Basic Concepts of Ventilation Design
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

<table>
<thead>
<tr>
<th>Quantity</th>
<th>To Convert</th>
<th>Into</th>
<th>Multiply By:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Volumetric Flow</td>
<td>cubic feet/minute (ft³/min)</td>
<td>cubic meters/second (m³/sec)</td>
<td>4.719 x 10⁻⁴</td>
</tr>
<tr>
<td>Velocity</td>
<td>feet/minute (fpm)</td>
<td>meters/second (m/s)</td>
<td>0.00508</td>
</tr>
<tr>
<td>Pressure</td>
<td>inches water (in w.g.)</td>
<td>Pascals (Pa)</td>
<td>249.1</td>
</tr>
</tbody>
</table>
Conservation of Mass

\[ Q = V \cdot A \]

Where

- \( Q \) = Volumetric Flow Rate, \( \text{ft}^3/\text{min} \)
- \( V \) = Air Velocity, \( \text{ft/min} \) or \( \text{fpm} \)
- \( A \) = Cross Sectional Area, \( \text{ft}^2 \) or \( \text{SF} \)

Flow rate at point 1 is called \( Q_1 \)
and is equal to
flow rate at point 2 which is called \( Q_2 \)

1 velocity = 50 FPM

Hood

2 velocity = 3000 fpm

Duct

Air Flow
Conservation of Mass

\[ Q = V \cdot A \]

1. 8 inch duct
2. 6 inch duct
3. 12 inch duct

\[ Q_1 + Q_2 = Q_3 \]

\[ V_1A_1 + V_2A_2 = V_3A_3 \]
AIR FLOW

- At standard temperature and pressure (STP):
  * 1 atmosphere & 70° 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 through a system only rising when going across a fan
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

- \( V = 1096 \left( \frac{V_p}{\rho} \right)^{0.5} \)
  where \( \rho = \) air density 
  @ STP \( \rho = 0.075 \text{ lb}_m/\text{ft}^3 \)

- \( V = 4005 \left( V_p \right)^{0.5} \)

- 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. Duct diameter = 3 inches
   Area = 0.0668 ft²

2. • Q₁ = Q₂
   • If Q desired is 300 cfm
   • Then Q = V A
     V = Q A
     V = (300) / (0.0068)
     V = 4490 fpm

3. • If there are no losses from the grinder hood entry then:
   \[ SP₁ + VP₁ = SP₂ + VP₂ \]
   but: \[ SP₁ = 0 \] and \[ VP₁ → 0 \]
   we then have:
   \[ 0 = SP₂ + VP₂ \]
   or \[ -VP₂ = SP₂ \]
Bench Grinder Exhaust Ventilation

1. Duct diameter = 3 inches
   Area = 0.0668 ft²
   V = 4490 fpm

2. 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 \( SP_2 = (-VP_2) \)

3. \( V = 4005 (VP)^{0.5} \)

4. \( VP_2 = (4490/4005)^2 \)

5. \( VP_2 = 1.26 \) in w.g.

6. then: \( SP_2 = (-VP_2) \)
   \( SP_2 = -1.26 \) in w.g.
However there are losses thru the grinder hood entry

\[ SP_2 = -(VP_2 + h_e) \]

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
  \[ h_e = (F_h) (VP) \]
- Now we define the static pressure at the hood as \( SP_h \)
- \( SP_h \) is also called the hood static suction and is the absolute value of \( SP_2 \)
Bench Grinder Exhaust Ventilation

Now add the hood entry loss:

\[ S_{Ph} = VP_2 + h_e = VP_2 + (F_h)(VP_2) \]

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

\[ S_{Ph} = 1.26 + (0.40)(1.26) = 1.76 \text{ in } \text{w.g.} \]
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

(a) Plain Duct End with $F_d = 0.93$
(b) Flanged Inlet with $F_d = 0.49$
(c) Bell-Mouth Inlet with $F_d = 0.04$
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 $F_d = 0.04$

<table>
<thead>
<tr>
<th>HODD TYPE</th>
<th>DESCRIPTION</th>
<th>COEFFICIENT OF ENTRY, $C_e$</th>
<th>ENTRY LOSS</th>
</tr>
</thead>
<tbody>
<tr>
<td>PLAIN OPENING</td>
<td></td>
<td>0.72</td>
<td>0.93 VP</td>
</tr>
<tr>
<td>FLANGED OPENING</td>
<td></td>
<td>0.82</td>
<td>0.49 VP</td>
</tr>
<tr>
<td>TAPER or CONE HOOD</td>
<td>Varies with angle of taper or cone. See Fig. 6-10</td>
<td></td>
<td></td>
</tr>
<tr>
<td>BELL MOUTH INLET</td>
<td></td>
<td>0.98</td>
<td>0.04 VP</td>
</tr>
<tr>
<td>ORIFICE</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>TYPICAL GRINDING HOOD</td>
<td>STRAIGHT TAKE-OFF</td>
<td>0.78</td>
<td>0.65 VP</td>
</tr>
<tr>
<td>TYPICAL GRINDING HOOD</td>
<td>TAPERED TAKE-OFF</td>
<td>0.85</td>
<td>0.40 VP</td>
</tr>
</tbody>
</table>
Hood Entry Coefficients

\[ C_e = \frac{\text{Actual Flow}}{\text{Hypothetical Flow \ no losses}} \]

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

\[ C_e = (VP/SP_h)^{0.5} \]
Hood Entry Coefficients

\[ C_e = (\frac{VP}{SP_h})^{0.5} \]

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

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

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

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

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

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

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

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

- See Industrial Ventilation Manual notes

Blowing vs. Exhausting
Air Flow Characteristics

Exhaust Hoods
Capture Velocity

From Dalla Valle’s empirical work

\[ V(x) = \frac{Q}{10x^2 + A} \]
Capture Velocity

\[ V(x) = \frac{Q}{10x^2 + A} \]

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?
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 7. Fan Characteristic Curve

- Fan Characteristic Curve for Model = M
- Size = P
- Speed = Y

Operating Point A

- System Characteristic Curve

Airflow rate, ACFM
Figure 8. Effect of a Change in the System Characteristic on the Operating Point
Questions?