perceptible air flow will exist except in the immediate area of the hood.

Exterior Hoods

Where enclosure of the process is impracticable, the air flow pattern in front of the hood must be such that capture velocities required to convey the contaminant to the hood opening will be maintained in the area of contaminant generation.

The method for determining, approximately, the quantity of air that must be exhausted from an unobstructed hood, without flanges, to produce design capture velocities at the point or origin, is given in Equation 1:

$$Q = V_X(10X^2 + A) \tag{1}$$

where

Q = quantity of air exhausted, cubic feet per minute.
V_X = air velocity at X-distance in feet from the hood and and on the center line of the hood, feet per minute.
X = distance along the hood center line, from the face of the hood to the point where the air velocity is V_X feet per minute, feet.
A = area of the hood opening, square feet.

Fig. 1 shows lines of equal velocities (velocity contours) for a rectangular hood opening with a side ratio of one-half. The velocities are expressed as percentages of the velocity at the opening. Studies have established the principle of *similarity of contours* which states that the positions of the velocity contours for any hood (when the contours are expressed in terms of the average velocity at the hood opening) are purely functions of the shape of the hood. Extensive studies[4, 11, 12] have revealed variations in values of such velocity contours for long narrow slots and for hoods with one or more planes shielded against air flow.

For smaller hoods, flanges which are usually 3 to 6 in. wide surrounding the hood opening usually will improve

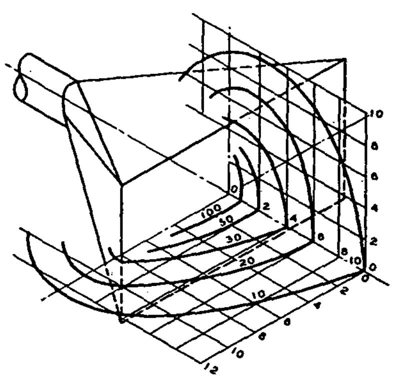

Fig. 1 .... Velocity Contours for Rectangular Opening with a Side Ratio of One-Half. Contours are Expressed as Percentages of the Velocity at the Opening

Table 2 .... Summary of Exhaust Rates Required with Common Types of Exterior Hoods Applied to Cold Processes Only[13]

X-Distance Inches	Simple Opening, with or without Flange or Taper, but No Plane Flanking Openings[a] Exhaust Rate Cfm	Simple Rectangular Opening Flanked by Plane Parallel to Axis[b] Exhaust Rate Cfm
2	25–75	25–50
4	50–100	50–75
6	100–200	75–150
8	200–400	150–300
10	300–600	200–400
12	400–800	300–600
15	600–1200	400–800
18	900–1500	600–900
21	1200–1800	700–1000
24	1500–2200	800–1200
30	2500–3500	1200–1800
36	3000–4500	1500–2200
42	4000–6000	2000–3000
48	5000–7000	2500–3500

X-Distance	Single slots flanked by parallel plane on which contamination process occurs For X-distance (i.e., width of plane) always less than ½ length of slot[c]
Less than 2 ft	Exhaust rate 125 to 150 cfm per sq ft of plane area.
More than 2 ft	Exhaust rate 75 to 125 cfm per sq ft of plane area.

[a] Refers to simple hood opening that performs in accordance with Equation 1.

[b] Refers to hood having a complete flanking plane parallel to axis which prevents flow of air from half of normal air supply zone, and performs in accordance with the equation, $Q = V_X(5X^2 + A)$. Example: rectangular hood opening resting on bench top. Includes single slots, so flanked, for X-distance greater than ½ slot length.

[c] If X-distance is greater than ½ slot length, consider it as a single opening flanked by a plane (see upper right column). With exhaust rate divided between 2 slots, an additional safety factor is provided.

From *Plant and Process Ventilation*, by W. C. L. Hemeon (Industrial Press, New York).

the air flow in front of the hood and will reduce the air volume required to provide desired capture velocities by as much as 25 percent.

The exhaust volume calculated for an exterior hood by different designers may vary greatly due to their selection of different empirical design velocities. Usually capture velocity design values are in the 50 to 200 *fpm* range.

The exhaust rates calculated for large exterior hoods may become needlessly large if the capture velocity is selected blindly. Whether on not an individual particle or a small wisp of gas or smoke in motion away from the hood is captured before it escapes from the zone of influence created by the hood is determined not only by the specific air velocity (design capture velocity) at the immediate point of contaminant release but also by the depth of the moving air curtain it must traverse before it gets beyond the influence of the hood. The capturing force of the air flow in front of any exhaust hood is the summation of all the separate velocities beyond (away from the hood) the point of contaminant release multiplied by the distance through which each velocity acts; i.e., force times distance, the force being a function of the velocity. For large hoods, or more accurately for large exhaust rates, the depth of the zone of influence is much greater than for small exhaust rates and, also for the former, the distance between successive velocity contours is greater. Further effectiveness for larger volumes can be obtained where the air supply source produces a *sweeping* flow toward the zone of contaminant release.

Table 2 presents design data by Hemeon[13] applicable to exterior hoods employed for exhausting contaminants originating in a *cold process* only. These values are based on Equation 1 wherein varying values of V_x have been employed according to a scale that takes account of velocity depth factors discussed in the preceding paragraph, and where X-distance is taken as the distance from the hood face to the point where any high velocity air currents engendered by the contaminating process itself have expended their energy. In addition, the ranges of exhaust volumes which have been found satisfactory in practice for many operations are reported in Table 3. These data provide a means of checking the calculated rates for specific applications.

SPECIAL EXHAUST REQUIREMENTS

It is important to note that certain operations may require exhaust volumes in excess of the quantities based on design data from Table 1 and Equation 1. Typical reasons for increased ventilation rates include:

1. Induced air flow caused by the thermal or stack effect from sources of extreme heat.
2. Induced air flow caused by falling granular material in large quantities through considerable height, or by internal rotating parts such as some types of crushers, knives, or macerators. (See discussion in later section Induced Air Flow and Table 4.)
3. Exhaust volumes *insufficient* to dilute mixtures of combustible vapor and air to less than 20 percent of the lower explosive limit of the combustible.
4. Room air currents caused by cross drafts, spot cooling, motion of machinery or operators. Cross drafts are of great significance in the exhaust of hot processes with high canopy hoods.
5. Design volumes, especially in the case of excess heat control and solvent vapors, may be based on dilution systems where exhaust volumes are selected to keep contaminant concentrations below design levels. See Reference 3, Chapter 10 of the 1961 GUIDE AND DATA BOOK.
6. Local or State regulations may specify larger exhaust volumes for specific operations.

Exhaust volume requirements for many specific operations have been listed in Tables 3, 5, 6, and 7.

Exhaust of Hot Processes

In designing local exhaust hoods for hot processes, it is necessary to estimate the rate of delivery of hot air to the hood by the convection column. The exhaust capacity should exceed this by an amount sufficient to create a velocity in the excess air flowing into the envelope surrounding the convection column that will prevent escape of the heated air at the edges of the hood. Unless distances from the heat source to the receiving hood are small (possibly under 3 ft), the quantity of heated air entering the hood is increased substantially by the induction and turbulent mixing of large quantities of room air as the distance above the heat source increases. Unless the exhaust hood can handle this total volume, spillage of contaminated mixture will occur.

The exhaust rate for satisfactory control of hot processes can be kept at a minimum by enclosing the hot processes as completely as possible and exhausting from the top of the enclosure.

Low canopy hoods rank next to enclosures in economy of air flow. High canopy hoods must handle greatly increased amounts of air over that in the heated stream itself, in order to receive and dispose of the air entrained by the convection column from far below. Where lateral exhaust ventilation must be employed, the air flow toward the hood has to overcome the tendency of the heated air to rise and also must sweep it into the hood. Consequently, the required exhaust air flow is very much larger than for hoods that simply capture the upward flow of heated air. A combination of an exhaust hood with a positive jet of air blown across the top of the equipment and directed into the hood may be required above large pieces of equipment.

Hemeon has suggested[13] design Equations 2 to 6 which follow for various types of hoods.

Equation 2 is suggested for estimating the flow of heated air rising from the top of a hot body:

$$q_o = 29\sqrt[3]{H_c A_p^2 l_b} \tag{2}$$

where

q_o = air flow rate at upper limits of hot body, cubic feet per minute.
A_p = cross-sectional area of air stream at upper limits of hot body, square feet.
l_b = height of hot body, feet.
H_c = convectional heat transfer rate, Btu per minute.

The area A_p may be approximated from the dimensions of the hot body. For horizontal rods, the width of the stream is practically the diameter of the rod. For vertical planes, the air stream appears to thicken at an angle of 4 to 5 deg with the plane. For horizontal planes, the area of the air stream may be taken as equal to the area of the plane itself.

In the case of horizontal plates, in the absence of better experimental information, l_b may be taken as equal to the horizontal diameter. Chapter 4 of the 1961 GUIDE AND DATA BOOK furnishes information which may be used in estimating the heat transfer rate H_c.

The flow rate, q_c, of heated air entering the hood after turbulent mixing and dilution have taken place may be approximated for low canopy hoods above horizontal surfaces and where no heat from steam is involved, as

$$q_c = 5.4\, A_s^{2} l_b (\Delta t)^{5/4} \tag{3}$$

where

A_s = surface area of hot body, square feet.
Δt = the temperature difference, hot body to room air, Fahrenheit degrees.

Similarly, where the heat is furnished by steam from a tank of hot water,

$$q_c = 290\, A_s \sqrt[3]{l_b G} \tag{4}$$

where

G = the rate of steam formation, pounds per (square foot water surface) (minute).

(Continued on p. 255)