ENVIRONMENTAL PROTECTION AGENCY
APTI 413: Control of Particulate Matter Emissions

Student Manual
Chapter 10
The preparation of this manual was overseen by the organizations of the Environmental Protection Agency (EPA) and the National Association of Clean Air Agencies (NACAA*), and the revision of materials was coordinated and managed by the Tidewater Operations Center of C² Technologies, Inc., Newport News, Virginia.

Valuable research and feedback was provided by an advisory group of subject matter experts composed of Dr. Jerry W. Crowder, TX; Dr. Tim Keener, OH; Dr. Douglas P. Harrison, LA., and Mr. Tim Smith, Senior Air Quality Specialist, EPA, Office of Air Quality Planning and Standards.

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*The National Association of Clean Air Agencies (NACAA) represents air pollution control agencies in 53 states and territories and over 165 major metropolitan areas across the United States.

State and local air pollution control officials formed NACAA (formerly STAPPA/ALAPCO) over 30 years ago to improve their effectiveness as managers of air quality programs. The associations serve to encourage the exchange of information among air pollution control officials, to enhance communication and cooperation among federal, state, and local regulatory agencies, and to promote good management of our air resources.
10.1 Hoods

The pollutants generated or released by process equipment must be captured so that they can be transported to the air pollution control device. In some cases particulate capture occurs in hoods (Figure 10-1). Hoods are often an integral part of the process equipment. The hood can consist of a simple, stationary plenum mounted above or to the side of the source, a large moveable plenum, or the process equipment itself.

If hoods do not capture pollutants generated by process equipment, the pollutants disperse directly into the plant air and eventually pass through roof vents and doors into the atmosphere. Evaluation of the ability of the hoods to capture pollutants at the point of generation is important in many inspections and engineering studies. The USEPA defines fugitive emissions as "emissions that (1) escape capture by process equipment exhaust hoods; (2) are emitted during material transfer; (3) are emitted to the atmosphere from the source area; and (4)
are emitted directly from process equipment.”

Fugitive emissions = Total emissions – Emissions captured by hood

Stack emissions = Emissions captured by hood $\times \left( \frac{100 - \eta}{100} \right)$

The importance of hood performance is illustrated by Examples 10-1 and 10-2, which are based on the simplified industrial process shown in Figure 10-2. This system consists of a process unit that generates pollutants, several hoods surrounding the process equipment, the ductwork, an air pollution control device, a fan, and a stack.

Example 10-1 Calculate the fugitive emissions and the stack emissions if the process equipment generates 100 lb$_m$/hr of particulate matter, the hood capture efficiency is 95%, and the collection efficiency of the air pollution control device is 95%.

Solution

Calculate fugitive emissions:

Fugitive emissions = total emissions – emissions captured by hood

$\frac{100 \text{ lb}_m}{\text{hr}} - 95 \frac{\text{lb}_m}{\text{hr}} = 5 \frac{\text{lb}_m}{\text{hr}}$

Calculate stack emissions:
Stack emissions =

\[
\text{Emissions captured by hood} \times \left( \frac{100 - \eta}{100} \right) = \left( 95 \frac{\text{lb} \text{m}}{\text{hr}} \right) \left( \frac{100 - 95}{100} \right) = 4.75 \frac{\text{lb} \text{m}}{\text{hr}}
\]

The capture of emissions by the hood is the key step in an air pollution control system. Example 10-1 shows that, even with high hood capture efficiency, fugitive emissions can be higher than emissions leaving the stack.

**Example 10-2** Calculate the fugitive emissions and the stack emissions if the process equipment generates 100 lb/hr of particulate matter, the hood capture efficiency is 90%, and the collection efficiency of the air pollution control device is 95%.

**Solution**

Calculate fugitive emissions:

\[
100 \frac{\text{lb} \text{m}}{\text{hr}} - 90 \frac{\text{lb} \text{m}}{\text{hr}} = 10 \frac{\text{lb} \text{m}}{\text{hr}}
\]

Calculate stack emissions:

Stack emissions =

\[
\text{Emissions captured by hood} \times \left( \frac{100 - \eta}{100} \right) = \left( 90 \frac{\text{lb} \text{m}}{\text{hr}} \right) \left( \frac{100 - 95}{100} \right) = 4.5 \frac{\text{lb} \text{m}}{\text{hr}}
\]

These two problems illustrate the importance of hoods. Slight changes in the ability of the hood to capture the pollutants can have a large impact on the total fugitive and stack emissions released into the atmosphere.

Unfortunately, it is not always possible to see the fugitive emissions. Gaseous and vapor emissions such as carbon monoxide, sulfur dioxide, hydrogen chloride, and nitric oxide are not visible. Even particulate emissions may be hard to see under the following circumstances:

- If there are numerous small fugitive sites
- If there is one major site that cannot be seen from normal areas accessible to personnel
- If the particulate matter is not in the size range that causes light scattering

Techniques for monitoring hood capture effectiveness are important because the quantities of fugitive emissions can be high, and these emissions are often hard to see.
Hood Operating Principles
Hoods are generally designed to operate under negative pressure. The air is drawn into the hood due to static pressures that are lower inside the hood than those in the process equipment and the surrounding air. Since air from all directions moves toward the low-pressure hood, the hood must be as close as possible to the process equipment in order to capture the pollutant-laden air and not just the surrounding air. At approximately one-hood diameter away from the hood entrance, the gas velocities are often less than 10% of the velocity at the hood entrance. Figure 10-3 illustrates how quickly the gas velocity decreases as distance from the hood increases.

Figure 10-3 indicates that the hood has very little influence on gas flow except in the area very close to the hood entrance. In order to ensure good capture of the pollutant-laden gas streams, the hood must be close to the emission source. The capture velocity of a hood is defined as the air velocity at any point in front of the hood or at the hood opening necessary to overcome opposing air currents and to capture the contaminated air at that point by pulling it into the hood. A general guide for appropriate capture velocities in several situations is provided in Table 10-1.

<table>
<thead>
<tr>
<th>Type of Material Release</th>
<th>Capture Velocity (ft/min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>With no velocity into quiet air</td>
<td>50-100</td>
</tr>
<tr>
<td>At low velocity into moderately still air</td>
<td>100-200</td>
</tr>
<tr>
<td>Active generation into zone of rapid air motion</td>
<td>200-500</td>
</tr>
<tr>
<td>With high velocity into zone of very rapid air</td>
<td>500-2,000</td>
</tr>
</tbody>
</table>

The following show conditions that would help determine what part of the capture velocity range should be used for a particular operation. In general, high toxicity contaminants released from small sources into rapidly moving air currents would
require higher capture velocities.

- The surrounding air currents
  - Minimal room air currents vs. disturbing room air currents
- The level of toxicity of the pollutant to be captured
  - Nuisance value only vs. high toxicity
- The amount of pollutant
  - Intermittent (low production) vs. high production (heavy use)
- Area of the hood opening
  - Large hood (large air mass in motion) vs. small hood (local control only)

The following flow/capture velocity equation for a freely suspended hood without a flange demonstrates the importance of the proximity of the hood to the source:

\[ Q = v_h (10X^2 + A_h) \]

Where

- \( Q \) = actual volumetric flow rate (ft\(^3\)/min)
- \( X \) = distance from hood face to farthest point of contaminant release (ft)
- \( v_h \) = hood capture velocity at distance \( X \) (ft/min)
- \( A_h \) = area of hood opening (ft\(^2\))

It should be noted that this correlation between distance, gas flow rate, and capture velocity should be used for estimation purposes only because the vacuum from a hood does not create equal velocity lines or points. Equation 10-1 is also limited to the distance \( X \) being less than or equal to 1.5 hood diameters. Capture velocity equations for a variety of hoods with different locations and arrangements can be obtained from the ACGIH, *Industrial Ventilation Manual*.

**Example 10-3**  
The recommended capture velocity for a certain pollutant entering a 16-inch diameter hood is 300 ft/min. What is the required volumetric flow rate for the following distances from the hood face \( X \)?

A. \( X = 12 \text{ in. (75\% of hood diameter)} \)
B. \( X = 24 \text{ in. (150\% of hood diameter)} \)

**Solution for Part A**

\[ Q = v_h (10X^2 + A_h) \]

Calculate the area of the hood opening:
Calculate the volumetric flow rate, $Q$, required to obtain the recommended capture velocity of 300 fpm at a distance of 12 inches from the hood:

$$Q = v_h (10X^2 + A_h) = 300 \frac{\text{ft}}{\text{min}} \left[ 10(1\text{ft})^2 + 1.40\text{ft}^2 \right] = 3,420 \frac{\text{ft}^3}{\text{min}}$$

**Solution for Part B** Calculate the volumetric flow rate, $Q$, required to obtain the recommended capture velocity of 300 fpm at a distance of 24 inches from the hood:

$$Q = v_h (10X^2 + A_h) = 300 \frac{\text{ft}}{\text{min}} \left[ 10(2\text{ft})^2 + 1.40\text{ft}^2 \right] = 12,420 \frac{\text{ft}^3}{\text{min}}$$

The volumetric flow rate requirements increased approximately four times when the distance between the hood and the contaminant source doubled.

**Hood Designs for Improved Performance**

There are many ways to design hoods to improve capture effectiveness. When the pollutant-laden gas stream is hot, the hood is often positioned above the point of pollutant release to take advantage of the buoyancy of the low-density hot gas stream.

Flanges can be used to block the movement of unwanted air into the hood. The recommended width of a flange for most situations should be equal to the square root of the hood area. The beneficial effect of a flange on the gas velocities near a hood entrance is shown in Figure 10-4.

Hood capture is greatly improved when the enclosure comprised of the hood and the side baffles can encompass the point of pollutant generation. These side baffles can be in the form of metal sheets, strips of fabric or plastic, or any other materials that block the movement of clean air into the low-pressure area of the hood. In addition to reducing the unintentional capture of clean air, these side baffles also prevent cross drafts, which can prevent the intended movement of the pollutant-laden gas into the hood.
Some process equipment inherently creates an entirely enclosed area for pollutant-laden gas capture. For example, coal-fired boilers generate pollutants in an enclosed furnace area that is maintained at a slightly negative static pressure of -0.05 to -0.25 in. WC. In this case, the boiler walls serve as the hood.

Another hood design that is used to improve capture effectiveness is called the push-pull hood. As shown in Figure 10-5, a high-velocity clean air stream is blown across the area of pollutant generation into the hood on the opposite side.

The high-velocity gas stream does not inherently disperse rapidly. Therefore, it flows toward the hood and is captured. The hood also effectively captures the pollutant-laden gas that is trapped in this strong cross draft. These types of hoods are sometimes used on open tanks and other sources where access from the top is necessary in order to operate the equipment. However, they may not be appropriate for tanks and other processes handling materials where the cross draft could significantly increase the quantities vaporized. Push-pull hoods can provide very high capture efficiencies where
they are applicable.

**Monitoring Hood Capture Effectiveness**

There are several effective ways to confirm that the hood capture effectiveness has not decreased since it was installed or tested. Visible emission observations for fugitive emissions should be conducted in the case of particulate sources. In general, you should confirm that the hood has not been moved away from the point of pollutant generation and that side baffles and other equipment necessary to maintain good operation have not been damaged or removed.

The hood static pressure should be monitored to ensure that the appropriate gas flow rate is maintained. The hood static pressure is simply the static pressure in the duct immediately downstream from the hood. This static pressure is usually negative and is entirely dependent on the hood geometry and the gas flow rate. As long as the hood has not been damaged or altered, the hood static pressure provides an indirect, but relatively accurate measurement of the gas flow rate. As indicated in Equation 10-4, the hood static pressure is determined from the velocity pressure in the duct from the hood and the hood entry loss. The loss of static pressure caused by air flowing into a system is referred to as entry loss.

\[
SP_h = VP_d + h_e
\]

Where
- \(SP_h\) = hood static pressure (in WC)
- \(VP_d\) = duct velocity pressure (in WC)
- \(h_e\) = hood entry loss (in WC)

The velocity pressure term is due to the energy necessary to accelerate the air from zero velocity to the velocity in the duct. The hood entry loss is usually expressed as some fraction of this velocity pressure:

\[
h_e = F_h VP_d
\]

Where
- \(F_h\) = hood entry loss coefficient (dimensionless)
- \(VP_d\) = duct velocity pressure (in WC)

Hood entry loss coefficients are tabulated in standard texts on hoods and ventilation systems.

When air enters a negative pressure duct, the airflow converges as shown in Figures 10-
6 through 10-8. The area where air converges upon entering a duct is referred to as *vena contracta*. After the *vena contracta*, the airflow expands to fill the duct and some of the velocity pressure converts to static pressure. The *vena contracta* is dependent on the hood geometry, which determines the resistance to airflow entering the hood. In general, the smoother the entry, the lower the entry loss coefficient.

The velocity pressure is related to the square of the gas velocity in the duct and the gas density:

\[(10-6)\]
\[ VP_d = p_g \left( \frac{v_d}{1,096.7} \right)^2 \]

Where

- \( VP_d \) = duct velocity pressure (in WC)
- \( v_d \) = duct gas velocity (ft/min)
- \( \theta_g \) = gas density (lb\( m^3 \)/ft\(^3 \))

As the gas flow rate into the hood increases, the hood static pressure increases. A decrease in hood static pressure (i.e., a less negative value) usually indicates that the gas flow rate entering the hood has decreased from previous levels. This may reduce the effectiveness of the hood by reducing the capture velocities at the hood entrance.

**Example 10-4**

A hood serving a paint dipping operation has a hood static pressure of 1.10 in WC. The baseline hood static pressure was 1.70 in WC. Estimate the gas flow rate under the following two conditions:

A. At present operating conditions
B. At baseline levels

Use the data provided below:

- \( F_h = 0.93 \)
- Temperature = 68°F
- Duct diameter 2 ft (inside diameter)

**Solution for Part A**

Calculate the velocity pressure in the duct:

\[
SP_h = (1 + F_h)VP_d
\]

\[
VP_d = \frac{SP_h}{1 + F_h} = \frac{1.10 \text{ in WC}}{1 + 0.93} = 0.57 \text{ in WC}
\]

Calculate the gas velocity in the duct:

\[
v_d = 1,096.7 \sqrt{\frac{VP_d}{p_g}} = 1,096.7 \sqrt{\frac{0.57 \text{ in WC}}{0.0747 \text{ lb}_m/\text{ft}^3}} = 3,029.5 \text{ ft/min}
\]

Calculate the gas flow rate:
\[ Q = v_d A_d = v_d \left( \frac{\pi D^2}{4} \right) = 3,029.5 \text{ ft}^3/\text{min} \left[ \frac{\pi (2\text{ft})^2}{4} \right] = 9,515.5 \text{ ft}^3/\text{min} \]

**Solution for Part B**

Calculate the velocity pressure in the duct:

\[ SP_h = (1 + F_h) VP_d \]

\[ VP_d = \frac{SP_h}{1 + F_h} = \frac{1.70 \text{ in WC}}{1 + 0.93} = 0.88 \text{ in WC} \]

Calculate the gas velocity in the duct:

\[ VP_d = p_g \left( \frac{v_d}{1,096.7} \right)^2 \]

\[ v_d = 1,096.7 \sqrt{\frac{VP_d}{p_g}} = 1,096.7 \sqrt{\frac{0.88 \text{ in WC}}{0.0747 \text{ lbm}^{-1}/\text{ft}^3}} = 3,764.2 \text{ ft}/\text{min} \]

Calculate the gas flow rate:

\[ Q = v_d A_d = v_d \left( \frac{\pi D^2}{4} \right) = 3,764.2 \text{ ft}^3/\text{min} \left[ \frac{\pi (2\text{ft})^2}{4} \right] = 11,819.9 \text{ ft}^3/\text{min} \]

The change in hood static pressure from 1.7 in WC to 1.1 in WC indicates a drop in the gas flow rate from 11,820 acfm to 9,518 acfm. This is nearly a 20% decrease in the gas flow rate.

**Transport Velocity**

When a particulate contaminant is captured by a hood system and enters the ductwork, a minimum *transport velocity* must be maintained to keep the contaminant from settling out of the gas stream and building up deposits in the ductwork. This would lead to decreased hood capture efficiencies and increased fugitive emissions. Systems with heavy particulate-laden gas streams should have clean-out ports installed to remove particles that have settled out. Typical transport velocities for different types of contaminants are shown in Table 10-2.
### Table 10-2. Transport Velocities

<table>
<thead>
<tr>
<th>Contaminant</th>
<th>Transport Velocity (ft/min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vapors, gases, smoke</td>
<td>Any (usually 1,000-2,000)</td>
</tr>
<tr>
<td>Fume</td>
<td>1,400-2,000</td>
</tr>
<tr>
<td>Very fine, light dusts</td>
<td>2,000-2,500</td>
</tr>
<tr>
<td>Dry dusts and powders</td>
<td>2,500-3,500</td>
</tr>
<tr>
<td>Average industrial dusts</td>
<td>3,500-4,000</td>
</tr>
<tr>
<td>Heavy dusts</td>
<td>4,000-4,500</td>
</tr>
<tr>
<td>Very heavy or moist dusts</td>
<td>&gt;4,500</td>
</tr>
</tbody>
</table>

Proper duct diameter is a key element when addressing minimum transport velocity. If a section of ductwork has a larger than necessary diameter, then settling out will most likely occur. If a section of ductwork is too small, the pressure drop will increase across this section, thus requiring the fan to handle more static pressure. Another concern when dealing with transport velocities is the abrasion of the ductwork, especially at elbows and the areas opposite entries. The amount of abrasion that occurs is dependent upon several factors, including the duct velocity, the amount and type of particulate matter in the gas stream, and the construction of the ductwork.

**Example 10-4**

A duct system transporting a dry dust requires a minimum transport velocity of 2,800 ft/min. The volumetric flow rate for the system is 978 acfm. What is the necessary duct diameter in inches for this section of ductwork to maintain the minimum transport velocity?

**Solution**

Calculate the duct area:

\[
A_d = \frac{Q}{v_d} = \frac{978 \text{ ft}^3/\text{min}}{2,800 \text{ ft/min}} = 0.349 \text{ ft}^2
\]

Calculate the duct diameter:

\[
D = \sqrt{\frac{4A_d}{\pi}} = \sqrt{\frac{4(0.349 \text{ ft}^2)}{\pi}} = 0.667 \text{ ft} = 8 \text{ in}
\]
Summary
Hoods are the first component of the air pollution control system and are of critical importance. If they fail to capture the pollutant, the overall collection efficiency of the system is reduced. Pollutants not captured by hoods become fugitive emissions. Many factors affect a hood's capture efficiency; however, one of the key factors is the distance between the pollutant source and the hood.

The geometry of a hood opening influences the hood entry loss coefficient and the hood static pressure due to the formation of the vena contracta. Comparing the hood static pressure against baseline condition provides a good indicator if the system has developed any problems.

Maintaining a system's minimum transport velocity is necessary to ensure that all of the captured pollutant reaches the air pollution control device and to prevent build-up of the pollutants in the ductwork.

10.2 Fans
Fans are the heart of the system. They control the gas flow rate at the point of pollutant generation in the process equipment and through the air pollution control devices. Fans provide the necessary energy for the gas stream to overcome the resistance to gas flow caused by the ductwork and air pollution control devices. Data concerning fan performance is important during inspections and all other technical evaluations of system performance.

Types of Fans and Fan Components
There are two main types of fans: axial and centrifugal. An axial fan is shown in Figure 10-9. The term, axial, refers to the use of a set of fan blades mounted on a rotating shaft aligned in the direction of air movement. A standard house ventilation fan is an axial fan.

A centrifugal fan has a wheel composed of a number of fan blades mounted around a hub. As shown in Figure 10-10, the hub turns on a shaft that passes through the fan housing. The gas enters from the side of the fan wheel, turns 90° and is accelerated as it passes over the fan blades. The term, centrifugal, refers to the trajectory of the gas stream as it passes out of the fan housing.

Centrifugal fans can generate high-pressure rises in the gas stream. Accordingly, they are well-suited for industrial processes and air pollution control systems. The remainder of this section concerns centrifugal fans.
The major components of a typical centrifugal fan include the fan wheel, fan housing, drive mechanism, and inlet dampers and/or outlet dampers. A wide variety of fan designs serve different applications.

The fan drive determines the speed of the fan wheel and the extent to which this speed can be varied. The types of fan drives can be grouped into three basic categories:

- Direct drive
- Belt drive
- Variable drive

In a direct drive arrangement, the fan wheel is linked directly to the shaft of the motor. This means that the fan wheel speed is identical to the motor rotational speed. With this type of fan drive, the fan speed cannot be varied.

*Belt driven fans* use multiple belts which rotate over a set of sheaves or pulleys mounted on the motor shaft and the fan wheel shaft. This type of drive mechanism is illustrated in Figure 10-11.
The belts transmit the mechanical energy from the motor to the fan. The fan wheel speed is simply the motor speed multiplied by the ratio of the motor wheel sheave diameter to the fan sheave diameter:

\[
\text{RPM}_{\text{fan}} = \text{RPM}_{\text{motor}} \times \frac{D_{\text{motor}}}{D_{\text{fan}}}
\]

Where

- \( \text{RPM}_{\text{fan}} \) = fan speed (revolutions per minute)
- \( \text{RPM}_{\text{motor}} \) = motor speed (revolutions per minute)
- \( D_{\text{fan}} \) = diameter of fan sheave (inches)
- \( D_{\text{motor}} \) = diameter of motor sheave (inches)

Fan wheel speeds in belt-driven arrangements are fixed unless the belts slip. Belt slippage normally reduces fan wheel speed several hundred rpm and creates a noticeable squeal. If it is necessary to change the fan wheel speed in a belt-driven arrangement, the motor and/or fan wheel sheaves must be replaced with units having different diameters. However, there are very definite safety limits to the extent to which the fan speed can be increased. If the fan rotational speed is excessive, the fan can disintegrate.

*Variable speed fans* use hydraulic or magnetic couplings that allow operator control of the fan wheel speed independent of the motor speed. The fan speed controls are often integrated into automated systems to maintain the desired fan performance over a variety of process operating conditions.

Fan dampers are used to control gas flow into and out of the centrifugal fan. These dampers can be on the inlet and/or the outlet side of the fan. Dampers on the outlet side simply impose a flow resistance that is used to control gas flow. Dampers on the inlet side are designed to control gas flow and to change how the gas enters the fan wheel at different operating conditions. Inlet dampers conserve fan energy due to their ability to affect the airflow pattern into the fan.

The fan wheel consists of a hub and a number of fan blades. The fan blades on the hub can be arranged in three different ways:

- Forward curved
- Backward curved
- Radial

*Forward curved* fans (Figure 10-12a) use blades that curve or slant toward the direction of
rotation of the fan wheel. These are especially sensitive to particle accumulation and are not used extensively in air pollution control systems.

![Types of fan wheels](image)

Figure 10-12. Types of fan wheels

*Backward curved* fans (Figure 10-12b) use straight plates, curved plates, or curved airfoils that angle away from the direction of rotation. These types of fan wheels are used in fans designed to handle gas streams with relatively low particulate loadings because they are prone to solids build-up. Backward curved fans are more energy efficient than radial fans.

*Radial* fans use fan wheel blades that extend straight out from the hub. A radial blade fan wheel, as shown in (Figure 10-12c), is often used on particulate-laden gas streams because it is the least sensitive to solids build-up.

**Centrifugal Fan Operating Principles**

A basic understanding of fan operating principles is necessary to evaluate the performance of an industrial ventilation system. The fan operating speed is one of the most important operating variables. Most fans can operate over a modest range of speeds. The flow rate of gas moving through the fan depends on the fan wheel rotational speed. As the speed increases, the gas flow rate increases proportionally. This relationship is expressed as one of the fan laws:

\[
Q_2 = Q_1 \left( \frac{\text{RPM}_2}{\text{RPM}_1} \right)
\]

Where

- \(Q_1\) = baseline gas flow rate (acfm)
- \(Q_2\) = present gas flow rate (acfm)
- \(\text{RPM}_1\) = baseline fan wheel rotational speed (revolutions per minute)
- \(\text{RPM}_2\) = present fan wheel rotational speed (revolutions per minute)

The rate of gas flow through a fan is always expressed in actual conditions. This is helpful because this value does not change, regardless of the gas density. In this respect, a fan is much like a shovel. It moves a specific volume of gas per minute, regardless of whether the gas is dense cold gas or light hot gas.
The gas stream moving through the fan has a static pressure rise due to the mechanical energy expended by the rotating fan wheel. As indicated in Figure 10-13, the static pressure at the outlet is always higher than the static pressure at the inlet. The static pressure rise across the fan is denoted as Fan SP:

\[
\text{Fan SP} = \text{SP}_{\text{out}} - \text{SP}_{\text{in}} - \text{VP}_{\text{in}}
\]

For conditions shown in the figure, Fan SP = 0.05 – (-10) – 0.50 = 9.55 in WC.

Fan SP is related to the square of the fan speed, as indicated in the second fan law:

\[
\text{Fan SP}_2 = \text{Fan SP}_1 \left( \frac{\text{RPM}_2}{\text{RPM}_1} \right)^2
\]

Where

- \( \text{Fan SP}_1 \) = baseline fan static pressure (in WC)
- \( \text{Fan SP}_2 \) = present fan static pressure (in WC)
- \( \text{RPM}_1 \) = baseline fan wheel rotational speed (revolutions per minute)
- \( \text{RPM}_2 \) = present fan wheel rotational speed (revolutions per minute)

The specific fan for an industrial ventilation system must be selected based on the specific air flow rate and fan static pressure rise needed to properly capture, transport, and control the emissions. Each industrial ventilation system includes one or more capture hoods, ductwork, air pollution control systems, the fan, and a stack. The gas flow rate through the ventilation system must be sufficient to provide adequate pollutant capture at the hoods and to ensure proper transport of the pollutant-laden air to the air pollution control systems. The fan static pressure rise must be sufficient to accelerate the air entering the hoods and to overcome the flow resistances of the hoods, ductwork, air pollution control systems, and stack.

The designer of an air pollution control system starts by specifying the air volumes and
velocities in the hoods, ductwork, air pollution control systems, and stack. These values are selected based on established engineering design principles to ensure high efficiency hood capture, proper operation of the air pollution control systems, and proper dispersion of the effluent gas stream from the stack. Using this information, the designer next calculates the flow resistance of the system and determines the fan static pressure needed to move the design volume.

The static pressure drop across each component of the system is related to the square of the air flow rate. This general relationship is shown in Figure 10-14 and is termed the system characteristic curve. The designer selects a fan that will deliver performance corresponding to one point on this curve – the required volume at the necessary fan static pressure. The data necessary for doing this are provided in multi-rating tables published by manufacturers for each specific fan model and size they produce. Based on these data, it is possible to select a fan model, the specific model size, the fan speed and the motor horsepower necessary to achieve the necessary air flow rate and fan static pressure condition. An excerpt from a multi-rating table for a centrifugal fan is shown below in Figure 10-15.
The match between the fan performance data and the system characteristic curve is illustrated in Figure 10-16 for the specific fan rotational speed chosen. As long as the overall system remains in good condition and the fan remains in good condition, the system will operate at the point shown in Figure 10-16. This is termed the operating point. When the total system static pressure drop and the fan static pressure rise are shown on the same graph, as in the case with Figure 10-16, it is convenient to simply delete the total system static pressure drop axis.

Figure 10-17 illustrates an example fan curve for a given fan speed. The multi-rating data used to select the fan represents a subset of the total data set that defines this fan curve. There is a specific fan curve for each fan model, model size, and speed. The intersection of the fan curve and the system characteristic curve is illustrated as Point A. This is the point that was determined by the system designer selecting the fan.
Air pollution control systems and other types of industrial ventilation systems, however, do not necessarily remain exactly at the conditions anticipated by the system designer and the fan manufacturer. A number of normal operating changes and operating problems can cause changes in the overall system air flow rates and the static pressure rises across the fan. The extent of the air flow and static pressure rise changes depend on the fan's performance conditions and the system characteristic curve.

If the gas flow resistance increases due to the build-up of dust in an air pollution control device or because a damper is closed, the system characteristic curve will shift upwards as indicated in Figure 10-18. With this increased gas flow resistance there will be a new operating point, labeled "B". At this new operating point, the fan static pressure rise will be slightly higher while the air flow rate will be slightly lower.

If the air flow resistance decreases due to changes in an air pollution control device or opening of a damper, the system characteristic curve will shift downwards. This results in a new operating point (labeled "C") that has a slightly reduced fan static pressure and increased air flow rate.

Some changes in the system characteristic curve are normal due to factor such as (1) air pollution control system cleaning cycles, (2) gradually increasing air infiltration between maintenance cycles, (3) the opening and closing of individual dampers on individual process sources, and (4) opening and closing of fan inlet or outlet dampers. The system must be designed to provide adequate pollutant capture even at the lowest normally occurring air flow rates.
When changes in the system characteristic curve are outside of the anticipated range, operators often have the option of modifying the fan to increase its capability. Most fans on industrial systems are selected to operate at a speed near the middle of its safe operating range. Slight increases in the fan speed can increase air flow rates and static pressure rises without exceeding the safe operating speed limits. The impact of a slight increase in the fan speed is illustrated in Figure 10-19.

It is apparent that the increased fan speed results in a new operating point (labeled “D”) having an air flow rate and fan static pressure rise that are both larger than the conditions represented by operating point "A." Not all fans can be easily adjusted to change the fan speed. For example, direct drive fans where the fan wheel shaft is directly driven by the fan motor operate only at the motor rotation speed and can not be adjusted. Belt driven fans can be adjusted but only by changing one or both of the sheaves on the fan and motor. Some large fans with hydraulic or magnetic drives have easily adjusted fan speeds.

Some inadvertent reductions in fan speed are possible for belt driven fans. If the drive
belts become slightly loose, they can slip as they move across the sheaves. This often results in a decreased air flow rate of 100 to 200 rpm. The decrease in the air flow rate is directly proportional to the decrease in the fan speed.

As noted earlier, the operating point of a system can be changed by the opening and closing of a fan inlet damper. This is a special damper mounted immediately ahead of the fan, and this damper changes how air enters the fan wheel. As the inlet damper is closed, the operating point shifts to the left, giving lower air flow rate at the higher fan static pressure. Opening the damper shifts the operating point to the right, giving higher air flow rate at the lower fan static pressure.

The fan inlet damper is often used to ensure safe operation of a fan that operates with air streams at elevated temperatures. During start-up when the air is cold, the fan inlet damper is kept partially closed to minimize the quantity of heavy cold air moved through the system. As the air heats and becomes less dense, the fan inlet damper opens to increase the air flow rate and fan static pressure rise. This approach minimizes the electrical power demand on the fan motor. Starting with the fan inlet dampers wide open would often exceed the safe current levels for the motor and thereby result in burnout of the motor windings. It is very important to avoid overloading fan motor currents.

**Example 10-5**

The static pressure drop across a ventilation system, measured at the fan inlet, is -16.5 in WC at a gas flow rate of 8,000 acfm. Estimate the static pressure drop if the flow rate is increased to 12,000 acfm.

**Solution**

\[
\frac{\Delta S_P_{\text{high flow}}}{\Delta S_P_{\text{low flow}}} = \left(\frac{Q_{\text{high flow}}}{Q_{\text{low flow}}}\right)^2
\]

\[
\Delta S_P_{\text{high flow}} = \Delta S_P_{\text{low flow}} \left(\frac{Q_{\text{high flow}}}{Q_{\text{low flow}}}\right)^2 = -16.5\text{in WC} \left(\frac{12,000\text{acfm}}{8,000\text{acfm}}\right)^2 = -37.1\text{in WC}
\]

**Note:** this solution is based on the assumption that there are no significant changes in gas density due to the increase in gas flow rate.

Decreased system resistance can sometimes be a problem. In this situation, the system operating point shifts to the right to a position of higher gas flow rate and lower static pressure rise. While this change would favor improved hood capture, it could reduce the collection efficiency of the air pollution control device. High gas velocities through certain types of air pollution control systems, such as fabric filters, electrostatic precipitators, carbon absorbers and catalytic oxidizers, can reduce efficiency.

It is helpful to be able to determine when the system characteristic curve has shifted. The most direct way to check the fan performance is to measure the gas flow rate. However, this is time consuming. The fan motor current data provides an indirect, but
sometimes very useful, indication of gas flow changes from the baseline conditions. If the system resistance has not changed, an increase in fan motor current is associated with an increase in the gas flow rate. Likewise, decreases in fan current occur when the gas flow rate drops in systems whose resistance has not changed. Unfortunately, the relationship between gas flow rate and motor current is not linear and system resistances can change. The nonlinear characteristic of the relationship is indicated by the brake horsepower curve shown in Figure 10-20. The fan motor current is directly proportional to the brake horsepower, as indicated by the following equation for a three-phase motor:

\[
BHP = \frac{1.73I \cdot E \cdot \text{Eff} \cdot PF}{745}
\]

Where

- \(BHP\) = brake horsepower
- \(I\) = fan motor current (amperes)
- \(E\) = voltage (volts)
- \(\text{Eff}\) = efficiency expressed as a decimal
- \(PF\) = power factor

While the shape of the horsepower curve varies for different types of fan wheels, this general relationship applies to all centrifugal fans in their normal operating range.

The brake horsepower is also related to the cube of the fan speed, as indicated by the third fan law:

\[
BHP_2 = BHP_1 \left(\frac{\text{RPM}_2}{\text{RPM}_1}\right)^3
\]

Where

- \(BHP_1\) = baseline brake horsepower
- \(BHP_2\) = present brake horsepower
- \(\text{RPM}_1\) = baseline fan wheel rotational speed (revolutions per minute)
- \(\text{RPM}_2\) = present fan wheel rotational speed (revolutions per minute)
Effect of Gas Temperature and Density

A fan operates like a high-speed shovel. Every rotation of the fan wheel at a given operating point moves a constant volume of air. While the volume is constant, the mass of the air being moved may not be constant. The density of the gas being handled by the fan is a function of the gas temperature. At high gas temperatures, the gas has a low density, and the gas is relatively light. When the gas temperature is cold, for example at ambient temperature, the gas is dense, and its weight is substantial. In addition to gas temperature, gas density is also a function of the absolute pressure.

The gas density has a direct effect on the fan motor current. The current will be high when the gas stream is cold, such as the times when the process is starting up. If steps are not taken to minimize gas flow during cold operating periods, the fan motor could burn out due to excessive current flow. To prevent this, the fan inlet or outlet dampers are usually partially closed during start-up to restrict the amount of dense air being handled. As the process heats up and the gas stream becomes less dense, the dampers can be opened to permit normal gas flow rates.

When using the fan motor current as an indicator of gas flow rate, it is important to correct the motor currents at the actual conditions back to standard conditions:

\[
I_{\text{STP}} = I_{\text{actual}} \left( \frac{\rho_{\text{STP}}}{\rho_{\text{actual}}} \right)
\]

Where

- \(I_{\text{STP}}\) = fan motor current at standard conditions (amperes)
- \(I_{\text{actual}}\) = fan motor current at actual conditions (amperes)
- \(\rho_{\text{STP}}\) = gas density at standard conditions (lbm/ft³)
ρ_{actual} = \text{density at actual conditions (lb}_m/ft^3)\]

**Example 10-7**  
A fan motor is operating at 80 amps and the gas flow rate through the system is 10,000 acfm at 300°F and -10 in WC (fan inlet). What is the motor current at standard conditions?

\[ I_{\text{STP}} = I_{\text{actual}} \left( \frac{\rho_{\text{STP}}}{\rho_{\text{actual}}} \right) \]

Calculate the gas density at actual condition:

\[ \rho = \frac{\text{PMW}}{RT} \]

\[ P = (407\text{ in WC } -10\text{ in WC}) \left( \frac{1_{\text{atm}}}{407\text{ in WC}} \right) = 0.975\text{ atm} \]

\[ T = 300°F + 460 = 760°\text{R} \]

\[ \rho = \frac{\text{PMW}}{RT} = \frac{(0.975\text{ atm})(29\text{ lb}_m}{\text{lb - mole}} \left( \frac{\text{atm} \cdot \text{ft}^3}{\text{lb - mole} \cdot °\text{R}} \right)(760°\text{R}) = 0.0510\text{ lb}_m/\text{ft}^3 \]

Calculate the motor current at standard conditions:

\[ I_{\text{STP}} = I_{\text{actual}} \left( \frac{\rho_{\text{STP}}}{\rho_{\text{actual}}} \right) = 80\text{amps} \left( \frac{0.0747\text{ lb}_m}{\text{ft}^3} \right) = 117\text{amps} \]

**Note 1:** The problem could have been solved quickly by using tabulated values of the gas density. However, this approach also reduces the risk of a gas density error caused by not taking into account the effect of pressure changes.

**Note 2:** The gas composition could be taken into account by calculating the weighted average molecular weights of the constituents rather than assuming 29 pounds per pound mole, which is close to the value for air. This correction is important when the gas stream has a high concentration of compounds such as carbon dioxide or water, which have molecular weights that are much different than air.

The gas temperature and pressure corrections for gas density must also be used when selecting a fan. The fan multi-rating tables are expressed in standard temperature (70°F) and pressure (1 atm). These corrections are needed to ensure that the fan will deliver...
the necessary gas flow rates and fan static pressure increases under the actual operating conditions anticipated in the process. As the gas flows through the fan, the pressure usually changes from negative to positive. This increase in pressure can cause the gas temperature to increase slightly.

**Summary**

Centrifugal fans are the most commonly used type of fan in air pollution control systems because of their ability to generate high pressure rises in the gas stream. The major components of a typical centrifugal fan include the fan wheel, fan housing, drive mechanism, and inlet dampers and/or outlet dampers.

The intersection of the fan characteristic curve and the system characteristic curve is called the operating point for the fan. The factors that affect the fan characteristic curve are the type of fan wheel and blade, the fan wheel rotational speed, and the shape of the fan housing. The system characteristic curve takes into account the energy losses throughout the ventilation system. These curves are helpful indicators in determining if a change in the system has occurred. A change in the system can also be detected through the fan motor current data that corresponds with the gas flow rate, provided the system resistance has not changed.

The fan laws can predict how a fan will be affected by a change in an operating condition. The fan laws apply to fans having the same geometric shape and operating at the same point on the fan characteristic curve.

A fan will move a constant volume of air; however the amount of work required to move the gas flow is dependent on the density of the gas. Two factors that affect density are temperature and pressure. The gas density has a direct effect on the fan motor current.

**10.3 Evaluating the Entire Industrial Process**

Industrial process systems consist of the process equipment, which generates the pollutants, the air pollution control equipment that removes them, and the fan that moves the gas stream.

The process equipment and the air pollutant control devices do not work independently. The operating conditions of all the system components are closely linked together by the fans, hoods, and ductwork.

Industrial Source System Flowcharts provide an important tool for evaluating the overall system.

Some reasons for understanding and evaluating the entire industrial process as a whole are given below:

- Changes in the process equipment can have a major impact on the efficiency
Changes in the air pollution control device can affect the ability of the process hoods to capture the pollutants at the point of generation.

• The operating data from one unit in the system can be valuable in evaluating the operating conditions in another unit in the system.

• Hoods and fans can influence the efficiency of the air pollution control equipment and the release of fugitive emissions from the process equipment.

Flowcharts
Flowcharts are a useful tool when you want to evaluate the performance of an entire system because they provide a means for organizing and presenting operating data. More specifically, flowcharts can be used for the following purposes:

• Evaluating process operating changes that are affecting control device performance
• Identifying instruments that are not working properly
• Identifying health and safety problems
• Communicating effectively

As discussed later (see lesson on Flowchart Diagrams) an expanded block diagram flowchart has been adopted for use in this Course. Major components such as baghouses are shown as a simple block rather than a complex sketch resembling the actual baghouse. A set of conventional instrument symbols and major equipment symbols have been adopted primarily from the conventional chemical engineering practice.

Most of the standard symbols are reproduced on the back of the flowchart sheet so that you do not need to remember any of the specific information included within this course. The form is basically “self contained.”

Flowchart Symbols
A complete flowchart consists of several symbols representing major and minor pieces of equipment and numerous material flow streams. It is important to be able to differentiate between the various types of material flow streams without sacrificing simplicity and clarity.

Material Streams
The recommended symbols selected for the material streams are presented in Figure 10-21.
Figure 10-21. Material stream symbols
Gas flow streams are shown as two parallel lines spaced slightly apart and therefore
appear larger than other streams. This size difference is important so that the inspector
can quickly scan the flowchart and differentiate between gas and liquid material flow
streams. Sections of ductwork connecting one major piece of equipment to another
are labeled with an alphabetic character.

Important liquid and solid material flow streams are shown as solid, single lines.
Diamonds with enclosed numbers are used to identify each of the streams.

To avoid cluttering the drawing, some of the liquid and solid material streams for
which operating data will not be necessary are unnumbered. These types of streams are
often called utility streams for a couple of reasons. They provide necessary materials to
the system being shown and the characteristics of these streams are relatively constant.
Typical utility streams for air pollution control equipment systems include make-up
water, cooling water, and low-pressure steam. Natural gas, oil, and other fossil fuels can
also be treated as utility streams to simplify the drawings. Instead of the numbered
diamonds, these utility streams are identified either by using one of the codes listed in
Table 10-3 or by a one- or two-word title. The codes or work titles are placed next to a
“stretched S” symbol, which is used to indicate that the source of the utility stream is
outside the scope of the drawing.

<table>
<thead>
<tr>
<th>Table 10-3. Codes for Utility Streams</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cal</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>CA</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>CD</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>CW</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>Gas</td>
</tr>
</tbody>
</table>

**Major Components of Systems**

A square or rectangle is used to denote major equipment such as the air pollution
control devices, tanks and vessels, or process equipment. Fans are denoted using a
relatively large circle with a set of tangential lines to indicate the discharge point. A
stack is shown as a slightly tapered rectangle. All of these symbols are shaded or filled
with crosshatched diagonal lines so that it is easy to pick out the major equipment
items from the gas handling ductwork and other streams leaving these units as shown
in Figure 10-22.
The items treated as major equipment depend on the overall complexity of the system being drawn and on individual preferences. These decisions are determined based primarily on the types of data and observations that are possible and the level of detail that is necessary to evaluate the performance of the overall system.

![Major equipment symbols](image)

![Identification of emission points](image)

The stack (or emission discharge point) is obviously important due to the visible emission observations and the presence of continuous emission monitors and stack sampling ports in some systems. The emission points, which should be subject to Method 9 or Method 22 visible emission observations, are identified by a set of inverted triangles immediately above the source as shown in Figure 10-23. These are numbered whenever there is any possibility of confusing different sources within a single industrial complex. The numbers used in the triangles should correspond with the emission point identification numbers used in the inspector’s working files. Typical identification numbers $E_1, E_2 \ldots E_n$ are used for enclosed emission points such as stacks and $F_1, F_2 \ldots F_n$ are used for fugitive emission points such as storage piles and material handling operations.

**Minor Components of Systems**

A number of small components in air pollution control systems should be shown on the block-diagram-type flowcharts in order to clarify how the system operates. A
partial list of these minor equipment components is provided in Table 10-4.

<table>
<thead>
<tr>
<th>Fabric Filters</th>
<th>Wet Scrubbers</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bypass dampers</td>
<td>Pumps</td>
</tr>
<tr>
<td>Relief dampers</td>
<td>Nozzles</td>
</tr>
<tr>
<td>Outlet dampers</td>
<td>Manual valves</td>
</tr>
<tr>
<td>Reverse air fans</td>
<td>Automatic valves</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Carbon Adsorbers and Oxidizers</th>
</tr>
</thead>
<tbody>
<tr>
<td>Indirect heat exchangers</td>
</tr>
<tr>
<td>Fans</td>
</tr>
</tbody>
</table>

Symbols for the minor components listed in Table 10-4 are shown in Figure 10-24. Note that all of these symbols are relatively simple and quick to draw.

**Instruments**

The presence of an instrument or a sampling port is indicated by a small circle connected to a stream line by a short dashed line as shown in Figure 10-25.

The type of instrument is indicated using the symbols listed in Table 10-5.
Table 10-5. Instrument Codes

<table>
<thead>
<tr>
<th>Code</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Motor current</td>
</tr>
<tr>
<td>CEM</td>
<td>Continuous emission monitor</td>
</tr>
<tr>
<td>Den</td>
<td>Density</td>
</tr>
<tr>
<td>F</td>
<td>Flow</td>
</tr>
<tr>
<td>L</td>
<td>Liquid level</td>
</tr>
<tr>
<td>LEL</td>
<td>Lower explosive limit</td>
</tr>
<tr>
<td>MP</td>
<td>Measurement port</td>
</tr>
<tr>
<td>Op</td>
<td>Opacity</td>
</tr>
<tr>
<td>P</td>
<td>Gas or liquid pressure</td>
</tr>
<tr>
<td>pH</td>
<td>Liquid or slurry pH</td>
</tr>
<tr>
<td>Δp</td>
<td>Static pressure drop</td>
</tr>
<tr>
<td>SP</td>
<td>Gas static pressure</td>
</tr>
<tr>
<td>SSP</td>
<td>Stack sampling port</td>
</tr>
<tr>
<td>T</td>
<td>Temperature</td>
</tr>
<tr>
<td>V</td>
<td>Vacuum gauge</td>
</tr>
<tr>
<td>VOC</td>
<td>Low concentration VOC monitor</td>
</tr>
<tr>
<td>W</td>
<td>Weight</td>
</tr>
</tbody>
</table>

Instruments such as manometers and dial-type thermometers can only be read at the gauge itself. These indicating gauges, shown in Figure 3-5, are simply denoted by the instrument circle and the instrument code. More sophisticated instruments with panel-mounted readout gauges (normally in the control room) are indicated using a line horizontally bisecting the instrument circle. In this case, the instrument code is placed directly above the line. When the instrument readout is a continuous strip chart recorder or data acquisition system, the letter “R” for “Recording” is placed below the line.

**Materials of Construction**

The materials of construction are relevant whenever there has been or may be a serious corrosion problem that could affect either system performance or safety. On a single-page-format-type of flowchart, it is impractical to specify the exact type of material and protective coatings on each vulnerable component because there are several hundred combinations of materials and coatings in common use. However, the general type of material in certain selected portions of the system may be important. For example, it would be helpful to know that a stack discharging high concentrations of sulfuric acid vapor is composed of carbon steel because this material is easily attacked by sulfuric acid. The stack platform and access ladders could be vulnerable to failure as the corrosion problem gets progressively worse. A small set of symbols is presented in Table 10-6 for identifying materials of construction.

<table>
<thead>
<tr>
<th>Material</th>
<th>Code</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon Steel</td>
<td>CS</td>
</tr>
<tr>
<td>Rubber lined</td>
<td>RL</td>
</tr>
<tr>
<td>Fiberglass reinforced plastic</td>
<td>FRP</td>
</tr>
<tr>
<td>Stainless steel</td>
<td>SS</td>
</tr>
<tr>
<td>Wood</td>
<td>WD</td>
</tr>
<tr>
<td>Nickel alloy</td>
<td>N</td>
</tr>
</tbody>
</table>

These symbols should be placed next to the major equipment item (e.g., stack, fan, air pollution control device) or the gas handling ductwork segment.

**Diagrams**

**Basic Flowcharting Techniques**

Flowcharts can serve many purposes and therefore many levels of sophistication in...
Flowchart preparation exist. Some of the most complex are design-oriented piping and instrumentation drawings (termed P&I drawings), which show every major component, valve, and pipe within the system. Even a drawing for a relatively simple system (or part of a system) can have more than 500 separate items shown on it. Conversely, a simple block diagram used as a field sketch may have only 3 to 5 symbols on the drawing.

Flowcharts for air pollution control studies should be relatively simple. Generally, you need more equipment detail than shown on a simple block diagram, but far less information than provided by the standard P&I drawing. The flowcharts should not be so cluttered with system design details that it is difficult to include present system operating conditions to help identify health and safety risks and performance problems. Since these are primarily “working” drawings, they must be small enough to be carried easily while walking around the facility. The flowcharts should not also require a lot of time to prepare or to revise.

For these reasons, an expanded block diagram flowchart has been adopted for use in this course. In this type of flowchart, only the system components directly relevant to the study are included. Major components such as baghouses are shown as a simple block rather than a complex sketch resembling the actual baghouse. Most minor components and material flow streams are omitted to avoid cluttering the drawing.

The size of the flowchart is designed so that it fits entirely on a single 8 ½ by 11 inch page and can be carried on a standard clipboard or in a notebook. Furthermore, most of the standard symbols are reproduced on the back of the flowchart sheet.

**Flowchart Diagrams**

An example flowchart for a relatively complicated air pollution source, a waste solvent incinerator, is shown in Figure 10-26. The process equipment in this example consists of a starved air modular incinerator with primary and secondary chambers. The air pollution control system consists of a venturi scrubber followed by a mist eliminator.
Figure 10-26. Example of flowchart of a waste solvent system

The primary and secondary chambers of the waste solvent incinerator have been shown separately because data from each chamber is important to the inspection. However, many components of the incinerator and wet scrubber systems have not been shown because their operating conditions are not central to the potential air pollution emission problems or health and safety problems.

The following problems illustrate how flowcharts can be helpful during the inspection of air pollution control systems. They serve as a tool for organizing relevant data and determining what needs further investigation.

1. Determine whether or not the operating data is consistent and logical.
2. Compare current data against site-specific baseline data.
3. Determine specific areas that may need emphasis during the inspection.
4. Determine potential health and safety problems that may be encountered during the inspection.

Example 10-8  A regulatory agency is conducting an inspection of a soil remediation unit at a hazardous waste site. This site is an abandoned chemical plan where several nonvolatile carcinogens (chlorinated organic compounds) are present in old lagoons. The plant uses a rotary kiln for destruction of the carcinogens and two side-by-side pulse jet fabric filters for control of particulate matter generated in the kiln. Based on the data shown in Figure 10-28 (Present Situation) and Table 10-7 (Baseline Data), determine the following:

A. Is the operating data for the system consistent and logical?
B. Do any important discrepancies exist between the current and baseline data?
C. What areas of the facility should be emphasized during the inspection?
D. What health and safety issues should be considered during the inspection?

Figure 10-27. Example flowchart of a hazardous waste incinerator and pulse jet baghouse system.

<table>
<thead>
<tr>
<th>Table 10-7. Baseline Data for the Hazardous Waste Incinerator</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Location</strong></td>
</tr>
<tr>
<td>Kiln hood</td>
</tr>
<tr>
<td>Evaporative cooler inlet</td>
</tr>
<tr>
<td>Evaporative cooler outlet</td>
</tr>
<tr>
<td>Baghouse inlet</td>
</tr>
<tr>
<td>Baghouse outlet</td>
</tr>
<tr>
<td>Duct E</td>
</tr>
<tr>
<td>Stack</td>
</tr>
</tbody>
</table>

**Solution Part A** Determine if the operating data for the system is consistent and logical. There should be logical trends in the gas temperatures, gas static pressures, gas oxygen concentrations (combustion sources) and other parameters along the direction of gas flow.

For this example, the gas temperature and static pressure data are listed in Tables 10-8 and 10-9 in the direction of gas flow.

<table>
<thead>
<tr>
<th>Table 10-8. Gas Temperature Profile for the Hazardous Waste Incinerator °C</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Location</strong></td>
</tr>
<tr>
<td>Kiln hood</td>
</tr>
<tr>
<td>Evaporative cooler inlet</td>
</tr>
<tr>
<td>Evaporative cooler outlet</td>
</tr>
<tr>
<td>Baghouse inlet</td>
</tr>
<tr>
<td>Baghouse outlet</td>
</tr>
</tbody>
</table>

In physics, the Joule-Thomson effect, or Joule-Kelvin effect is a process in which the temp. of a real gas is either decreased or increased by letting the gas expand freely at constant enthalpy.
### Table 10-9. Gas Static Pressure Profit for the Hazardous Waste Incinerator (in WC)

<table>
<thead>
<tr>
<th>Location</th>
<th>Present</th>
<th>Baseline</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kiln hood</td>
<td>-0.10</td>
<td>-0.1</td>
</tr>
<tr>
<td>Evaporative cooler inlet</td>
<td>-1.0</td>
<td>-1.0</td>
</tr>
<tr>
<td>Evaporative cooler outlet</td>
<td>No Data</td>
<td>No Data</td>
</tr>
<tr>
<td>Baghouse inlet</td>
<td>No Data</td>
<td>No Data</td>
</tr>
<tr>
<td>Baghouse outlet</td>
<td>-3.2</td>
<td>-5.1</td>
</tr>
<tr>
<td>Duct E</td>
<td>+0.4</td>
<td>-1.5</td>
</tr>
<tr>
<td>Stack</td>
<td>-0.1</td>
<td>-1.0</td>
</tr>
</tbody>
</table>

The gas temperature and static pressure trends through the system are both logical. The gas temperatures are at a maximum at the discharge of the combustion source, and they decrease throughout the system. The gas temperature at the fan outlet is not provided for this example. Note that sometimes gas temperature at the fan outlet is higher than that at the fan inlet due to compression that occurs as the gas moves through the fan (the Joule-Thomson effect). The static pressures become progressively more negative as the gas approaches the fan. After the fan, the static pressure of the system significantly increases, as expected. Since the set of plan instruments provides consistent and logical profiles through the system, they are probably relatively accurate.

**Solution Part B**  Compare the current data against the site-specific baseline data to the extent that it is available.

**Step 1.** Compare the current temperature data against the site-specific baseline data.
a. Evaluate the temperature data for Duct B using Figure 10-28 and Table 10-8

The 160°C temperature drop (from 819°C to 659°C) in the short duct (B) between the kiln and the evaporative cooler is relatively new. The baseline data indicates that the previous temperature drop was 25°C due to radiative and convective heat losses from the refractory-lined duct. The significantly higher temperature drop presently occurring across this short section of ductwork indicates that air infiltration is probably happening. This air infiltration could reduce the amount of combustion being pulled from the kiln and thereby cause fugitive emissions from the kiln. A check for fugitive emissions should be included in the scope of the inspection.

b. Evaluate the destruction efficiency of the rotary kiln using the kiln outlet temperature data using Figure 10-28 and Table 10-8.

The primary function of this portable plant is to incinerate the contaminated soil. It is apparent from the flowchart that the most useful single parameter for evaluating the destruction efficiency of the rotary kiln system is the kiln outlet temperature monitored by the temperature gauge on the left side of duct B. The present value of 819°C is well with the baseline data obtained during the trial burn tests in which the unit demonstrated good performance. Accordingly, it appears that the unit is presently in compliance.
c. Evaluate the temperature data for the evaporative cooler. See Figure 10-28 and Table 10-8.

The evaporative cooler is important primarily because it protects the temperature-sensitive Nomex® bags used in the downstream pulse jet baghouses. It is clear from the flowchart that presently there is a gas temperature drop of 425°C across the evaporative cooler. This fact combined with an observed outlet gas temperature of 234°C demonstrates that this unit is operating as intended. It is not necessary to climb to the top of the unit to check the spray nozzles.

d. Evaluate the temperature data for the baghouse. See Figure 10-28 and Table 10-9

The flowchart data indicates there is a severe temperature drop across the baghouse (28°C). This should be included in the field evaluation.

Step 2. Compare the current pressure drop data against the site-specific baseline data.

a. Evaluate the static pressure data across the kiln using Figure 10-29 and Table 10-8.

The pressure readings are in agreement for the baseline data and the present data.

b. Evaluate the static pressure data from the evaporative cooler inlet to the baghouse outlet. See Figure 10-27 and Table 10-9.

The baseline static pressure drop is 4.1 in WC compared with a present pressure drop reading of 2.2 in WC. Pressure drops across evaporative coolers tend to remain constant. However, the pressure drop across baghouses can vary due to changes in emission loading or a malfunction. Emission loading is directly related to a pressure drop increase. A decrease in pressure drop may result from air inleakage at the bag connections points. Air inleakage can also occur due to worn or torn bags.

c. Evaluate the static pressure data from the baghouse exit to the stack. See Figure 10-27 and Table 10-9.

The static pressure increase created by the fan (3.6 in WC) is similar for the baseline and present conditions. The static pressure drop from the fan exit to the stack is also in agreement.

**Solution Part C** Determine the areas that should be emphasized during inspection. They are as follows:

1. Check for air infiltration in Duct B.

2. Check for fugitive emissions from rotary kiln.

3. Investigate reasons for temperature drop of the pulse jet baghouses.

4. Check for air inleakage across the pulse jet baghouse.
**Solution Part D** Determine what health and safety issues should be considered during the inspection.

The pulse jet baghouse should be one of the main areas evaluated during the field portion of the inspection. However, this work must be conducted carefully in order to minimize safety hazards. The roof of the unit should be avoided because it is an uninsulated metal surface at 175°C (349°F). The soles of safety shoes could begin to melt and thereby cause a fall. Furthermore, there is a slight possibility of falling through the roof of the baghouse. The gas temperature drop of 28°C across the baghouse indicates severe air infiltration that may be caused by corrosion. If so, the roof may have been weakened. Corrosion is very likely in this process due to the formation of hydrochloric acid and water vapor in the kiln.

The waste being burned in this portable plan includes several suspected carcinogens. This should be noted on the flowchart to serve as a reminder to stay out of areas where inhalation problems or skin absorption hazards could exist.

**Summary of Health and Safety Issues**

1. Avoid roof of pulse jet baghouse.

2. Remain aware that chemicals in process are possible carcinogens. Avoid areas where inhalation or absorption may become dangerous.

**Example 10-9** A company is routinely evaluating the performance of a venturi scrubber serving a hazardous waste incinerator. They are using an Enhanced Monitoring Protocol that is based on the static pressure drop gauge across the venturi. Answer the following questions based on the data shown in Figure 10-29 (below).

a. Is there any reason to believe that the venturi scrubber pressure drop gauge is malfunctioning?

b. Is there any reason to be concerned about fugitive emissions from the emergency bypass stack? (The emergency bypass stack has the stack cap covering the outlet.)

The present data and the corresponding baseline data are provided in Tables 10-10 and 10-11.
Figure 10-29. Example flowchart of a hazardous waste incinerator and venturi scrubber system

<table>
<thead>
<tr>
<th>Table 10-9. Static Pressures and Static Pressure Drops (in WC)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Stack Pressures</strong></td>
</tr>
<tr>
<td>Incinerator primary chamber</td>
</tr>
<tr>
<td>Duct B</td>
</tr>
<tr>
<td>Mist eliminator inlet</td>
</tr>
<tr>
<td>Fan Inlet (Duct D)</td>
</tr>
<tr>
<td>Stack</td>
</tr>
<tr>
<td><strong>Stack Pressure Drop</strong></td>
</tr>
<tr>
<td>Venturi Scrubber</td>
</tr>
<tr>
<td>Mist eliminator</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 10-9. Gas Temperature (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Present</strong></td>
</tr>
<tr>
<td>Incinerator secondary chamber</td>
</tr>
<tr>
<td>Duct B</td>
</tr>
<tr>
<td>Fan Inlet</td>
</tr>
<tr>
<td>Stack</td>
</tr>
</tbody>
</table>
Solution Part A  First, evaluate the quality of data before attempting to evaluate the system. There should be logical trends for the static pressures, gas temperatures, and other relevant parameters.

The static pressure and pressure drop data have been combined into a single graph (Figure 10-31), which can be used to evaluate the static pressures along the entire gas flow path. The present static pressure drop data for the venturi scrubber does not make sense. The present mist eliminator inlet static pressure and fan inlet static pressure data suggest that the static pressure drop across the venturi scrubber should be higher than indicated by the gauge. It is possible that the venturi scrubber pressure drop gauge is malfunctioning and that the actual static pressure drop is relatively similar to the baseline value of 36 in WC.

Solution Part B  There is no reason to suspect fugitive emissions from the emergency bypass stack. The static pressures upstream and downstream of the bypass stack are negative. Accordingly, ambient air could leak into a poorly sealed stack. Untreated combustion gas could not escape through gaps in the stack seal.

Flowcharts Summary
A flowchart of the process system can be used to:

- Identify changes in control device performance due to process changes
- Identify instruments that are not consistent with other similar instruments in the system
- Communicate effectively with other personnel
- Avoid potential health and safety hazards

Flowcharts used for agency inspections should be prepared prior to or in the early stages of the inspection. If flowcharts for the system being inspected have been prepared previously, they should be reviewed prior to the on-site work and updated as necessary.
Review Problems

1. Calculate the hood static pressure if the hood coefficient of entry is 0.49 and the gas flow rate through a 1.5-foot diameter duct from the hood is 6,200 ft³/min. Use standard temperatures and pressures.
   a. 2.1 in WC
   b. 1.15 in WC
   c. 0.38 in WC
   d. 0.85 in WC

2. Find the farthest distance away that a flanged hood, 6 in by 12 in, can be placed away from the contaminant source and maintain the capture velocity of 300 fpm and a volumetric flow rate of 2,000 acfm. The equation for a flanged hood is:
   \[ Q = 0.75v_b(10X^2 + A_h) \]
   a. 15 inches
   b. 24 inches
   c. 3 inches
   d. 11 inches

3. Estimate the rotational speed of a belt-driven centrifugal fan based on the following data:
   Motor rotational speed, RPM\(_{Motor}\) = 1778 rpm
   Motor sheave diameter, D\(_{Motor}\) = 8 in.
   Fan sheave diameter, D\(_{Fan}\) = 14 in.
   a. 1,239 rpm
   b. 2,000 rpm
   c. 400 rpm
   d. 1,016 rpm

4. A system consists of the following components (in order): hood, fabric filter, centrifugal fan, and stack. The fabric filter static pressure drop has increased from 4.5 inches of water to 6.5 inches of water. If the fan dampers do not move to compensate for this change, what will happen to the hood static pressure?
   a. It will be less negative (closer to zero)
   b. It will be more negative
   c. It will become positive
   d. It will remain unchanged

5. A centrifugal fan is moving 1,000 cubic feet of air per minute at a temperature of 450°F and a fan inlet pressure of -15 inches of water. What will the actual air flow rate be if the gas temperature decreases to 68°F, the inlet pressure remains the same, and the fan rotational speed remains the same?
   a. The air flow rate will increase to 1,723 acfm
   b. The air flow rate will decrease to 580 acfm
   c. The air flow rate will remain at 1,000 acfm
6. A centrifugal fan is operating with a motor current of 120 amps. The gas density entering the fan during normal operation is 0.045 pounds per cubic foot. Estimate the motor current at standard conditions when the gas density is approximately 0.075 pounds per cubic foot.
   a. 500 amps
   b. 200 amps
   c. 159 amps
   d. 90 amps

7. The static pressure drop through a section of ductwork is -1.2 inches of water when the gas flow rate is 5,000 acfm. Estimate the static pressure drop across this section if the gas flow rate increases to 8,000 acfm. Assume that there are no gas density changes associated with the increased gas flow rate.
   a. -3.07 in WC
   b. -1.1 in WC
   c. -2 in WC
   d. -4 in WC

8. The hood capture efficiency is 92% and the wet scrubber control system has a collection efficiency of 95%. If the process served by this system is generating 140 pounds of pollutant per hour, calculate the fugitive emissions and the stack emissions.
   a. 20.50 lb/hr fugitive emissions and 9.80 lb/hr stack emissions
   b. 1.50 lb/hr fugitive emissions and 0.80 lb/hr stack emissions
   c. 11.2 lb/hr fugitive emissions and 6.4 lb/hr stack emissions
   d. 14.0 lb/hr fugitive emissions and 3.54 lb/hr stack emissions

9. Assume a fan is presently operating with the following conditions: 20,000 acfm, 2.5 in WC fan static pressure, 400 rpm, and 12 brake horsepower. Using the fan laws determine the new rpm, brake horsepower, and static pressure when the volumetric increases to 22,500 acfm.
   a. 490 rpm, 19.1 bhp, 5.7 in WC
   b. 350 rpm, 9.5 bhp, 3.9 in WC
   c. 400 rpm, 10.2 bhp, 2.8 in WC
   d. 450 rpm, 17.1 bhp, 3.2 in WC

10. What would happen to the desired operating point of a fan if a hole developed in the inlet ductwork. Which characteristic curve will shift, what will happen to the operating point, volumetric flow rate, and hood static pressure?
    a. The system characteristic curve will shift down, the volumetric flow rate will increase, and the hood static pressure will increase.
    b. The system characteristic curve will shift down, the volumetric flow rate will decrease, and the hood static pressure will decrease.
    c. The system characteristic curve will shift down, the volumetric flow rate will increase, and the hood static pressure will decrease.
    d. The system characteristic curve will shift up, the volumetric flow rate will increase, and the hood static pressure will decrease.
Use Figure 10-32 to answer questions 11 and 12.

![Example flowchart](image-url)

Figure 10-31. Example flowchart

11. Which static pressure reading appears to be illogical according to the flowchart?
   a. Duct a
   b. Duct b
   c. Duct c
   d. They all appear logical

12. The temperature in Duct A was checked by plant personnel and determined to be correct. Which of the other temperature readings appears to be illogical according to the flowchart?
   a. Duct B
   b. Duct C
   c. Duct D
   d. They all appear logical.
Review Problem Solutions

1. Calculate the hood static pressure if the hood coefficient of entry is 0.49 and the gas flow rate through a 1.5-foot diameter duct from the hood is 6,200 ft³/min. Use standard temperatures and pressures.

   b. 1.15 in WC

   **Solution**

   \[ SP_h = V_{p_d} + h_c = (1 + F_h) V_{p_d} \]

   Calculate the velocity pressure:

   \[ v = \frac{Q}{A} = \frac{6,200 \text{ ft}^2/\text{min}}{\pi (1.5 \text{ ft})^2 / 4} = 3,508 \text{ ft}^2/\text{min} \]

   \[ V_{p_d} = p_g \left( \frac{v_d}{1,096.7} \right)^2 = 0.075 \text{ ft}^2/\text{min} \left( \frac{3,508 \text{ ft}^2/\text{min}}{1,096.7} \right)^2 = 0.77 \text{ in WC} \]

   Calculate the hood static pressure:

   \[ SP_h = (1 + 0.49)0.77 \text{ in WC} = 1.15 \text{ in WC} \]

2. Find the farthest distance away that a flanged hood, 6 in by 12 in, can be placed away from the contaminant source and maintain the capture velocity of 300 fpm and a volumetric flow rate of 2,000 acfm. The equation for a flanged hood is:

   \[ Q = 0.75v_h (10X^2 + A_h) \]

   **Solution**

   \[ x = \sqrt{\frac{Q}{0.75v_h} - A_h} = \sqrt{\frac{2,000 \text{ ft}^3/\text{min}}{0.75 (300 \text{ ft}^2/\text{min})} - \frac{6\text{in}(12\text{in})}{144 \text{ in}^2/\text{ft}^2}} = 0.92 \text{ ft} = 11 \text{ in} \]
3. Estimate the rotational speed of a belt-driven centrifugal fan based on the following data:
   - Motor rotational speed, \( \text{RPM}_{\text{Motor}} = 1778 \text{ rpm} \)
   - Motor sheave diameter, \( D_{\text{Motor}} = 8 \text{ in.} \)
   - Fan sheave diameter, \( D_{\text{Fan}} = 14 \text{ in.} \)

   d. \( 1,016 \text{ rpm} \)

   **Solution**
   
   \[
   \text{RPM}_{\text{Fan}} = \frac{\text{RPM}_{\text{Motor}} \times D_{\text{Motor}}}{D_{\text{Fan}}} = \frac{1,778 \text{ rpm} \times 8 \text{ in.}}{14 \text{ in.}} = 1,016 \text{ rpm}
   \]

4. A system consists of the following components (in order): hood, fabric filter, centrifugal fan, and stack. The fabric filter static pressure drop has increased from 4.5 inches of water to 6.5 inches of water. If the fan dampers do not move to compensate for this change, what will happen to the hood static pressure?
   a. It will be less negative (closer to zero)

5. A centrifugal fan is moving 1,000 cubic feet of air per minute at a temperature of 450°F and a fan inlet pressure of -15 inches of water. What will the actual air flow rate be if the gas temperature decreases to 68°F, the inlet pressure remains the same, and the fan rotational speed remains the same?
   c. The air flow rate will remain at 1,000 acfm

6. A centrifugal fan is operating with a motor current of 120 amps. The gas density entering the fan during normal operation is 0.045 pounds per cubic foot. Estimate the motor current at standard conditions when the gas density is approximately 0.075 pounds per cubic foot.
   b. 200 amps

   **Solution**
   
   \[
   I_{\text{STP}} = I_{\text{actual}} \left( \frac{P_{\text{STP}}}{P_{\text{actual}}} \right) = 120 \text{amps} \left( \frac{0.075 \text{ lb/m}^3}{0.045 \text{ lb/m}^3} \right) = 200 \text{amps}
   \]

7. The static pressure drop through a section of ductwork is -1.2 inches of water when the gas flow rate is 5,000 acfm. Estimate the static pressure drop across this section if the gas flow rate increases to 8,000 acfm. Assume that there are no gas density changes associated with the increased gas flow rate.
   a. -3.07 in WC

   **Solution**
\[ \Delta SP_{\text{high flow}} = \Delta SP_{\text{low flow}} \left( \frac{Q_{\text{high flow}}}{Q_{\text{low flow}}} \right)^2 = -1.2 \text{ in WC} \left( \frac{8,000 \text{ acfm}}{5,000 \text{ acfm}} \right)^2 = -3.07 \text{ in WC} \]

8. The hood capture efficiency is 92% and the wet scrubber control system has a collection efficiency of 95%. If the process served by this system is generating 140 pounds of pollutant per hour, calculate the fugitive emissions and the stack emissions.

\[ \text{c. 11.2 lb}_m/\text{hr fugitive emissions and 6.4 lb}_m/\text{hr stack emissions} \]

**Solution**

Fugitive emissions = \((1 - 0.92) \times 1.40 \text{ lb}_m/\text{hr} = 11.2 \text{ lb}_m/\text{min} \]

Stack emissions = \((1 - 0.95)(0.92) \times 140 \text{ lb}_m/\text{min} = 6.4 \text{ lb}_m/\text{min} \]

9. Assume a fan is presently operating with the following conditions: 20,000 acfm, 2.5 in WC fan static pressure, 400 rpm, and 12 brake horsepower. Using the fan laws determine the new rpm, brake horsepower, and static pressure when the volumetric increases to 22,500 acfm.

\[ \text{d. 450 rpm, 17.1 bhp, 3.2 in WC} \]

**Solution**

\[ \frac{Q_1}{Q_2} = \frac{\text{RPM}_1}{\text{RPM}_2} \]

\[ \text{RPM}_1 = \text{RPM}_2 \cdot \frac{Q_1}{Q_2} = 400 \text{ rpm} \left( \frac{22,500 \text{ acfm}}{20,000 \text{ acfm}} \right) = 450 \text{ rpm} \]

\[ \frac{\text{BHP}_1}{\text{BHP}_2} = \left( \frac{\text{RPM}_1}{\text{RPM}_2} \right)^3 \]

\[ \text{BHP}_2 = \text{BHP}_1 \left( \frac{\text{RPM}_1}{\text{RPM}_2} \right)^3 = 12 \text{ bhp} \left( \frac{450 \text{ rpm}}{400 \text{ rpm}} \right)^3 = 17.1 \text{ bhp} \]

\[ \frac{\text{SP}_1}{\text{SP}_2} = \left( \frac{\text{RPM}_1}{\text{RPM}_2} \right)^2 \]

\[ \text{SP}_2 = \text{SP}_1 \left( \frac{\text{RPM}_1}{\text{RPM}_2} \right)^2 = 2.5 \text{ in WC} \left( \frac{450 \text{ rpm}}{400 \text{ rpm}} \right)^2 = 3.2 \text{ in WC} \]
10. What would happen to the desired operating point of a fan if a hole developed in the inlet ductwork. Which characteristic curve will shift, what will happen to the operating point, volumetric flow rate, and hood static pressure?

c. The system characteristic curve will shift down, the volumetric flow rate will increase, and the hood static pressure will decrease.

11. Which static pressure reading appears to be illogical according to the flowchart?
   d. They all appear logical

12. The temperature in Duct A was checked by plant personnel and determined to be correct. Which of the other temperature readings appears to be illogical according to the flowchart?
   a. Duct B
References

