The New Jersey Section of the American Industrial Hygiene Association

NJ AIHA CIH Review
Industrial Ventilation

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Course Objectives

At the end of this session, you should:

1. Be familiar with the industrial ventilation topics you may encounter on the ABIH CIH Exam
2. Identify your strengths & areas for improvement
3. Identify additional resources to prepare for the exam
4. Practice solving ‘typical’ ventilation questions

Properties of Air

Ventilation Reference Values

Temperature 70°F

For "Absolute" Rankine = °F+ 460

°C+ 273

Pressure 29.92 in Hg

1 inch Hg = 13.6 inch H_2O

Density 0.075 lbs/ft^3

MW ~ 29 (79% N, 20.9% O_2)

Other Reference Temperature & Pressure Values

Work Place and Ambient Air:

25° C and 760 mm Hg [77° F and 29.92 in Hg]

Stack Sampling (Air Pollution)

20 ° C and 760 mm Hg [68° F and 29.92 in Hg]

Pressure

1 psi = 27.7 in H_20 = 2.03 in Hg

1 in Hg = 13.6 in H_20

Principles

Air moves from higher to lower pressure gradients created by fans

<u>Assumptions:</u>

- Air assumed to be "incompressible" because exhaust system pressures < 3% (12 inches of water) above or below atmospheric pressure
- Airflow assumed to be turbulent, expressed as the Reynolds Number

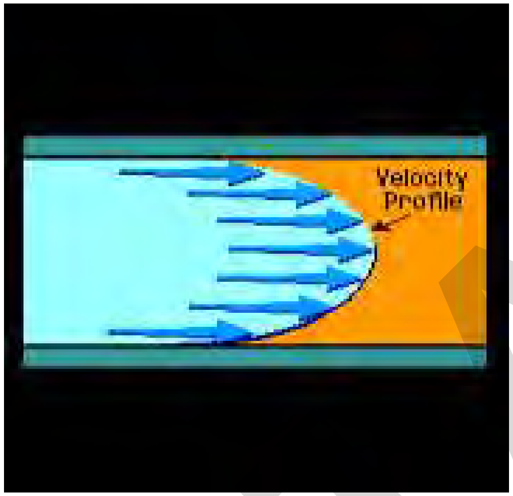
Reynolds Number (N_{Re})

A dimensionless parameter used to characterize fluid flow

Ratio of the inertial force which is causing gas movement to the viscous force which is restricting movement

Airflow Patterns in a Duct

Laminar Flow



Turbulent Flow

Classification of Gas Flow Types

Type Gas Flow	Laminar	Transitional	Turbulent
Reynolds Number	< 2,000	2,000 – 10,000	> 10,000

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Conservation of Energy – Bernoulli's Equation

Total Energy in = Total Energy out

- Units: (inches of water)

Two types of energy

- Potential: (Static Pressure SP)
- Kinetic: (Velocity Pressure VP)

Conservation of Energy

SP and VP are mutually convertible

- If velocity in a system decreases, then some of the kinetic (velocity pressure) energy is converted to potential (static pressure) energy and vice versa
- Conversion is never 100% because of system losses; however, the total pressure with the exception of these losses remains constant
- These pressures are compared relative to atmospheric pressure as either (+) or (-) and position WRT the fan

Total Pressure

Total pressure (TP) is the algebraic sum of the velocity pressure and static pressure (can be positive or negative depending upon position WRT fan)

Total Pressure (TP) = VP + SP

Static Pressure

Static Pressure: pressure exerted in all directions

- Example: Tire pressure
- Value can be negative or positive, depending upon position in the ventilation system
  - Positive on the exhaust side of the fan, and
  - Negative on the suction side of the fan

Velocity Pressure

Velocity pressure (VP) is directional pressure or kinetic energy resulting from air flow

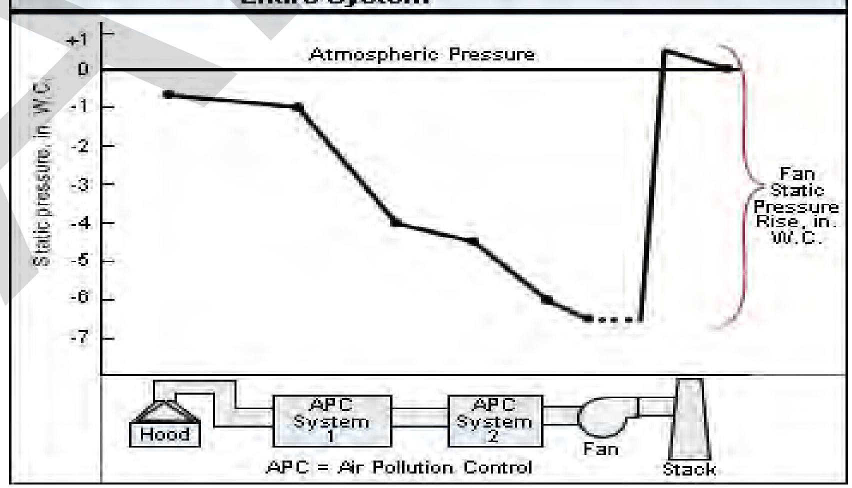
- Velocity pressure is always positive
- Example: Wind

Pressure Changes WRT Exhaust Fan

	Total Pressure	Static Pressure	Velocity Pressure
Suction Side of Fan	-	-	+
Exhaust Side of Fan	+	+	+

Illustration of static pressure drop downstream of an exhaust fan from a hood through two air pollution control systems. Upon discharge from the fan to the environment, static pressure changes from negative to positive value

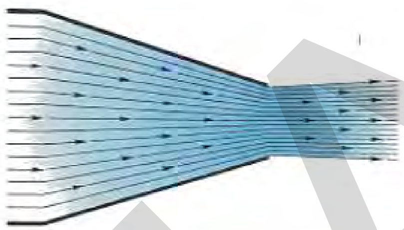
Figure 5. Static Pressure Drop and Rise Across Entire System



Conservation of Mass

Conservation of Mass

- Q = AV   Q = AV



- Air volume Flow Rate
  $V = 4005\ (VP)^{1/2}$

5 Components of a Local Exhaust System

1. Hood
   - Locate as close to source as possible
   - Provide the necessary capture and/or control velocities for contaminants of interest
2. Duct
   - Provide necessary transport velocity for contaminant
     - 3500 fpm for most industrial dusts
   - Ensure proper size to:
     - Maintain Q and transport velocity
     - Considers friction loss

Components of a Local Exhaust System

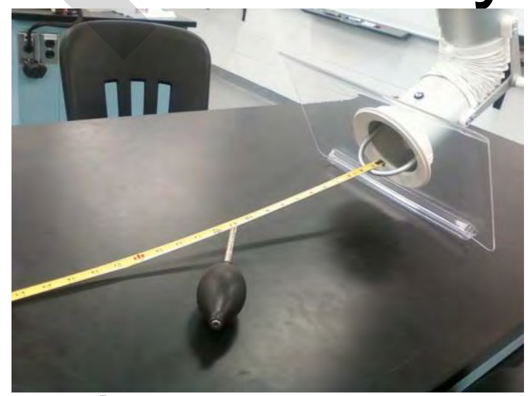
3. Air cleaner – Selection Criteria
   - Consider contaminant (effluent) characteristics
   - Determine desired contaminant removal and/or destruction efficiencies to achieve compliance criteria
   - Calculate pressure losses of each to the system
4. Fan – Selection Criteria
   - Achieve desired flow Q (cfm)
   - Determine required capture, control and duct transport velocities to counteract system resistance (i.e., SP losses) on the suction & exhaust side of the system

5 Components of a Local Exhaust System

5. Exhaust Stack
   - Must be high enough to avoid gas re-entrainment
   - Typically project 6 – 10 feet above the highest part of the roof
   - Must have vibration isolation

Hoods

Suction openings that provide points of entry into the local exhaust system



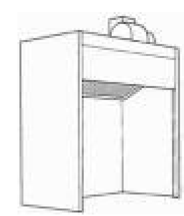
Three Hood Types:

1. Exterior hoods
   - Draws air away from the contaminant source but does not surround the source at the point of release
   - Round or rectangular openings

Hoods

2. Receiving Hoods – designed to take advantage of natural movement of contaminant from its source
   - Canopy hoods over hot processes
   - Capture hoods for grinders

3. Enclosing hoods – glove box provides complete enclosure, booth hoods have one side open for access



Hood Capture Velocity

The air velocity at any point in front of the hood or at the hood opening necessary to overcome opposing air currents and draw the contaminant into the hood

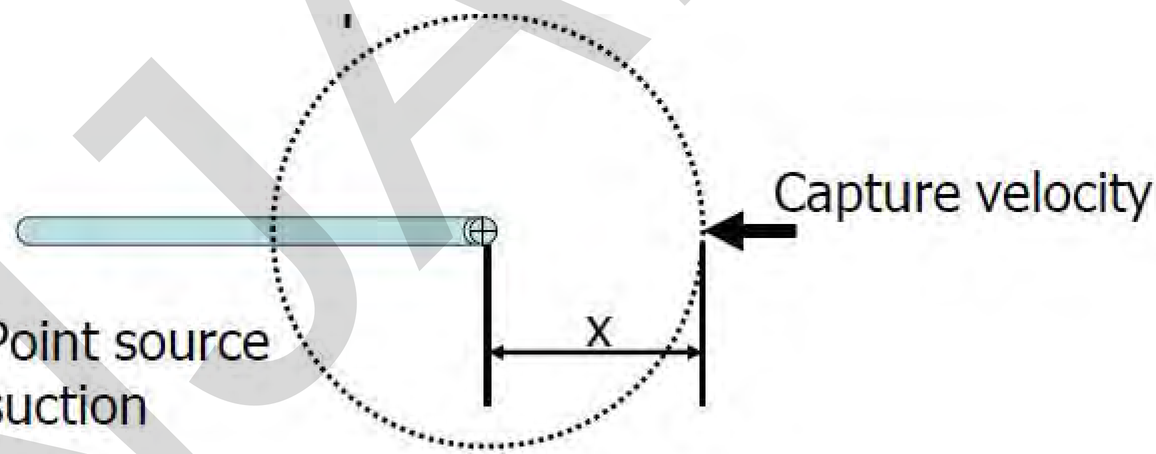
- Air drawn into a hood (all directions) by lower SP
- At approximately one-hood-diameter away from the hood entrance, the gas velocities are often <10% of the velocity at the hood entrance
- High terminal make-up air velocities at the hood opening necessitates higher capture velocities
- Suction is less efficient than blowing

Theoretical Point Source Suction

Required Airflow (CFM)

$Q = V\,(10X^2 + A)$

Area of Round Duct: $A = \pi(d^2/4)$



Chapter 3: ACGIH Industrial Ventilation Handbook

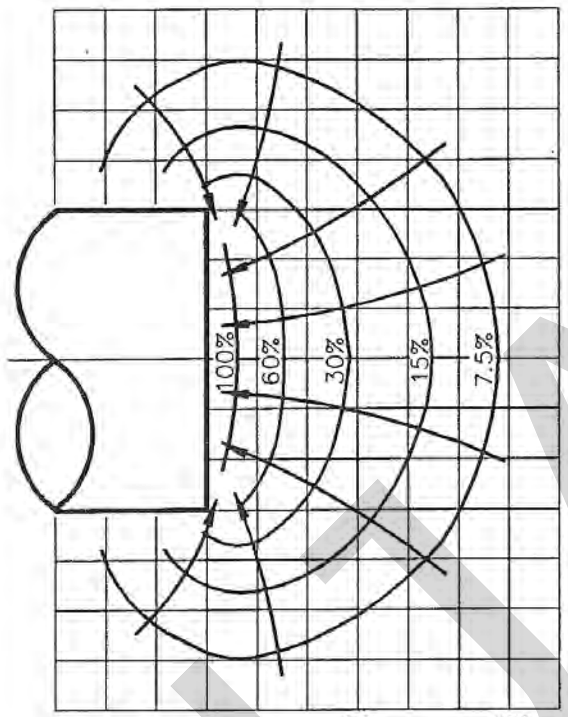
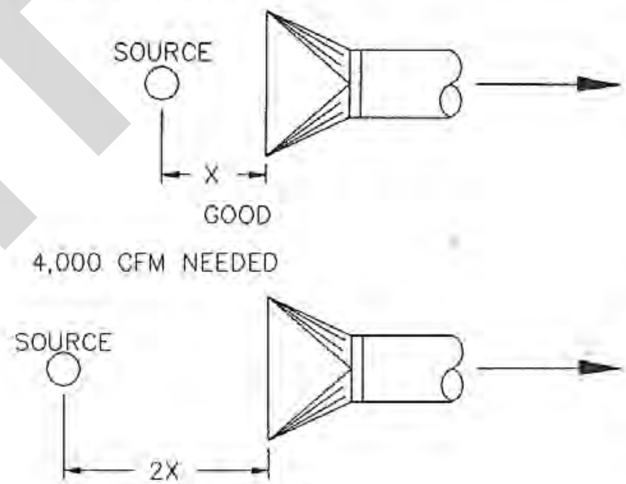


FIGURE 3–6 VELOCITY CONTOURS – PLAIN CIRCULAR OPENING – % OF OPENING VELOCITY

It can be seen from Equation 3.3 that velocity decreases inversely with the square of the distance from the hood (see



LOCATION

PLACE HOOD AS CLOSE TO THE SOURCE OF CONTAMINANT AS POSSIBLE. THE REQUIRED VOLUME VARIES WITH THE SQUARE OF THE DISTANCE FROM THE SOURCE.

FIGURE 3–5 FLOW RATE AS DISTANCE FROM HOOD

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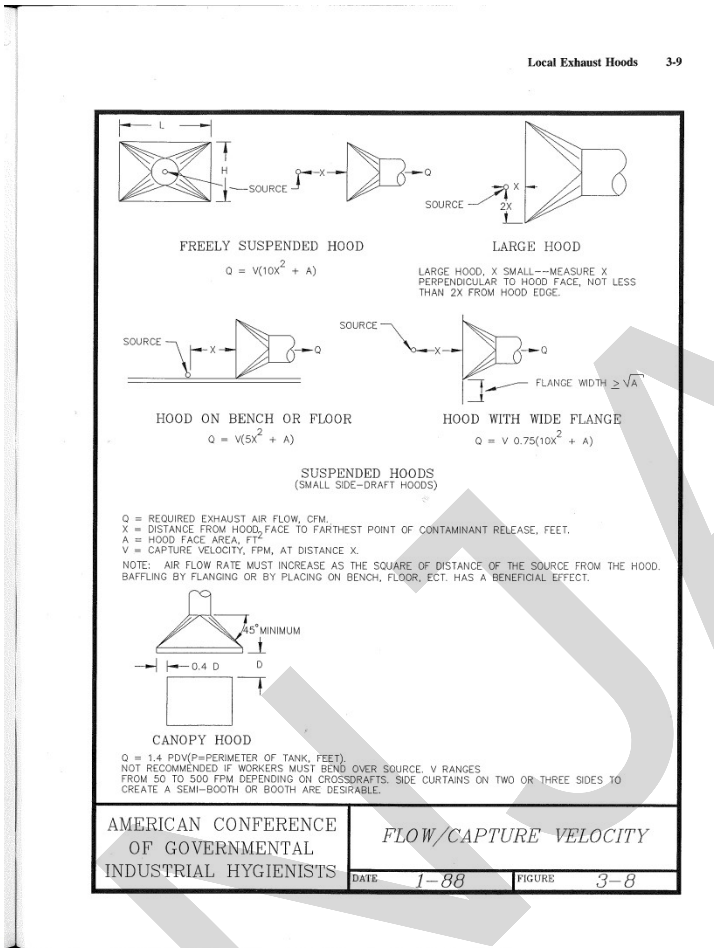

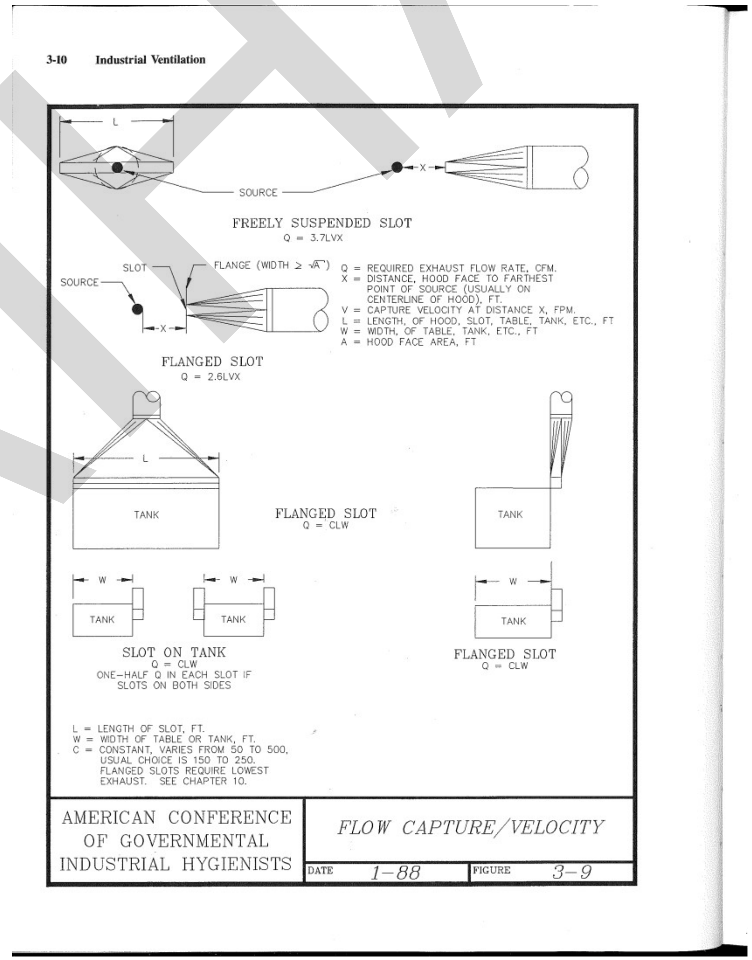

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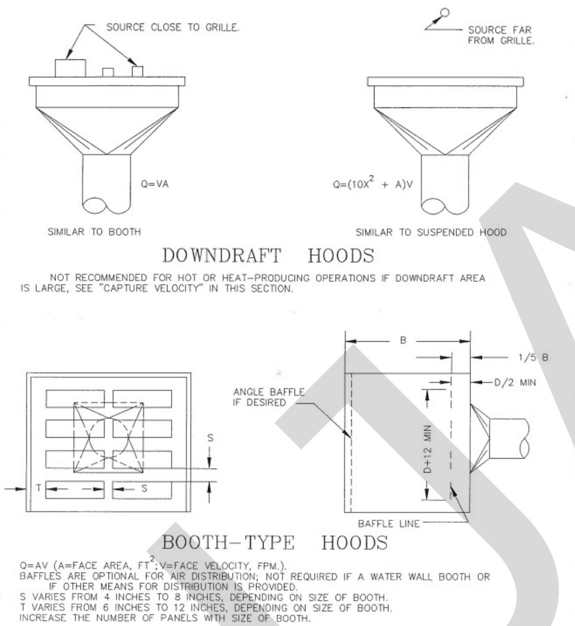
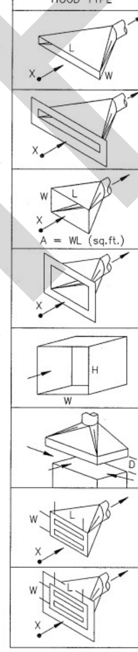


AMERICAN CONFERENCE OF GOVERNMENTAL INDUSTRIAL HYGIENISTS	FLOW/CAPTURE VELOCITY
	DATE 1-88 FIGURE 3-10



HOOD TYPE	DESCRIPTION	ASPECT RATIO, W/L	AIR FLOW
	SLOT	0.2 OR LESS	Q = 3.7 LVX
	FLANGED SLOT	0.2 OR LESS	Q = 2.6 LVX
A = WL (sq.ft.)	PLAIN OPENING	0.2 OR GREATER AND ROUND	$Q = V(10X^2+A)$
	FLANGED OPENING	0.2 OR GREATER AND ROUND	$Q = 0.75V(10X^2+A)$
	BOOTH	TO SUIT WORK	Q = VA = VWH
	CANOPY	TO SUIT WORK	Q = 1.4 PVD SEE VS-99-03 P = PERIMETER D = HEIGHT ABOVE WORK
	PLAIN MULTIPLE SLOT OPENING 2 OR MORE SLOTS	0.2 OR GREATER	$Q = V(10X^2+A)$
	FLANGED MULTIPLE SLOT OPENING 2 OR MORE SLOTS	0.2 OR GREATER	$Q = 0.75V(10X^2+A)$

AMERICAN CONFERENCE OF GOVERNMENTAL INDUSTRIAL HYGIENISTS	HOOD TYPES
	DATE 1-88 FIGURE 3-11

Capture Velocity Considerations

TABLE 3-1. Range of Capture Velocities[3.1,3.2]

Condition of Dispersion of Contaminant	Example	Capture Velocity, fpm
Released with practically no velocity into quiet air.	Evaporation from tanks; degreasing, etc.	50–100
Released at low velocity into moderately still air.	Spray booths; intermittent container filling; low speed conveyor transfers; welding; plating; pickling.	100–200
Active generation into zone of rapid air motion.	Spray painting in shallow booths; barrel filling; conveyor loading; crushers.	200–500
Released at high initial velocity into zone at very rapid air motion.	Grinding; abrasive blasting; tumbling	500–2000

In each category above, a range of capture velocity is shown. The proper choice of values depends on several factors:

Lower End of Range

1. Room air currents minimal or favorable to capture.
2. Contaminants of low toxicity or of nuisance value only.
3. Intermittent, low production.
4. Large hood-large air mass in motion.

Upper End of Range

1. Disturbing room air currents.
2. Contaminants of high toxicity.
3. High production, heavy use.
4. Small hood-local control only.

Source: Chapter 3, ACGIH Industrial Ventilation Handbook

Methods to Reduce Hood Airflow

Flanges: Surface placed parallel to hood face to prevent unwanted airflow from behind hood

Baffles: Surface placed to prevent unwanted airflow from the front or sides of a hood

Push-Pull systems

Transport Velocity

The minimum velocity that will transport captured particles in a duct with little settling

Recommended Transport Velocities

Pollutant Type	Recommended Transport Velocity (ft/min)
Gaseous gases	~1,000 – 2,000
Light particulate loading	~3,000 – 3,500
Normal particulate loading	~3,500 – 4,500

Losses of Energy or Pressure

Friction Loss: The static pressure loss in a system caused by the friction between moving air and the duct wall, expressed as:

Inches w.g./100 feet

Fractions of VP/100 feet of duct

- Varies directly with:

Duct length

Interior roughness of duct

- Square of velocity $(\text{velocity})^2$
- Varies inversely with duct diameter

Losses of Energy or Pressure

Dynamic Loss: A loss due to turbulence due to the change in direction or velocity in a duct or hood

- **Hood Entry Loss (h_e) –** turbulence losses due to air entry into a hood or duct
- **Coefficient of Entry (C_e) –** efficiency of conversion of hood SP to velocity pressure

Losses of Energy or Pressure

Acceleration Loss: A loss of energy required to accelerate air from zero velocity to a given velocity; this value is always positive and equal to 1 VP

Combination Loss: A loss occurring as a result of friction and dynamic losses that occurs in elbows, contractions, expansions, branch entries, and air cleaners

Hood Static Pressure (SP_h)

The static pressure required to accelerate air at rest into the hood and up to the duct velocity (1 VP) and the hood entry loss (Duct VP + h_e loss)

- Expressed in terms of SP:

  SP_h = VP (inches water) + he (inches water)

- Expressed in terms of VP:

  SP_h = acceleration loss + hood entry loss

  = 1 VP + Fd(VP)

  Where Fd(VP) is hood entry loss factor

Hood Entry Loss (h_e)

The loss of pressure (inches of water) when air enters a duct through a hood. Most hood entry loss is associated with a vena contracta formed in the duct, which varies with the opening type and is always expressed in absolute terms

$he = SP_h + VP_d$

Where:

SP_h = hood static pressure, inches W.C.

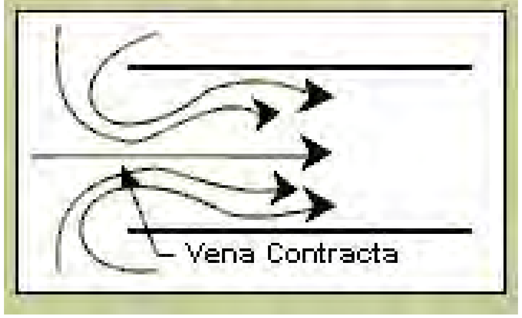
VP_h = hood velocity pressure, inches W.C.

h_e = Overall hood entry loss, inches W.C.

Vena Contracta

- An area of air convergence upon entering a duct
- As air passes through the vena contracta, its velocity increases; then the airflow expands and some of the velocity pressure converts to static pressure. The hood static pressure and entry loss are related to the size of the vena contracta.
- The hood geometry determines the size of the vena contracta by influencing how smoothly airflow enters the hood (circular vs. square)

Figure 2. Air Flow Convergence in a Duct



Coefficient of Entry (C_e)

A measure of the efficiency of a hood's ability to convert static pressure to velocity pressure; the ratio of actual to ideal flow

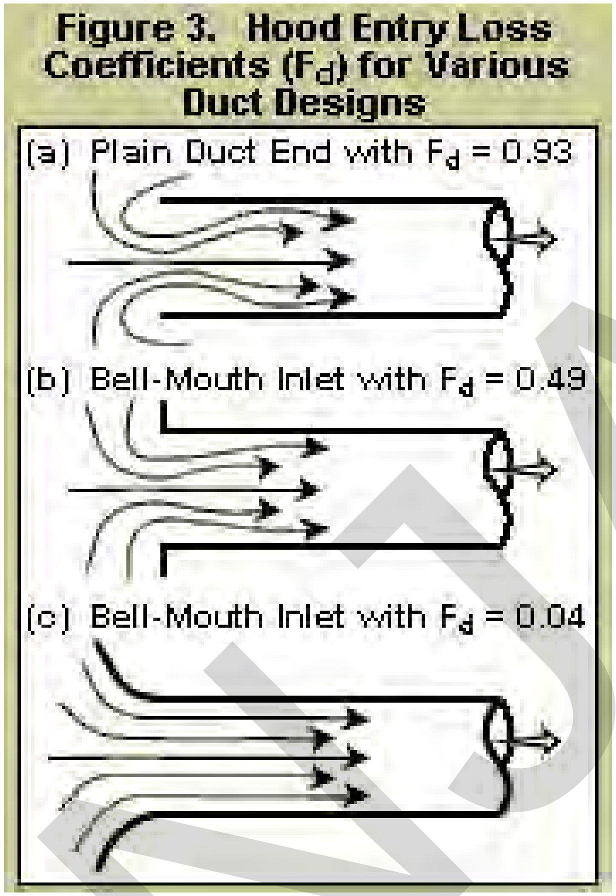
C_e = actual flow rate / theoretical flow rate

OR

$C_e = (VP)^{1/2} / (SP)^{1/2}$

All are expressed in positive terms (absolute value)

Hood Entry Loss Factors (F_d)* for Several Duct Designs



$$*h_e = F_d(VP)$$

Friction Losses Ducts and Elbows

Friction in straight ducts

- Round ducts preferred
- Rectangular ducts use equivalent diameter

Elbows have friction & turbulence loss

- Long radius elbows have less loss

Turbulence occurs in contractions and expansions

Friction Losses Ducts and Elbows

To calculate air flow pressure drop in typical HVAC ducting, use friction charts

- Chart shows the friction pressure loss in terms of inches of water/100 feet of duct equivalent

For elbows, use charts that express losses or resistance as a fraction of VP or to equivalent length of straight duct

Other Losses in System Design

- Branch duct entry
- Air cleaners
- Stacks

Use reference data when calculating these system losses

Static Pressure Measurements

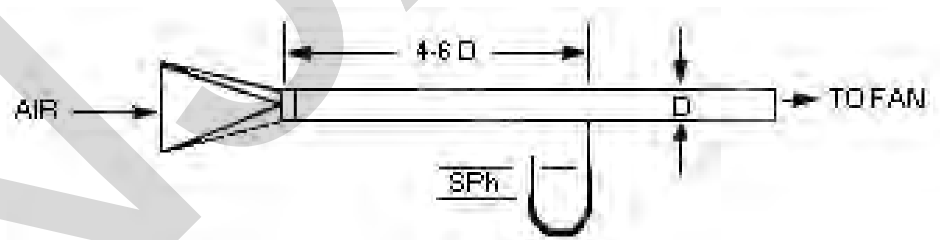
U Tube manometer

- Simplest; usually filled with water or oil

Magnehelic gauge – aneroid gauge

Measurement locations

- Hoods: 3 duct diameters downstream from plain or flanged hood
- Ducts: Location where airflow parallel to duct wall



Velocity Pressure Measurement

Pitot Tube: Consists of two concentric tubes

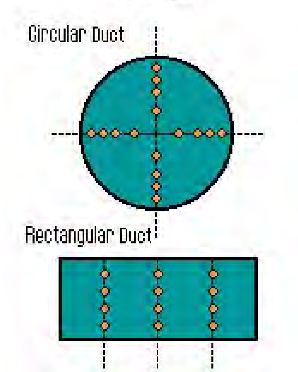
- One detects Total Pressure (TP)
- The other detects Static Pressure (SP)
- Generally not used at velocities < 600 FPM (0.02 inches water VP)
- Standard type does not require calibration, but S-type does

Velocity Pressure Measurement

Pitot-Traverse used for an accurate velocity profile

- Location must have uniform flow
  - At least 7.5 duct diameters downstream from major turbulence
  - 2.5 duct diameters upstream from disturbance or stack
- For round ducts, concentric areas of *equal area* with a specific number of points
  - Ducts < 6” diameter, 6 points per traverse
  - Ducts 6 – 48” diameter, 10 points per traverse
  - Ducts > 48” diameter, 20 points per traverse

Figure 5. Measurement Points in a Duct



Pitot Tube Measurement

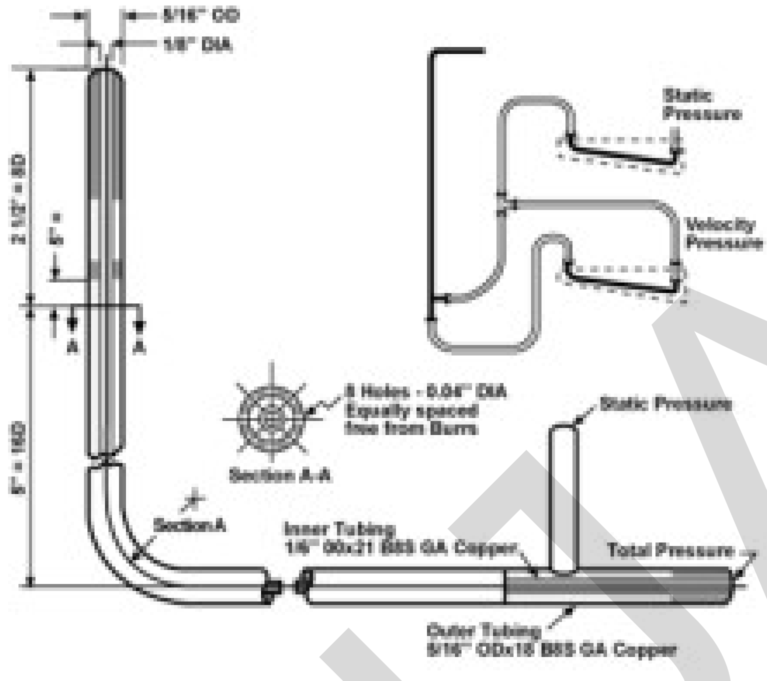
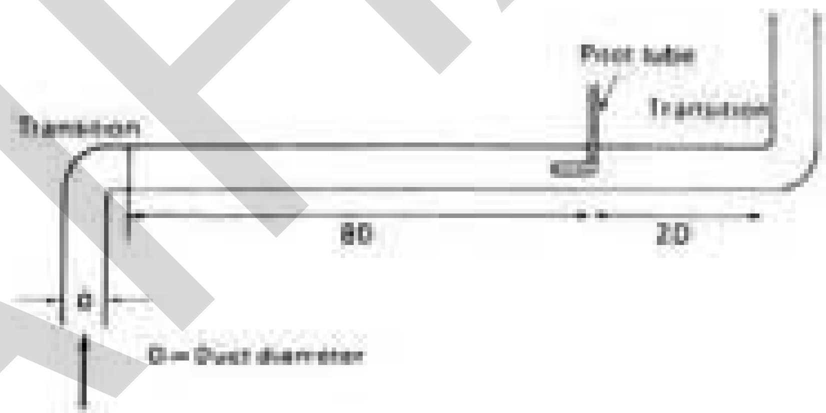


Figure 2.15 - The Pitot Tube



Ideal Location for
Pitot Tube Measurement

Velocity Pressure Measurement

Swinging Vane Anemometer

- Different scales, up to 10,000 fpm
- Frequent calibration required, not good with high dust loading

Hot wire thermoanemometer

- 10 – 8,000 fpm
- Fragile; do not use in dusty or combustible environments

Rotating Vane

- 200 – 2,000 fpm
- Frequent calibration required; not good with high dust loading

Fans

Must provide required flow and static pressure

Fan Types

1. Axial (propeller and vane axial fans)
   - Flow parallel to fan propeller shaft (axis of rotation)
   - Used for general or dilution ventilation for clean air, not local exhaust
   - Relatively high noise levels
   - Vane axial fans used in mining, tunnels, small areas
2. Centrifugal
   - Airflow perpendicular to axis of rotation; commonly used for local exhaust ventilation

Types of Centrifugal Fans

Classified by impeller blade shape

1. Radial
   - Most common; least sensitive to particulate laden gas streams
   - Can handle high static pressures; medium noise
2. Backward curve
   - For low particulate loads; higher efficiency for fluctuating pressures but accumulates solids; quiet operation
3. Forward curve
   - Used for heating & AC systems
   - Minimal use in industrial ventilation

Fan Law Basics

Static pressure rise across fan increases rapidly as fan speed is increased

Relationship between gas flow rate (Q) and fan motor current is not linear; the non-linear characteristic of the relationship is indicated by the brake horsepower curve

The fan motor current is directly proportional to the brake horsepower

Fan Law #1

1. As the speed increases (rpm), the airflow rate increases (Q)

$$Q_2 = Q_1\ (rpm_2/rpm_1)$$

<u>Where:</u>

Q_1 = baseline airflow rate, CFM

Q_2 = new airflow rate, CFM

Rpm_1 = baseline fan wheel rotational speed, revolutions per minute

Rpm_2 = new fan wheel rotational speed, revolutions per minute

Fan Law #2

2. Fan static pressure is proportional to rpm^2

$$\text{Fan } SP_2 = \text{Fan } SP_1 \, (rpm_2/rpm_1)^2$$

Where:

Fan SP_1 = baseline fan static pressure, in. W.C.

Fan SP_2 = new fan static pressure, in. W.C.

Rpm_1 = baseline fan wheel rotational speed, revolutions per minute

Rpm_2 = new fan wheel rotational speed, revolutions per minute

Fan Static Pressure

The air stream moving through the fan experiences a static pressure rise due to the mechanical energy expended

Static pressure at the outlet is always higher than the static pressure at the inlet

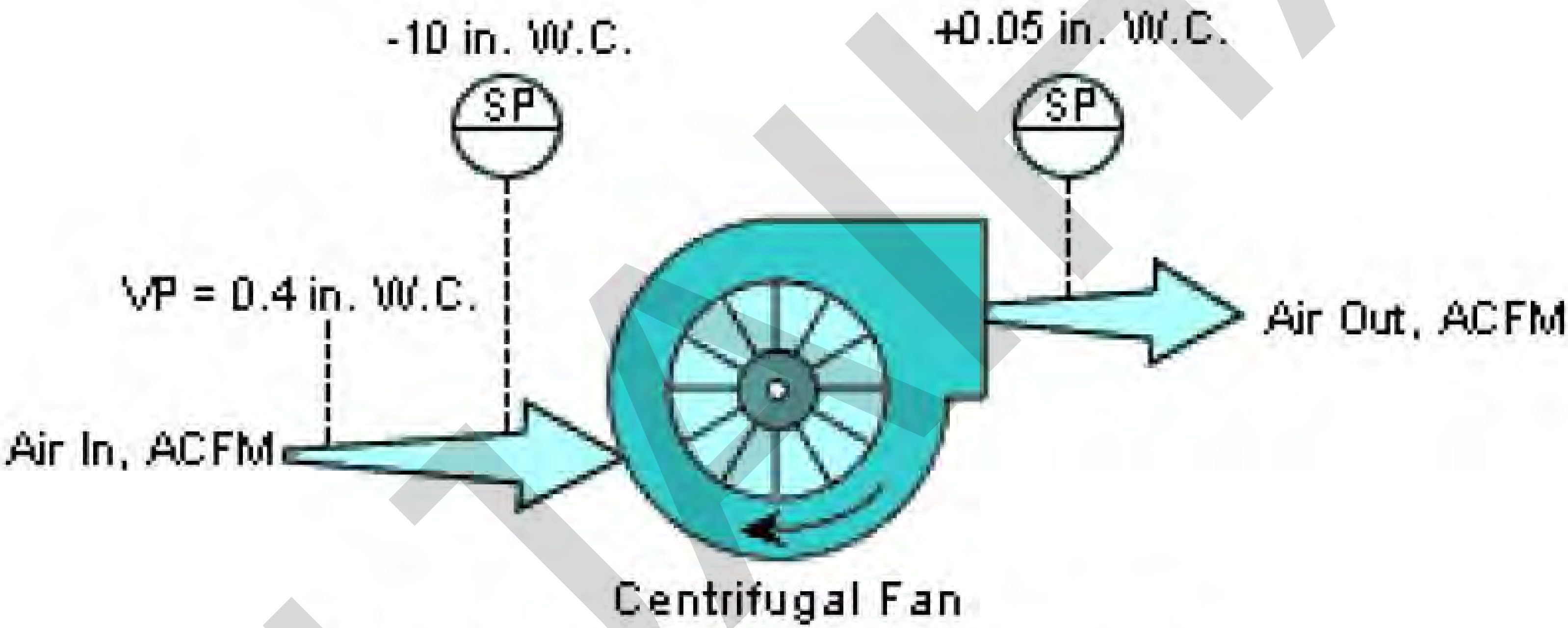
$$\text{Fan } \Delta SP_2 = SP_{fan\ outlet} - SP_{fan\ inlet} - Vp_{fan\ inlet}$$

$SP_{fan\ outlet}$ = static pressure at fan outlet, in W.C.

$SP_{fan\ inlet}$ = static pressure at fan inlet, in W.C.

$Vp_{fan\ inlet}$ = velocity pressure at fan outlet, in W.C.

Static Pressure Rise Across a Fan



Fan ΔSP = [0.05 - (-10) - 0.4] in. W.C.
= 9.65 in. W.C.

Brake Horsepower (bhp)

The actual horsepower required to move air through a ventilation system against a fixed total pressure plus losses in the fan

$$\text{Fan bhp} = \text{ahp} * (1/\text{efficiency})$$

Where:

ahp = air horsepower or theoretical hp required to drive a fan with no losses (100% efficient)

eff = fan mechanical efficiency

Brake Horsepower

$$BHP = I * E * 1.73 * Eff * P.F./746$$

Where:

BHP = Brake horsepower (total power consumed by fan)

I = Fan motor current, amperes

E = Voltage applied, volts

Eff = Efficiency, expressed as a decimal

P.F. = Power factor (phase relationship between current and voltage waveforms)

Brake Horsepower

$$BHP = Q(TP)/(6362)ME$$

Where:

Q = flow (cfm)

TP = total pressure

ME = mechanical advantage of fan (usually 0.5 – 0.65)

Fan Law #3

3. Brake horsepower is related to the cube of the fan speed

$$BHP_2 = BHP_1 (rpm_2/rpm_1)^3$$

<u>Where:</u>

Fan BHP_1 = baseline brake horsepower

Fan BHP_2 = new brake horsepower

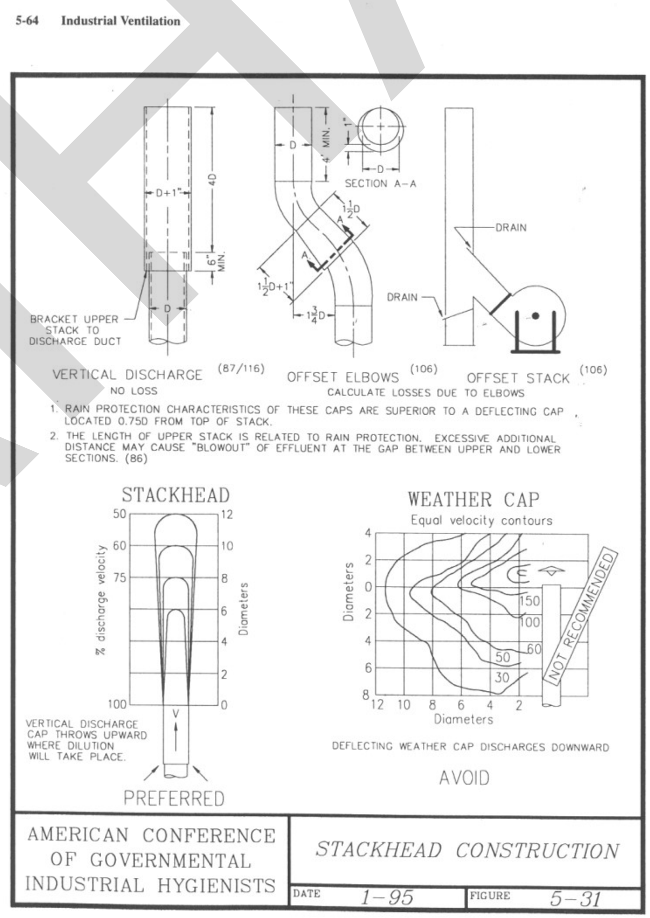
Rpm_1 = baseline fan wheel rotational speed, revolutions per minute

Rpm_2 = new fan wheel rotational speed, revolutions per minute

Exhaust Stack Design

Considerations:

- Actual vs. Effective height
- Resistance (SP)
- Prevailing wind speed & direction (wind rose)
- Roof obstructions
- ASHRAE Fundamentals Chapter 16



ASHRAE Airflow Around Buildings

Airflow around buildings affects:
- Safety
- Processes & Operations
- Building environmental factors

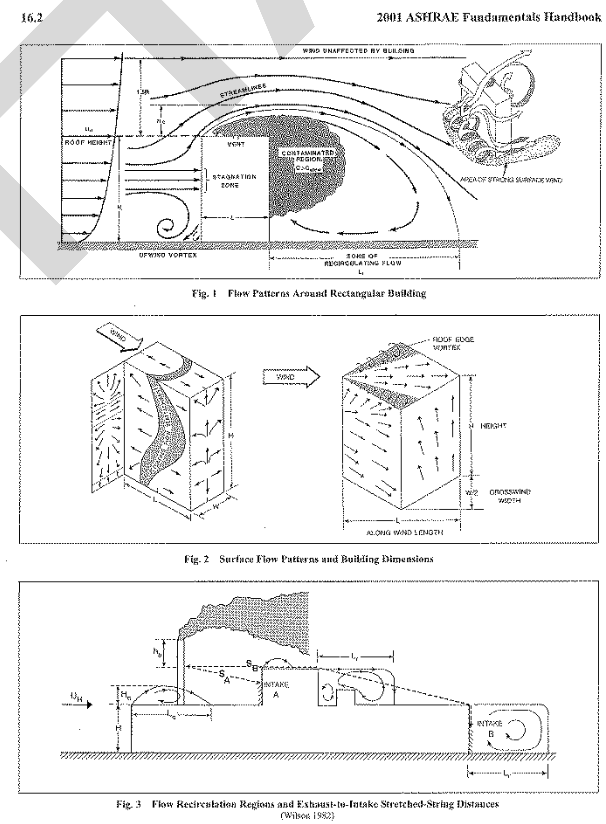
Scaling Height (R)

$$R = B_s 0.^{67} B_l^{0.33}$$

Where:

B_s = smaller of upwind building dimensions

B_l = larger of upwind building dimensions



General Ventilation

Dilution ventilation is a form of exposure control that involves providing enough air in the workplace to dilute the concentration of airborne contaminants to acceptable levels

Dilution Ventilation

Appropriate when:

- Emission sources contain materials of relatively low hazard, as assessed by the toxicity, dose rate, and low individual susceptibility
- Emission sources are primarily vapors or gases or small, respirable-size aerosols not likely to settle
- Emissions occur uniformly
- Emissions are widely dispersed

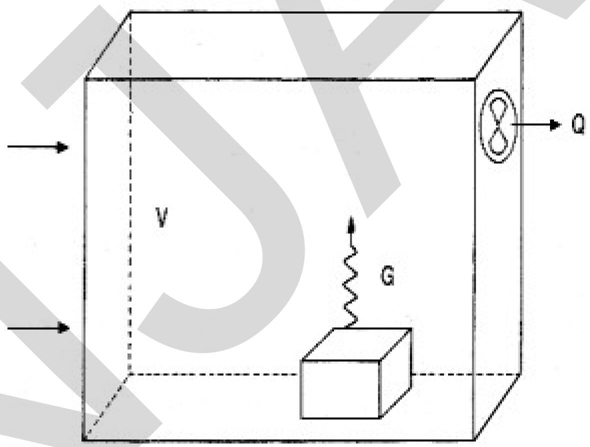
Dilution Ventilation

A simple case assumes:

– Perfect mixing of air in the room

– Constant generation rate of contaminant(s)

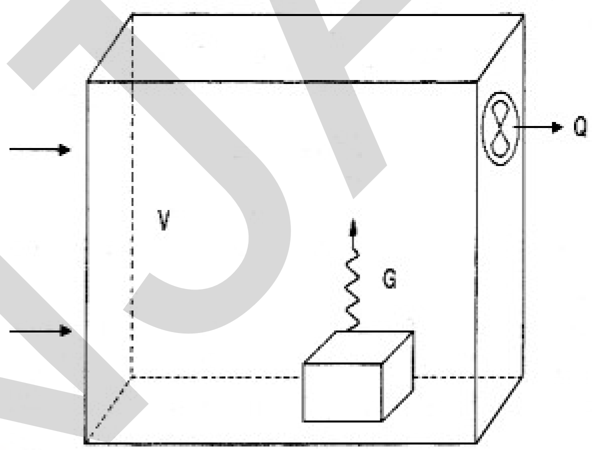
Example: Dip tank emissions



General Exhaust Ventilation

Box Model

- C = concentration as a function of time (ppm)
- G = contaminant generation rate (constant)
- Q = ventilation rate (cfm)
- V = room volume (ft^3)



General Exhaust Ventilation

To estimate the flowrate needed to achieve TLV assuming constant emission and constant room ventilation rates

$$Q = \frac{403 * SG * ER * K * 10^6}{MW * TLV}$$

Where:

C = contaminant concentration (ppm)

ER = emission rate (pints/minute)

MW = molecular weight (gm/gm-mol)

TLV = desired value and/or TLV (ppm)

SG = specific gravity

K = mixing factor (3 – 10)

Q = ventilation rate (cfm)

General Exhaust Ventilation

To calculate for transient concentrations (releases):

Purging: Decreases from an initial concentration with zero generation

$$C_t = C_0 * e^{-tQ/KV}$$

OR

$$C_t = C_0 * e^{-t\,N_{changes}}$$

Where:

T = time (minutes)

K = mixing factor

$N_{changes}$ = number of air changes per hour

Q = ventilation rate (cfm)

V = room volume (ft^3)

Purge Example

Find the purge time required to safely enter a room 12.5' x 10' x 8' with an initial contaminant concentration of 3000 ppm. Assume the ventilation rate is 500 cfm, a mixing factor of 3, the TLV is 100 ppm.

Answer: 20.41 minutes

General Exhaust Ventilation

Accumulating example – concentrations increase from initial with a generation rate > 0

$$C = G/Q(1-e^{Nt/60}) * 10^6$$

Or

$$C = G/Q(1-e^{Qt/V}) * 10^6$$

Where:

C = contaminant concentration (ppm)

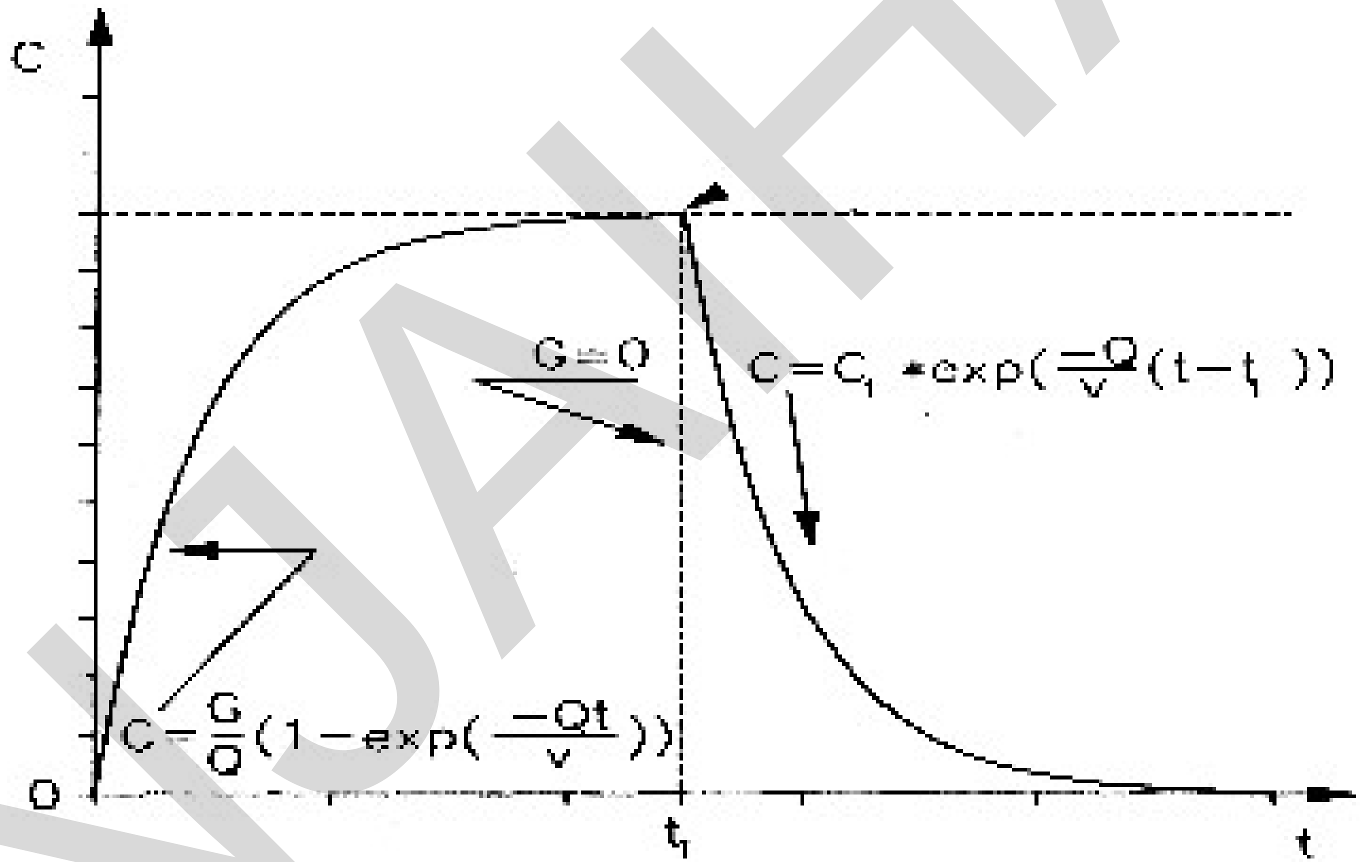
N = number of air changes per hour

G = vapor generation rate (mg/min)

T = time (minutes)

Q = ventilation rate (cfm) ** effective (Q/K)

Accumulation and Purge Combined



Biological Safety Cabinets (BSC’s)

Serve as the primary containment device designed to provide personnel, environmental, and product protection when following appropriate work practices and procedures

BSC’s use HEPA filtered supply (not class I) and exhaust systems

Three biological safety cabinet classes:

- Class I
- Class II
- Class III

Class I Biosafety Cabinets

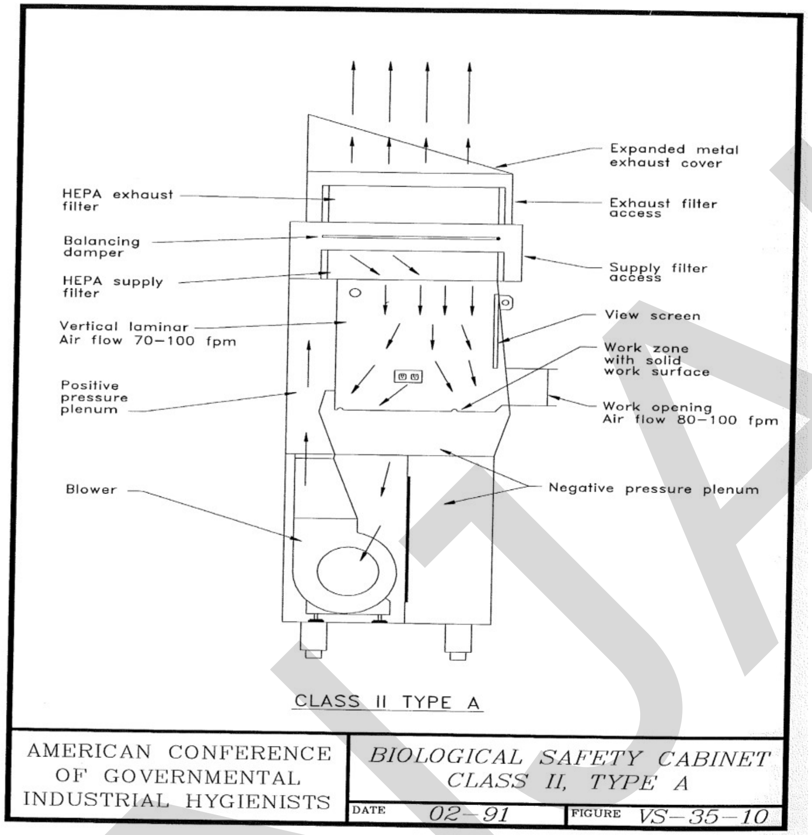
Draws unfiltered air in through the opening.

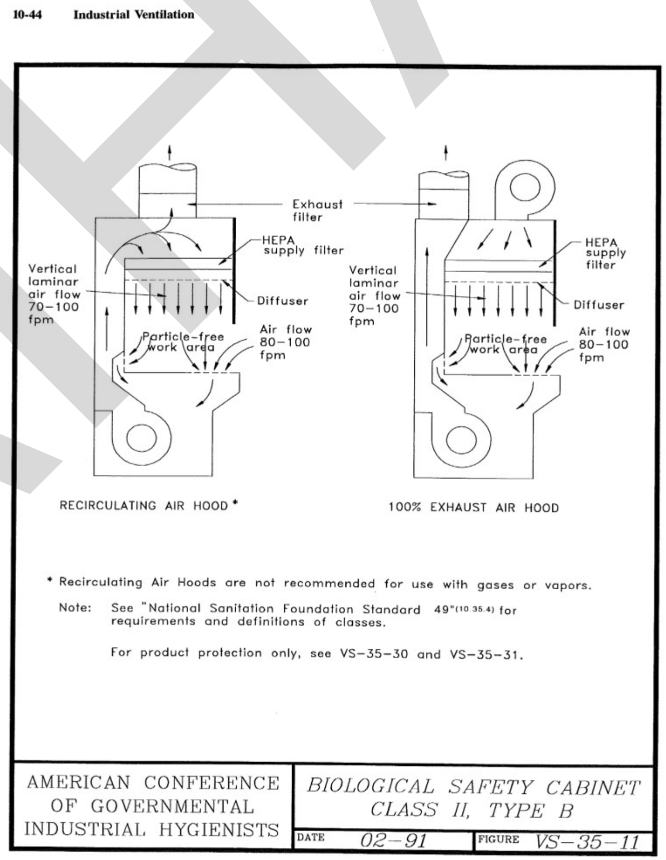
- Provides no product protection
- Used to enclose aerosol generating procedures or equipment or animal cage changing
- Often hard ducted

Class II Biosafety Cabinets

- Airflow drawn into a HEPA filter along front grill of unit to provide product protection
- Unidirectional air moving parallel to the opening provides personal protection; however, disruptions in air currents increase the potential for contaminant release
- Exhaust HEPA filtered, can be discharged to the laboratory or to a ‘thimble’ exhaust connection
- Subdivided further into classes ‘A’ or ‘B’; see NIOSH BMBL

Class II Biological Safety Cabinets

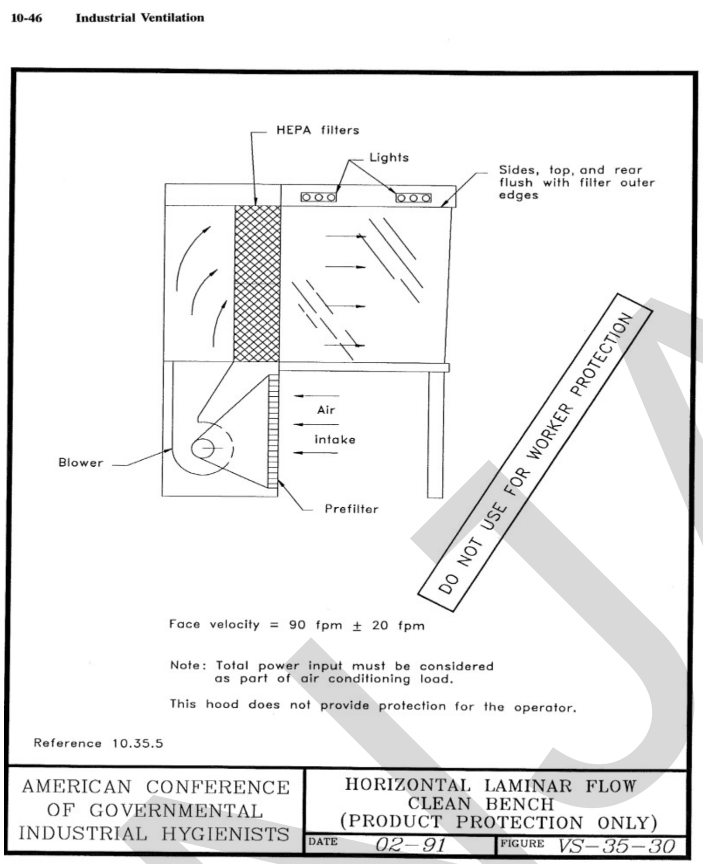


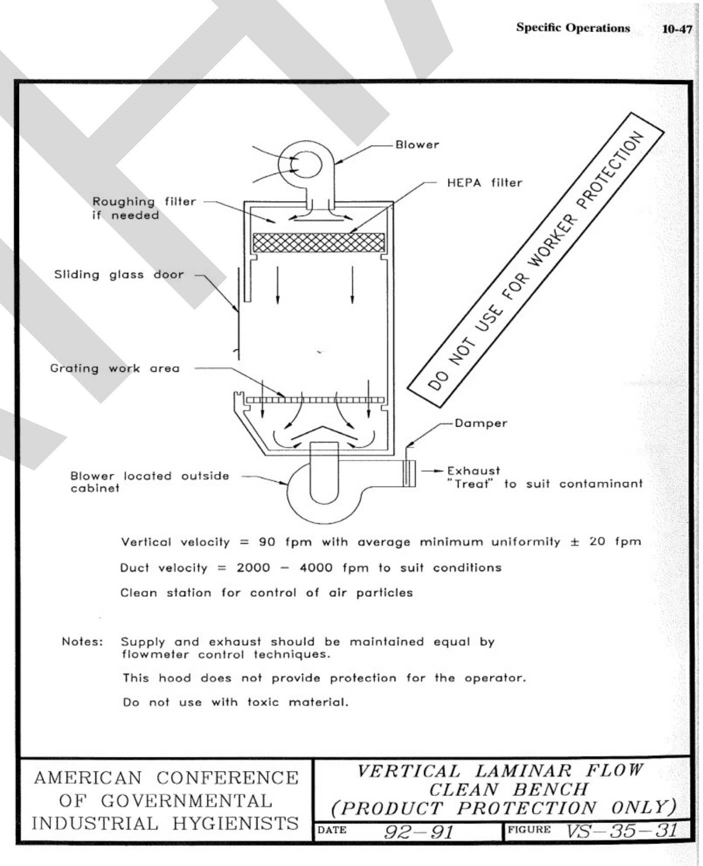


Class III Biosafety Cabinet

- Designed for work with highly infectious agents; provides maximum worker and environmental protection
- Material access provided via a dunk tank or double door pass through box
- Supply and exhaust air HEPA filtered
- Long, heavy duty rubber gloves into the cabinet enable manipulation of isolated materials inside.

Laminar Flow Hoods (Non Protective)





Fume Hood Acceptance Testing

ASHRAE 110 – 1995 establishes a procedure to assess containment (leakage rate) of laboratory fume hoods

- Assumes proper laboratory design (low terminal supply air velocity at hood opening, hoods installed away from windows & heavy traffic areas)
- Establishes qualitative & quantitative criteria to assess hood performance
  - Qualitative: Smoke reflux at opening (small & large source)
  - Quantitative: Face velocity, tracer gas concentration at hood opening
- Considers sash movement effect for VAV exhaust systems

ASHRAE Acceptance Criteria

Typical Acceptance Criteria expressed as:

4.0 AM/AI/AU 0.05

Where:

AM = as manufactured

AI = as installed

AU = as used

4.0 = release rate of SF_6, liters/minute

0.05 = acceptable control (leakage) level, ppm

Additional References

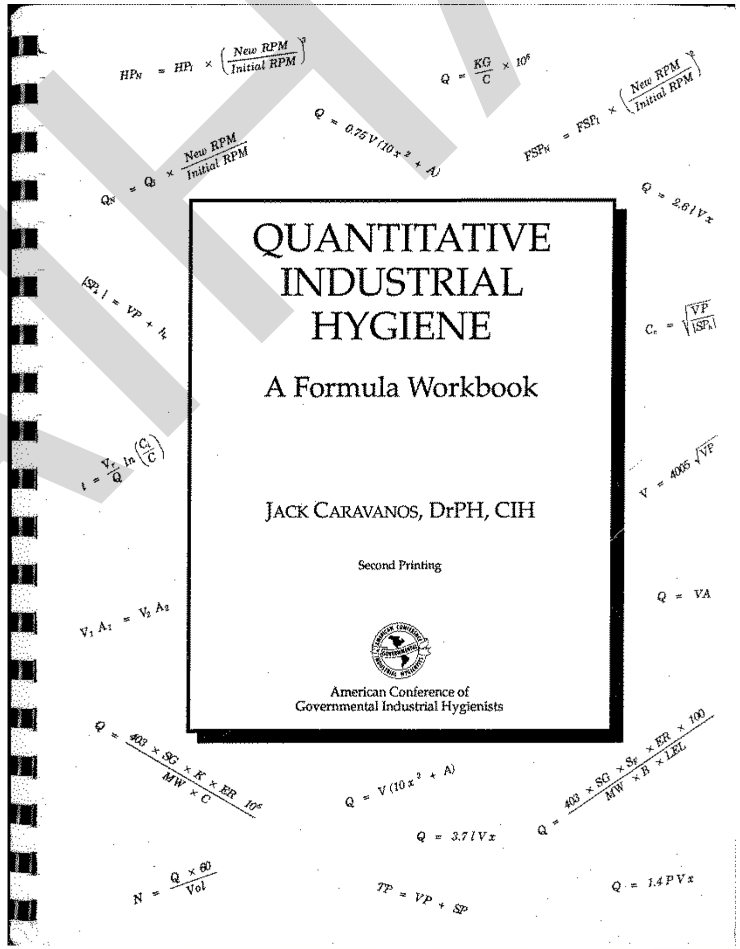
ACGIH. Industrial Ventilation: A Manual of Recommended Practice

ASHRAE 110 – 1995 Fume Hood Test Method

National Science Foundation Standard 49 (Biosafety Cabinet)

Biosafety in Microbiological and Biomedical Laboratories – CDC

ABIH website – equation sheet http://www.abih.org/sites/default/files/downloads/Equation%20Sheet%20Vent%20Plates%202018.pdf



Part 2: Questions & Practice Problems

Question #1

A velocity traverse should be made at what point of how many duct diameters down stream and upstream respectively from major disturbances?

a. 7.5 and 2.5
b. 2.5 and 7.5
c. 8.5 and 1.5
d. 1.5 and 8.5
e. Anywhere in the duct

Answer, Question #1

A velocity traverse should be made at what point of how many duct diameters down stream and upstream respectively from major disturbances?

Answer: a, 7.5 duct diameters down stream and 2.5 upstream from major disturbances

Question #2

The minimum distance from the throat of a flanged duct end that hood static pressure is measured is:

a. 1 duct diameter
b. 2 duct diameters
c. 3 duct diameters
d. 10 duct diameters
e. 7 duct diameters

Answer, Question #2

The minimum distance from the throat of a flanged duct end that hood static pressure is measured is:

Answer: 3 duct diameters

Question #3

The fan with the medium tip speed that can be used with a heavy dust load passing through is:

a. Forward-curved blade centrifugal
b. Backward blade centrifugal
c. Vane axial fan
d. Disc type axial
e. Radial blade centrifugal

Answer, Question #3

The fan with the medium tip speed that can be used with a heavy dust load passing through is:

Answer: e, Radial blade centrifugal

Question #4

If fan speed is increased with constant air density, the static pressure:

a. Varies as fan speed
b. Varies as the square root of fan speed
c. Varies as the square of fan speed
d. Varies as the cube of fan speed
e. None of the above

Answer, Question #4

If fan speed is increased with constant air density, the static pressure:

Answer: c, Varies as the square of fan speed

Question #5

If fan size is increased and the tip speed remains constant, the pressure:

a. Increases
b. Decreases
c. Remains the same
d. Pressure not applicable
e. Converts to heat

Answer, Question #5

If fan size is increased and the tip speed remains constant, the pressure:

Answer: c, remains the same

Question #6

The most important factor in the production of noise by fans is:

a. Characteristic of gas
b. Size of fan motor
c. Blade tip speed
d. Type of fan housing material
e. Characteristics of particulates

Answer, Question #6

The most important factor in the production of noise by fans is:

Answer: c, blade tip speed

Problem #1

In a processing room of a plasticizer plant, two pints of a solvent are evaporated per 8 hour shift. What volume of dilution air is required per shift to maintain the general air concentration at the TLV of 200 ppm. Assume SG of 0.87, K factor of 4, MW of 92.

a. 15,000 ft^3/shift
b. 32,541 ft^3/shift
c. 172,840 ft^3/shift
d. 152,439 ft^3/shift
e. 232,524 ft^3/shift

Answer to Problem #1

d, 152,439 ft^3/shift

$$Q = [(403 * SG * 1x10^6 * K)/(MW * TLV)] * ER$$

$$= [(403 * 0.87 * 1x10^6 * 4)/(92 * 200)] * 2$$

$$= 152,439\ ft^3/shift$$

Note: Typically we use an evaporation rate of pints per minute, so if in an hour time duration divide by 60. Here, the units remained per shift, so multiply by 2. WATCH YOUR UNITS!

Problem #2

Room air at 70° F enters at a rate of 100 CFM in each of 10 enclosures where enamel frit is fused. The air temperature rises to 500 ° F before leaving the enclosures. What volume of air must the fan handle?

a. 1811 CFM
b. 181 CFM
c. 1000 CFM
d. 552.1 CFM
e. 7142.8 CFM

Answer to Problem #2

a, 1811 CFM air volume the fan must handle

Notes:

10 enclosures @ 100 CFM each (1000 CFM)

The temperature increases to 500 ° F

Temperature must be in units of Rankine (°F + 460)

T_2/T_1 = (500 + 460) / (70 + 460) = 1.811 * 1000 CFM

= 1811 CFM required for the fan

Problem #3

A 10 point pitot tube traverse for air at 70 ° F and 1 atmosphere in a 12” circular duct yielded a flow rate of 1444 CFM. If the Ce at the inlet of this duct is 0.69, what is the static pressure of this plain end round duct?

a. 0.21
b. 0.11
c. 0.69
d. 0.44
e. 0.30

Answer to Problem #3

d, 0.44

$Q = V*A$ $1444 = V*(0.785)$; $V = 1839$ fpm

$V = 4005\,(VP)^{.5}$ or $1839/4005 = (VP)^{.5}$ $VP = 0.21$

$C_e = (VP/SP)^{.5}$ $0.69 = (0.21/SP)^{.5}$

$0.476 = (0.21/SP)$, $SP = 0.441$ inches

Problem #4

An air velocity measurement made in a 8 inch diameter round duct was 1266 fpm. What was the velocity pressure?

a. 1.0 in. H_20
b. 0.1 in. H_20
c. 442 CFM
d. 63641 CFM
e. 2.0 in. H_20

Answer Problem #4

b, 0.1 inches H_20

$V = 4005 (VP)^{.5}$ or $1266/4005 = (VP)^{.5}$

$0.316 = (VP)^{.5}$, $VP = 0.099$ or 0.1

Problem #5

1000 CFM is to be exhausted through a flanged hood 4 inches by 7 inches. What is the expected capture velocity at 6 inches at the centerline in front of the hood?

a. 495 fpm
b. 333 fpm
c. 250 fpm
d. 560 fpm
e. 1000 fpm

Answer, Problem #5

a, 495 fpm

$$Q = 0.75V * [10X^2 + A]$$

$$1000 = 0.75\ V * [10(.5^2) + 0.19]$$

$$1333 = V\ (2.69)$$

$$495 = V$$

Problem #6

A hood is found to have a coefficient of entry, Ce = 0.65. If the duct area is 0.2 ft^2, and 800 CFM flows through the hood, what is the static pressure just upstream of the hood?

a. -1.65 inches H2O
b. -0.65 inches H2O
c. -1.54 inches H2O
d. -2.36 inches H2O
e. -1.00 inches H20

Answer to Problem #6

d, -2.36 inches H2O

$Q = V * A$ or $800 = V * 0.2$; $V = 4000$ fpm

$V = 4005 (VP)^{.5}$; $4000/4005 = (VP)^{.5}$

$VP = 0.997$ or 1

$C_e = [VP/SP]^{.5}$ or $0.65 = [1/SP]^{.5}$

$SP = - 2.36$

Problem #7

A hood 6 inches in diameter (area 0.196 ft^2) has a hood entry loss factor of 0.50 VP with a hood static pressure of 2 inches water. The air velocity in the hood is:

a. 1519 fpm
b. 2143 fpm
c. 4619 fpm
d. 5231 fpm
e. 6417 fpm

Answer to Problem #7

c, 4619 fpm

$SP = VP + h_e$ or $2 = VP + 0.5VP$

$2 = 1.5\ VP$; $VP = 1.33$

$V = 4005\ (VP)^{.5}$ or $4005 * (1.33\ )^{.5}$

$V = 4619$ fpm

Problem #8

The coefficient of entry for the hood described above is:

a. 0.78
b. 0.62
c. 0.98
d. 0.54
e. 0.82

Answer to Problem #8

e, 0.82

$C_e = (VP/SP)^{.5}$ or $(1.3/2)^{.5}$

$C_e = 0.82$

Problem #9

Static pressure immediately downstream from a 24 inch wheel grinder was measured to be 1.69 inches of water. The duct diameter was 6 inches and the hood had a coefficient of entry of 0.78. What air flow (rounded off) would be obtained?

a. 400 CFM
b. 800 CFM
c. 1200 CFM
d. 1600 CFM
e. 2000 CFM

Answer to Problem #9

b,   800 CFM

$C_e = (VP/SP)^{.5}$ or $0.78 = (VP/1.69)^{.5}$

$0.608 = VP/1.69$; $VP = 1.03$

$V = 4005\ (VP)^{.5}$; $4005\ (1.03)^{.5}$; $V = 4061$ fpm

$Q = V * A$ or $[4061 * 0.196] = 796$ CFM

Problem #10

If brass turnings were the contaminant, the design transport velocity would be at least 4500 fpm. What duct size (diameter in inches) is required to deliver 500 CFM?

a. 3.5 inches
b. 4.0 inches
c. 4.5 inches
d. 5.0 inches
e. 5.5 inches

Answer to Problem # 10

c, 4.5 inches

Q = V * A; V = 4500 fpm, Q = 500 CFM

500 = 4500 * A; A = 0.111

Since A = 3.14 ($d^2/4$) with d in units of feet

Select 4” or 0.33 feet as d, then A = 0.085 and it must be 4.5”

Problem #11

If a velocity pressure of 0.6 inches H_2O and an impact pressure of 0.2 inches H_2O were measured in a duct on the suction side of the fan, what is the static pressure?

a. 0.8 inches H_2O
b. -0.4 inches H_2O
c. 0.4 inches H_2O
d. 0.8 inches H_2O
e. 0.6 inches H_2O

Answer to Problem # 11

b, -0.4 inches H_2O

VP = 0.6", TP (Impact) = 0.2"

TP = SP + VP or 0.2 = SP + 0.6

SP = -0.4 inches water

OR look for the only negative value!

Problem #12

A flanged hood 6 inches in diameter has a flow rate of 800 CFM and a C_e = 0.82. What is the hood entry loss (h_e) in inches H_2O?

a. 1.54 inches H_2O
b. 1.04 inches H_2O
c. 0.98 inches H_2O
d. 0.76 inches H_2O
e. 0.50 inches H_2O

Answer to Problem #12

e, 0.50 inches H_2O

Q = V*A or 800 = V*(0.196); V = 4081 fpm

V = 4005 $(VP)^{.5}$; 4081 = 4005 $(VP)^{.5}$; VP = 1.03

Ce = $(VP/SP)^{.5}$ or 0.82 = $(1.03/SP)^{.5}$; SP = 1.54

SP = VP + h_e or 1.54 = 1.03 h_e; h_e = 0.51

Thank You!

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