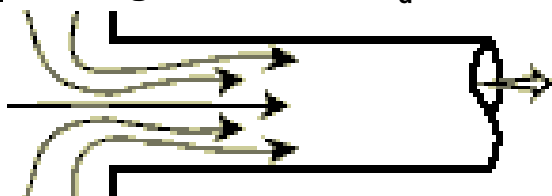


**Figure 3. Hood Entry Loss Coefficients ( $F_d$ ) for Various Duct Designs**

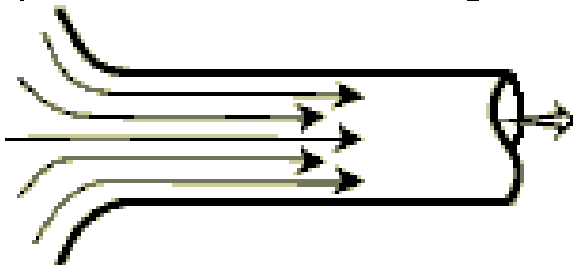
(a) Plain Duct End with  $F_d = 0.93$

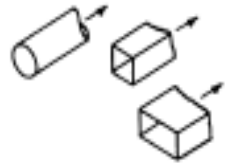
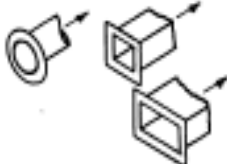
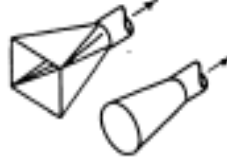
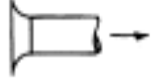




(b) Flanged Inlet with  $F_d = 0.49$



(c) Bell-Mouth Inlet with  $F_d = 0.04$



HOOD TYPE	DESCRIPTION	COEFFICIENT OF ENTRY, $C_e$	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

**Actual Flow**

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

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

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

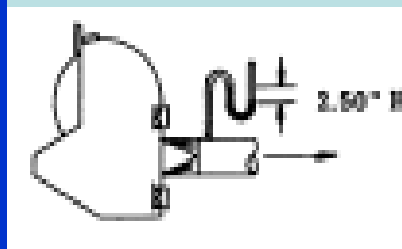
# Hood Entry Coefficients

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

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<sup>2</sup>
- $C_e$  bench grinder hood =  $0.78$

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

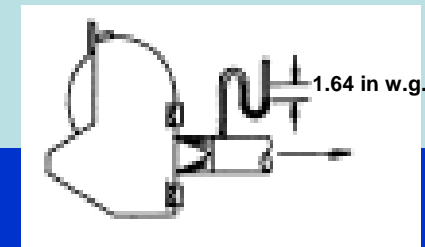
$$(VP/SP_h) = (0.78)^2$$

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

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

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

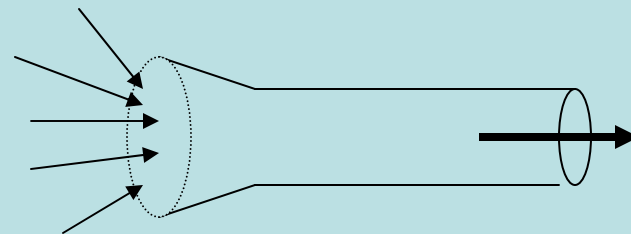
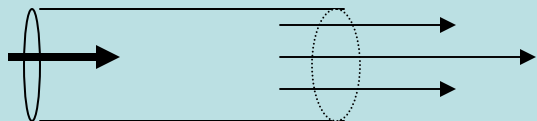
$$VP = 0.998 \text{ in w.g.}$$



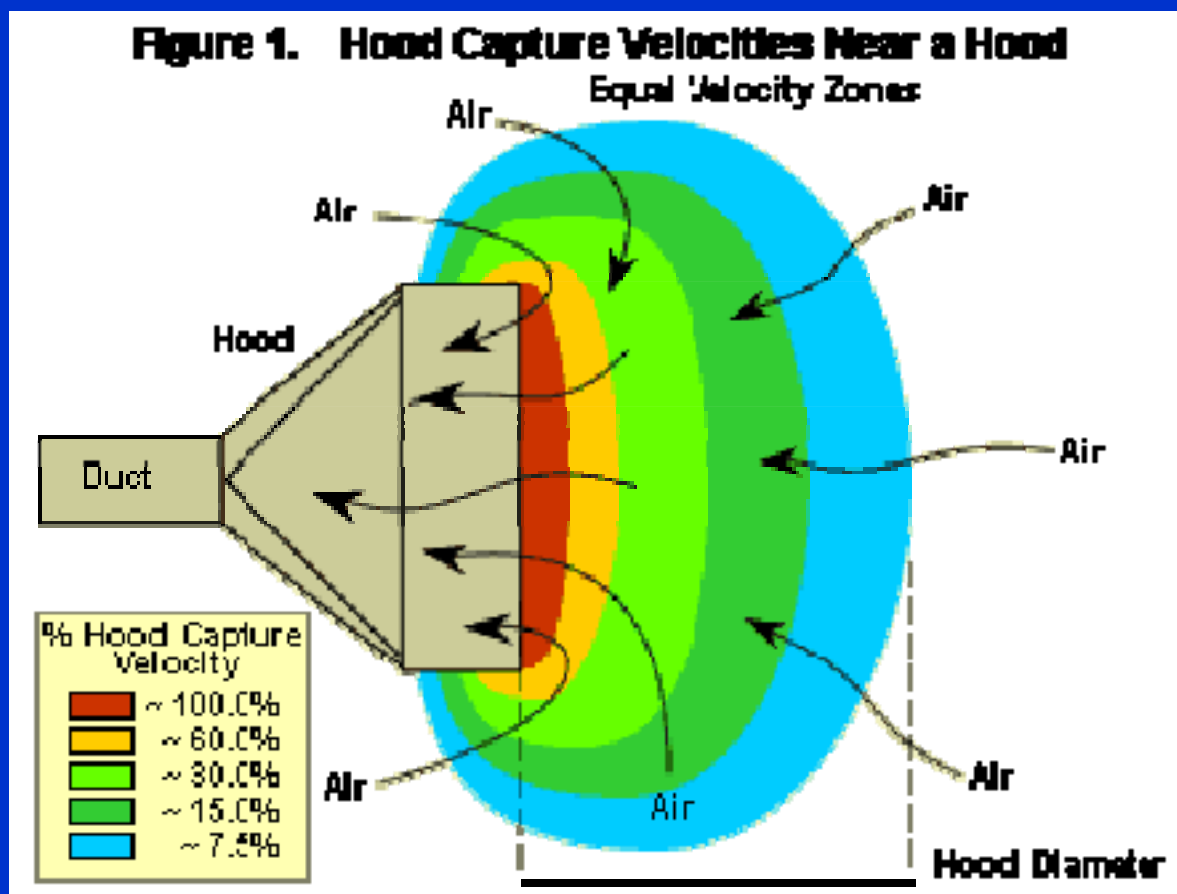
# Air Flow Characteristics

- See Industrial Ventilation Manual notes

## Blowing vs. Exhausting



# Air Flow Characteristics



Exhaust Hoods

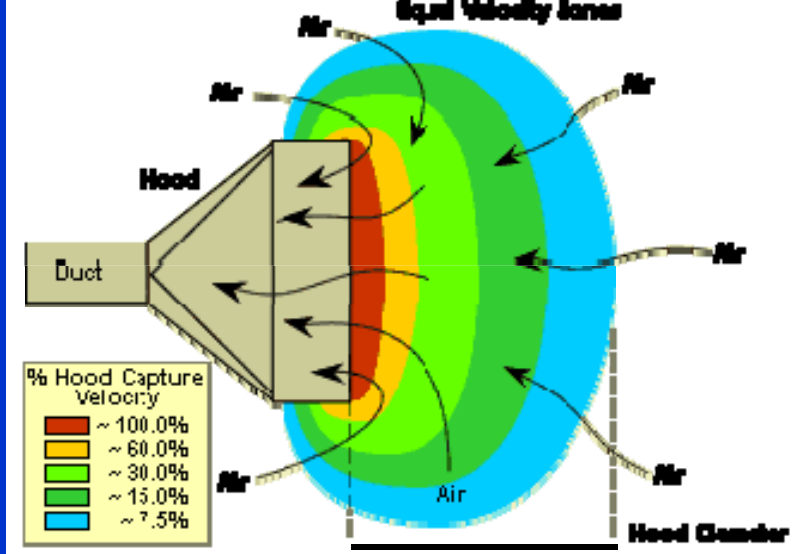
Capture Velocity

From Dalla Valle's empirical work

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



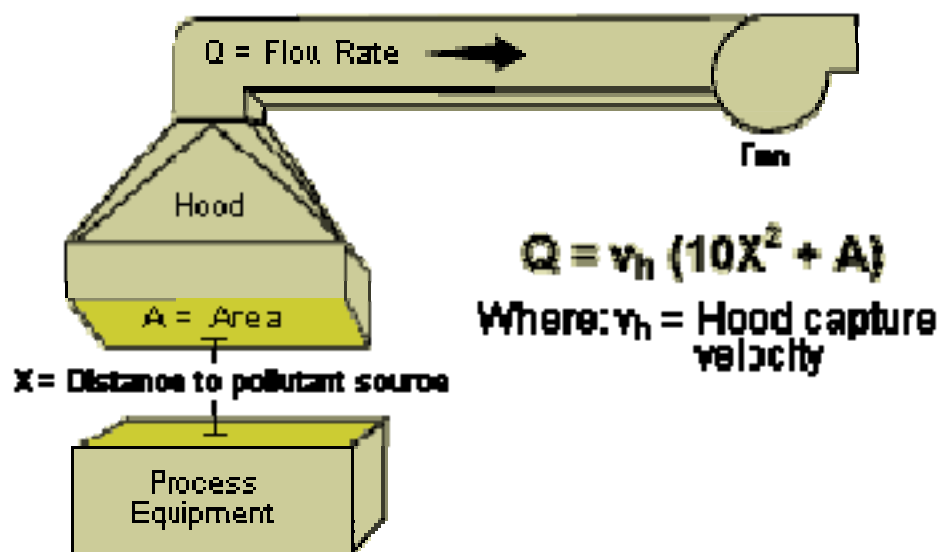
**Figure 1. Hood Capture Velocities Near a Hood Equal Velocity Zones**



# Capture Velocity

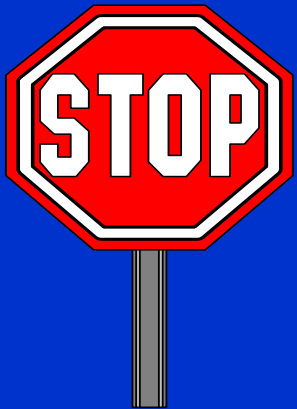
$$V_{(x)} = Q / (10 X^2 + A)$$

**Figure 2. Hood Capture Velocity Equation (without Flange)**



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 ?

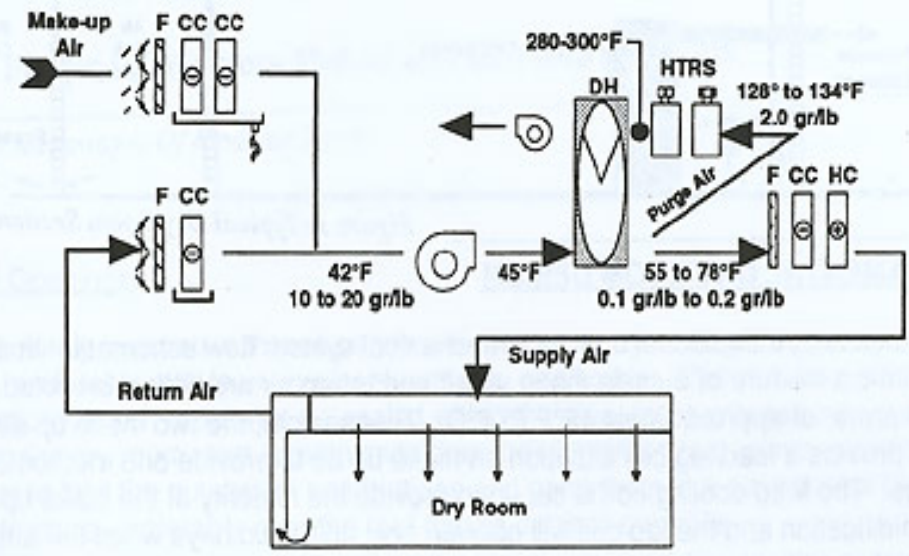
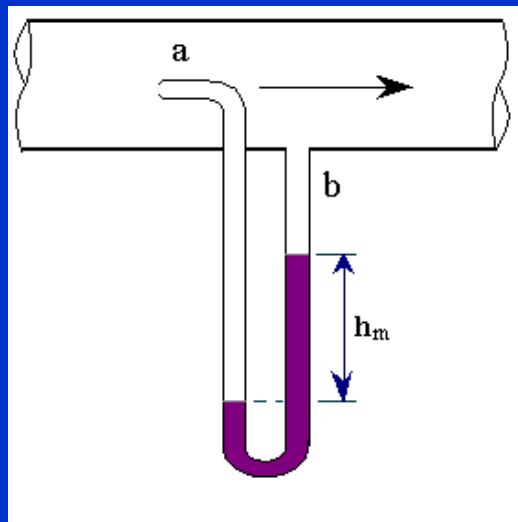
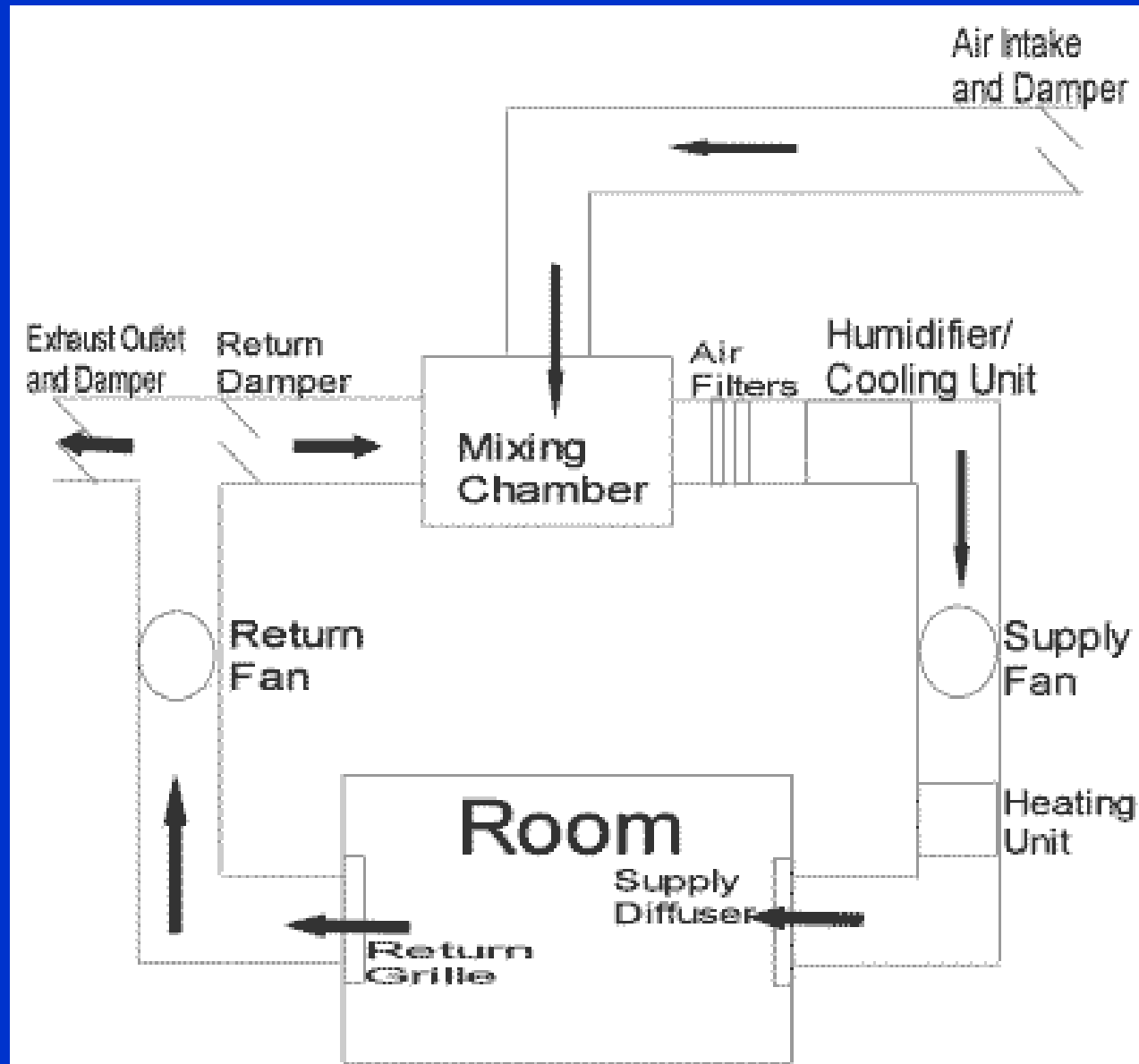


Figure 4- Typical Dry Room Mechanical System

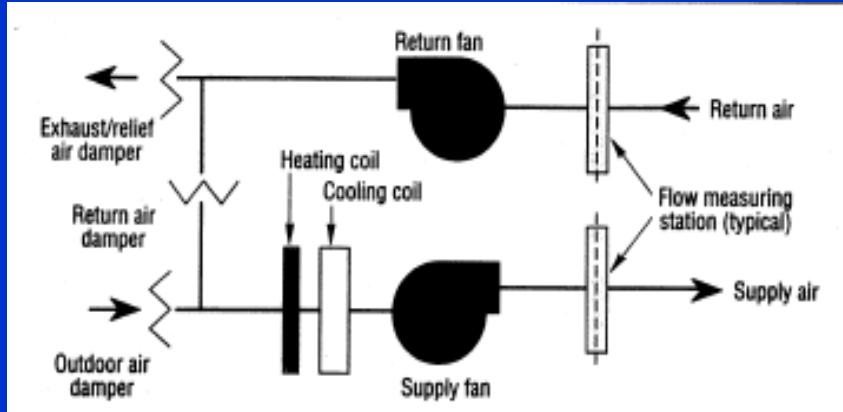
## *In-Place Filter Testing Workshop*

# **Ventilation Systems: Operation and Testing**

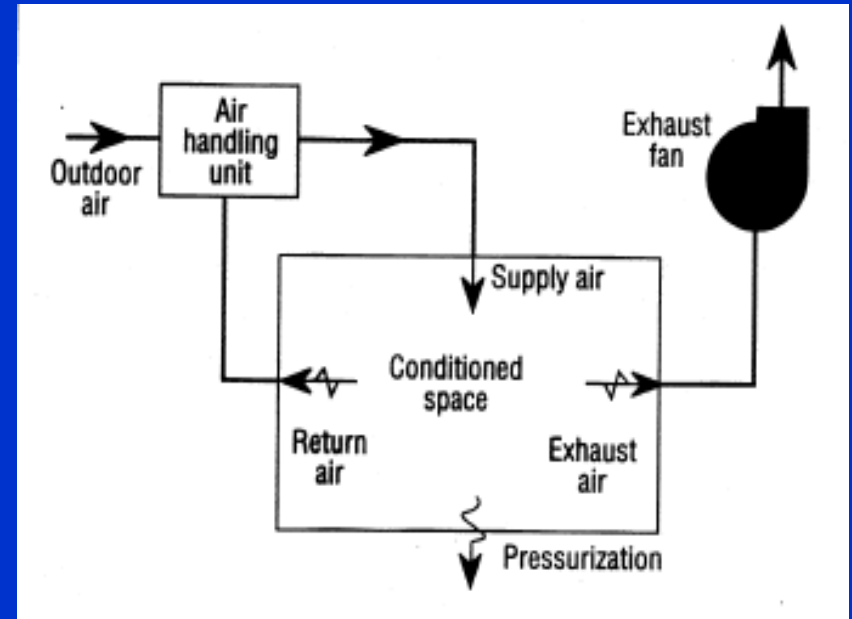
# HVAC Systems



# HVAC Systems

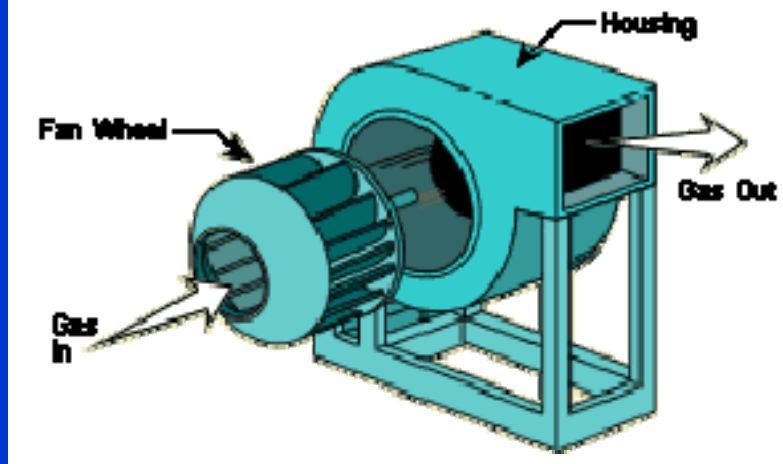


**Air Handling System with Economizer**

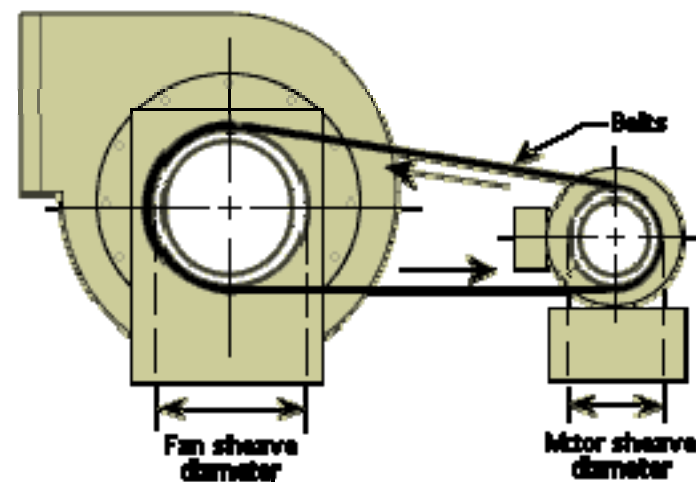


**Air Balance in a Conditioned Space**

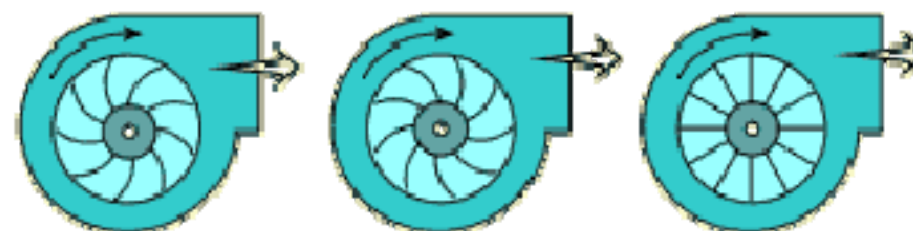
**Figure 2. Centrifugal Fan Components**



**Figure 3. Centrifugal Fan and Motor Sheaves**



**Figure 4. Types of Fan Wheels:**



(a) Forward Curved

(b) Backward Curved

(c) Radial

Figure 3. Total System Static Pressure Drop

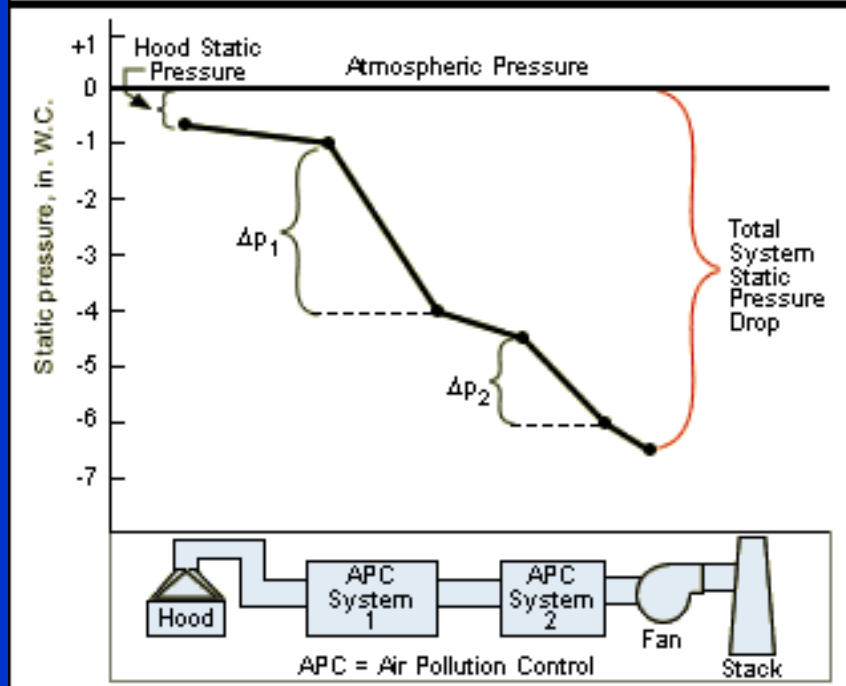


Figure 4. System Characteristic Curve

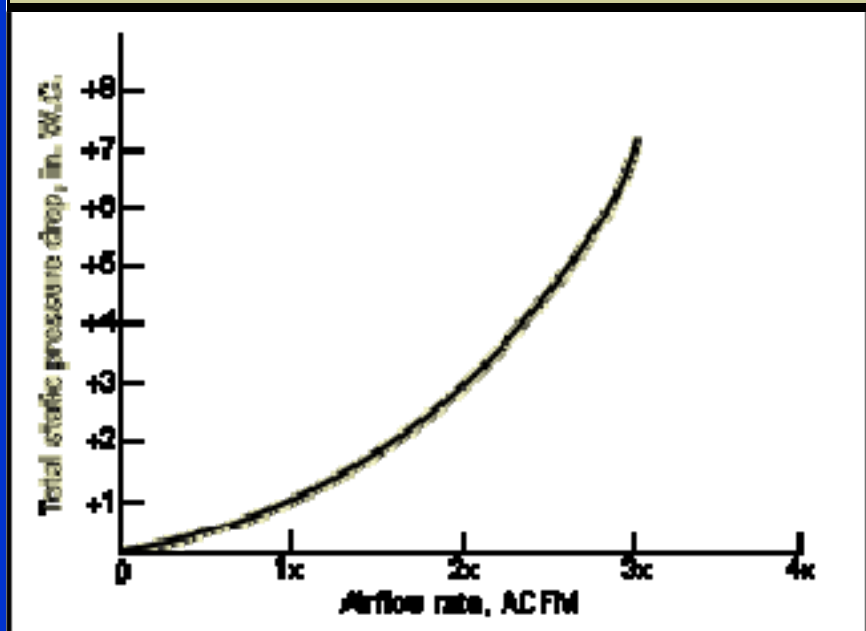
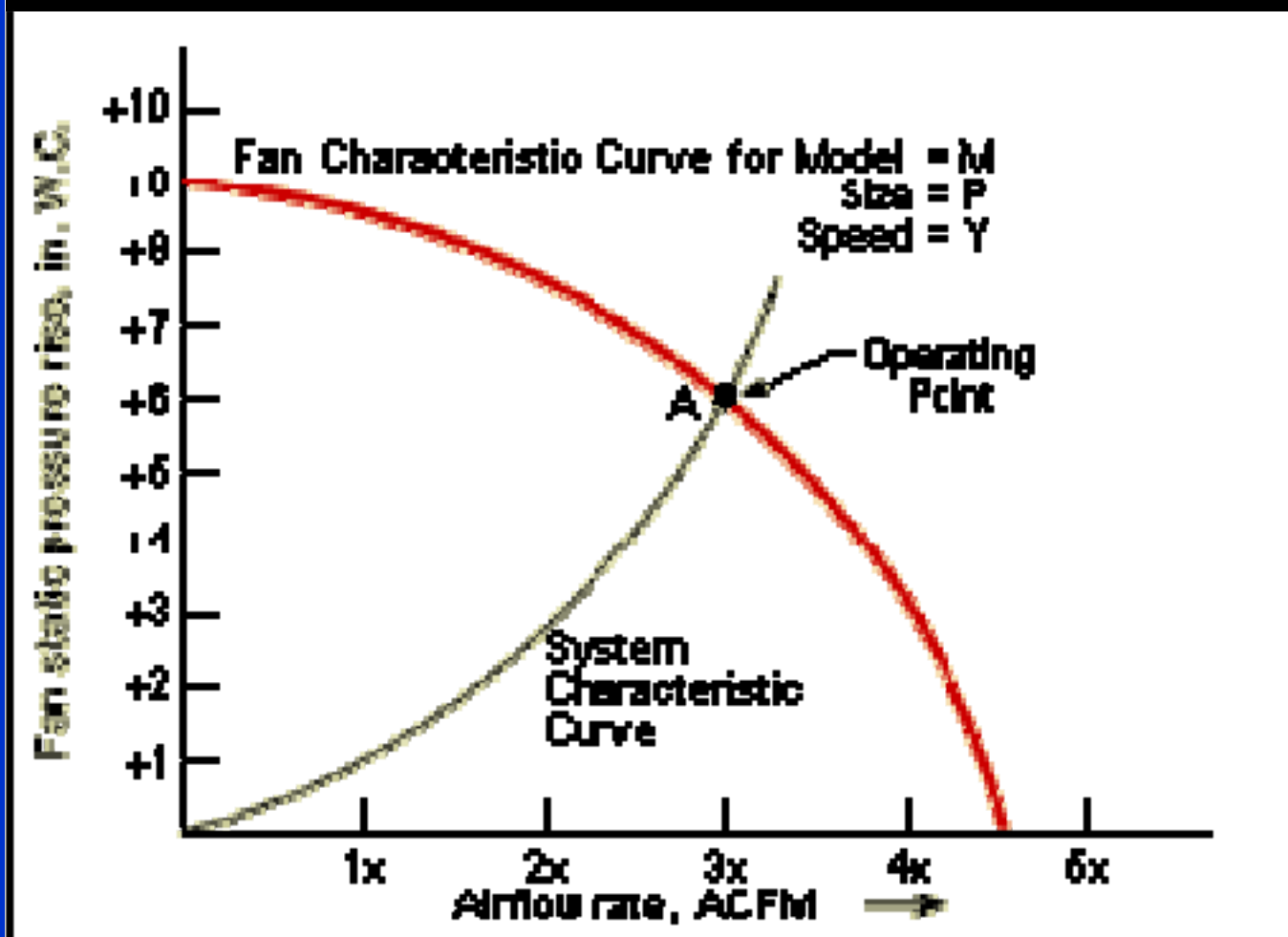


Figure 7. Fan Characteristic Curve





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

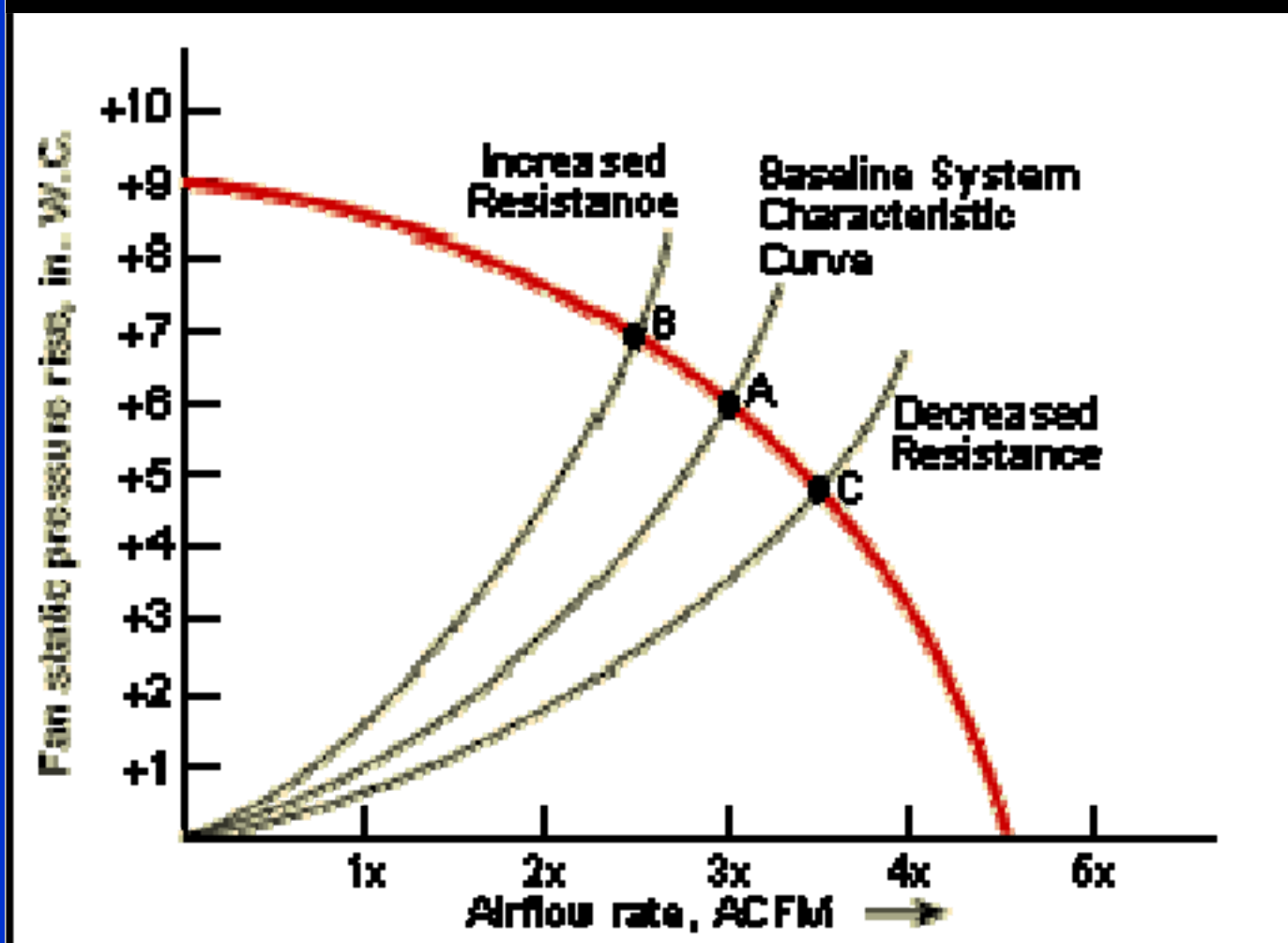
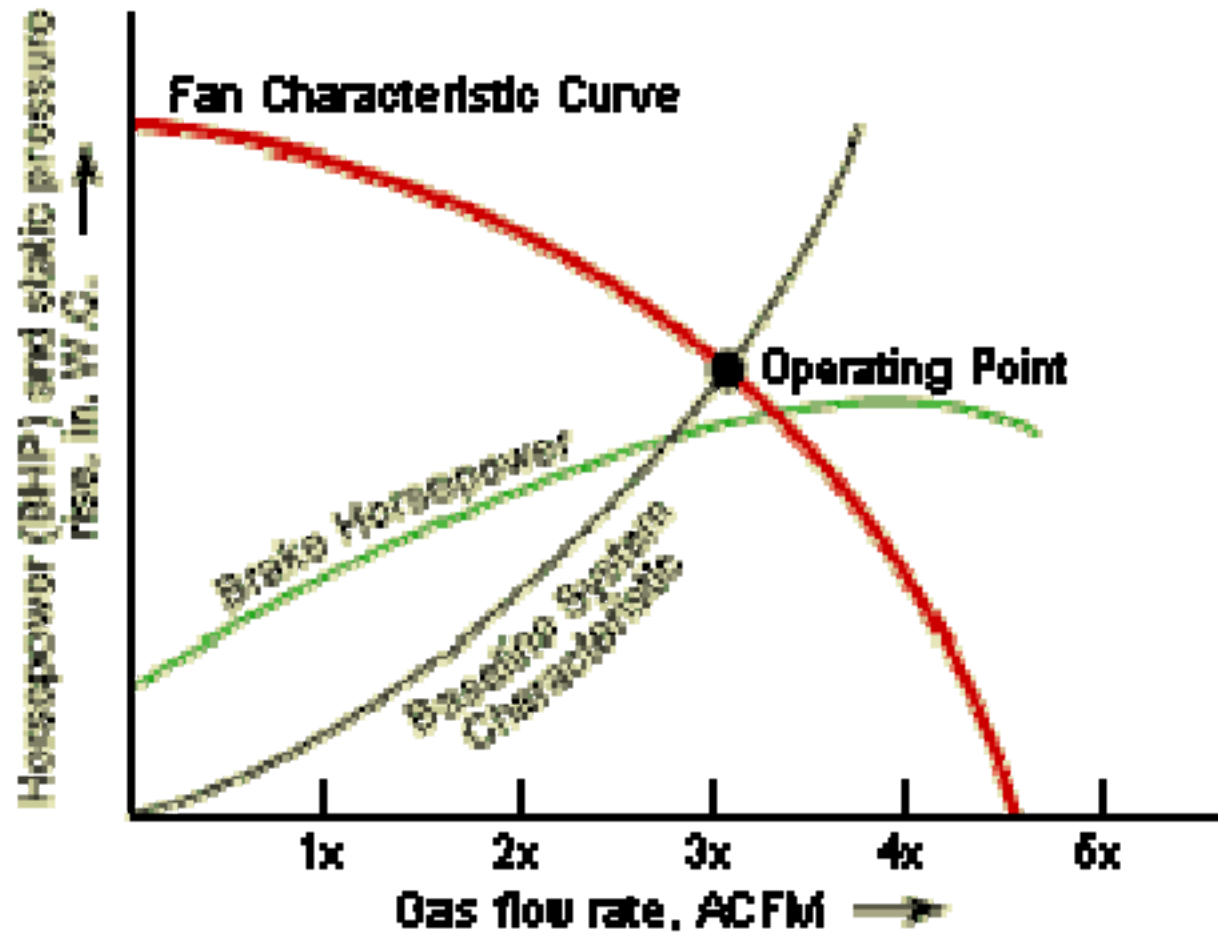
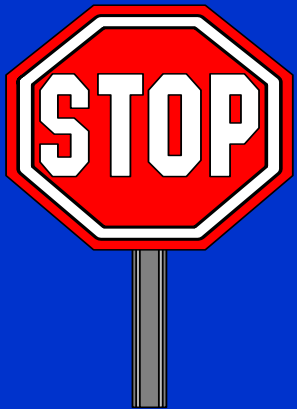


Figure 1. Example of a Brake Horsepower Curve





# Questions ?

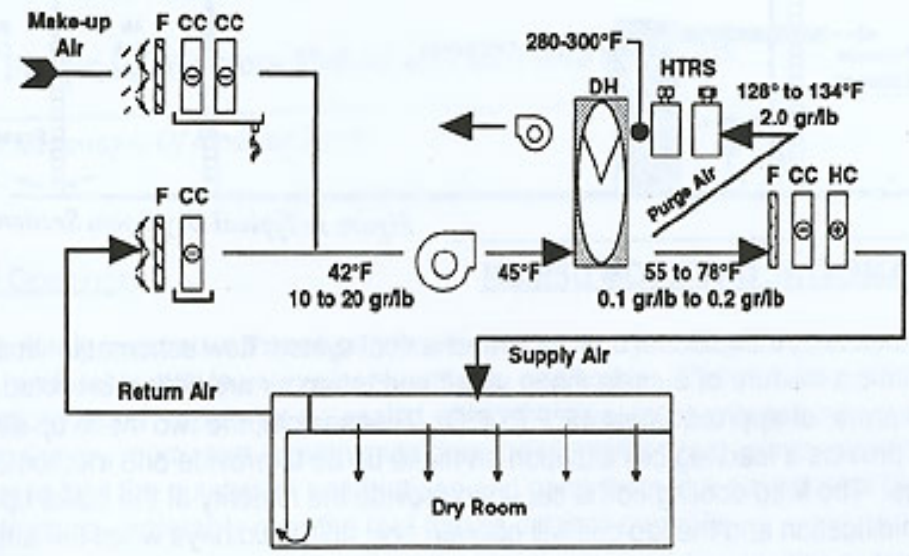
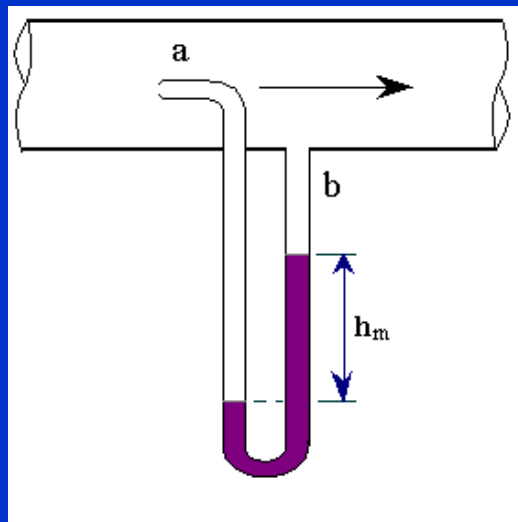
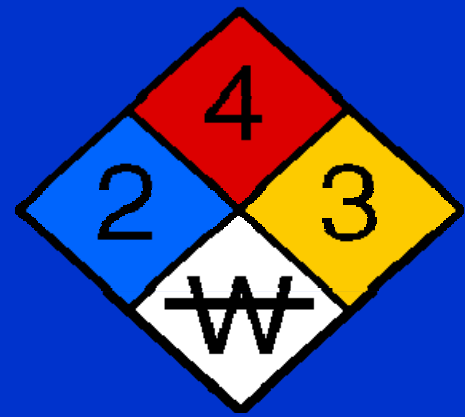


Figure 4- Typical Dry Room Mechanical System

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