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INDUSTRIAL VENTILATION

A Manual of Recommended Practice

23rd Edition

1998

American Conference of Governmental Industrial Hygienists
1330 Kemper Meadow Drive
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This Edition is Dedicated in Memory of
KNOWLTON J. CAPLAN, PE, CIH, CSP

June 23, 1919 - April 11, 1997

Knowlton J. Caplan, while at the Division of Occupational Health, Michigan Department of Health, supervised the preparation of a field manual on industrial ventilation. That manual became the basis of the first edition of *Industrial Ventilation* in 1951. For the next forty-six years, the Ventilation Committee has felt Caplan's presence as we published the "Vent Manual." This 23rd edition is no different. Although "Cap" has not been an active member of the Committee for the past eleven years, his presence was felt at almost every meeting. Frequently we punctuated discussions with a quotation from Cap or a reference to one of his published works. Because of his influence, we proudly dedicate this edition to Knowlton J. Caplan.

During his 50-year career, Cap was a pioneer in the fields of industrial hygiene, industrial ventilation, and air pollution control. He conducted basic research on cyclone and fabric filter dust collectors and holds several patents for these devices. As an associate professor in the public health department of the University of Minnesota, Cap advised numerous Master's degree students in industrial hygiene, occupational health and air pollution control. He has been an instructor at the industrial ventilation conferences at Michigan State University (thirty years) and the University of Washington (ten years). As an author of more than 70 technical papers, he was a frequent presenter at the American Industrial Hygiene Conference and the American Society of Heating Refrigeration and Air Conditioning Engineers (ASHRAE) meeting. In addition he wrote chapters in *Air Pollution* by Stern, *Industrial Hygiene and Toxicology* by Patty, *Uranium Production Technology* by Harrington and Rueble, and was the Associate Editor of *Industrial Hygiene Aspects of Plant Operations: Volume 3 - Engineering Considerations in Equipment Selec-*

tion, Layout and Building Design by Crawley and Crawley.

Besides his innovative ventilation design, Cap developed a method for testing laboratory fume hoods which won the Best Paper of the Year award of the Michigan Industrial Hygiene Society in 1982, which later became the basis for the ASHRAE Standard 110-1995, "Method of Testing Laboratory Fume Hood Performance." Cap was a significant participant in the development of the ANSI Standard Z9.5-1992, "American National Standard for Laboratory Ventilation." He was the first to employ "clean air islands" to supplement local exhaust ventilation where necessary.

Cap has been active in several societies: ACGIH (Committee on Industrial Ventilation), Air Pollution Control Association (Committee on Dust, Fume, and Mist Control), American Industrial Hygiene Association (Board of Directors, Air Pollution Control Committee), ASHRAE (Industrial Ventilation, Industrial Process Air Cleaning), American National Standards Committee (Air Pollution Committee, Health and Safety Committee), American Board of Industrial Hygiene.

Cap was born in St. Louis, Missouri. He earned his bachelor's and master's degree in chemical engineering from Washington University in the 1940s. He served in the Commissioned Corps of the U.S. Public Health Services. He worked as a chemical engineer and ventilation engineer at Ralston Purina Company and Mallinckrodt Chemical, Uranium Division. He also worked for the St. Louis County Health Department and the Michigan Department of Health as an industrial hygienist. In addition, Cap did consulting work, primarily as a ventilation engineer for Industrial Health Engineering Associates (co-founder), Pace Incorporated, and Rust Environment and Infrastructure.

FOREWORD

Industrial Ventilation: A Manual of Recommended Practice is the outgrowth of years of experience by members of the ACGIH Industrial Ventilation Committee members and a compilation of research data and information on design, maintenance, and evaluation of industrial exhaust ventilation systems. The Manual attempts to present a logical method of designing and testing these systems. It has found wide acceptance as a guide for official agencies, as a standard for industrial ventilation designers, and as a textbook for industrial hygiene courses.

The Manual is not intended to be used as law, but rather as a guide. Because of new information on industrial ventilation becoming available through research projects, reports from engineers, and articles in various periodicals and journals, review and revision of each section of the Manual is an ongoing Committee project. The Manual is available as a hardbound book and on CD-ROM. In a constant effort to present the latest techniques and data, the Committee desires, welcomes, and actively seeks comments and suggestions on the accuracy and adequacy of the information presented herein.

In this 23rd edition, the Committee has made a number of minor revisions. Chapter 5 includes updated duct calculation sheets designed to aid in calculations. The "3 eye" duct friction charts have been replaced with tables to permit easier determination of the duct friction factor. The metric supplement has been

deleted and the Committee has developed a separate metric manual.

This publication is designed to present accurate and authoritative information with regard to the subject matter covered. It is distributed with the understanding that neither the Committee nor its members collectively or individually assume any responsibility for any inadvertent misinformation, omissions, or for the results in the use of this publication.

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ACKNOWLEDGMENTS

Industrial Ventilation is a true Committee effort. It brings into focus in one source useful, practical ventilation data from all parts of the country. The Committee membership of industrial ventilation and industrial hygiene engineers represents a diversity of experience and interest that ensures a well-rounded, cooperative effort.

From the First Edition in 1951, this effort has been successful as witnessed by the acceptance of the "Ventilation Manual" throughout industry, by governmental agencies, and as a worldwide reference and text.

The present Committee is grateful for the faith and firm foundation provided by past Committees and members listed below. Special acknowledgment is made to the Division of

Occupational Health, Michigan Department of Health, for contributing their original field manual which was the basis of the First Edition, and to Mr. Knowlton J. Caplan who supervised the preparation of that manual.

The Committee is grateful also to those consultants who have contributed so greatly to the preparation of this and previous editions of *Industrial Ventilation* and to Mrs. Norma Donovan, Secretary to the Committee, for her untiring zeal in our efforts.

To many other individuals and agencies who have made specific contributions and have provided support, suggestions, and constructive criticism, our special thanks.

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DEFINITIONS

Aerosol: An assemblage of small particles, solid or liquid, suspended in air. The diameter of the particles may vary from 100 microns down to 0.01 micron or less, e.g., dust, fog, smoke.

Air Cleaner: A device designed for the purpose of removing atmospheric airborne impurities such as dusts, gases, vapors, fumes, and smoke. (Air cleaners include air washers, air filters, electrostatic precipitators, and charcoal filters.)

Air Filter: An air cleaning device to remove light particulate loadings from normal atmospheric air before introduction into the building. Usual range: loadings up to 3 grains per thousand cubic feet (0.003 grains per cubic foot). Note: Atmospheric air in heavy industrial areas and in-plant air in many industries have higher loadings than this, and dust collectors are then indicated for proper air cleaning.

Air Horsepower: The theoretical horsepower required to drive a fan if there were no losses in the fan, that is, if its efficiency were 100 percent.

Air, Standard: Dry air at 70 F and 29.92 in (Hg) barometer. This is substantially equivalent to 0.075 lb/ft³. Specific heat of dry air = 0.24 btu/lb/F.

Aspect Ratio: The ratio of the width to the length; $AR = W/L$.

Aspect Ratio of an Elbow: The width (W) along the axis of the bend divided by depth (D) in plane of bend; $AR = W/D$.

Blast Gate: Sliding damper.

Blow (throw): In air distribution, the distance an air stream travels from an outlet to a position at which air motion along the axis reduces to a velocity of 50 fpm. For unit heaters, the distance an air stream travels from a heater without a perceptible rise due to temperature difference and loss of velocity.

Brake Horsepower: The horsepower actually required to drive a fan. This includes the energy losses in the fan and can be determined only by actual test of the fan. (This does not include the drive losses between motor and fan.)

Capture Velocity: The air velocity at any point in front of the hood or at the hood opening necessary to overcome opposing air currents and to capture the contaminated air at that point by causing it to flow into the hood.

Coefficient of Entry: The actual rate of flow caused by a given hood static pressure compared to the theoretical flow which would result if the static pressure could be converted to velocity pressure with 100 percent efficiency. It is the ratio of actual to theoretical flow.

Comfort Zone (Average): The range of effective temperatures over which the majority (50% or more) of adults feel comfortable.

Convection: The motion resulting in a fluid from the differences in density and the action of gravity. In heat transmission, this meaning has been extended to include both forced and natural motion or circulation.

Density: The ratio of the mass of a specimen of a substance to the volume of the specimen. The mass of a unit volume of a substance. When weight can be used without confusion, as synonymous with mass, density is the weight of a unit volume of a substance.

Density Factor: The ratio of actual air density to density of standard air. The product of the density factor and the density of standard air (0.075 lb/ft³) will give the actual air density in pounds per cubic foot; $d \times 0.075 = \text{actual density of air, lbs/ft}^3$.

Dust: Small solid particles created by the breaking up of larger particles by processes crushing, grinding, drilling, explosions, etc. Dust particles already in existence in a mixture of materials may escape into the air through such operations as shoveling, conveying, screening, sweeping, etc.

Dust Collector: An air cleaning device to remove heavy particulate loadings from exhaust systems before discharge to outdoors. Usual range: loadings 0.003 grains per cubic foot and higher.

Entry Loss: Loss in pressure caused by air flowing into a duct or hood (inches H₂O).

Fumes: Small, solid particles formed by the condensation of vapors of solid materials.

Gases: Formless fluids which tend to occupy an entire space uniformly at ordinary temperatures and pressures.

Gravity, Specific: The ratio of the mass of a unit volume of a substance to the mass of the same volume of a standard substance at a standard temperature. Water at 39.2 F is the standard substance usually referred to. For gases, dry air, at the same temperature and pressure as the gas, is often taken as the standard substance.

Hood: A shaped inlet designed to capture contaminated air and conduct it into the exhaust duct system.

Humidity, Absolute: The weight of water vapor per unit volume, pounds per cubic foot or grams per cubic centimeter.

Humidity, Relative: The ratio of the actual partial pressure of the water vapor in a space to the saturation pressure of pure water at the same temperature.

Inch of Water: A unit of pressure equal to the pressure exerted by a column of liquid water one inch high at a standard temperature.

Lower Explosive Limit: The lower limit of flammability or explosibility of a gas or vapor at ordinary ambient temperatures expressed in percent of the gas or vapor in air by volume. This limit is assumed constant for temperatures up to 250 F. Above these temperatures, it should be decreased by a factor of 0.7 since explosibility increases with higher temperatures.

Manometer: An instrument for measuring pressure; essentially a U-tube partially filled with a liquid, usually water, mercury or a light oil, so constructed that the amount of displacement of the liquid indicates the pressure being exerted on the instrument.

Micron: A unit of length, the thousandth part of 1 mm or the millionth of a meter (approximately 1/25,000 of an inch).

Minimum Design Duct Velocity: Minimum air velocity required to move the particulates in the air stream, fpm.

Mists: Small droplets of materials that are ordinarily liquid at normal temperature and pressure.

Plenum: Pressure equalizing chamber.

Pressure, Static: The potential pressure exerted in all directions by a fluid at rest. For a fluid in motion, it is measured in a direction normal to the direction of flow. Usually expressed in inches water gauge when dealing with air. (The tendency to either burst or collapse the pipe.)

Pressure, Total: The algebraic sum of the velocity pressure and the static pressure (with due regard to sign).

Pressure, Vapor: The pressure exerted by a vapor. If a vapor is kept in confinement over its liquid so that the vapor can accumulate above the liquid, the temperature being held constant, the vapor pressure approaches a fixed limit called the maximum or saturated vapor pres-

sure, dependent only on the temperature and the liquid. The term vapor pressure is sometimes used as synonymous with saturated vapor pressure.

Pressure, Velocity: The kinetic pressure in the direction of flow necessary to cause a fluid at rest to flow at a given velocity. Usually expressed in inches water gauge.

Radiation, Thermal (Heat) Radiation: The transmission of energy by means of electromagnetic waves of very long wave length. Radiant energy of any wave length may, when absorbed, become thermal energy and result in an increase in the temperature of the absorbing body.

Replacement Air: A ventilation term used to indicate the volume of controlled outdoor air supplied to a building to replace air being exhausted.

Slot Velocity: Linear flow rate of contaminated air through slot, fpm.

Smoke: An air suspension (aerosol) of particles, usually but not necessarily solid, often originating in a solid nucleus, formed from combustion or sublimation.

Temperature, Effective: An arbitrary index which combines into a single value the effect of temperature, humidity, and air movement on the sensation of warmth or cold felt by the human body. The numerical value is that of the temperature of still, saturated air which would induce an identical sensation.

Temperature, Wet-Bulb: Thermodynamic wet-bulb temperature is the temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature. Wet-bulb temperature (without qualification) is the temperature indicated by a wet-bulb psychrometer constructed and used according to specifications.

Threshold Limit Values (TLVs): The values for airborne toxic materials which are to be used as guides in the control of health hazards and represent time-weighted concentrations to which nearly all workers may be exposed 8 hours per day over extended periods of time without adverse effects (see Appendix).

Transport (Conveying) Velocity: See Minimum Design Duct Velocity.

Vapor: The gaseous form of substances which are normally in the solid or liquid state and which can be changed to these states either by increasing the pressure or decreasing the temperature.

ABBREVIATIONS

A area
 acfm flow rate at actual condition
 AHP air horsepower
 AR aspect ratio
 A_s Slot area
 B barometric pressure
 bhp brake horsepower
 bhp_a brake horsepower, actual
 bhp_s brake horsepower, standard air
 btu British thermal unit
 btuh btu/hr
 Ce coefficient of entry
 cfm cubic feet per minute
 CLR centerline radius
 D diameter
 df density factor
 ET effective temperature
 F degree, Fahrenheit
 F_d duct entry loss coefficient
 F_{el} elbow loss coefficient
 F_{en} entry loss coefficient
 fpm feet per minute
 fps feet per second
 F_s slot loss coefficient
 ft² square foot
 ft³ cubic foot
 g gravitational force, ft/sec/sec
 gpm gallons per minute
 gr grains
 h_d duct entry loss
 h_e overall hood entry loss
 h_{el} elbow loss
 h_{en} entry loss
 HEPA high-efficiency particulate air filters
 h_f loss in straight duct run
 H_f duct loss coefficient
 hp horsepower
 hr hour
 h_s slot or opening entry loss
 in inch

in² square inch
 "wg inches water gauge
 lb pound
 lbm pound mass
 LEL lower explosive limit
 ME mechanical efficiency
 mg milligram
 min minute
 mm millimeter
 MRT mean radiant temperature
 MW molecular weight
 p density of air in lb/ft³
 ppm parts per million
 psi pounds per square inch
 PWR power
 Q flow rate in cfm
 Q_{corr} corrected flow rate at a junction
 R degree, Rankin
 RH relative humidity
 rpm revolutions per minute
 scfm flow rate at standard condition
 sfpm surface feet per minute
 sp gr specific gravity
 SP static pressure
 SP_{gov} higher static pressure at junction of 2 ducts
 SP_h hood static pressure
 SP_s Sp, system handling standard air
 STP standard temperature and pressure
 TLV Threshold Limit Value
 TP total pressure
 V velocity, fpm
 V_d duct velocity
 VP velocity pressure
 VP_d duct velocity pressure
 VP_r resultant velocity power
 VP_s slot velocity pressure
 V_s Slot velocity
 V_t duct transport velocity
 W watt

Chapter 1

GENERAL PRINCIPLES OF VENTILATION

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1.1 INTRODUCTION

The importance of clean uncontaminated air in the industrial work environment is well known. Modern industry with its complexity of operations and processes uses an increasing number of chemical compounds and substances, many of which are highly toxic. The use of such materials may result in particulates, gases, vapors, and/or mists in the workroom air in concentrations that exceed safe levels. Heat stress can also result in unsafe or uncomfortable work environments. Effective, well-designed ventilation offers a solution to these problems where worker protection is needed. Ventilation can also serve to control odor, moisture, and other undesirable environmental conditions.

The health hazard potential of an airborne substance is characterized by the *Threshold Limit Value (TLV®)*. The TLV refers to the airborne concentration of a substance and represents conditions under which it is believed that nearly all workers may be exposed day after day without adverse health effects. The *time-weighted average (TWA)* is defined as the time-weighted average concentration for a conventional 8-hour workday and a 40-hour workweek which will produce no adverse health effects for nearly all workers. The TLV-TWA is usually used to determine a safe exposure level. TLVs are published annually by the American Conference of Governmental Industrial Hygienists (ACGIH); revisions and additions are made regularly as information becomes available. Appendix A of this Manual provides the current TLV list for chemical substances as of the date of publication.

Ventilation systems used in industrial plants are of two generic types. The SUPPLY system is used to supply air, usually tempered, to a work space. The EXHAUST system is used to remove the contaminants generated by an operation in order to maintain a healthful work environment.

A complete ventilation program must consider both the supply and the exhaust systems. If the overall quantity of air exhausted from a work space is greater than the quantity of outdoor air supplied to the space, the plant interior will experience a lower pressure than the local atmospheric pressure. This may be desirable when using a dilution ventilation system to control or isolate contaminants in a specific area of the overall plant. Often, this condition occurs simply because local exhaust systems are installed and consideration is not given to the corresponding replacement air systems. Air will then enter the plant in an uncontrolled manner through cracks, walls, windows, and doorways. This typically results in: 1) employee discomfort in winter months for those working near the plant perimeter, 2) exhaust system performance degradation, possibly leading to loss of contaminant control and a potential health hazard, and 3) higher heating and cooling costs. Chapter 7 of this Manual discusses these points in more detail.

1.2 SUPPLY SYSTEMS

Supply systems are used for two purposes: 1) to create a

comfortable environment in the plant (the HVAC system); and 2) to replace air exhausted from the plant (the REPLACE-MENT system). Many times, supply and exhaust systems are coupled, as in dilution control systems (see Section 1.3 and Chapter 2.)

A well-designed supply system will consist of an air inlet section, filters, heating and/or cooling equipment, a fan, ducts, and register/grilles for distributing the air within the work space. The filters, heating and/or cooling equipment and fan are often combined into a complete unit called an airhouse or air supply unit. If part of the air supplied by a system is recirculated, a RETURN system is used to bring the air back to the airhouse.

1.3 EXHAUST SYSTEMS

Exhaust ventilation systems are classified in two generic groups: 1) the GENERAL exhaust system and 2) the LOCAL exhaust system.

The general exhaust system can be used for heat control and/or removal of contaminants generated in a space by flushing out a given space with large quantities of air. When used for heat control, the air may be tempered and recycled. When used for contaminant control (the dilution system), enough outdoor air must be mixed with the contaminant so that the average concentration is reduced to a safe level. The contaminated air is then typically discharged to the atmosphere. A supply system is usually used in conjunction with a general exhaust system to replace the air exhausted.

Dilution ventilation systems are normally used for contaminant control only when local exhaust is impractical, as the large quantities of tempered replacement air required to offset the air exhausted can lead to high operating costs. Chapter 2 describes the basic features of general ventilation systems and their application to contaminant and fire hazard control.

Local exhaust ventilation systems operate on the principle of capturing a contaminant at or near its source. It is the preferred method of control because it is more effective and the smaller exhaust flow rate results in lower heating costs compared to high flow rate general exhaust requirements. The present emphasis on air pollution control stresses the need for efficient air cleaning devices on industrial ventilation systems, and the smaller flow rates of the local exhaust system result in lower costs for air cleaning devices.

Local exhaust systems are comprised of up to four basic elements: the hood(s), the duct system (including the exhaust stack and/or recirculation duct), the air cleaning device, and the fan. The purpose of the hood is to collect the contaminant generated in an air stream directed toward the hood. A duct system must then transport the contaminated air to the air cleaning device, if present, or to the fan. In the air cleaner, the contaminant is removed from the air stream. The fan must overcome all the losses due to friction, hood entry, and fittings

in the system while producing the intended flow rate. The duct on the fan outlet usually discharges the air to the atmosphere in such a way that it will not be re-entrained by the replacement and/or HVAC systems. In some situations, the cleaned air is returned to the plant. Chapter 7 discusses whether this is possible and how it may be accomplished.

This Manual deals with the design aspects of exhaust ventilation systems, but the principles described also apply to supply systems.

1.4 BASIC DEFINITIONS

The following basic definitions are used to describe air flow and will be used extensively in the remainder of the Manual.

The density (ρ) of the air is defined as its mass per unit volume and is normally expressed in pounds mass per cubic foot (lbm/ft^3). At standard atmospheric pressure (14.7 psia), room temperature (70 F) and zero water content, its value is normally taken to be $0.075 \text{ lbm}/\text{ft}^3$, as calculated from the perfect gas equation of state relating pressure, density, and temperature:

$$p = \rho RT \quad [1.1]$$

where:

p = the absolute pressure in pounds per square foot absolute (psfa)

ρ = the density, lbm/ft^3

R = the gas constant for air and equals $53.35 \text{ ft-lb}/\text{lbm-degrees Rankine}$

T = the absolute temperature of the air in degrees Rankine

Note that degrees Rankine = degrees Fahrenheit + 459.7.

From the above equation, density varies inversely with temperature when pressure is held constant. Therefore, for any dry air situation (see Chapter 5 for moist air calculations),

$$\rho T = (\rho T)_{\text{STD}}$$

or

$$\rho = \rho_{\text{STD}} \frac{T_{\text{STD}}}{T} = 0.075 \frac{530}{T} \quad [1.2]$$

For example, the density of dry air at 250 F would be

$$\rho = 0.075 \frac{530}{460 + 250} = 0.056 \text{ lbm}/\text{ft}^3$$

The volumetric flow rate, many times referred to as "volumes," is defined as the volume or quantity of air that passes a given location per unit of time. It is related to the average velocity and the flow cross-sectional area by the equation

$$Q = VA \quad [1.3]$$

where:

Q = volumetric flow rate, cfm

V = average velocity, fpm

A = cross-sectional area, ft^2

Given any two of these three quantities, the third can readily be determined.

Air or any other fluid will always flow from a region of higher total pressure to a region of lower total pressure in the absence of work addition (a fan). There are three different but mathematically related pressures associated with a moving air stream.

Static pressure (SP) is defined as the pressure in the duct that tends to burst or collapse the duct and is expressed in inches of water gage ("wg). It is usually measured with a water manometer, hence the units. SP can be positive or negative with respect to the local atmospheric pressure but must be measured perpendicular to the air flow. The holes in the side of a Pitot tube (see Figure 9-9) or a small hole carefully drilled to avoid internal burrs that disturb the air flow (never punched) into the side of a duct will yield SP.

Velocity pressure (VP) is defined as that pressure required to accelerate air from zero velocity to some velocity (V) and is proportional to the kinetic energy of the air stream. The relationship between V and VP is given by

$$V = 1096 \sqrt{\frac{VP}{\rho}}$$

or

$$VP = \rho \left(\frac{V}{1096} \right)^2 \quad [1.4]$$

where:

V = velocity, fpm

VP = velocity pressure, "wg

If standard air is assumed to exist in the duct with a density of $0.075 \text{ lbm}/\text{ft}^3$, this equation reduces to

$$V = 4005 \sqrt{VP}$$

or

$$VP = \left(\frac{V}{4005} \right)^2 \quad [1.5]$$

VP will only be exerted in the direction of air flow and is *always* positive. Figure 1-1 shows graphically the difference between SP and VP.

Total pressure (TP) is defined as the algebraic sum of the static and velocity pressures or

$$TP = SP + VP \quad [1.6]$$

Total pressure can be positive or negative with respect to

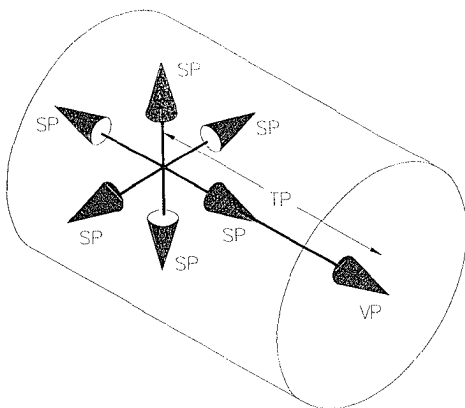


FIGURE 1-1. SP, VP, and TP at a point

atmospheric pressure and is a measure of the energy content of the air stream, always dropping as the flow proceeds downstream through a duct. The only place it will rise is across the fan.

Total pressure can be measured with an impact tube pointing directly upstream and connected to a manometer. It will vary across a duct due to the change of velocity across a duct and therefore single readings of TP will not be representative of the energy content. Chapter 9 illustrates procedures for measurement of all pressures in a duct system.

The significance of these pressures can be illustrated as follows. Assume a duct segment with both ends sealed was pressurized to a static pressure of 0.1 psi above the atmospheric pressure as shown in Figure 1-2. If a small hole (typically 1/16" to 3/32") were drilled into the duct wall and connected to one side of a U-tube manometer, the reading would be approximately 2.77 "wg. Note the way the left-hand manometer is deflected. If the water in the side of the manometer exposed to the atmosphere is higher than the water level in the side connected to the duct, then the pressure read by the gauge is positive (greater than atmospheric). Because there is no velocity, the velocity pressure is 0 and $SP = TP$. A probe which faces the flow is called an impact tube and will

measure TP. In this example, a manometer connected to an impact tube (the one on the right) will also read 2.77 "wg. Finally, if one side of a manometer were connected to the impact tube and the other side were connected to the static pressure opening (the center one), the manometer would read the difference between the two pressures. As $VP = TP - SP$, a manometer so connected would read VP directly. In this example, there is no flow and hence $VP = 0$ as indicated by the lack of manometer deflection.

If the duct ends were removed and a fan placed midway in the duct, the situation might change to the one shown on Figure 1-3. Upstream of the fan, SP and TP are negative (less than atmospheric). This is called the *suction side*. Downstream of the fan, both SP and TP are positive. This is called the *pressure side*. Regardless of which side of the fan is considered, VP is always positive. Note that the direction in which the manometers are deflected shows whether SP and TP are positive or negative with respect to the local atmospheric pressure.

1.5 PRINCIPLES OF AIR FLOW

Two basic principles of fluid mechanics govern the flow of air in industrial ventilation systems: conservation of mass and conservation of energy. These are essentially bookkeeping laws which state that all mass and all energy must be completely accounted for. A coverage of fluid mechanics is not in the purview of this manual; reference to any standard fluid mechanics textbook will show the derivation of these principles. However, it is important to know what simplifying assumptions are included in the principles discussed below. They include:

1. Heat transfer effects are neglected. If the temperature inside the duct is significantly different than the air temperature surrounding the duct, heat transfer will occur. This will lead to changes in the duct air temperature and hence in the volumetric flow rate.
2. Compressibility effects are neglected. If the overall pressure drop from the start of the system to the fan is greater than about 20 "wg, then the density will change

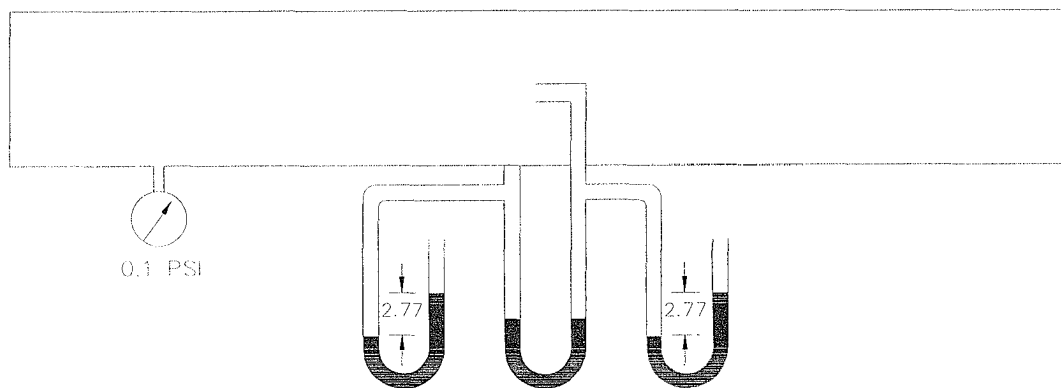


FIGURE 1-2. Measurement of SP, VP, and TP in a pressurized duct

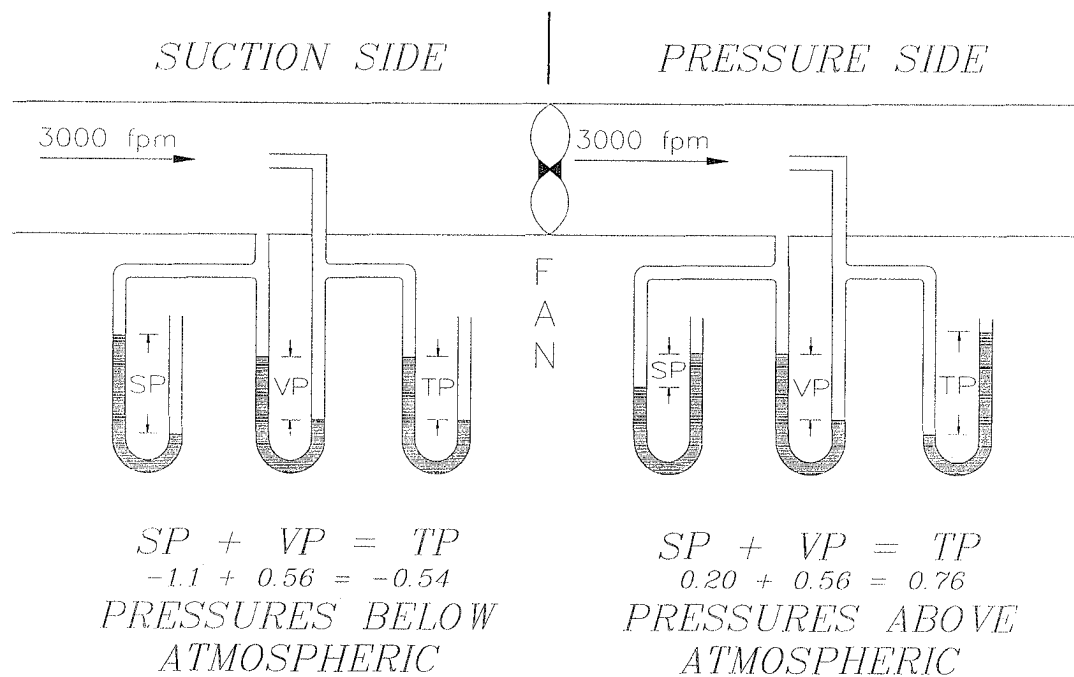


FIGURE 1-3. SP, VP, and TP at points in a ventilation system

change (see Chapter 5).

3. The air is assumed to be dry. Water vapor in the air stream will lower the air density, and correction for this effect, if present, should be made. Chapter 5 describes the necessary psychrometric analysis.
4. The weight and volume of the contaminant in the air stream is ignored. This is permissible for the contaminant concentrations in typical exhaust ventilation systems. For high concentrations of solids or significant amounts of gases other than air, corrections for this effect should be included.

Conservation of mass requires that the net change of mass flow rate must be zero. If the effects discussed above are negligible, then the density will be constant and the net change of volumetric flow rate (Q) must be zero. Therefore, the flow rate that enters a hood must be the same as the flow rate that passes through the duct leading from the hood. At a branch entry (converging wye) fitting, the sum of the two flow rates that enter the fitting must leave it. At a diverging wye, the flow rate entering the wye must equal the sum of the flow rates that leave it. Figure 1-4 illustrates these concepts.

Conservation of energy means that all energy changes must be accounted for as air flows from one point to another. In terms of the pressures previously defined, this principle can be expressed as:

$$TP_1 = TP_2 + h_f$$

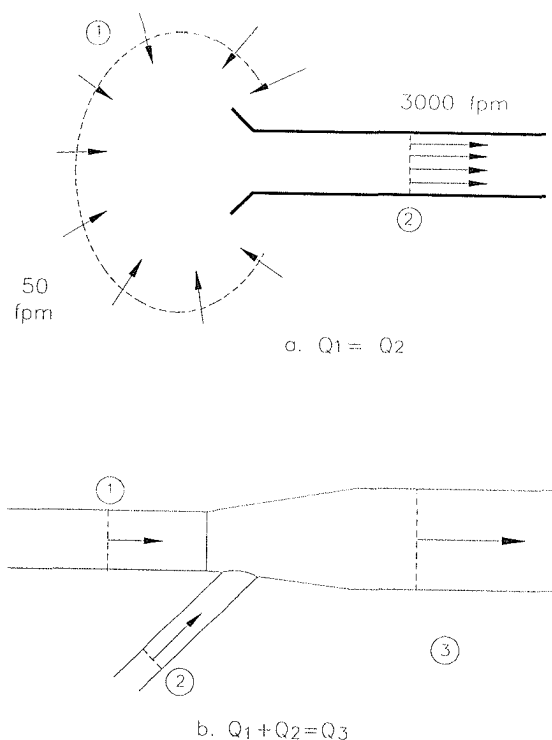


FIGURE 1-4. Volumetric flow rates in various situations. a. Flow through a hood; b. Flow through a branch entry

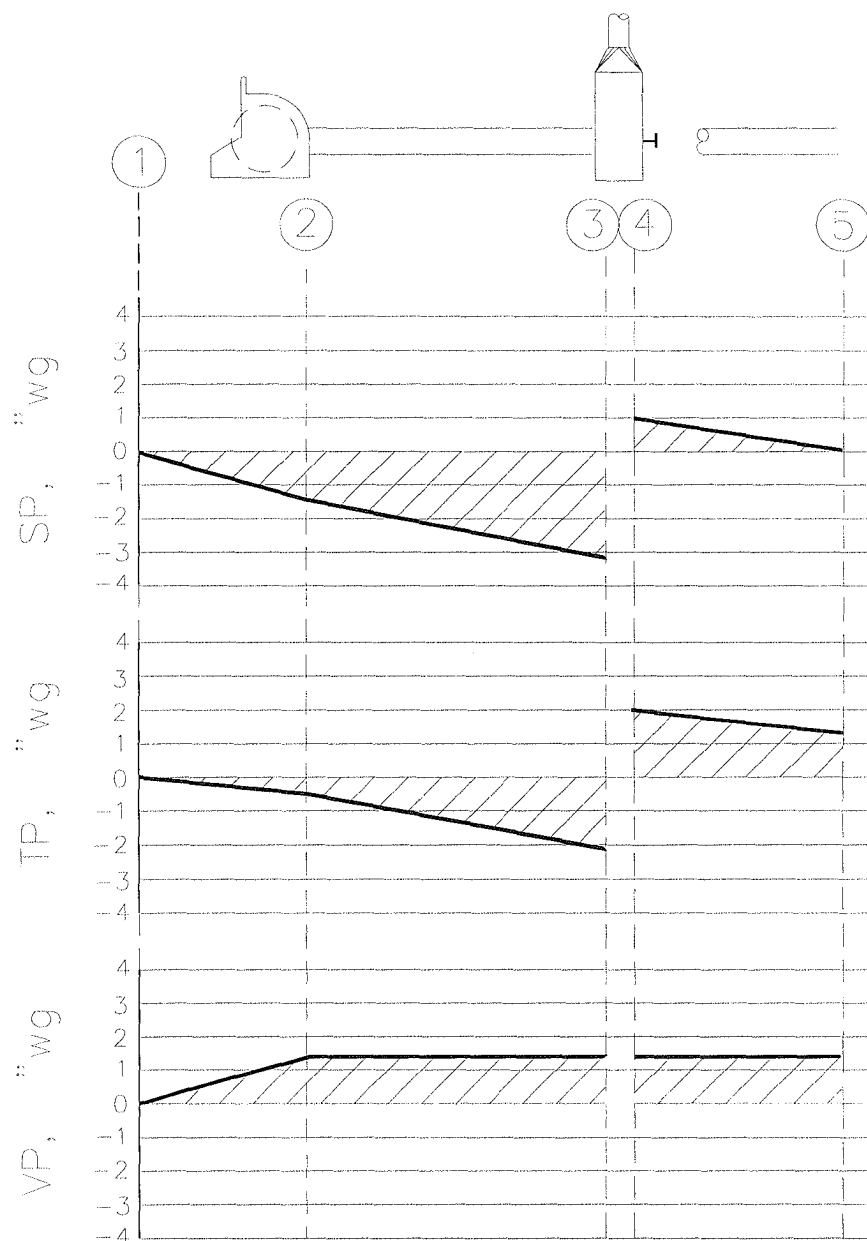


FIGURE 1-5. Variation of SP, VP, and TP through a ventilation system

$$SP_1 + VP_1 = SP_2 + VP_2 + h_1 \quad [1.7]$$

where:

subscript 1 = some upstream point

subscript 2 = some downstream point

h_1 = all energy losses encountered by the air as it flows from the upstream to the downstream point

Note that, according to this principle, *the total pressure must fall in the direction of flow.*

The application of these principles will be demonstrated by

an analysis of the simple system shown in Figure 1-5. The normally vertical exhaust stack is shown laying horizontally to facilitate graphing the variation of static, total, and velocity pressures. The grinder wheel hood requires 300 cfm and the duct diameter is constant at 3.5 inches (0.0668 ft² area).

1.6 ACCELERATION OF AIR AND HOOD ENTRY LOSSES

Air flows from the room (point 1 of Figure 1-5) through the hood to the duct (point 2 of Figure 1-5) where the velocity can be calculated by the basic equation:

$$V = \frac{Q}{A} = \frac{300}{0.0668} = 4490 \text{ fpm}$$

This velocity corresponds to a velocity pressure of 1.26 "wg, assuming standard air.

If there are no losses associated with entry into a hood, then applying the energy conservation principle (Equation 1.7) to the hood yields

$$SP_1 + VP_1 = SP_2 + VP_2$$

This is the well known Bernoulli principle of fluid mechanics. Subscript 1 refers to the room conditions where the static pressure is atmospheric ($SP_1 = 0$) and the air velocity is assumed to be very close to zero ($VP_1 = 0$). Therefore, the energy principle yields

$$SP_2 = -VP_2 = -1.26 \text{ "wg}$$

Even if there were no losses, *the static pressure must decrease due to the acceleration of air to the duct velocity.*

In reality, there are losses as the air enters the hood. These hood entry losses (h_d) are normally expressed as a loss coefficient (F_d) multiplied by the duct velocity pressure; so $h_d = F_d VP_d$ (where $VP_d = VP_2$). The energy conservation principle then becomes

$$SP_2 = -(VP_2 + h_d) \quad [1.8]$$

(See 3.5.1, 3.5.2, and Figure 5-1 for a discussion of h_d and h_e .)

The absolute value of SP_2 is known as the hood static suction (SP_h). Then

$$SP_h = -SP_2 = VP_2 + h_e \quad [1.9]$$

For the example in Figure 1-5, assuming an entry loss coefficient of 0.40,

$$\begin{aligned} SP_h &= VP_2 + F_d VP_2 \\ &= 1.26 + (0.40)(1.26) \\ &= 1.26 + 0.50 = 1.76 \text{ "wg} \end{aligned}$$

In summary, the static pressure downstream of the hood is negative (less than atmospheric) due to two effects:

1. Acceleration of air to the duct velocity; and
2. Hood entry losses.

From the graph, note that $TP_2 = -h_e$, which confirms the premise that total pressure decreases in the flow direction.

An alternate method of describing hood entry losses is by the hood entry coefficient (C_e). This coefficient is defined as the square root of the ratio of duct velocity pressure to hood static suction, or

$$C_e = \sqrt{\frac{VP}{SP_h}} \quad [1.10]$$

If there were no losses, then $SP_h = VP$ and $C_e = 1.00$. However, as hoods always have some losses, C_e is always less than 1.00. In Figure 1-5,

$$C_e = \sqrt{\frac{VP}{SP_h}} = \sqrt{\frac{1.26}{1.76}} = 0.845$$

An important feature of C_e is that it is a constant for any given hood. It can, therefore, be used to determine the flow rate if the hood static suction is known. This is because

$$Q = VA = 1096A \sqrt{\frac{VP}{\rho}} = 1096A C_e \sqrt{\frac{SP_h}{\rho}} \quad [1.11]$$

For standard air, this equation becomes

$$Q = 4005 A C_e \sqrt{SP_h} \quad [1.12]$$

For the example in Figure 1-5,

$$Q = 4005(0.0668)(0.845) \sqrt{1.76} = 300 \text{ cfm}$$

By use of C_e and a measurement of SP_h , the flow rate of a hood can be quickly determined and corrective action can be taken if the calculated flow rate does not agree with the design flow rate.

1.7 DUCT LOSSES

There are two components to the overall total pressure losses in a duct run: 1) friction losses and 2) fitting losses.

1.7.1 Friction Losses. Losses due to friction in ducts are a complicated function of duct velocity, duct diameter, air density, air viscosity, and duct surface roughness. The effects of velocity, diameter, density, and viscosity are combined into the *Reynolds number* (R_e), as given by

$$R_e = \frac{\rho dv}{\mu} \quad [1.13]$$

where:

- ρ = density, lbm/ft³
- d = diameter, ft
- v = velocity, ft/sec
- μ = the air viscosity, lbm/s-ft

The effect of surface roughness is typically given by the *relative roughness*, which is the ratio of the absolute surface roughness height (k), defined as the average height of the roughness elements on a particular type of material, to the duct diameter. Some standard values of absolute surface roughness used in ventilation systems are given in Table 1-1.

L. F. Moody^(1.1) combined these effects into a single chart commonly called the *Moody diagram* (see Figure 1-6). With a knowledge of both the Reynolds number and the relative roughness, the *friction coefficient* (f), can be found.

TABLE 1-1. Absolute Surface Roughness

Duct Material	Surface Roughness (k), feet
Galvanized metal	0.00055
Black iron	0.00015
Aluminum	0.00015
Stainless steel	0.00015
Flexible duct (wires exposed)	0.01005
Flexible duct (wires covered)	0.00301

The above roughness heights are design values. It should be noted that significant variations from these values may occur, depending on the manufacturing process.

Once determined, the friction coefficient is used in the *Darcy-Weisbach friction coefficient equation* to determine the overall duct friction losses:

$$h_f = f \frac{L}{d} VP \quad [1.14]$$

where:

h_f = friction losses in a duct, "wg

f = Moody diagram friction coefficient (dimen-

sionless)

L = duct length, ft

d = duct diameter, ft

VP = duct velocity pressure, "wg

There are many equations available for computer solutions to the Moody diagram. One of these is that of Churchill,^(1,2) which gives accurate (to within a few percent) results over the entire range of laminar, critical, and turbulent flow, all in a single equation. This equation is:

$$f = 8 \left[\left(\frac{8}{Re} \right)^{12} + (A + B)^{-3/2} \right]^{1/12} \quad [1.15]$$

where:

$$A = \left\{ -2.457 \ln \left[\left(\frac{7}{Re} \right)^{0.9} + \left(\frac{k}{3.7D} \right) \right] \right\}^{16}$$

$$B = \left(\frac{37,530}{Re} \right)^{16}$$

While useful, this equation is quite difficult to use without a computer. Several attempts have been made to simplify the determination of friction losses for specialized situations. For

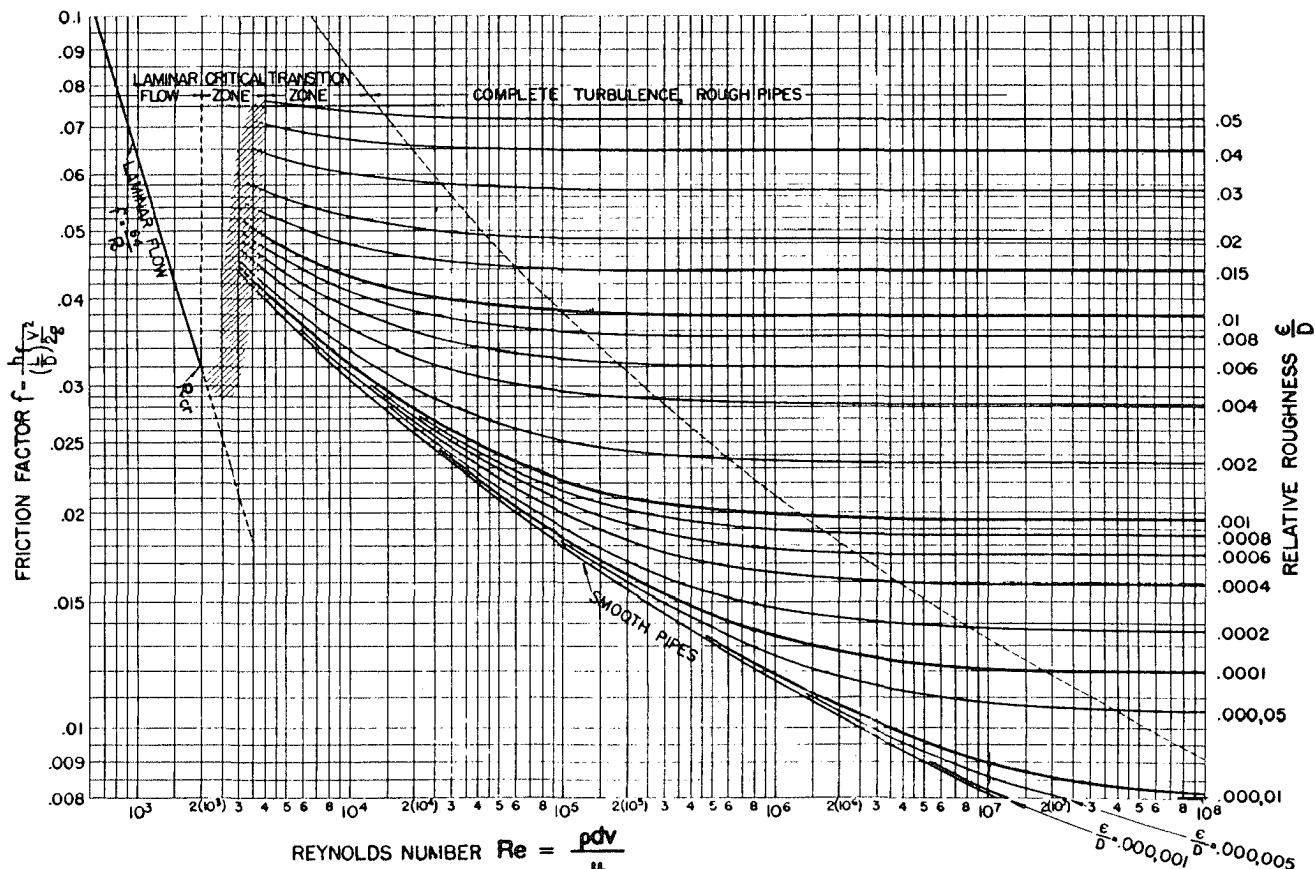


FIGURE 1-6. Moody diagram (adapted from reference 1.1)

many years, charts based on the Wright^(1.3) equation have been used in ventilation system design:

$$h_f = 2.74 \frac{(V/1000)^{1.9}}{D^{1.22}} \quad [1.16]$$

where:

V = duct velocity, fpm

D = duct diameter, inches

This equation gives the friction losses, expressed as "wg per 100 feet of pipe, for standard air of 0.075 lbm/ft³ density flowing through average, clean, round galvanized pipe having approximately 40 slip joints per 100 feet ($k = 0.0005$ ft).

The later work by Loeffler^(1.4) presented equations for use in the "velocity pressure" calculation method. Using the standard values of surface roughness, equations were obtained that could be used with the Darcy-Weisbach equation in the form:

$$h_f = \left(12 \frac{f}{D}\right) L \quad VP = H_f L \quad VP \quad [1.17]$$

where the "12" is used to convert the diameter D in inches to feet.

Simplified equations were determined for the flow of standard air through various types of duct material with good accuracy (less than 5% error). The equations thus resulting were:

$$H_f = 12 \frac{f}{D} = \frac{aV^b}{Q^c} \quad [1.18]$$

where the constant "a" and the exponents "b" and "c" vary as a function of the duct material as shown in Table 1-2. Note that no correlation was made with the extremely rough flexible duct with wires exposed. This equation, using the constants from Table 1-2 for galvanized sheet duct, were used to develop the friction Tables 5-5 and 5-6. Note that the value obtained from the chart or from equation 1.18 must be multiplied by both the length of duct and the velocity pressure.

1.7.2 Fitting Losses. The fittings (elbows, entries, etc.) in a duct run will also produce a loss in total pressure. These losses are given in Chapter 5.

The fitting losses are given by a loss coefficient (F) multiplied by the duct velocity pressure. Thus,

$$h_{en} = F_{en} \quad VP \quad [1.19]$$

In contractions, entries, or expansions, there are several different velocity pressures. The proper one to use with the loss coefficient will be identified where the coefficients are listed.

In Figure 1-5, 15 feet of straight, constant diameter galva-

TABLE 1-2. Correlation Equation Constants

Duct Material	k, Ft	a	b	c
Aluminum, black iron, stainless steel	0.00015	0.0425	0.465	0.602
Galvanized sheet duct	0.00051	0.0307	0.533	0.612
Flexible duct, fabric wires covered	0.0035	0.0311	0.604	0.639

nized duct connects the hood to a fan inlet. Because the duct area is constant, the velocity, and therefore the velocity pressure, is also constant for any given flow rate. The energy principle is:

$$SP_2 + VP_2 = SP_3 + VP_3 + h_f$$

where subscript 3 refers to the fan inlet location. Because $VP_2 = VP_3$, the losses will appear as a reduction in static pressure (there will, of course, be a corresponding reduction in total pressure). The friction loss can be found from Equation 1.17 with the aid of Equation 1.18:

$$\begin{aligned} H_f &= 0.0307 \frac{V^{0.533}}{Q^{0.612}} \\ &= 0.0307 \frac{4490^{0.533}}{300^{0.612}} = 0.0828 \end{aligned}$$

From Equation 1.17, $h_f = (0.0828)(15)(1.26) = 1.56$ "wg. Using this in the energy principle,

$$SP_3 = SP_2 - h_f = -1.76 \text{ "wg} - 1.56 \text{ "wg} = -3.32 \text{ "wg}$$

Another 10 feet of straight duct is connected to the discharge side of the fan. The losses from the fan to the end of the system would be about 1.04 "wg. Because the static pressure at the end of the duct must be atmospheric ($SP_5 = 0$), the energy principle results in

$$SP_4 = SP_5 + h_f = 0 \text{ "wg} + 1.04 \text{ "wg} = 1.04 \text{ "wg}$$

Therefore, the static pressure at the fan outlet must be higher than atmospheric by an amount equal to the losses in the discharge duct.

1.8 MULTIPLE-HOOD EXHAUST SYSTEMS

Most exhaust systems are more complicated than the preceding example. It is usually more economical to purchase a single fan and air cleaner to service a series of similar operations than to create a complete system for each operation. For example, the exhaust from 10 continuously used grinders can be combined into a single flow which leads to a common air cleaner and fan. This situation is handled similarly to a simple system, but with some provision to ensure that the air flow from each hood is as desired (see Chapter 5).

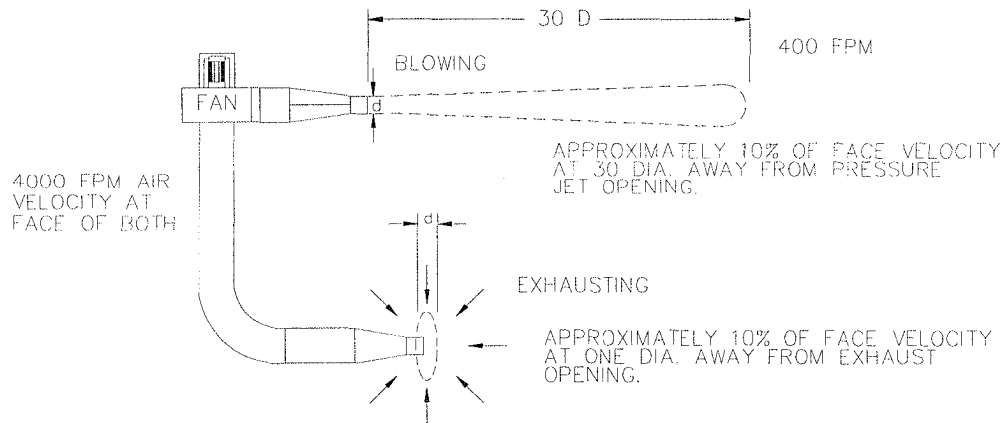


FIGURE 1-7. Blowing vs. exhausting

1.9 AIR FLOW CHARACTERISTICS OF BLOWING AND EXHAUSTING

Air blown from a small opening retains its directional effect for a considerable distance beyond the plane of the opening. However, if the flow of air *through the same opening* were reversed so that it operated as an exhaust opening handling the same volumetric flow rate, the flow would become almost non-directional and its range of influence would be greatly reduced. For this reason, local exhaust must not be contemplated for any process that cannot be conducted in the *immediate* vicinity of the hood. Also, because of this effect, every effort should be made to enclose the operation as much as possible. Figure 1-7 illustrates the fundamental difference between blowing and exhausting.

This effect also shows how the supply or replacement air discharge grilles can influence an exhaust system. If care is

not taken, the discharge pattern from a supply grille could seriously affect the flow pattern in front of an exhaust hood.

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Chapter 2

GENERAL INDUSTRIAL VENTILATION

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2.1 INTRODUCTION

"General industrial ventilation" is a broad term which refers to the supply and exhaust of air with respect to an area, room, or building. It can be divided further into specific functions as follows:

1. *Dilution Ventilation* is the dilution of contaminated air with uncontaminated air for the purpose of controlling potential airborne health hazards, fire and explosive conditions, odors, and nuisance-type contaminants. Dilution ventilation also can include the control of airborne contaminants (vapors, gases, and particulates) generated within tight buildings.

Dilution ventilation is not as satisfactory for health hazard control as is local exhaust ventilation. Circumstances may be found in which dilution ventilation provides an adequate amount of control more economically than a local exhaust system. One should be careful, however, not to base the economical considerations entirely upon the first cost of the system since dilution ventilation frequently exhausts large amounts of heat from a building, which may greatly increase the energy cost of the operation.

2. *Heat Control Ventilation* is the control of indoor atmospheric conditions associated with hot industrial environments such as are found in foundries, laundries, bakeries, etc., for the purpose of preventing acute discomfort or injury.

2.2 DILUTION VENTILATION PRINCIPLES

The principles of dilution ventilation system design are as follows:

1. Select from available data the amount of air required for satisfactory dilution of the contaminant. The values tabulated on Table 2-1 assume perfect distribution and dilution of the air and solvent vapors. These values must be multiplied by the selected K value (see Section 2.3.1).
2. Locate the exhaust openings near the sources of contamination, if possible, in order to obtain the benefit of "spot ventilation."
3. Locate the air supply and exhaust outlets such that the air passes through the zone of contamination. The operator should remain between the air supply and the source of the contaminant.
4. Replace exhausted air by use of a replacement air system. This replacement air should be heated during cold weather. Dilution ventilation systems usually handle large quantities of air by means of low pressure fans. Replacement air must be provided if the system is to operate satisfactorily.
5. Avoid re-entry of the exhausted air by discharging the

exhaust high above the roof line or by assuring that no window, outdoor air intakes, or other such openings are located near the exhaust discharge.

2.3 DILUTION VENTILATION FOR HEALTH

The use of dilution ventilation for health has four limiting factors: 1) the quantity of contaminant generated must not be too great or the air flow rate necessary for dilution will be impractical; 2) workers must be far enough away from the contaminant source or the evolution of contaminant must be in sufficiently low concentrations so that workers will not have an exposure in excess of the established TLV; 3) the toxicity of the contaminant must be low; and 4) the evolution of contaminants must be reasonably uniform.

Dilution ventilation is used most often to control the vapors from organic liquids with a TLV of 100 ppm or higher. In order to successfully apply the principles of dilution to such a problem, factual data are needed on the rate of vapor generation or on the rate of liquid evaporation. Usually such data can be obtained from the plant if any type of adequate records on material consumption are kept.

2.3.1 General Dilution Ventilation Equation: The ventilation rate needed to maintain a constant concentration at a uniform generation rate is derived by starting with a fundamental material balance and assuming no contaminant in the air supply,

$$\text{Rate of Accumulation} = \text{Rate of Generation} - \text{Rate of Removal}$$

or

$$VdC = Gdt - Q'Cdt \quad [2.1]$$

where:

V = volume of room
G = rate of generation
Q' = effective volumetric flow rate
C = concentration of gas or vapor
t = time

At a steady state, $dC = 0$

$$Gdt = Q'Cdt$$

$$\int_{t_1}^{t_2} Gdt = \int_{t_1}^{t_2} Q'Cdt$$

At a constant concentration, C, and uniform generation rate, G,

$$G(t_2 - t_1) = Q'C(t_2 - t_1)$$

$$Q' = \frac{G}{C} \quad [2.2]$$

Due to incomplete mixing, a K value is introduced to the rate of ventilation; thus:

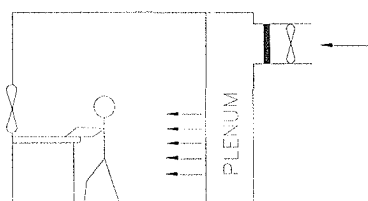
TABLE 2-1. Dilution Air Volumes for Vapors

The following values are tabulated using the TLV values shown in parentheses, parts per million. TLV values are subject to revision if further research or experience indicates the need. If the TLV value has changed, the dilution air requirements must be recalculated. The values on the table must be multiplied by the evaporation rate (pts/min) to yield the effective ventilation rate (Q') (see Equation 2.5).

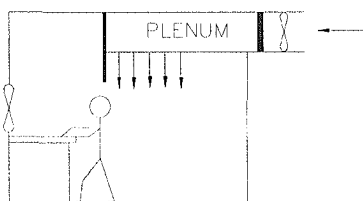
Liquid (TLV in ppm)**	<u>Ft³ of Air (STP) Required for Dilution to TLV*</u>
	Per Pint Evaporation
Acetone (500)	11,025
n-Amyl acetate (100)	27,200
Benzene (0.5)	NOT RECOMMENDED
n-Butanol (butyl alcohol) (50)	88,000
n-Butyl acetate (150)	20,400
Butyl Cellosolve (2-butoxyethanol) (25)	NOT RECOMMENDED
Carbon disulfide (10)	NOT RECOMMENDED
Carbon tetrachloride (5)	NOT RECOMMENDED
Cellosolve (2-ethoxyethanol) (5)	NOT RECOMMENDED
Cellosolve acetate (2-ethoxyethyl acetate) (5)	NOT RECOMMENDED
Chloroform (10)	NOT RECOMMENDED
1-2 Dichloroethane (ethylene dichloride) (10)	NOT RECOMMENDED
1-2 Dichloroethylene (200)	26,900
Dioxane (25)	NOT RECOMMENDED
Ethyl acetate (400)	10,300
Ethyl alcohol (1000)	6,900
Ethyl ether (400)	9,630
Gasoline (300)	REQUIRES SPECIAL CONSIDERATION
Isoamyl alcohol (100)	37,200
Isopropyl alcohol (400)	13,200
Isopropyl ether (250)	11,400
Methyl acetate (200)	25,000
Methyl alcohol (200)	49,100
Methyl n-butyl ketone (5)	NOT RECOMMENDED
Methyl Cellosolve (2-methoxyethanol) (5)	NOT RECOMMENDED
Methyl Cellosolve acetate (2-methoxyethyl acetate) (5)	NOT RECOMMENDED
Methyl chloroform (350)	11,390
Methyl ethyl ketone (200)	22,500
Methyl isobutyl ketone (50)	64,600
Methyl propyl ketone (200)	19,900
Naphtha (coal tar)	REQUIRES SPECIAL CONSIDERATION
Naphtha VM&P (300)	REQUIRES SPECIAL CONSIDERATION
Nitrobenzene (1)	NOT RECOMMENDED
n-Propyl acetate (200)	17,500
Stoddard solvent (100)	30,000–35,000
1,1,2,2-Tetrachloroethane (1)	NOT RECOMMENDED
Tetrachloroethylene (25)	159,400
Toluene (50)	75,700
Trichloroethylene (50)	90,000
Xylene (100)	33,000

*The tabulated dilution air quantities must be multiplied by the selected K value.

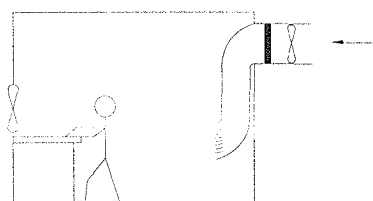
**See Threshold Limit Values 1997 Appendix A.



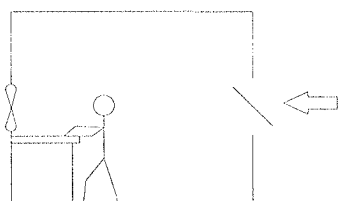
BEST AIR INLET
BEST EXHAUST
 $K = 1.0$ MINIMUM



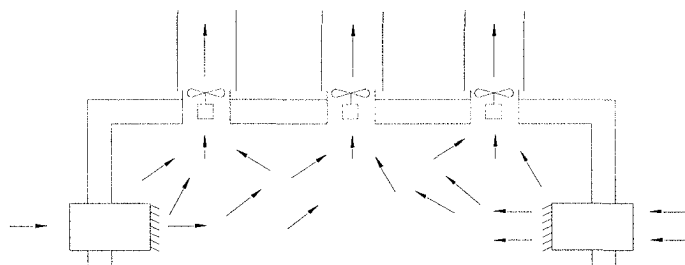
BEST AIR INLET
BEST EXHAUST
 $K = 1.0$ MINIMUM



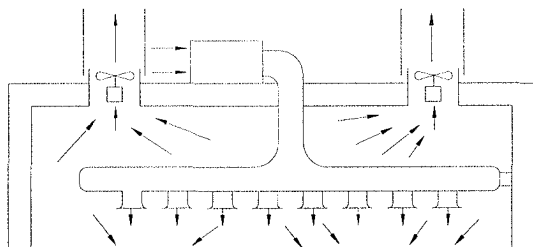
BEST AIR INLET
BEST EXHAUST
 $K = 1.5$ MINIMUM



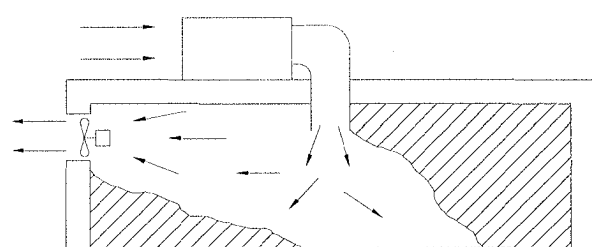
FAIR AIR INLET
BEST EXHAUST
 $K = 2.5$ MINIMUM



FAIR
 $K = 2$ TO 5
REF. 2.2



GOOD
 $K = 1.5$ TO 2
REF. 2.2



POOR
 $K = 5$ TO 10
REF. 2.2

NOTE: THE K FACTORS LISTED HERE CONSIDER ONLY THE INLET AND EXHAUST LOCATIONS AND ARE JUDGEMENTAL. TO SELECT THE K FACTOR USED IN THE EQUATION, THE NUMBER AND LOCATION OF THE EMPLOYEES, THE SOURCE OF THE CONTAMINANT, AND THE TOXICITY OF THE CONTAMINANT MUST ALSO BE CONSIDERED.

AMERICAN CONFERENCE
OF GOVERNMENTAL
INDUSTRIAL HYGIENISTS

" K " FACTORS
SUGGESTED FOR INLET
AND EXHAUST LOCATIONS

DATE 1-88

FIGURE 2-1

$$Q' = \frac{Q}{K} \quad [2.3]$$

where:

Q = actual ventilation rate, cfm

Q' = effective ventilation rate, cfm

K = a factor to allow for incomplete mixing

Equation 2.2 then becomes:

$$Q = \left(\frac{G}{C} \right) K \quad [2.4]$$

This K factor is based on several considerations:

1. The efficiency of mixing and distribution of replacement air introduced into the room or space being ventilated (see Figure 2-1).
2. The toxicity of the solvent. Although TLV and toxicity are not synonymous, the following guidelines have been suggested for choosing the appropriate K value:

Slightly toxic material: TLV > 500 ppm

Moderately toxic material: TLV ≤ 100–500 ppm

Highly toxic material: TLV < 100 ppm

3. A judgement of any other circumstances which the industrial hygienist determined to be of importance based on experience and the individual problem. Included in these criteria are such considerations as:
 - a. Duration of the process, operational cycle, and normal locations of workers relative to sources of contamination.
 - b. Location and number of points of generation of the contaminant in the workroom or area.
 - c. Seasonal changes in the amount of natural ventilation.
 - d. Reduction in operational effectiveness of mechanical air moving devices.
 - e. Other circumstances which may affect the concentration of hazardous material in the breathing zone of the workers.

The K value selected, depending on the above considerations, ranges from 1 to 10.

2.3.2 Calculating Dilution Ventilation for Steady-State Concentration: The concentration of a gas or vapor at a steady state can be expressed by the material balance equation

$$Q' = \frac{G}{C}$$

Therefore, the rate of flow of uncontaminated air required to maintain the atmospheric concentration of a hazardous material at an acceptable level can be easily calculated if the generation rate can be determined. Usually, the acceptable concentration (C) expressed in parts per million (ppm) is

considered to be the TLV. For liquid solvents, the rate of generation is

$$G = \frac{\text{CONSTANT} \times \text{SG} \times \text{ER}}{\text{MW}}$$

where:

G = generation rate, cfm

CONSTANT = the volume in ft³ that 1 pt of liquid, when vaporized, will occupy at STP, ft³/pt

SG = Specific gravity of volatile liquid

ER = evaporation rate of liquid, pts/min

MW = molecular weight of liquid

Thus, $Q' = G \div C$ can be expressed as

$$Q' = \frac{403 \times 10^6 \times \text{SG} \times \text{ER}}{\text{MW} \times C} \quad [2.5]$$

EXAMPLE PROBLEM

Methyl chloroform is lost by evaporation from a tank at a rate of 1.5 pints per 60 minutes. What is the effective ventilation rate (Q') and the actual ventilation rate (Q) required to maintain the vapor concentration at the TLV?

TLV = 350 ppm, SG = 132, MW = 133.4, Assume K = 5

Assuming perfect dilution, the effective ventilation rate (Q') is

$$Q' = \frac{(403)(10^6)(132)(15/60)}{(133.4)(350)}$$

For incomplete mixing, the actual ventilation rate (Q) is

$$Q = \frac{(403)(10^6)(132)(15/60)(5)}{(133.4)(350)}$$

2.3.3 Contaminant Concentration Buildup (see Figure 2-2): The concentration of a contaminant can be calculated after any change of time. Rearranging the differential material balance results in

$$\frac{dC}{G - Q'C} = \frac{dt}{V}$$

which can be integrated to yield

$$\ln \left(\frac{G - Q'C_2}{G - Q'C_1} \right) = - \frac{Q'(t_2 - t_1)}{V} \quad [2.6]$$

where subscript 1 refers to the initial condition and subscript 2 refers to the final condition. If it is desired to calculate the time required to reach a given concentration, then rearranging $t_2 - t_1$, or Δt , gives

$$\Delta t = - \frac{V}{Q'} \left[\ln \left(\frac{G - Q'C_2}{G - Q'C_1} \right) \right] \quad [2.7]$$

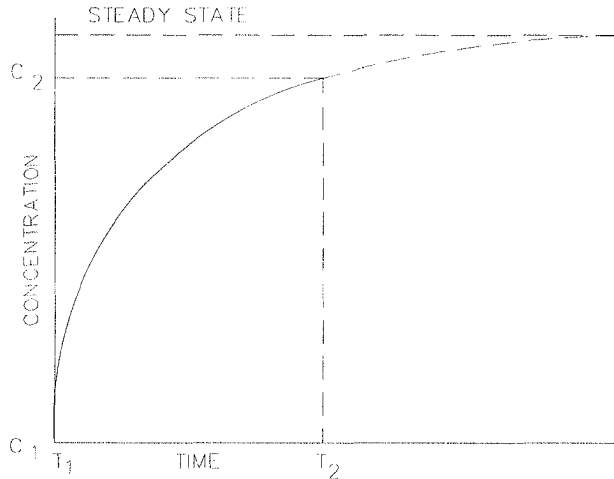


FIGURE 2-2. Contaminant concentration buildup

If $C_1 = 0$, then the equation becomes

$$\Delta t = -\frac{V}{Q'} \left[\ln \left(\frac{G - Q'C_2}{G} \right) \right] \quad [2.8]$$

Note: the concentration C_2 is ppm or parts/ 10^6 (e.g., if $C_2 = 200$ ppm, enter C_2 as $200 \div 10^6$).

If it is desired to determine the concentration level (C_2) after a certain time interval, $t_2 - t_1$ or Δt , and if $C_1 = 0$, then the equation becomes

$$C_2 = \frac{G \left[1 - e \left(-\frac{Q'\Delta t}{V} \right) \right]}{Q'} \quad [2.9]$$

Note: to convert C_2 to ppm, multiply the answer by 10^6 .

EXAMPLE

Methyl chloroform vapor is being generated under the following conditions: $G = 1.2$ cfm; $Q' = 2000$ cfm; $V = 100,000$ cu ft; $C_1 = 0$; $K = 3$. How long before the concentration (C_2) reaches 200 ppm or $200 \div 10^6$?

$$\Delta t = -\frac{V}{Q'} \left[\ln \left(\frac{G - Q'C_2}{G} \right) \right] = 20.3 \text{ min}$$

Using the same values as in the preceding example, what will be the concentration after 60 minutes?

$$C_2 = \frac{G \left[1 - e \left(-\frac{Q'\Delta t}{V} \right) \right]}{Q'} \times 10^6 = 419 \text{ ppm}$$

2.3.4 Rate of Purging (see Figure 2-3): Where a quantity of air is contaminated but where further contamination or generation has ceased, the rate of decrease of concentration

over a period of time is as follows:

$$VdC = -Q'Cdt$$

$$\int_{C_1}^{C_2} \frac{dC}{C} = -\frac{Q'}{V} \int_{t_1}^{t_2} dt$$

$$\ln \left(\frac{C_2}{C_1} \right) = -\frac{Q'}{V} (t_2 - t_1)$$

or,

$$C_2 = C_1 e^{\left[-\frac{Q'(t_2 - t_1)}{V} \right]} \quad [2.10]$$

EXAMPLE

In the room of the example in Section 2.3.3, assume that ventilation continues at the same rate ($Q' = 2000$ cfm) but that the contaminating process is interrupted. How much time is required to reduce the concentration from 100 (C_1) to 25 (C_2) ppm?

$$t_1 - t_2 = -\frac{V}{Q'} \ln \left(\frac{C_2}{C_1} \right) = 69.3 \text{ min}$$

In the problem above, if the concentration (C_1) at t_1 is 100 ppm, what will concentration (C_1) be after 60 minutes (Δt)?

$$C_2 = C_1 e^{\left(-\frac{Q'\Delta t}{V} \right)} = 30.1 \text{ ppm}$$

2.4 MIXTURES—DILUTION VENTILATION FOR HEALTH

In many cases, the evaporating liquid for which dilution ventilation rates are being designed will consist of a mixture of solvents. The common procedure used in such instances is as follows.

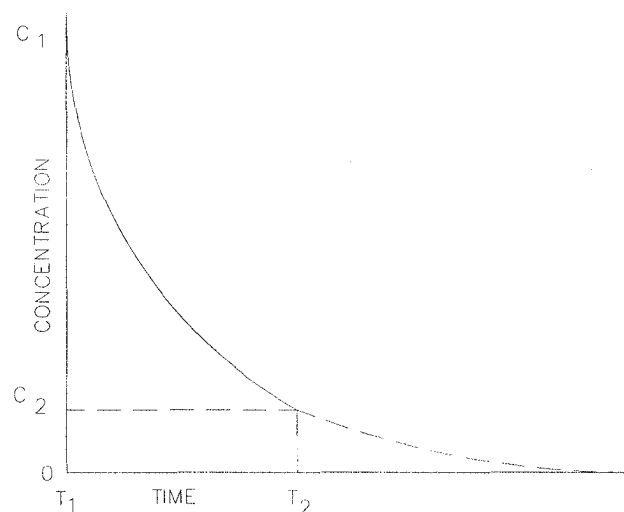


FIGURE 2-3. Rate of purging

When two or more hazardous substances are present, their combined effect, rather than that of either individually, should be given primary consideration. *In the absence of information to the contrary, the effects of the different hazards should be considered as additive.* That is, if the sum of the following fractions,

$$\frac{C_1}{TLV_1} + \frac{C_2}{TLV_2} + \dots + \frac{C_n}{TLV_n} \text{ etc.} \quad [2.11]$$

exceeds unity, then the threshold limit of the mixture should be considered as being exceeded. "C" indicates the observed atmospheric concentration and "TLV" the corresponding threshold limit. In the absence of information to the contrary, the dilution ventilation therefore should be calculated on the basis that the effect of the different hazards is additive. The air quantity required to dilute each component of the mixture to the required safe concentration is calculated, and the *sum* of the air quantities is used as the required dilution ventilation for the mixture.

Exceptions to the above rule may be made when there is good reason to believe that the chief effects of the different harmful substances are not additive but independent, as when purely local effects on different organs of the body are produced by the various components of the mixture. In such cases, the threshold limit ordinarily is exceeded only when at least one member of the series itself has a value exceeding unity, e.g.,

$$\frac{C_1}{TLV_1} \text{ or } \frac{C_2}{TLV_2}$$

Therefore, where two or more hazardous substances are present and it is known that the effects of the different substances are not additive but act independently on the different organs of the body, the required dilution ventilation for each component of the mixture should be calculated and the highest cfm thus obtained used as the dilution ventilation rate.

EXAMPLE PROBLEM

A cleaning and gluing operation is being performed; methyl ethyl ketone (MEK) and toluene are both being released. Both have narcotic properties and the effects are considered additive. Air samples disclose concentrations of 150 ppm MEK and 50 ppm toluene. Using the equation given, the sum of the fractions $[(150 \div 200) + (50 \div 50) = 1.75]$ is greater than unity and the TLV of the mixture is exceeded. The volumetric flow rate at standard conditions required for dilution of the mixture to the TLV would be as follows:

Assume 2 pints of each is being released each 60 min. Select a K value of 4 for MEK and a K value of 5 for toluene; sp gr for MEK = 0.805, for toluene = 0.866; MW for MEK = 72.1, for toluene = 92.13.

$$Q \text{ for MEK} = \frac{(403)(0.805)(10^6)(4)(2/60)}{72.1 \times 200} = 3000 \text{ cfm}$$

$$Q \text{ for toluene} = \frac{(403)(0.866)(10^6)(5)(2/60)}{92.13 \times 50} = 12,627 \text{ cfm}$$

$$Q \text{ for mixture} = 3000 + 12,627 = 15,627 \text{ cfm}$$

2.5 DILUTION VENTILATION FOR FIRE AND EXPLOSION

Another function of dilution ventilation is to reduce the concentration of vapors within an enclosure to below the lower explosive limit. It should be stressed that this concept is never applied in cases where workers are exposed to the vapor. In such instances, dilution rates for health hazard control are always applied. The reason for this will be apparent when comparing TLVs and lower explosive limits (LELs).

The TLV of xylene is 100 ppm. The LEL of xylene is 1% or 10,000 ppm. An atmosphere of xylene safe-guarded against fire and explosion usually will be kept below 25% of the LEL or 2500 ppm. Exposure to such an atmosphere may cause severe illness or death. However, in baking and drying ovens, in enclosed air drying spaces, within ventilation ductwork, etc., dilution ventilation for fire and explosion is used to keep the vapor concentration to below the LEL.

Equation 2.5 can be modified to yield air quantities to dilute below the LEL. By substituting LEL for TLV:

$$Q = \frac{(403)(\text{sp gr liquid})(100)(ER)(S_f)}{(\text{MW liquid})(LEL)(B)} \text{ (for Standard Air)} \quad [2.12]$$

Note 1. Since LEL is expressed in % (parts per 100) rather than ppm (parts per million as for the TLV), the coefficient of 1,000,000 becomes 100.

2. S_f is a safety coefficient which depends on the percent of the LEL necessary for safe conditions. In most ovens and drying enclosures, it has been found desirable to maintain vapor concentrations at not more than 25% of the LEL at all times in all parts of the oven. In properly ventilated continuous ovens, a S_f coefficient of 4 (25% of the LEL) is used. In batch ovens, with good air distribution, the existence of peak drying rates requires an S_f coefficient of 10 or 12 to maintain safe concentrations at all times. In non-recirculating or improperly ventilated batch or continuous ovens, larger S_f coefficients may be necessary.
3. B is a constant which takes into account the fact that the lower explosive limit of a solvent vapor or air mixture decreases at elevated temperatures. $B = 1$ for temperatures up to 250 F; $B = 0.7$ for temperatures above 250 F.

EXAMPLE PROBLEM

A batch of enamel-dipped shelves is baked in a recirculating oven at 350 F for 60 minutes. Volatiles in the enamel applied to the shelves consist of two pints of xylene. What oven ventilation rate, in cfm, is required to dilute the xylene vapor concentration within the oven to a safe limit at all times?

For xylene, the LEL = 1.0%; sp gr = 0.88; MW = 106; Sf = 10; B = 0.7. From Equation 2.12:

$$Q = \frac{(403)(0.88)(2/60)(100)(10)}{(106)(10)(0.7)} = 159 \text{ cfm}$$

Since the above equation is at standard conditions, the air flow rate must be converted from 70 F to 350 F (operating conditions):

$$Q_A = (Q_{STP}) (\text{Ratio of Absolute Temperature})$$

$$= (Q_{STP}) \frac{(460 \text{ F} + 350 \text{ F})}{(460 \text{ F} + 70 \text{ F})}$$

$$Q_A = 159 \left(\frac{810}{530} \right)$$

$$= 243 \text{ cfm}$$

EXAMPLE PROBLEM

In many circumstances, solvent evaporation rate is non-uniform due to the process temperature or the manner of solvent use.

A 6 ft diameter muller is used for mixing resin sand on a 10-minute cycle. Each batch consists of 400 pounds of sand, 19 pounds of resin, and 8 pints of ethyl alcohol (the ethyl alcohol evaporates in the first two minutes). What ventilation rate is required?

For ethyl alcohol, LEL = 3.28%; sp gr = 0.789; MW = 46.07; Sf = 4; B = 1

$$Q = \frac{(403)(0.789)(3/2)(100)(4)}{(46.07)(3.28)(1)} = 3367 \text{ cfm}$$

Another source of data is the National Board of Fire Underwriters' Pamphlet #86, *Standard for Class A Ovens and Furnaces*.^(2,3) This contains a more complete list of solvents and their properties. In addition, it lists and describes a number of safeguards and interlocks which must always be considered in connection with fire dilution ventilation. See also Reference 2.4.

2.6 FIRE DILUTION VENTILATION FOR MIXTURES

It is common practice to regard the entire mixture as consisting of the components requiring the highest amount of dilution per unit liquid volume and to calculate the required air quantity on that basis. [This component would be the one

with the highest value for sp gr/(MW)(LEL).]

2.7 VENTILATION FOR HEAT CONTROL

Ventilation for heat control in a hot industrial environment is a specific application of general industrial ventilation. The primary function of the ventilation system is to prevent the acute discomfort, heat-induced illness and possible injury of those working in or generally occupying a designated hot industrial environment. Heat-induced occupational illnesses, injuries, or reduced productivity may occur in situations where the total heat load may exceed the defenses of the body and result in a heat stress situation. It follows, therefore, that a heat control ventilation system or other engineering control method must follow a physiological evaluation in terms of potential heat stress for the occupant in the hot industrial environment.

Due to the complexity of conducting a physiological evaluation, the criteria presented here are limited to general considerations. It is strongly recommended, however, that the NIOSH Publication No. 86-113, *Criteria for a Recommended Standard, Occupational Exposure to Hot Environments*,^(2,5) be reviewed thoroughly in the process of developing the heat control ventilation system.

The development of a ventilation system for a hot industrial environment usually includes the control of the ventilation air flow rate, velocity, temperature, humidity, and air flow path through the space in question. This may require inclusion of certain phases of mechanical air-conditioning engineering design which is outside the scope of this manual. The necessary engineering design criteria that may be required are available in appropriate publications of the American Society of Heating, Refrigeration and Air-Conditioning Engineers (ASHRAE) handbook series.

2.8 HEAT BALANCE AND EXCHANGE

An essential requirement for continued normal body function is that the deep body core temperature be maintained within the acceptable range of about 37 C (98.6 F) \pm 1 C (1.8 F). To achieve this, body temperature equilibrium requires a constant exchange of heat between the body and the environment. The rate and amount of the heat exchange are governed by the fundamental laws of thermodynamics of heat exchange between objects. The amount of heat that must be exchanged is a function of 1) the total heat produced by the body (metabolic heat), which may range from about 1 kilocalorie (kcal) per kilogram (kg) of body weight per hour (1.16 watts) at rest to 5 kcal/kg body weight/hour (7 watts) for moderately hard industrial work; and 2) the heat gained, if any, from the environment. The rate of heat exchange with the environment is a function of air temperature and humidity; skin temperature; air velocity; evaporation of sweat; radiant temperature; and type, amount, and characteristics of the clothing worn, among other factors. Respi-

ratory heat loss is of little consequence in human defenses against heat stress.

The basic heat balance equation is:

$$\Delta S = (M - W) \pm C \pm R - E \quad [2.13]$$

where:

ΔS = change in body heat content

$(M - W)$ = total metabolism - external work performed

C = convective heat exchange

R = radiative heat exchange

E = evaporative heat loss

To solve the equation, measurement of metabolic heat production, air temperature, air water vapor pressure, wind velocity, and mean radiant temperature are required.

The major modes of heat exchange between man and the environment are convection, radiation, and evaporation. Other than for brief periods of body contact with hot tools, equipment, floors, etc., which may cause burns, conduction plays a minor role in industrial heat stress. Because of the typically small areas of contact between either body surfaces or clothing and hot or cold objects, heat exchange by thermal conduction is usually not evaluated in a heat balance equation for humans. The effect of heat exchange by thermal conduction in human thermal regulation is important when large areas of the body are in contact with surfaces that are at temperatures different from average skin temperature (nominally 95 F), e.g., when someone is prone or supine for long periods. It is also important when even small body areas are in contact with objects that provide steep thermal gradients for heat transfer, e.g., when someone is standing on very cold or very hot surfaces.

The equations for calculating heat exchange by convection, radiation, and evaporation are available in Standard International (SI) units, metric units, and English units. In SI units heat exchange is in watts per square meter of body surface (W/m^2). The heat exchange equations are available in both metric and English units for both the seminude individual and the worker wearing conventional long-sleeved work shirt and trousers. The values are in kcal/h or British thermal units per hour (Btu/h) for the "standard worker" defined as one who weighs 70 kg (154 lbs) and has a body surface area of 1.8 m² (19.4 ft²).

2.8.1 Convection: The rate of convective heat exchange between the skin of a person and the ambient air immediately surrounding the skin is a function of the difference in temperature between the ambient air (t_a), the mean weighted skin temperature (t_{sk}) and the rate of air movement over the skin (V_a). This relationship is stated algebraically for the "standard worker" wearing the customary one-layer work clothing ensemble as:

$$C = 0.65 V_a^{0.6} (t_a - t_{sk}) \quad [2.14]$$

where:

C = convective heat exchange, Btu/h

V_a = air velocity, fpm

t_a = air temperature, F

t_{sk} = mean weighted skin temperature, usually assumed to be 95 F

When $t_a > 95$ F there will be a gain in body heat from the ambient air by convection. When $t_a < 95$ F, heat will be lost from the body to the ambient air by convection.

2.8.2 Radiation: Infrared radiative heat exchange between the exposed surfaces of a person's skin and clothing varies as a function of the difference between the fourth power of the absolute temperature of the exposed surfaces and that of the surface of the radiant source or sink, the exposed areas and their emissivities. Heat is gained by thermal radiation if the facing surface is warmer than the average temperature of the exposed skin and clothing, and *vice versa*. A practical approximation for infrared radiant heat exchange for a person wearing conventional clothing is:

$$R = 15.0 (t_w - t_{sk}) \quad [2.15]$$

where:

R = radiant heat exchange, Btu/h

t_w = mean radiant temperature, F

t_{sk} = mean weighted skin temperature

2.8.3 Evaporation: The evaporation of water (sweat) or other liquids from the skin or clothing surfaces results in a heat loss from the body. Evaporative heat loss for humans is a function of air flow over the skin and clothing surfaces, the water vapor partial pressure gradient between the skin surface and the surrounding air, the area from which water or other liquids are evaporating and mass transfer coefficients at their surfaces.

$$E = 2.4 V_a^{0.6} (\rho_{sk} - \rho_a) \quad [2.16]$$

where:

E = evaporative heat loss, Btu/h

V_a = air velocity, fpm

ρ_a = water vapor pressure of ambient air, mmHg

ρ_{sk} = water vapor pressure on the skin, assumed to be 42 mm Hg at a 95 F skin temperature

2.9 ADAPTIVE MECHANISM OF THE BODY

Even people in generally good health can adjust physiologically to thermal stress only over a narrow range of environmental conditions. Unrestricted blood flow to the skin, an unimpeded flow of dry, cool air over the skin surface and sweating are prime defenses in heat stress. Although heat

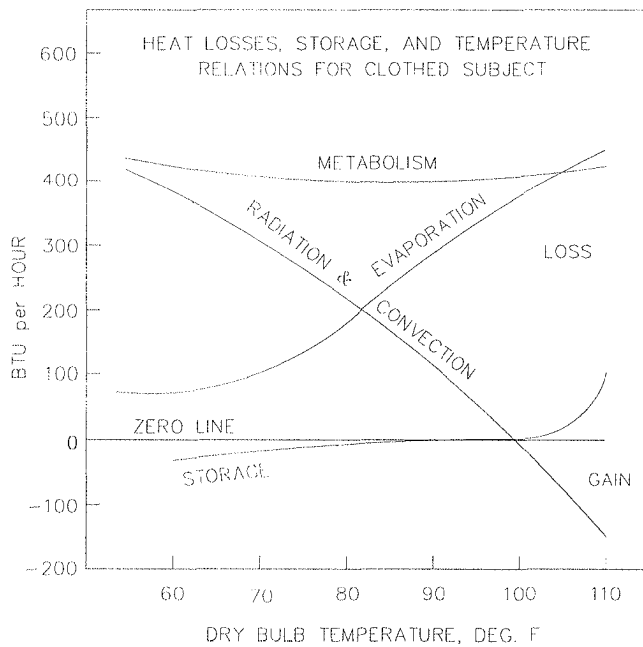


FIGURE 2-4. Heat losses, storage, and temperature relations

produced by muscle activity reduces the impact of cold stress, it can add substantially to the total challenge during heat stress. Diminished health status, medications, limited prior thermal exposure, among other factors, increase danger from thermal stresses.

The reflex control of blood flow is the body's most effective and important first line of defense in facing either cold or heat stress. Reducing blood flow to the skin of the hands, feet, fingers and toes is an important measure for reducing heat loss in a cold environment. Blood flow to the skin, however, increases many-fold during heat stress. Its effect is to increase rates of heat distribution in the body and maximize conductive, convective, radiative and evaporative heat losses to the environment (Figure 2-4). Its cost is often to reduce perfusion of other organs, especially the brain, and reduce systemic arterial blood pressure, leading to reduced consciousness, collapse, heat exhaustion and other heat-induced illnesses.

Reflex sweating during the physical activities of exercise, work and/or heat stress brings often large volumes of body water and electrolytes (salts) to the skin surface. Heat is lost when the water in sweat evaporates. Whether the electrolytes remain on the skin surface or are deposited in clothing, they are nonetheless permanently lost to the body. The electrolyte content of a typical American diet usually provides adequate electrolyte replacement for these losses. Electrolyte replacement fluids, however, may be necessary for people on salt-restricted diets and those who commonly sustain periods of prolonged and profuse sweating. It is essential for everyone that the lost body water and electrolytes are replaced in the same volume and proportion as lost in sweat. Muscle spasms, cramps, gastrointestinal disturbances and general malaise,

among other signs and symptoms, commonly develop when they are not.

2.10 ACCLIMATIZATION

People in generally good health normally develop heat acclimatization in a week or so after intermittently working or exercising in high heat. Its effect is to improve the comfort and safety of the heat exposure. It occurs because of an increase in total circulating blood volume, an improved ability to maintain systemic arterial blood pressure during heat stress, and a developed ability to produce larger volumes of more dilute sweat at rates of production more precisely matched to the heat load. Heat acclimatization rapidly diminishes even after a day or so of discontinued activity in the heat. Most is lost after about a week.

2.11 ACUTE HEAT DISORDERS

A variety of heat disorders can be distinguished clinically when individuals are exposed to excessive heat. A brief description of these disorders follows.

2.11.1 Heat Stroke: Heat stroke (also known as "sun stroke") is a **life-threatening** condition which without exception demands immediate emergency medical care and hospitalization. Before medical care arrives, move the person to a shaded area, check for other injuries, ensure there is an unobstructed airway, remove or loosen clothing, and flood the body surface with free-flowing, tepid (not cold) water. Vigorous fanning helps cooling. Heat stroke develops when body heat gains from exercise, work and/or a hot environment overwhelm normal thermoregulatory defenses. Characteristically, sweating has ceased, the skin is hot and dry, and deep body temperature is above about 104 F. The person may be either diaphoretic, semiconscious, unconscious or agitated, delirious and in convulsions. Demand medical care even if consciousness returns—lethal effects may develop in the next 24 to 72 hours.

2.11.2 Heat Exhaustion: Heat exhaustion (also called "exercise-induced heat exhaustion" and "heat syncope") most commonly occurs in people who are not heat acclimatized and who are in poor physical condition, obese, inappropriately dressed, and exercising or working energetically in the heat at unaccustomed and/or demanding tasks. It is characterized by lightheadedness, dizziness, vision disturbances, nausea, vague flu-like symptoms, tinnitus, weakness, and occasionally, collapse. The person's deep body temperature is typically in a normal range or only slightly elevated; the skin is moist and cool but may be reddened by its high rate of blood flow. Heat exhaustion develops when there is reflex demand for blood flow to the skin to dissipate body heat and a simultaneous reflex demand for blood flow to exercising muscles to meet metabolic needs of increased activity. These peripheral distributions of blood volume reduce systemic arterial pressure and brain blood flow, causing most of the

TABLE 2-2. Estimating Energy Cost of Work by Task Analysis^(2,6)

A. Body position and movement		kcal/min*
Sitting		0.3
Standing		0.6
Walking		2.0-3.0
Walking uphill		Add 0.8/meter rise
B. Type of Work		Average kcal/min Range kcal/min
Hand work – light		0.4 0.2-1.2
Hand work – heavy		0.9
Work one arm – light		1.0 0.7-2.5
Work one arm – heavy		1.7
Work both arms – light		1.5 1.0-3.5
Work both arms – heavy		2.5
Work whole body – light		3.5 2.5-15.0
Work whole body – moderate		5.0
Work whole body – heavy		7.0
Work whole body – very heavy		9.0
C. Basal metabolism		1.0
D. Sample calculation **		
Assembling work with heavy hand tools		
1. Standing		0.6
2. Two-arm work		3.5
3. Basal metabolism		1.0
TOTAL		5.1 kcal/min

*For standard worker of 70 kg body weight (154 lbs) and 1.8 m² body surface (19.4 ft²).

**Example of measuring metabolic heat production of a worker when performing initial screening.

symptoms of heat exhaustion. Resting in a cool environment where there is free flowing, dry air usually remedies symptoms quickly. Although heat exhaustion is debilitating and uncomfortable, it is not often a long-term health threat. There are considerable dangers, of course, for anyone operating machinery when consciousness is impaired because of heat exhaustion or for any other reason.

2.11.3 Heat Cramps and Heat Rash: Heat cramps (also known as "muscle cramps") are spontaneous, involuntary, painful and prolonged muscle contractions that commonly occur in otherwise healthy people when both body water and electrolyte levels have not been restored after extended periods of heavy sweating during exercise and/or heat stress. Full recovery can be expected in about 24 hours with the use of electrolyte replacement fluids and rest. Heat rash (also known as "prickly heat" or "miliaria rubia") is an acute, inflammatory skin disease characterized by small red, itchy or tingling lesions, commonly in areas of skin folds or where there is

abrasive clothing. It commonly disappears when these areas are kept dry, unabraded and open to free flowing, dry air.

2.12 ASSESSMENT OF HEAT STRESS AND HEAT STRAIN

Heat Stress is defined by environmental measurements of air temperature, humidity, air flow rate, the level of radiant heat exchange and evaluation of a person's metabolic heat production rate from exercise and/or work. Heat stress is the load on thermoregulation. *Heat Strain* is defined as the cost to each person facing heat stress. Although all people working at the same intensity in the same environment face the same level of heat stress, each is under a unique level of heat strain. Almost any environmental thermal exposure will be comfortable and safe for some, but endangering, even lethal to others. Because disabilities, danger and death arise directly from heat strain, no measure of heat stress is a reliable indicator of a particular person's heat strain or the safety of the exposure.

2.12.1 Evaluation of Heat Stress: Dry-bulb air temperature (DB: so-called "dry-bulb" temperature) is measured by calibrated thermometers, thermistors, thermocouples and similar temperature-sensing devices which themselves do not produce heat and which are protected from the effects of thermal conduction, evaporation, condensation and radiant heat sources and sinks. Relative humidity is evaluated psychrometrically as a function of the steady-state difference between dry-bulb temperature and that indicated by the temperature of a sensor covered with a freely evaporating, water-saturated cotton wick. Such a measure reports "NWB"

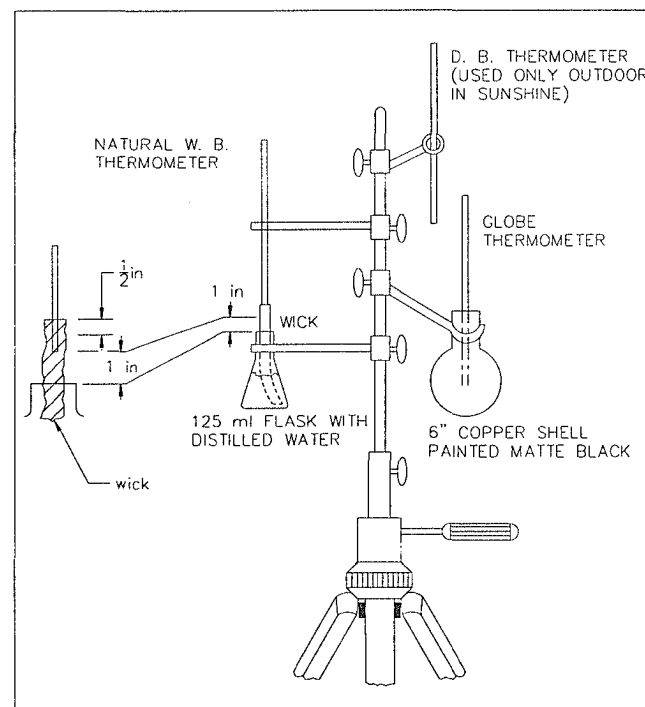


FIGURE 2-5. Determination of wet-bulb globe temperature

(natural wet-bulb temperature) when the wetted sensor is affected only by prevailing air movement, and "WB" (when it is exposed to forced convection). Free air movement is measured with an unobstructed anemometer. Infrared radiant "heat transfer" is typically measured by a temperature sensor at the center of a 6-inch, hollow, copper sphere painted flat ("matte") black. Such a measure reports "GT" (globe temperature) (Figure 2-5). A person's metabolic heat production is usually evaluated from an estimated level of average physical activity (Table 2-2).

Although there are a number of different indices for evaluating heat stress, none is reliable as a sole indicator of heat strain for a specific person. Dry-bulb temperature is the least valuable measure of heat stress because it provides no information about ambient relative humidity, or heat exchange by convection or radiation, and gives no estimate of the metabolic heat production. Wet-bulb, globe temperature (WBGT) is often used as an index of heat stress. When there is a source of radiant heat transfer (solar radiation, hot surfaces of machinery):

$$\text{WBGT} = 0.7 t_{\text{nw}} + 0.2 t_g + 0.1 t_a \quad [2.17]$$

where

t_{nw} = natural wet-bulb temperature

t_g = globe temperature

When radiant heat transfer is negligible:

$$\text{WBGT} = 0.7 t_{\text{nw}} + 0.3 t_g \quad [2.18]$$

WBGT evaluates more factors contributing to heat stress than does dry-bulb temperature alone. It does not, however, effectively evaluate the importance of energy transfer from human skin by convection which is essential for the removal of heat from the skin surface and the formation of water vapor from secreted sweat. Nor does WBGT evaluate the importance of metabolic heat production in the heat stress. Under some environmental conditions, heat produced by metabolism is the predominant stressor.

2.12.2 Evaluation of Heat Strain: The incidence and severity of heat strain will vary greatly among people exposed to the same level of heat stress. Paying attention to the early signs and symptoms of heat strain is the best first line of defense against debilitating heat-induced discomfort and injuries. It is dangerous, inappropriate and irresponsible to consider a heat stress as safe for all when some exposed to it show heat strain signs and symptoms, while others do not. Acute heat strain is indicated by:

Visible Sweating: Thermoregulatory reflexes normally fine-tune with precision the rate of sweating to the rate at which body heat must be lost to maintain homeostasis. Normally, there is no liquid water on the skin surface in a tolerable heat stress because water brought to the skin surface by sweating readily forms

invisible water vapor in the process of evaporative cooling. Although an all too common occurrence in the workplace, liquid sweat either on the skin surface, or soaked into clothing, is a sure sign of heat strain. It indicates the level of sweating required to keep body temperature in a normal range cannot be matched by the rate of water evaporation from the skin surface to the environment. It is necessary either to increase the air flow rate over skin and clothing surfaces, lower ambient temperature and relative humidity, reduce radiative heat gain, and/or reduce metabolic heat production if progressive heat disabilities are to be avoided.

Discontinued Sweating: A hot, dry skin for someone exposed to heat stress is a dangerous sign. It indicates suppression of sweating, perhaps exacerbated by prescription or over-the-counter medications. The appearance of a hot, dry skin for someone in a heat stress demands immediate attention and corrective actions.

Elevated Heart Rate: Short-term increases in heart rate are normal for episodic increases in work load. In a heat stress, however, a sustained heart rate greater than 160/min for those younger than about 35 years, or 140/min for those who are older, is a sign of heat strain.

Elevated Deep Body Temperature: A sustained deep body temperature greater than 100.4 F is a sign of heat strain in someone exposed to heat stress.

Decreased Systemic Arterial Blood Pressure: A fall in blood pressure of more than about 40 Torr in about 3.5 minutes for someone working in a heat stress indicates a heat-induced disability. Reduced consciousness, feeling of weakness, vision disturbances, and other signs and symptoms are likely to follow.

Personal Discomfort: Heat strain may be indicated in some heat-stressed individuals by severe and sudden fatigue, nausea, dizziness, lightheadedness, or fainting. Others may complain of irritability; mental confusion; clumsiness; forgetfulness; general malaise; the development of sometimes vague, flu-like symptoms; and paradoxical chills and shivering.

Infrequent Urination: Urinating less frequently than normal and the voiding of a small volume of dark-colored urine is a sign of whole body dehydration. Dehydration compromises the body's ability to maintain a large enough circulating blood volume so that normal blood pressure is maintained in the face of the combined stressors of exercise and heat exposure. People who work or exercise in the heat need to develop the habit of drinking adequate volumes of water at frequent enough intervals to maintain the same patterns of urination they have when not heat stressed. Those who sweat heavily for long periods need also to discuss

with their physicians a possible need for using electrolyte replacement fluids.

2.13 WORKER PROTECTION

There is improved safety, comfort and productivity when those working in the heat are:

1. In generally good physical condition and not obese, are heat acclimatized, and are experienced in the heat stressing job. They also need to know how to select clothing and maintain whole body hydration and electrolyte levels to provide the greatest comfort and safety.
2. In areas that are well-ventilated and shielded from infrared radiant heat sources.
3. Knowledgeable about the effects of their medications on cardiovascular and peripheral vascular function, blood pressure control, body temperature maintenance, sweat gland activity, metabolic effects and levels of attention or consciousness.
4. Appropriately supervised when there is a history of abuse or recovery from abuse of alcohol or other intoxicants.
5. Provided accurate verbal and written instructions, frequent training programs and other information about heat stress and strain.

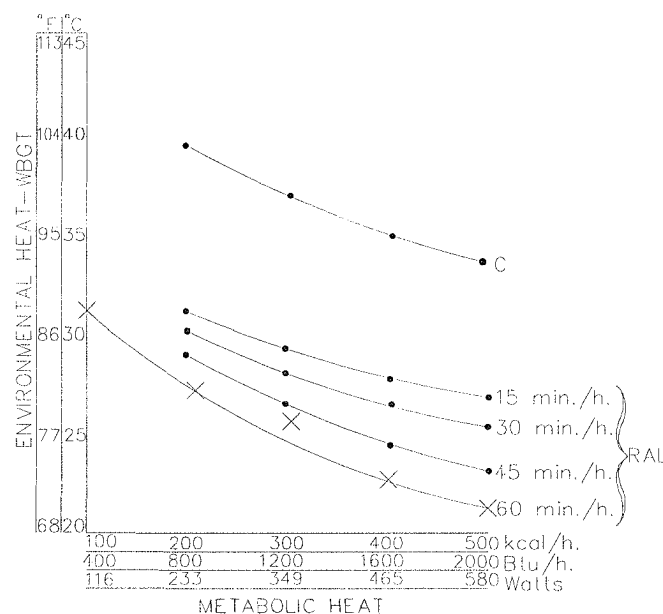
6. Able to recognize the signs and symptoms of heat strain in themselves and others exposed to heat stress and know the appropriately effective steps for their remediation (Figures 2-6 and 2-7).

2.14 VENTILATION CONTROL

The control method presented here is limited to a general engineering approach. Due to the complexity of evaluating a potential heat stress-producing situation, it is essential that the accepted industrial hygiene method of recognition, evaluation, and control be utilized to its fullest extent. In addition to the usual time-limited exposures, it may be necessary to specify additional protection which may include insulation, baffles, shields, partitions, personal protective equipment, administrative control, and other measures to prevent possible heat stress. Ventilation control measures may require a source of cooler replacement air, an evaporative or mechanically cooled source, a velocity cooling method, or any combination thereof. Specific guidelines, texts, and other publications or sources should be reviewed for the necessary data to develop the ventilation system.

2.15 VENTILATION SYSTEMS

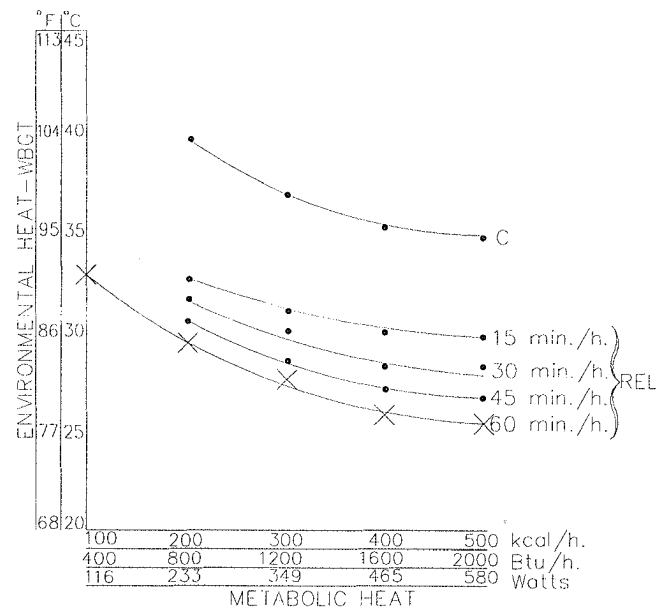
Exhaust ventilation can be used to remove excessive heat and/or humidity if a replacement source of cooler air is available. If it is possible to enclose the heat source, such as is the case with ovens or certain furnaces, a gravity or forced



C = CEILING LIMIT
RAL = RECOMMENDED ALERT LIMIT

*FOR "STANDARD WORKER" OF 70 kg (154 lbs) BODY WEIGHT AND
1.8 m² (19.4 ft²) BODY SURFACE.

FIGURE 2-6. Recommended heat-stress alert limits, heat-unacclimatized workers



C = CEILING LIMIT

REL = RECOMMENDED EXPOSURE LIMIT

*FOR "STANDARD WORKER" OF 70 kg (154 lbs) BODY WEIGHT AND 1.8 m² (19.4 ft²) BODY SURFACE.

FIGURE 2-7. Recommended heat-stress exposure limits, heat-acclimatized workers

air stack may be all that is necessary to remove excessive heat from the workroom. If a partial enclosure or local hood is indicated, control velocities should be used as described in Chapter 3.

Many operations do not lend themselves to local exhaust. General ventilation may be the only alternative. To determine the required general ventilation, the designer must estimate the acceptable temperature or humidity rise. The first step in determining the required volumetric flow is to determine the sensible and latent heat load. Next, determine the volumetric flow to dissipate the sensible heat and the volumetric flow to dissipate the latent heat. The required general ventilation is the larger of the two volumetric flows.

The sensible heat rise can be determined by the following:

$$H_s = Q_s \times \rho \times c_p \times \Delta T \times (60 \text{ min/hr}) \quad [2.19]$$

where:

H_s = Sensible heat gain, BTU/hr

Q_s = Volumetric flow for sensible heat, cfm

ρ = Density of the air, lbm/ft³

c_p = Specific heat of the air, BTU/lbm- F

ΔT = Change in temperature, F

For air $c_p = 0.24$ BTU/lbm - F and $\rho = 0.075$ lbm/ft³;

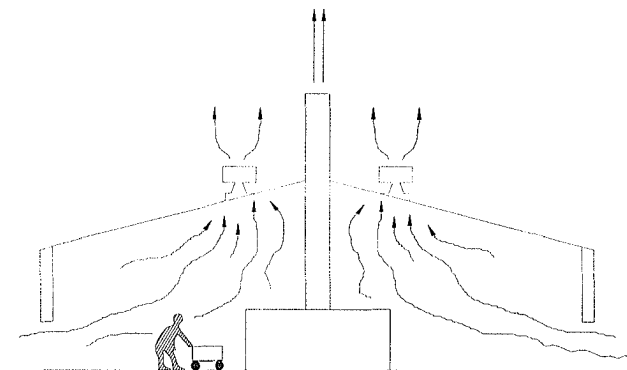


FIGURE 2-8. Good natural ventilation and circulation

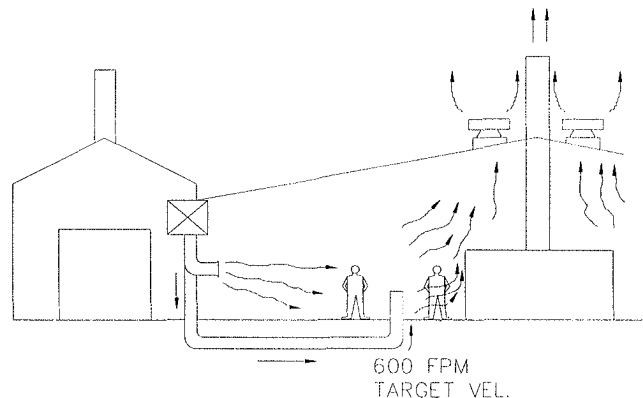


FIGURE 2-9. Good mechanically supplied ventilation

consequently, the equation becomes

$$H_s = 108 \times Q_s \times \Delta T$$

or

$$Q_s = H_s \div (108 \times \Delta T) \quad [2.20]$$

In order to use this equation, it is necessary to first estimate the heat load. This will include loads from the sun, people, lights, and motors, as well as other particular sources of heat. Of these, sun load, lights, and motors are all completely sensible. The people heat load is part sensible and part latent. In the case of hot processes which give off both sensible and latent heat, it will be necessary to estimate the amounts or percents of each. In using the above equation for sensible heat, one must decide the amount of temperature rise which will be permitted. Thus, in a locality where 90 F outdoor dry bulb may be expected, if it is desired that the inside temperature not exceed 100 F, or a 10-degree rise, a certain air flow rate will be necessary. If an inside temperature of 95 F is required, the air flow rate will be doubled.

For latent heat load, the procedure is similar, although more difficult. If the total amount of water vapor is known, the heat load can be estimated from the latent heat of vaporization, 970 BTU/lb. In a manner similar to the sensible heat calculations, the latent heat gain can be approximated by:

$$H_l = Q_l \times \rho \times c_l \times \Delta h \times (60 \text{ min/hr}) \times (1 \text{ lb} / 7000 \text{ grains})$$

where:

H_l = Latent heat gain, BTU/hr

Q_l = Volumetric flow for latent heat, cfm

ρ = Density of the air, lbm/ft³

c_l = Latent heat of vaporization, BTU/lbm

Δh = Change in absolute humidity of the air, grains-water/lbm-dry air

For air, c_l is approximately 970 BTU/lb and $\rho = 0.075$ lbm/ft³. Consequently, the equation becomes

$$H_l = 0.62 \times Q_l \times \Delta h$$

or

$$Q_l = \frac{H_l}{0.62 \times \Delta h} \quad [2.21]$$

If the rate of moisture released, M in pounds per hours, is known, then

$$M = Q_l \times \rho \times \Delta h \times (1 \text{ lb} / 7000 \text{ gr}) \times (60 \text{ min/hr})$$

$$= Q_l \times \rho \times \Delta h \div (116.7)$$

or

$$Q_l = \frac{116.7 \times M}{\rho \times \Delta h} \quad [2.22]$$

The value of the "grains-water per pound-air difference" is read from a psychrometric chart or table. It represents the difference in moisture content of the outdoor air and the conditions acceptable to the engineer designing the exhaust system. The air quantities calculated from the two equations above should not be added to arrive at the required quantity. Rather, the higher quantity should be used since both sensible and latent heat are absorbed simultaneously. Furthermore, in the majority of cases the sensible heat load far exceeds the latent heat load, so the design usually can be calculated on the basis of sensible heat alone.

The ventilation should be designed to flow through the hot environment in a manner that will efficiently control the excess heat. Figures 2-8 and 2-9 illustrate this principle.

2.16 VELOCITY COOLING

If the air dry-bulb or wet-bulb temperatures are lower than 95–100 F, the worker may be cooled by convection or evaporation. When the dry bulb temperature is higher than 95–100 F, increased air velocity may add heat to the worker by convection. If the wet bulb temperature is high also, evaporative heat loss may not increase proportionately and the net result will be an increase in the worker's heat burden. Many designers consider that supply air temperature should not exceed 80 F for practical heat relief.

Current practice indicates that air velocities in Table 2-3 can be used successfully for direct cooling of workers. For best results, provide directional control of the air supply (Figure 2-10) to accommodate daily and seasonal variations in heat exposure and supply air temperature.

2.17 RADIANT HEAT CONTROL

Since radiant heat is a form of heat energy which needs no medium for its transfer, radiant heat cannot be controlled by

TABLE 2-3. Acceptable Comfort Air Motion at the Worker

	Air Velocity, fpm*
Continuous Exposure	
Air conditioned space	50–75
Fixed work station, general ventilation	
or spot cooling: Sitting	75–125
Standing	100–200
Intermittent Exposure, Spot Cooling or Relief Stations	
Light heat loads and activity	1000–2000
Moderate heat loads and activity	2000–3000
High heat loads and activity	3000–4000

*Note: Velocities greater than 1000 fpm may seriously disrupt the performance of nearby local exhaust systems. Care must be taken to direct air motion to prevent such interference.

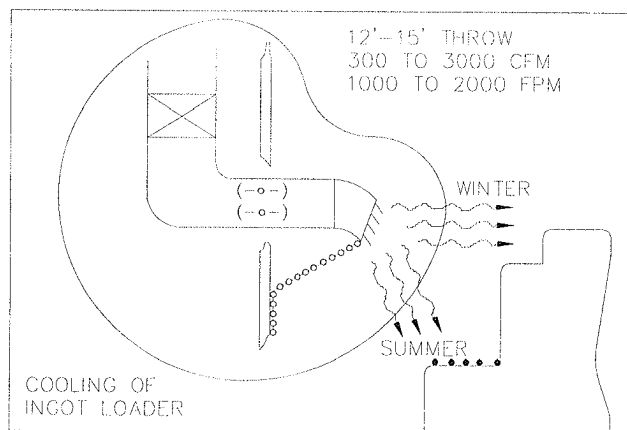


FIGURE 2-10. Spot cooling with volume and directional control

ventilation. Painting or coating the surface of hot bodies with materials having low radiation emission characteristics is one method of reducing radiation.

For materials such as molten masses of metal or glass which cannot be controlled directly, radiation shields are effective. These shields can consist of metal plates, screens, or other material interposed between the source of radiant heat and the workers. Shielding reduces the radiant heat load by reflecting the major portion of the incident radiant heat away from the operator and by re-emitting to the operator only a portion of that radiant heat which has been absorbed. Table 2-4 indicates the percent of both reflection and emission of radiant heat associated with some common shielding materials. Additional ventilation will control the sensible heat load but will have only a minimal effect, if any, upon the radiant heat load. See Figure 2-11.

2.18 PROTECTIVE SUITS FOR SHORT EXPOSURES

For brief exposures to very high temperatures, insulated aluminized suits and other protective clothing may be worn. These suits reduce the rate of heat gain by the body but provide no means of removing body heat; therefore, only short exposures may be tolerated.

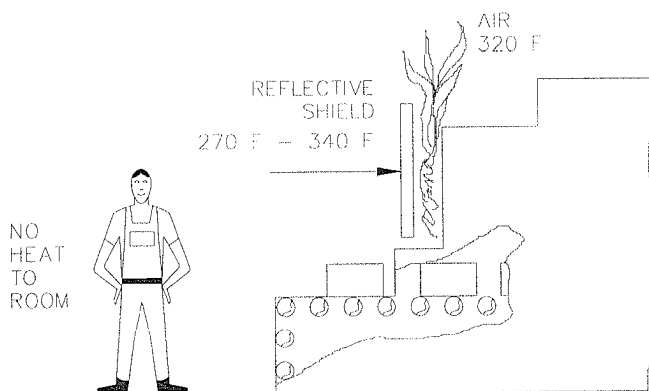


FIGURE 2-11. Heat Shielding

TABLE 2-4. Relative Efficiencies of Common Shielding Materials

Surface of Shielding	Reflection of Radiant Heat Incident Upon Surface	Emission of Radiant Heat from Surface
Aluminum, bright	95	5
Zinc, bright	90	10
Aluminum, oxidized	84	16
Zinc, oxidized	73	27
Aluminum paint, new, clean	65	35
Aluminum paint, dull, dirty	40	60
Iron, sheet, smooth	45	55
Iron, sheet, oxidized	35	65
Brick	20	80
Lacquer, black	10	90
Lacquer, white	10	90
Asbestos board	6	94
Lacquer, flat black	3	97

2.19 RESPIRATORY HEAT EXCHANGERS

For brief exposure to air of good quality but high temperature, a heat exchanger on a half-mask respirator face piece is available. This device will bring air into the respiratory passages at a tolerable temperature but will not remove contaminants nor furnish oxygen in poor atmospheres.

2.20 REFRIGERATED SUITS

Where individuals must move about, cold air may be blown into a suit or hood worn as a portable enclosure. The usual refrigeration methods may be used with insulated tubing to the suit. It may be difficult, however, to deliver air at a sufficiently low temperature. If compressed air is available, cold air may be delivered from a vortex tube worn on the suit. Suits of this type are commercially available.

2.21 ENCLOSURES

In certain hot industries, such as in steel mills, it is impractical to control the heat from the process. If the operation is such that remote control is possible, an air conditioned booth or cab can be utilized to keep the operators reasonably comfortable in an otherwise intolerable atmosphere.

2.22 INSULATION

If the source of heat is a surface giving rise to convection, insulation at the surface will reduce this form of heat transfer. Insulation by itself, however, will not usually be sufficient if the temperature is very high or if the heat content is high.

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Chapter 3

LOCAL EXHAUST HOODS

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3.1 INTRODUCTION

Local exhaust systems are designed to capture and remove process emissions prior to their escape into the workplace environment. The local exhaust hood is the point of entry into the exhaust system and is defined herein to include all suction openings regardless of their physical configuration. The primary function of the hood is to create an air flow field which will effectively capture the contaminant and transport it into the hood. Figure 3-1 provides nomenclature associated with local exhaust hoods.

3.2 CONTAMINANT CHARACTERISTICS

3.2.1 Inertial Effects: Gases, vapors, and fumes will not exhibit significant inertial effects. Also, fine dust particles, 20 microns or less in diameter (which includes respirable particles), will not exhibit significant inertial effects. These materials will move solely with respect to the air in which they are mixed. In such cases, the hood needs to generate an air flow pattern and capture velocity sufficient to control the motion of the contaminant-laden air plus extraneous air currents caused by room cross-drafts, vehicular traffic, etc.

3.2.2 Effective Specific Gravity: Frequently, the location of exhaust hoods is mistakenly based on a supposition that the contaminant is "heavier than air" or "lighter than air." In most health hazard applications, this criterion is of little value (see Figure 3-2). Hazardous fine dust particles, fumes, vapors, and gases are truly airborne, following air currents, and are not subject to appreciable motion either upward or downward because of their own density. Normal air movement will assure an even mixture of these contaminants. Exception to these observations may occur with very hot or very cold operations or where a contaminant is generated at very high levels and control is achieved before the contaminant becomes diluted.

3.2.3 Wake Effects: As air flows around an object, a phenomenon known as "boundary layer separation" occurs. This results in the formation of a turbulent wake on the downstream side of the object similar to what is observed as a ship moves through the water. The wake is a region of vigorous mixing and recirculation. If the object in question is a person who is working with, or close to, a contaminant-generating source, recirculation of the contaminant into the breathing zone is likely. An important consideration in the design of ventilation for contaminant control is minimizing this wake around the human body and, to the extent possible, keeping contaminant sources out of these recirculating regions (see also Section 3.4.6.)

3.3 HOOD TYPES

Hoods may be of a wide range of physical configurations but can be grouped into two general categories: enclosing and exterior. The type of hood to be used will be dependent on the

physical characteristics of the process equipment, the contaminant generation mechanism, and the operator/equipment interface (see Figure 3-3).

3.3.1 Enclosing Hoods: Enclosing hoods are those which completely or partially enclose the process or contaminant generation point. A complete enclosure would be a laboratory glove box or similar type of enclosure where only minimal openings exist. A partial enclosure would be a laboratory hood or paint spray booth. An inward flow of air through the enclosure opening will contain the contaminant within the enclosure and prevent its escape into the work environment.

The enclosing hood is preferred wherever the process configuration and operation will permit. If complete enclosure is not feasible, partial enclosure should be used to the maximum extent possible (see Figure 3-3).

3.3.2 Exterior Hoods: Exterior hoods are those which are located adjacent to an emission source without enclosing it. Examples of exterior hoods are slots along the edge of the tank or a rectangular opening on a welding table.

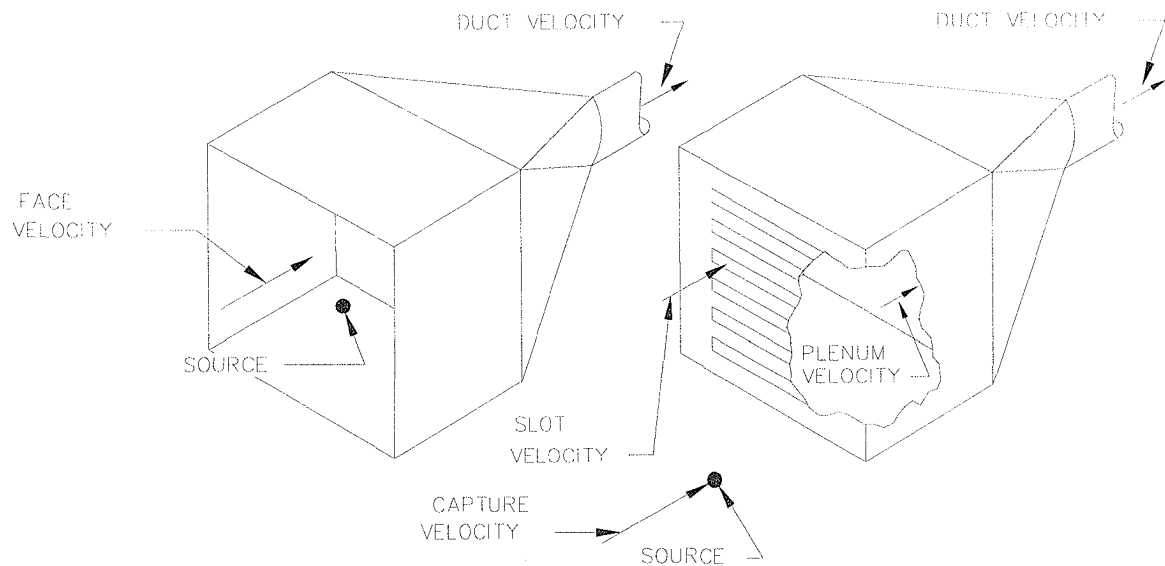
Where the contaminant is a gas, vapor, or fine particulate and is not emitted with any significant velocity, the hood orientation is not critical. However, if the contaminant contains large particulates which are emitted with a significant velocity, the hood should be located in the path of the emission. An example would be a grinding operation (see Chapter 10, VS-80-11).

If the process emits hot contaminated air, it will rise due to thermal buoyancy. Use of a side draft exterior hood (located horizontally from the hot process) may not provide satisfactory capture due to the inability of the hood-induced air flow to overcome the thermally induced air flow. This will be especially true for very high temperature processes such as a melting furnace. In such cases, a canopy hood located over the process may be indicated (see Section 3.9).

A variation of the exterior hood is the push-pull system (Section 3.8). In this case, a jet of air is pushed across a contaminant source into the flow field of a hood. Contaminant control is primarily achieved by the jet. The function of the exhaust hood is to receive the jet and remove it. The advantage of the push-pull system is that the push jet can travel in a controlled manner over much greater distances than air can be drawn by an exhaust hood alone. The push-pull system is used successfully for some plating and open surface vessel operations but has potential application for many other processes. However, the push portion of the system has potential for increasing operator exposure if not properly designed, installed, or operated. Care must be taken to ensure proper design, application, and operation.

3.4 HOOD DESIGN FACTORS

Capture and control of contaminants will be achieved by



- CAPTURE VELOCITY**— 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 CAUSING IT TO FLOW INTO THE HOOD.
- FACE VELOCITY**— AIR VELOCITY AT THE HOOD OPENING.
- SLOT VELOCITY**— AIR VELOCITY THROUGH THE OPENINGS IN A SLOT-TYPE HOOD. IT IS USED PRIMARILY AS A MEANS OF OBTAINING UNIFORM AIR DISTRIBUTION ACROSS THE FACE OF THE HOOD.
- PLENUM VELOCITY**— AIR VELOCITY IN THE PLENUM. FOR GOOD AIR DISTRIBUTION WITH SLOT-TYPES OF HOODS, THE MAXIMUM PLENUM VELOCITY SHOULD BE 1/2 OF THE SLOT VELOCITY OR LESS.
- DUCT VELOCITY**— AIR VELOCITY THROUGH THE DUCT CROSS SECTION. WHEN SOLID MATERIAL IS PRESENT IN THE AIR STREAM, THE DUCT VELOCITY MUST BE EQUAL TO OR GREATER THAN THE MINIMUM AIR VELOCITY REQUIRED TO MOVE THE PARTICLES IN THE AIR STREAM.

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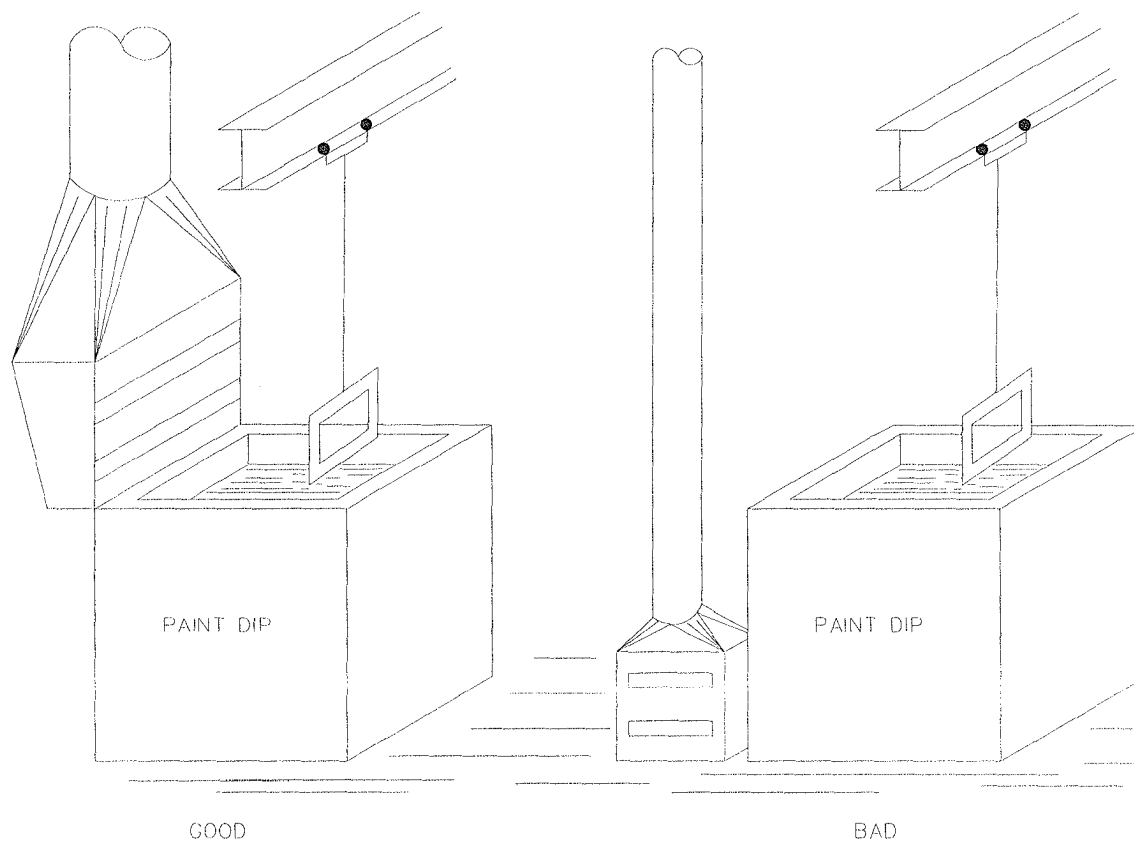
*HOOD NOMENCLATURE
LOCAL EXHAUST*

DATE

4-96

FIGURE

3-1



LOCATION

SOLVENT VAPORS IN HEALTH HAZARD CONCENTRATIONS ARE NOT APPRECIABLY HEAVIER THAN AIR. EXHAUST FROM THE FLOOR USUALLY GIVES FIRE PROTECTION ONLY.

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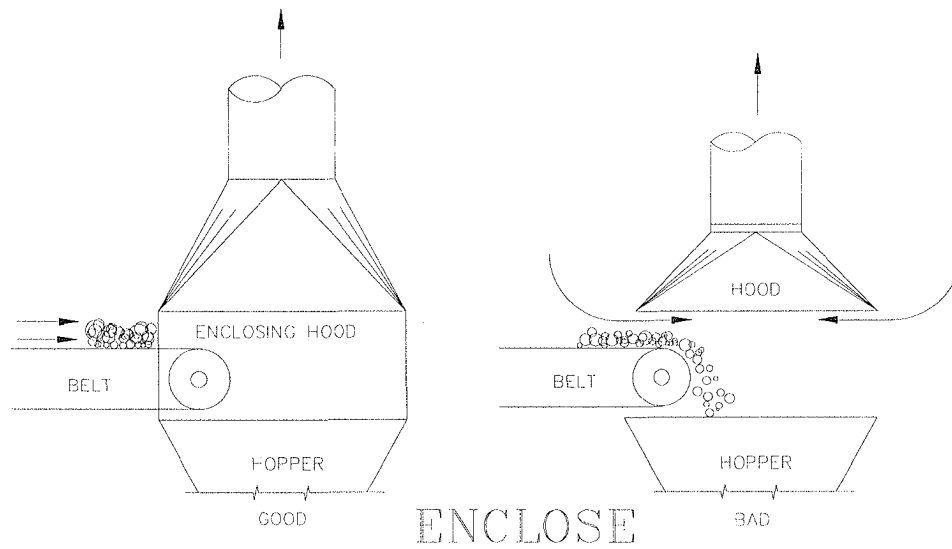
*EFFECTS OF
SPECIFIC GRAVITY*

DATE

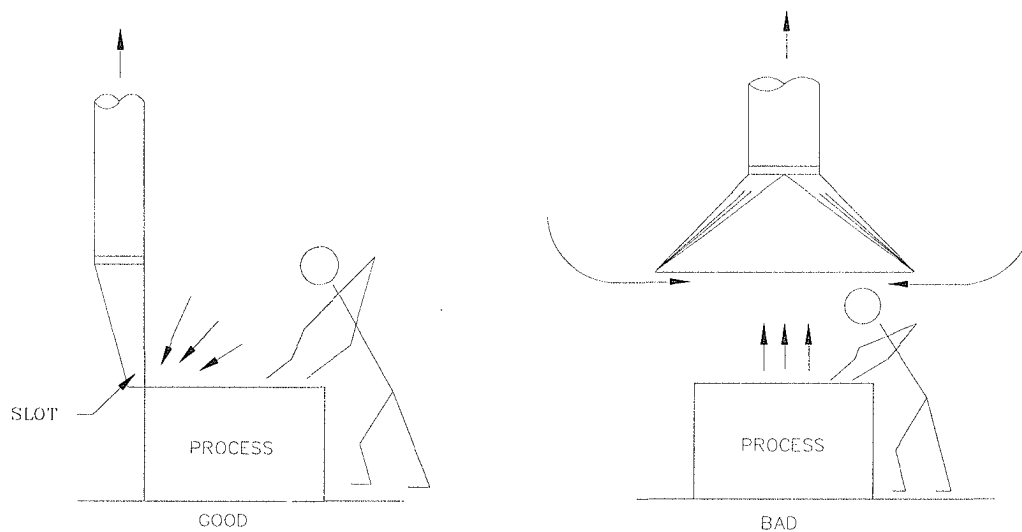
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FIGURE

3-2



ENCLOSE THE OPERATION AS MUCH AS POSSIBLE. THE MORE COMPLETELY ENCLOSED THE SOURCE, THE LESS AIR REQUIRED FOR CONTROL.



LOCATE THE HOOD SO THE CONTAMINANT IS REMOVED AWAY FROM THE BREATHING ZONE OF THE OPERATOR.

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*ENCLOSURE AND OPERATOR/
EQUIPMENT INTERFACE*

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FIGURE

3-3

TABLE 3-1. Range of Capture Velocities^(3.1, 3.2)

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

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

Lower End of Range	Upper End of Range
1. Room air currents minimal or favorable to capture.	1. Disturbing room air currents.
2. Contaminants of low toxicity or of nuisance value only.	2. Contaminants of high toxicity.
3. Intermittent, low production.	3. High production, heavy use.
4. Large hood-large air mass in motion.	4. Small hood-local control only.

the inward air flow created by the exhaust hood. Air flow toward the hood opening must be sufficiently high to maintain control of the contaminant until it reaches the hood. External air motion may disturb the hood-induced air flow and require higher air flow rates to overcome the disturbing effects. Elimination of sources of external air motion is an important factor in achieving effective control without the need for excessive air flow and its associated cost. Important sources of air motion are

- Thermal air currents, especially from hot processes or heat-generating operations.
- Motion of machinery, as by a grinding wheel, belt conveyor, etc.
- Material motion, as in dumping or container filling.
- Movements of the operator.
- Room air currents (which are usually taken at 50 fpm minimum and may be much higher).
- Rapid air movement caused by spot cooling and heating equipment.

The shape of the hood, its size, location, and rate of air flow are important design considerations.

3.4.1 Capture Velocity: The minimum hood-induced air velocity necessary to capture and convey the contaminant into the hood is referred to as capture velocity. This velocity will be a result of the hood air flow rate and hood configuration.

Exceptionally high air flow hoods (example, large foundry side-draft shakeout hoods) may require less air flow than would be indicated by the capture velocity values recommended for small hoods. This phenomenon may be ascribed to:

- The presence of a large air mass moving into the hood.

- The fact that the contaminant is under the influence of the hood for a much longer time than is the case with small hoods.
- The fact that the large air flow rate affords considerable dilution as described above.

Table 3-1 offers capture velocity data. Additional information is found in Chapter 10.

3.4.2 Hood Flow Rate Determination: Within the bounds of flanges, baffles, adjacent walls, etc., air will move into an opening under suction from all directions. For an enclosure, the capture velocity at the enclosed opening(s) will be the exhaust flow rate divided by the opening area. The capture velocity at a given point in front of the exterior hood will be established by the hood air flow through the geometric surface which contains the point.

As an example, for a theoretical unbounded point suction source, the point in question would be on the surface of a sphere whose center is the suction point (Figure 3-4).

The surface area of a sphere is $4\pi X^2$. Using $V = Q/A$

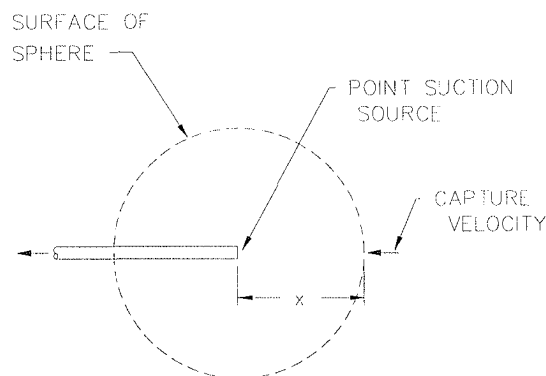


FIGURE 3-4. Point suction source

(Equation 1.3), the velocity at point X on the sphere's surface can be given by

$$Q = V(4\pi X^2) = 12.57 VX^2 \quad [3.1]$$

where:

Q = air flow into suction point, cfm

V = velocity at distance X, fpm

A = $4\pi X^2$ = area of sphere, ft²

X = radius of sphere, ft

Similarly, if an unbounded line source were considered, the surface would be that of a cylinder and the flow rate (neglecting end effects) would be

$$Q = V(2\pi XL^2) = 6.28 VXL \quad [3.2]$$

where:

L = length of line source, ft

Equations 3.1 and 3.2 illustrate, on a theoretical basis, the relationship between distance, flow, and capture velocity and can be used for gross estimation purposes. In actual practice, however, suction sources are not points or lines, but rather have physical dimensions which cause the flow surface to deviate from the standard geometric shape. Velocity contours have been determined experimentally. Flow^(3.3) for round hoods, and rectangular hoods which are essentially square, can be approximated by

$$Q = V(10X^2 + A) \quad [3.3]$$

where:

Q = air flow, cfm

V = centerline velocity at X distance from hood, fpm

X = distance outward along axis in ft. (NOTE: equation is accurate only for limited distance of X, where X is within 1.5 D)

A = area of hood opening, ft²

D = diameter of round hoods or side of essentially square hoods, ft

Where distances of X are greater than 1.5 D, the flow rate increases less rapidly with distance than Equation 3.3 indicates.^(3.4,3.5)

It can be seen from Equation 3.3 that velocity decreases inversely with the square of the distance from the hood (see Figure 3-5.)

Figures 3-6 and 3-7 show flow contours and streamlines for plane and flanged circular hood openings. Flow contours are lines of equal velocity in front of a hood. Similarly, streamlines are lines perpendicular to velocity contours. (The tangent to a streamline at any point indicates the direction of air flow at that point.)

Flow capture velocity equations for various hood configurations are provided in Figures 3-8, 3-9, 3-10, and 3-11.

3.4.3 Effects of Flanges and Baffles: A flange is a surface at and parallel to the hood face which provides a barrier to unwanted air flow from behind the hood. A baffle is a surface which provides a barrier to unwanted air flow from the front or sides of the hood.

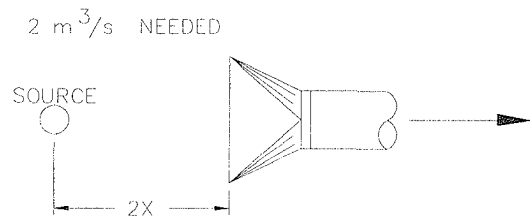
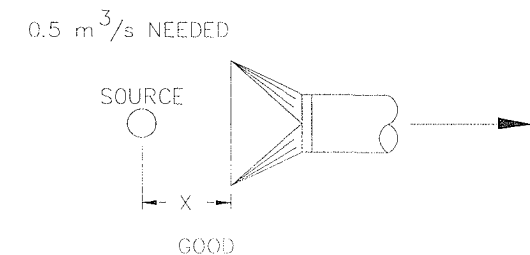
If the suction source were located on a plane, the flow area would be reduced (1/2 in both cases), thereby decreasing the flow rate required to achieve the same velocity. A flange around a hood opening has the same effect of decreasing the required flow rate to achieve a given capture velocity. In practice, flanging can decrease flow rate (or increase velocity) by approximately 25% (see Figures 3-6, 3-7, and 3-11). For most applications, the flange width should be equal to the square root of the hood area (\sqrt{A}).

Baffles can provide a similar effect. The magnitude of the effect will depend on the baffle location and size.

Figure 3-11 illustrates several hood types and gives the velocity/flow formulas which apply.

A summary of other equations for hood velocity and the impact of cross-drafts on hood performance can be found in Reference 3.25.

3.4.4 Air Distribution: Slot hoods are defined as hoods with an opening width-to-length ratio (W/L) of 0.2 or less. Slot hoods are most commonly used to provide uniform



LOCATION

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

FIGURE 3-5. Flow rate as distance from hood

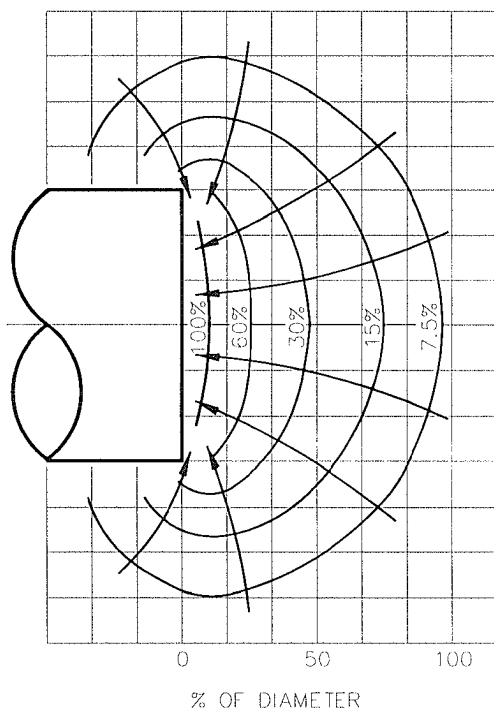


FIGURE 3-6. Velocity contours — plain circular opening — % of opening velocity

exhaust air flow and an adequate capture velocity over a finite length of contaminant generation, e.g., an open tank or over the face of a large hood such as a side-draft design. The function of the slot is solely to provide uniform air distribution. Slot velocity does not contribute toward capture velocity. A high slot velocity simply generates high pressure losses. Note that the capture velocity equation (Figure 3-11) shows that capture velocity is related to the exhaust volume and the slot length, not to the slot velocity.

Slot hoods usually consist of a narrow exhaust opening and a plenum chamber. Uniform exhaust air distribution across the slot is obtained by sizing slot width and plenum depth so that velocity through the slot is much higher than in the plenum. Splitter vanes may be used in the plenum; however, in most industrial exhaust systems, vanes are subject to corrosion and/or erosion and provide locations for material to accumulate. Adjustable slots can be provided but are subject to tampering and maladjustment. The most practical hood is the fixed slot and unobstructed plenum type. The design of the slot and plenum is such that the pressure loss through the slot is high compared with the pressure loss through the plenum. Thus, all portions of the slot are subjected to essentially equal suction and the slot velocity will be essentially uniform.

There is no straightforward method for calculating the pressure drop from one end to the other of a slot-plenum combination. A very useful approximation, applicable to most

hoods, is to design for a maximum plenum velocity equal to one-half of the slot velocity. For most slot hoods, a 2000 fpm slot velocity and 1000 fpm plenum velocity is a reasonable choice for uniformity of flow and moderate pressure drop. Centered exhaust take-off design results in the smallest practical plenum size since the air approaches the duct from both directions. Where large, deep plenums are possible, as with foundry shake-out hoods, the slot velocity may be as low as 1000 fpm with a 500 fpm plenum velocity.

3.4.5 Rectangular and Round Hoods: Air distribution for rectangular and round hoods is achieved by air flow within the hood rather than by pressure drop as for the slot hood. The plenum (length of hood from face to tapered hood to duct connection) should be as long as possible. The hood take-off should incorporate a 60° to 90° total included tapered angle. Multiple take-offs may be required for long hoods. End take-off configurations require large plenum sizes because all of the air must pass in one direction.

Figures 3-12 and 3-13 provide a number of distribution techniques.

3.4.6 Worker Position Effect: The objective of industrial ventilation is to control the worker's exposure to toxic airborne pollutants in a safe, reliable manner. As one of the main

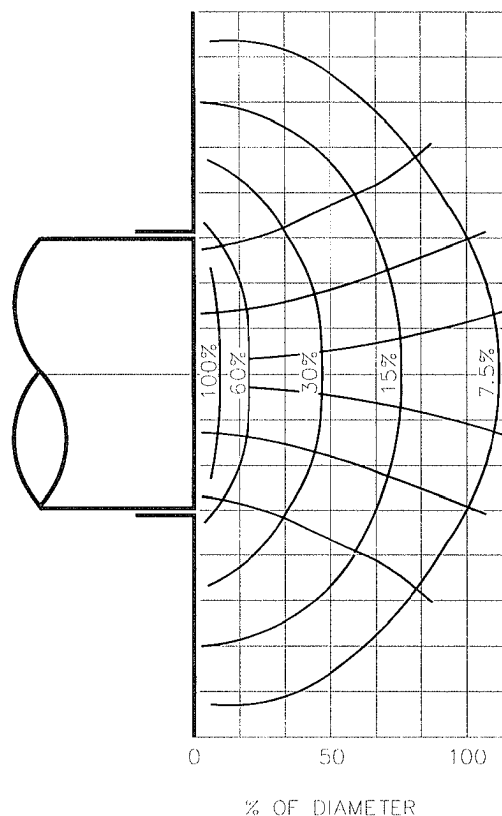
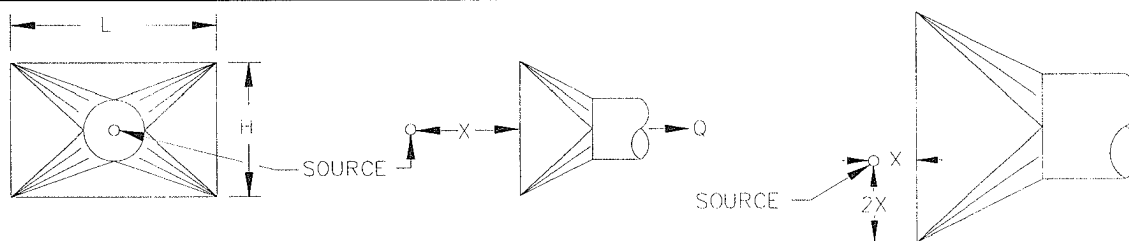


FIGURE 3-7. Velocity contours — flanged circular opening — % of opening velocity

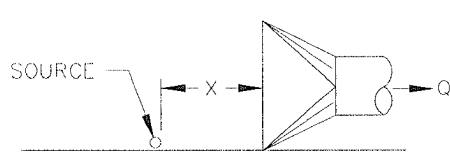


FREELY SUSPENDED HOOD

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

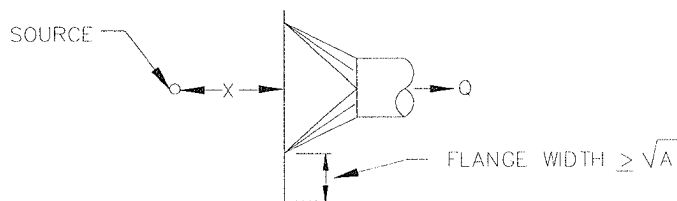
LARGE HOOD

LARGE HOOD, X SMALL--MEASURE X PERPENDICULAR TO HOOD FACE, NOT LESS THAN 2X FROM HOOD EDGE.



HOOD ON BENCH OR FLOOR

$$Q = V(5X^2 + A)$$



HOOD WITH WIDE FLANGE

$$Q = V 0.75(10X^2 + A)$$

SUSPENDED HOODS (SMALL SIDE-DRAFT HOODS)

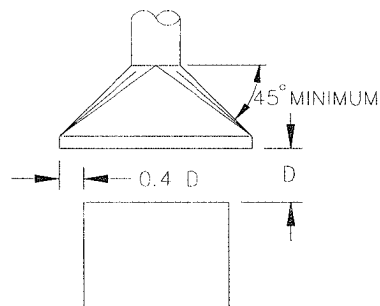
Q = REQUIRED EXHAUST AIR FLOW, CFM.

X = DISTANCE FROM HOOD FACE TO FARTHEST POINT OF CONTAMINANT RELEASE, FT.

A = HOOD FACE AREA, FT².

V = CAPTURE VELOCITY, FPM, AT DISTANCE X.

NOTE: AIR FLOW RATE MUST INCREASE AS THE SQUARE OF DISTANCE OF THE SOURCE FROM THE HOOD. BAFFLING BY FLANGING OR BY PLACING ON BENCH, FLOOR, ECT. HAS A BENEFICIAL EFFECT.



CANOPY HOOD

$Q = 1.4 PDV$ (P=PERIMETER OF TANK, FT).

NOT RECOMMENDED IF WORKERS MUST BEND OVER SOURCE. V RANGES FROM 50 TO 500 FPM DEPENDING ON CROSSDRAFTS. SIDE CURTAINS ON TWO OR THREE SIDES TO CREATE A SEMI-BOOTH OR BOOTH ARE DESIRABLE.

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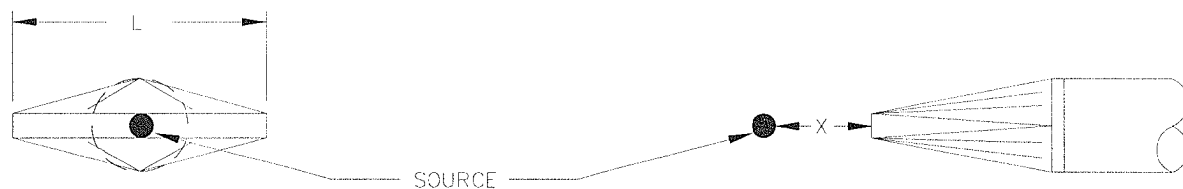
FLOW/CAPTURE VELOCITY

DATE

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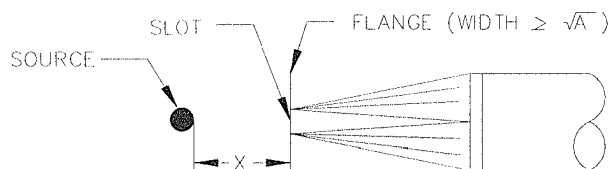
FIGURE

3-8



FREELY SUSPENDED SLOT

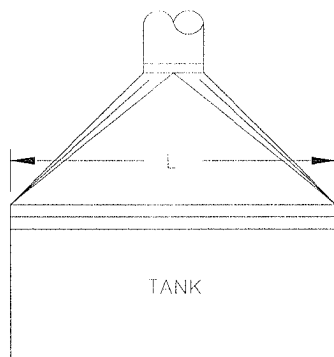
$$Q = 3.7LVX$$



FLANGED SLOT

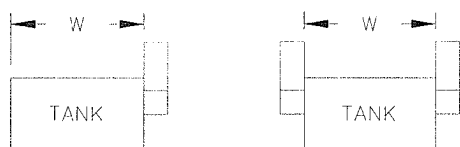
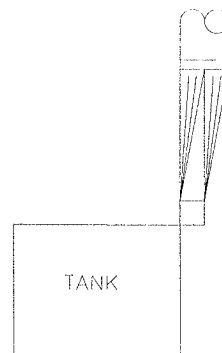
$$Q = 2.6LVX$$

Q = REQUIRED EXHAUST FLOW RATE, CFM
X = DISTANCE, HOOD FACE TO FARTHEST POINT OF SOURCE (USUALLY ON CENTERLINE OF HOOD), FT.
V = CAPTURE VELOCITY AT DISTANCE X, FPM.
L = LENGTH, OF HOOD, SLOT, TABLE, TANK, ETC., FT.
W = WIDTH, OF TABLE, TANK, ETC., FT.
A = HOOD FACE AREA, FT.



FLANGED SLOT

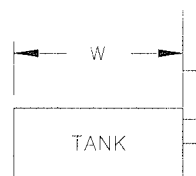
$$Q = CLW$$



SLOT ON TANK

$$Q = CLW$$

ONE-HALF Q IN EACH SLOT IF SLOTS ON BOTH SIDES



FLANGED SLOT

$$Q = CLW$$

L = LENGTH OF SLOT, FT.
W = WIDTH OF TABLE OR TANK, FT.
C = CONSTANT, VARIES FROM 50 TO 500, USUAL CHOICE IS 150 TO 250.
FLANGED SLOTS REQUIRE LOWEST EXHAUST. SEE CHAPTER 10.

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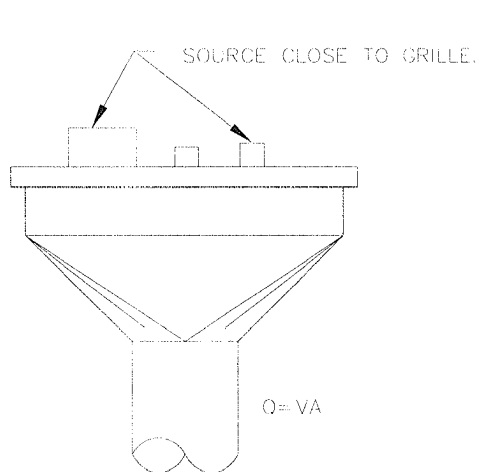
FLOW/CAPTURE VELOCITY

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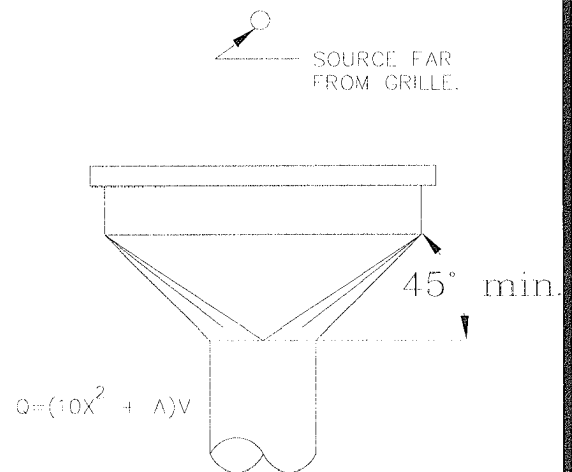
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FIGURE

3-9



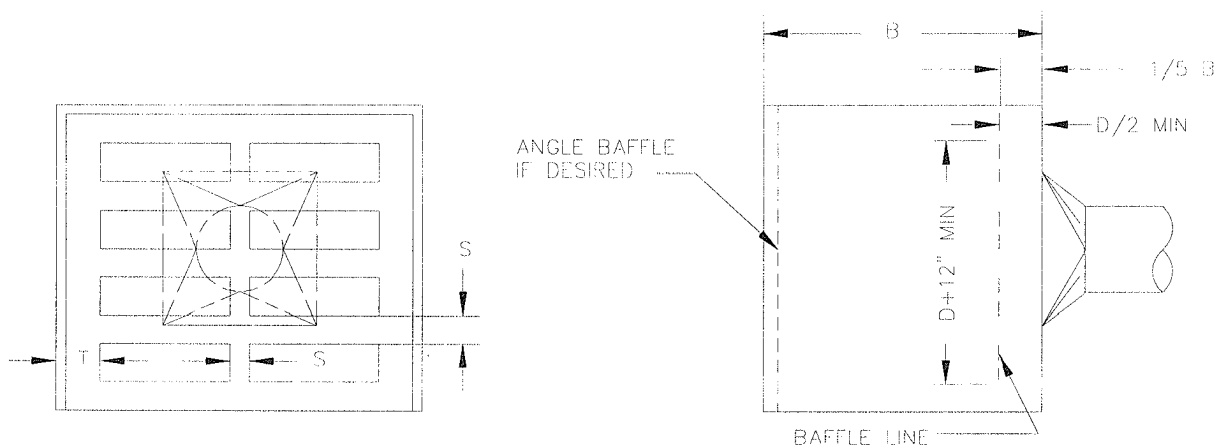
SIMILAR TO BOOTH



SIMILAR TO SUSPENDED HOOD

DOWNDRAFT HOODS

NOT RECOMMENDED FOR HOT OR HEAT-PRODUCING OPERATIONS IF DOWNDRAFT AREA IS LARGE, SEE "CAPTURE VELOCITY" IN THIS SECTION.



BOOTH-TYPE HOODS

$Q=AV$ (A=FACE AREA, FT^2 ; V=FACE VELOCITY, FPM).

BAFFLES ARE OPTIONAL FOR AIR DISTRIBUTION; NOT REQUIRED IF A WATER WALL BOOTH OR IF OTHER MEANS FOR DISTRIBUTION IS PROVIDED.

S VARIES FROM 4 INCHES TO 8 INCHES, DEPENDING ON SIZE OF BOOTH.

T VARIES FROM 6 INCHES TO 12 INCHES, DEPENDING ON SIZE OF BOOTH.

INCREASE THE NUMBER OF PANELS WITH SIZE OF BOOTH.

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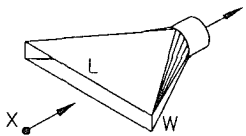
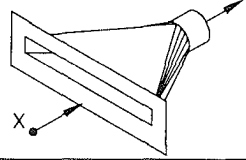
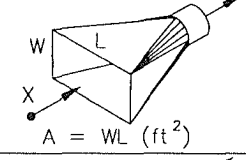
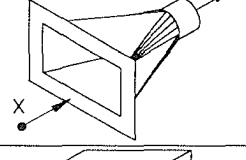
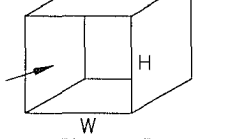
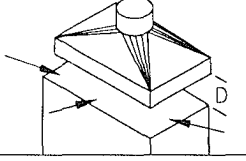
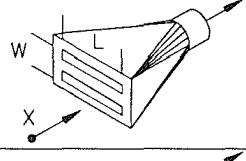
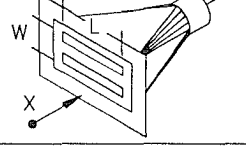
FLOW/CAPTURE VELOCITY

DATE

1-88

FIGURE

3-10

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

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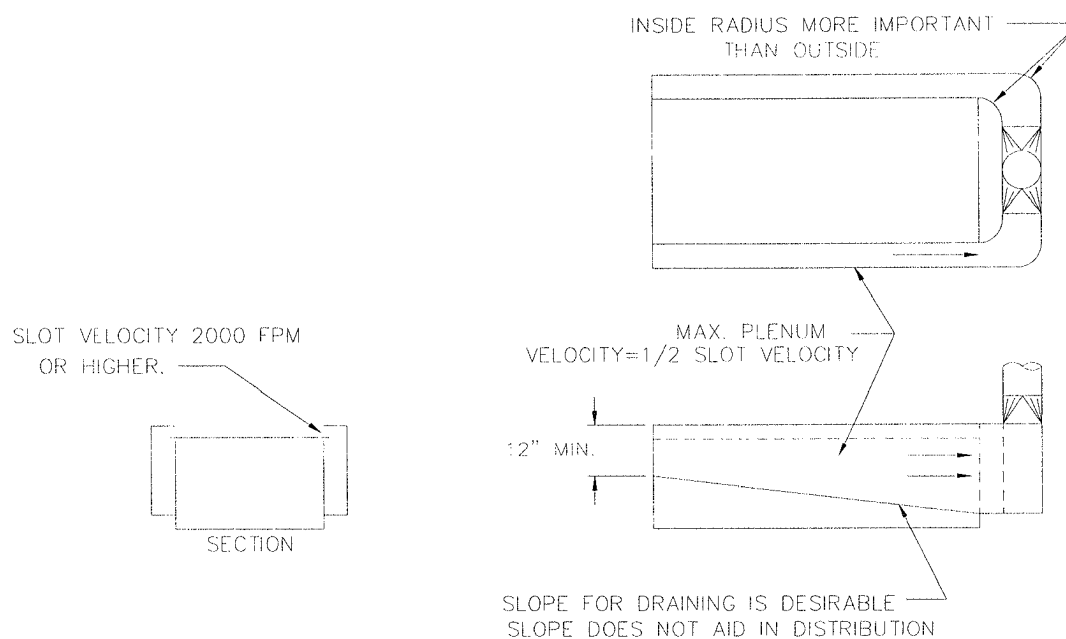
HOOD TYPES

DATE

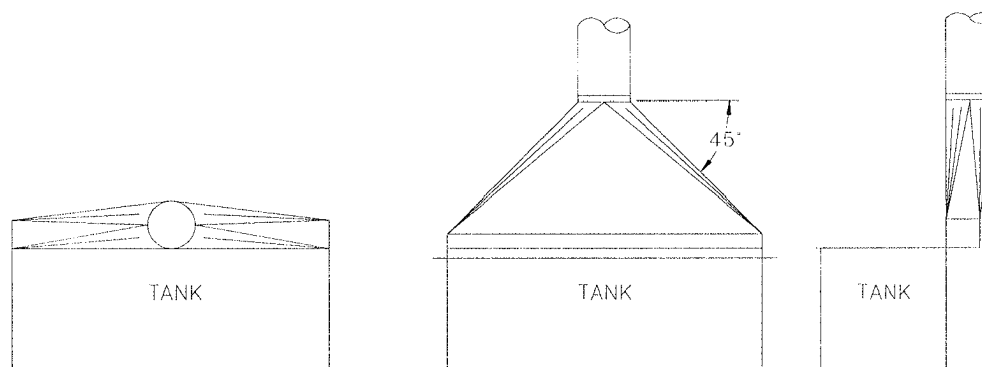
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FIGURE

3-11

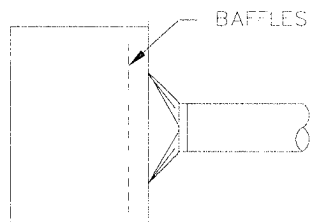
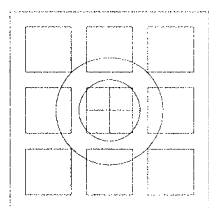


DISTRIBUTION BY SLOT RESISTANCE

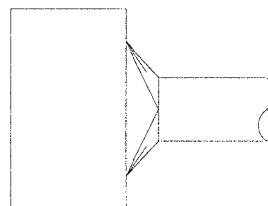
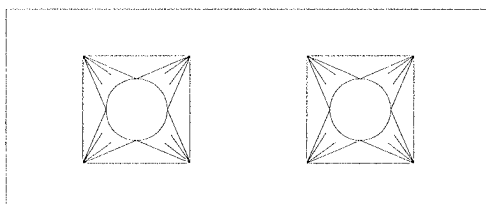


DISTRIBUTION BY FISH TAIL

WITH LOW PLENUM VELOCITIES AND HIGH SLOT VELOCITIES, GOOD DISTRIBUTION IS OBTAINED.
SLOTS OVER 10 FEET TO 12 FEET IN LENGTH USUALLY NEED MULTIPLE TAKE-OFFS.



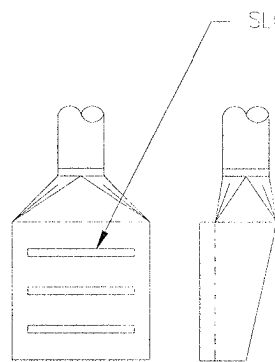
DISTRIBUTION BY BAFFLES
SEE FIG. 3-10



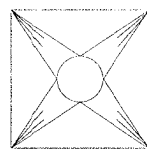
LONG BOOTHS - DISTRIBUTION BY MULTIPLE TAKE-OFFS AND TAPERS

BOOTH CANOPY

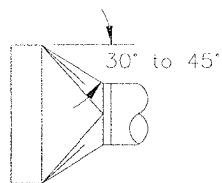
(SAME PRINCIPLES APPLY TO CANOPY TYPE)



DISTRIBUTION BY
SLOT (OR BAFFLES)



DISTRIBUTION BY TAPER



SIDE-DRAFTS AND SUSPENDED HOODS

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DISTRIBUTION TECHNIQUES

DATE

1-88

FIGURE

3-13

engineering controls, local exhaust ventilation is designed to be near the point of contaminant generation. Often, consideration is not given to how the workers will position themselves with respect to the air flow. Studies^(3,6-3,9) show that the position of the worker with respect to the flow direction is an important parameter in determining the breathing zone concentration.

Figure 3-14, Position 2, shows a worker oriented with his back to the air flow. Immediately downstream of the worker, a zone of reverse flow and turbulent mixing occurs due to boundary layer separation. Contaminant released into this region (e.g., from a hand-held or proximal source) will be mixed into the breathing zone resulting in exposure. Figure 3-14, Position 1, shows a worker oriented at 90° to the flow direction; here, the reverse flow zone forms to the side and there is less opportunity for the entrainment of contaminant into the breathing zone.

Studies suggest that this phenomenon is important when large booth-type hoods are employed or in situations where there is a reasonably uniform air flow. Exposure studies^(3,10) using a tailor's mannequin to simulate an operator in a booth-type hood used for the transfer of powders showed, in all cases, that exposures for Position 1 were less than those in Position 2 by at least a factor of 2000.

A second case study^(3,6) reported women who used a spray and brush application of a chloroform-based adhesive were significantly exposed despite working in a ventilated booth. A 50% reduction in exposure was found when the workers stood side-on to the air flow (Position 1). Subsequent modification of spray practices resulted in a determination that a 30° angle to the air flow and holding the nozzle in the downstream hand seemed optimal. No alterations to the actual design or air flow of the booth were needed to achieve acceptable exposure levels.

The preceding discussion assumes that the worker is not in the wake of an upstream object and that the contaminant source has negligible momentum. In cases where the contaminant source has significant momentum (e.g., high-pressure compressed air paint spray operations), the effect of position on exposure may be reversed — i.e., Position 1 in Figure 3-14 may produce higher exposures. This is associated with the deflection of the spray upstream of the worker and subsequent recirculation through the breathing zone. Further research and field studies are needed to evaluate the tendency for reverse flow to occur in more complex situations. Although the importance of boundary layer separation effects with smaller local exhaust hoods has not been thoroughly explored, three studies^(3,11-3,13) suggest that the 90° orientation is beneficial even in this instance. It is recommended that the side orientation (i.e., Position 1) be the preferred orientation in situations where feasible. Down-draft configurations may provide similar benefits under certain conditions.

It is recommended that the side orientation (i.e., Position

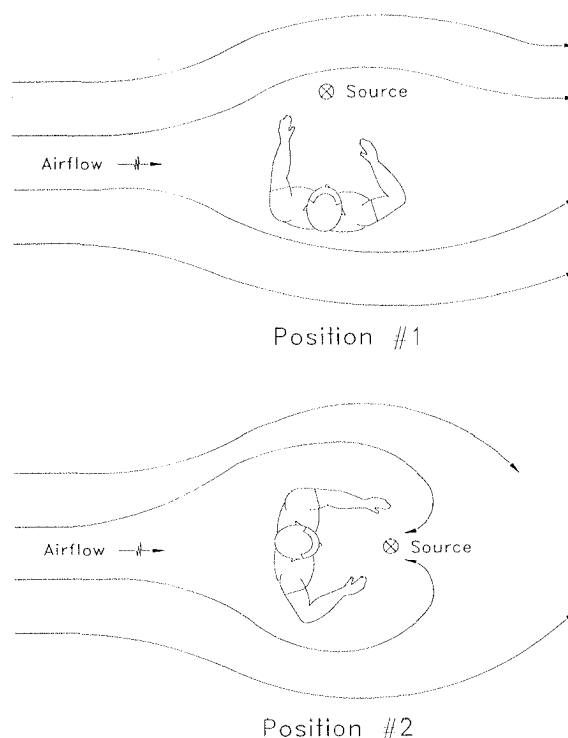


FIGURE 3-14. Worker position effect

1) be investigated as a preferred work practice where feasible. It is important to assess the exposure with personal sampling pumps to confirm the benefits of one position versus another as other factors may complicate the issue.

3.5 HOOD LOSSES

Plain duct openings, flanged duct openings, canopies, and similar hoods have only one significant energy loss. As air enters the duct, a vena contracta is formed and a small energy loss occurs first in the conversion of static pressure to velocity pressure (see Figure 3-15.) As the air passes through the vena contracta, the flow area enlarges to fill the duct and velocity pressure converts to static pressure. At this point, the uncontrolled slow down of the air from the vena contracta to the downstream duct velocity results in the major portion of the entry loss. The more pronounced the vena contracta, the greater will be the energy loss and hood static pressure.

Compound hoods are hoods which have two or more points of significant energy loss and must be considered separately and added together to arrive at the total loss for the hood. Common examples of hoods having double entry losses are slot-type hoods and multiple-opening, lateral draft hoods commonly used on plating, paint dipping and degreasing tanks, and foundry side-draft shakeout ventilation.

The hood entry loss (h_e) can be expressed, therefore, in terms of hood loss coefficients (F_f) which, when multiplied by the slot or duct velocity pressure (VP), will give the entry loss (h_e) in inches of water. The hood static pressure is equal

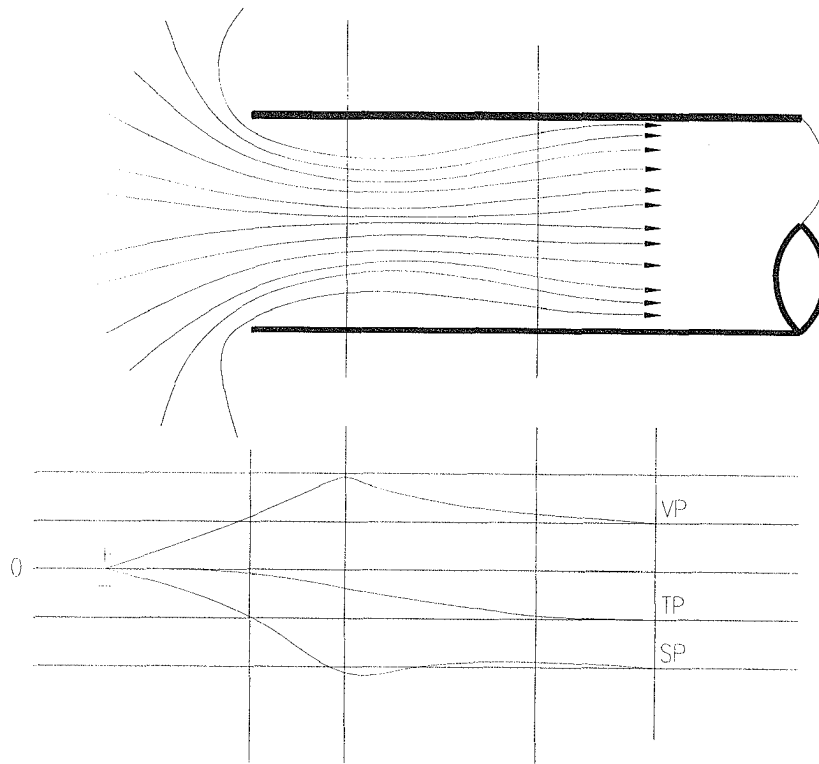


FIGURE 3-15. Air flow at the vena contracta

to the hood entry loss plus the velocity pressure in the duct. The hood entry loss represents the energy necessary to overcome the losses due to air moving through and into the duct. The velocity pressure represents the energy necessary to accelerate the air from rest to duct velocity (see Chapter 1, Section 1.6, "Acceleration of Air and Hood Entry Losses.") This may be expressed as:

$$SP_h = h_e + VP_d \quad [3.4]$$

$$SP_h = (F_s)(VP_s) + (F_d)(VP_d) + VP_d \quad [3.5]$$

where:

h_e = overall hood entry loss = $h_s + h_d$, "wg

h_s = slot or opening loss = $(F_s)(VP_s)$, "wg

h_d = duct entry loss = $(F_d)(VP_d)$, "wg

F_s = loss coefficient for slot

F_d = loss coefficient for duct entry

VP_s = slot or opening velocity pressure, "wg

VP_d = duct velocity pressure, "wg

One exception can occur when the slot velocity (or other hood entry velocity) is higher than is the duct velocity. In such case, the acceleration velocity pressure used in determining SP is the higher slot or opening velocity pressure.

Figures 3-16 and 5-13 give hood entry loss coefficients for several typical hood types.

3.5.1 Simple Hoods: A simple hood is shown in Figure 3-17. If the hood face velocity for such a simple hood is less than 1000 fpm, h_s will be negligible and the loss will be dependent on h_d only. If the hood face velocity is greater than 1000 fpm, both h_s and h_d should be considered. Face velocities greater than 1000 fpm will usually only occur with relatively small hood face areas (0.25 to 0.50 ft²).

EXAMPLE PROBLEM

$$\text{Given: Face Velocity } (V_f) = \frac{Q}{A_f} = 250 \text{ fpm}$$

$$\text{Duct Velocity } (V_d) = \frac{Q}{A_f} = 3000 \text{ fpm}$$

$$VP_d = \left(\frac{V_d}{4005} \right)^2 = 0.56 \text{ "wg}$$

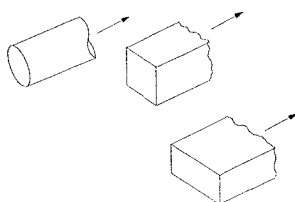
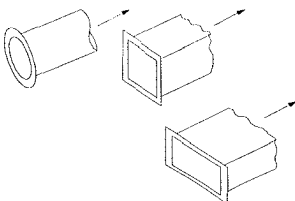
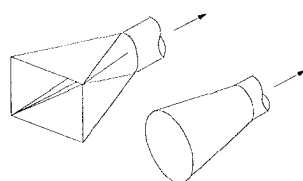
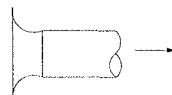
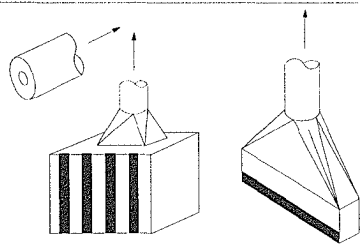
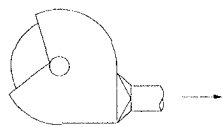
$F_d = 0.25$ as shown in Figure 5-12

$$SP_h = h_d + VP_d$$

$$= (0.25)(0.56) + 0.56$$

$$= 0.70 \text{ "wg}$$

3.5.2 Compound Hoods: Figure 3-18 illustrates a double entry loss hood. This is a single slot hood with a plenum and a transition from the plenum to the duct. The purpose of the

HOOD TYPE	DESCRIPTION	HOOD ENTRY LOSS (F_{in})
	PLAIN OPENING	0.93
	FLANGED OPENING	0.49
	TAPER OR CONE HOOD	SEE CHAPTER 10
	BELL MOUTH INLET	0.04
	ORIFICE	SEE CHAPTER 10
	TYPICAL GRINDING HOOD	(STRAIGHT TAKEOFF) 0.65
		(TAPERED TAKEOFF) 0.40

AMERICAN CONFERENCE OF GOVERNMENTAL INDUSTRIAL HYGIENISTS	<i>HOOD LOSS FACTORS</i>	
	DATE 4-96	FIGURE 3-16

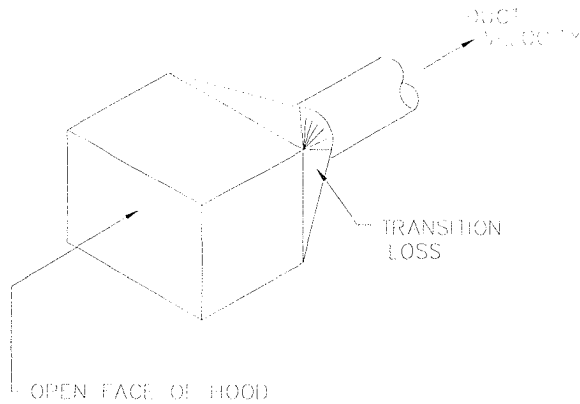


FIGURE 3-17. Simple hood

plenum is to give uniform velocity across the slot opening. Air enters the slot, in this case a sharp-edged orifice, and loses energy due to the vena contracta at this point. The air then continues through the plenum where the greater portion of the slot velocity is retained because the air stream projects itself across the plenum in a manner similar to the "blowing" supply stream shown in Figure 1-7. (The retention of velocity in the plenum is characteristic of most local exhaust hoods because of the short plenum length.) In the case of very large hoods or exhausted closed rooms, however, the velocity loss must be taken into account. Finally, the air converges into the duct through the transition where the second significant energy loss occurs. For this type hood, both h_s and h_h must be considered.

EXAMPLE PROBLEM

Given: Slot Velocity (V_s) = 2000 fpm

Duct Velocity (V_d) = 3500 fpm

(V_d is greater than V_s ; therefore, use VP_d as the acceleration VP)

$$VP_s = \left(\frac{V_s}{4005} \right)^2 = 0.25 \text{ "wg}$$

F_s for slot = 1.78 from Figure 5-12

$$VP_d = \left(\frac{V_d}{4005} \right)^2 = 0.76 \text{ "wg}$$

F_d = 0.25 as shown in Figure 5-13

$$\begin{aligned} SP_h &= h_s + h_d + VP_d \\ &= (1.78)(0.25) + (0.25)(0.76) + 0.76 \\ &= 1.40 \text{ "wg} \end{aligned}$$

3.6 MINIMUM DUCT VELOCITY

The velocity pressure, VP_d , utilized to determine hood losses in the previous examples is determined from the air velocity in the duct immediately downstream of the hood to duct connection. This velocity is determined by the type of

material being transported in the duct.

For systems handling particulate, a minimum design velocity is required to prevent settling and plugging of the duct. On the other hand, excessively high velocities are wasteful of power and may cause rapid abrasion of ducts.^(3,14-3,21) Minimum recommended design velocities are higher than theoretical and experimental values to protect against practical contingencies such as:

1. Plugging or closing one or more branch will reduce the total flow rate in the system and correspondingly will reduce the velocities in at least some sections of the duct system.
2. Damage to ducts, by denting for example, will increase the resistance and decrease the flow rate and velocity in the damaged portion of the system.
3. Leakage of ducts will increase flow rate and velocity downstream of the leak but will decrease air flow upstream and in other parts of the system.
4. Corrosion or erosion of the fan wheel or slipping of a fan drive belt will reduce flow rates and velocities.
5. Velocities must be adequate to pick up or re-entrain dust which may have settled due to improper operation of the exhaust system.

The designer is cautioned that for some conditions such as sticky materials, condensing conditions in the presence of dust, strong electrostatic effects, etc., velocity alone may not be sufficient to prevent plugging, and other special measures may be necessary.

Some typical duct velocities are provided in Table 3-2. The use of minimum duct velocity is treated in detail in Chapter 5.

3.7 SPECIAL HOOD REQUIREMENTS

3.7.1 Ventilation of Radioactive and High Toxicity

Processes: Ventilation of radioactive and high toxicity processes requires a knowledge of the hazards, the use of extraordinarily effective control methods, and adequate maintenance which includes monitoring. Only the basic principles

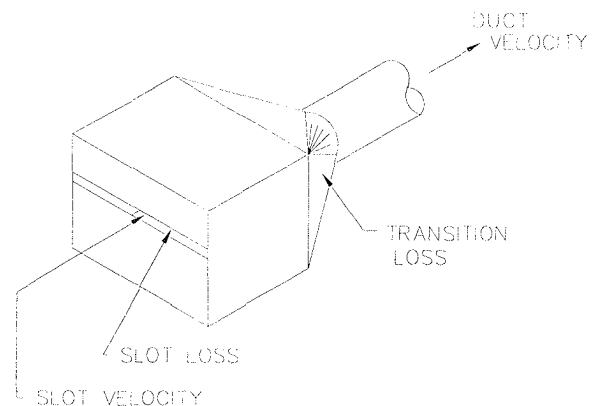


FIGURE 3-18. Compound hood

TABLE 3-2. Range of Minimum Duct Design Velocities

Nature of Contaminant	Examples	Design Velocity
Vapors, gases, smoke	All vapors, gases, and smoke	Any desired velocity (economic optimum velocity usually 1000–2000 fpm)
Fumes	Welding	2000–2500
Very fine light dust	Cotton lint, wood flour, litho powder	2500–3000
Dry dusts & powders	Fine rubber dust, Bakelite molding powder dust, jute lint, cotton dust, shavings (light), soap dust, leather shavings	3000–4000
Average industrial dust	Grinding dust, buffing lint (dry), wool jute dust (shaker waste), coffee beans, shoe dust, granite dust, silica flour, general material handling, brick cutting, clay dust, foundry (general), limestone dust, packaging and weighing asbestos dust in textile industries	3500–4000
Heavy dusts	Sawdust (heavy and wet), metal turnings, foundry tumbling barrels and shake-out, sand blast dust, wood blocks, hog waste, brass turnings, cast iron boring dust, lead dust	4000–4500
Heavy or moist	Lead dusts with small chips, moist cement dust, asbestos chunks from transite pipe cutting machines, buffing lint (sticky), quick-lime dust	4500 and up

can be covered here. For radioactive processes, reference should be made to the standards and regulations of the nuclear regulatory agencies.

Local exhaust hoods should be of the enclosing type with the maximum enclosure possible. Where complete or nearly complete enclosure is not possible, control velocities from 50 to 100% higher than the minimum standards in this manual should be used. If the enclosure is not complete and an operator must be located at an opening, such as in front of a laboratory hood, the maximum control velocity should not exceed 125 fpm. Air velocities higher than this value will create eddies in front of the operator which may pull contaminant from the hood into the operator's breathing zone. Replacement air should be introduced at low velocity and in a direction that does not cause disruptive cross drafts at the hood opening.

3.7.2 Laboratory Operations: Glove boxes should be used for high-activity alpha or beta emitters as well as highly toxic and biological materials. The air locks used with the glove box should be exhausted if they open directly to the room.

For low-activity radioactive laboratory work, a laboratory fume hood may be acceptable. For such hoods, an average face velocity of 80–100 fpm is recommended. See Section 10.35, VS-35-01, -02, -04, and -20.

For new buildings, it is frequently necessary to estimate the air conditioning early — before the detailed design and equipment specifications are available. For early estimating, the guidelines provided in Section 10.35 for hood air flow and replacement air flow can be used.

3.8 PUSH-PULL VENTILATION

Push-pull ventilation consists of a push nozzle and an exhaust hood to receive and remove the push jet. Push-pull is used most commonly on open surface vessels such as plating tanks^(3.22) but may be effectively used elsewhere (see VS-70-10). The advantage of push-pull is that a push jet will maintain velocity over large distances, 20–30 ft or more, whereas the velocity in front of an exhaust hood decays very rapidly as the distance from the hood increases. Properly used, the push jet intercepts contaminated air and carries it relatively long distances into the exhaust hood, thus providing control where it may be otherwise difficult or impossible.

3.8.1 Push Jet: Ambient air is entrained in the push jet and results in a jet flow at the exhaust hood several times greater than the push nozzle flow rate. The jet velocity will decay with distance from the nozzle. The entrainment ratio for a long thin slot-(or pipe) type nozzle may be approximated by:^(3.23)

$$\frac{Q_x}{Q_o} = 12 \sqrt{\left(\frac{ax}{b_o}\right) + 0.41} \quad [3.6]$$

The velocity ratio may be approximated by:^(3.23)

$$\frac{V_x}{V_o} = \frac{12}{\sqrt{\frac{ax}{b_o} + 0.41}} \quad [3.7]$$

where:

Q_o = the push nozzle supply flow

Q_x = the jet flow rate at a distance x from the nozzle

V_o = the push nozzle exit air velocity

V_x = the peak push jet velocity at a distance x

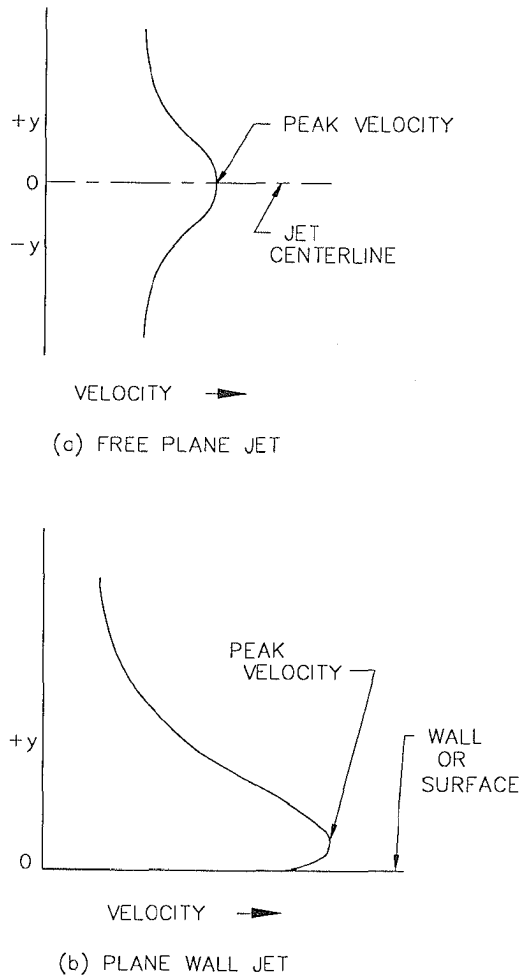


FIGURE 3-19. Jet velocity profile

a = a coefficient characteristic of the nozzle (0.13 for slots and pipes)

x = distance from the nozzle

b_o = the slot width*

[*If the nozzle is freely suspended (free plane jet), b_o is equal to one-half the total slot width. If the nozzle is positioned on or very near a plane surface (wall jet), b_o is equal to the full slot width. For pipes with holes, b_o is the width of a slot with equivalent area.]

Typical jet velocity profiles are shown in Figure 3-19.

Obstructions in the jet path should be minimized near the jet. Objects with small cross-sections, such as parts hangers, will cause serious problems; however, large flat surface objects should be avoided. At further distances from the nozzle where the jet has expanded, larger objects may be acceptable if they are located within the jet.

The nozzle may be constructed as a long thin slot, a pipe with holes or individual nozzles. The total nozzle exit area should not exceed 50% of the nozzle plenum cross-sectional

area to assure even flow distribution. Slot width can range from 0.125–0.25 inch for short push length such as plating tanks (4–8 ft). Hole size should be 0.25 inch on 3 to 8 diameter spacing. The nozzle momentum factor, which is proportional to nozzle exit flow per foot of nozzle length times nozzle exit velocity ($Q_o \times V_o$), must be sufficient to result in an effective jet but not so strong that the exhaust hood is overpowered. A $Q_o V_o$ range should be approximately 50,000–75,000 per foot of nozzle length for short distances of 4–8 feet.

3.8.2 Pull Hood: The pull hood will accept and remove the push jet flow. The same design considerations regarding flow distribution, hood entry losses, etc., used for a normal pull-only hood should be used. The hood pull flow should be approximately 1.5–2.0 times the push flow which reaches the hood. If design criteria specifying pull flow rate are not available, Equation 3.6 can be used.

The hood opening height should be the same as the width of the expanded jet, if possible. However, smaller opening heights are acceptable if the hood flow rate meets the 1.5–2.0 times jet flow criteria.

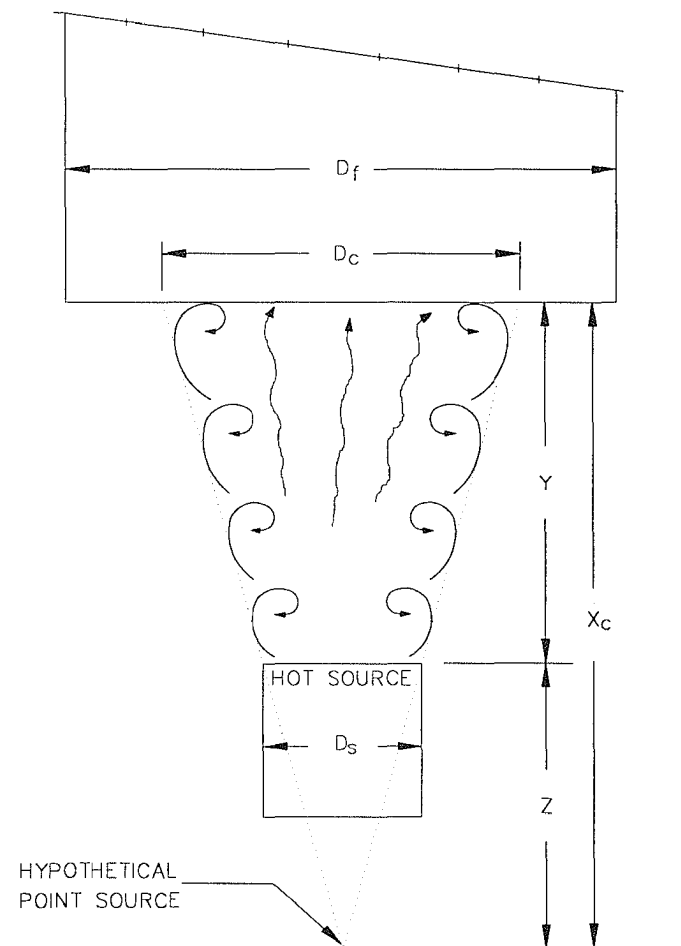


FIGURE 3-20. Dimensions used to design high-canopy hoods for hot sources (Ref. 3.24)

Each push-pull application will necessitate special attention. Wherever possible, a pilot system should be evaluated prior to final installation.

3.8.3 Push-Pull System Design: Specific design criteria have been developed experimentally for plating, cleaning, or other open surface vessels and are provided in VS-70-10, VS-70-11, and VS-70-12. Where such specific design criteria are not available, the criteria provided in Sections 3.8.1 and 3.8.2 can be used. When designing with Equation 3.7, a push jet velocity (V_x) of 150–200 fpm at the exhaust hood face should be specified.

3.9 HOT PROCESSES

Design of hooding for hot processes requires different considerations than design for cold processes.^(3,24) When significant quantities of heat are transferred to the air above and around the process by conduction and convection, a thermal draft is created which causes an upward air current with air velocities as high as 400 fpm. The design of the hood and exhaust rate must take this thermal draft into consideration.

3.9.1 Circular High Canopy Hoods: As the heated air rises, it mixes turbulently with the surrounding air. This results in an increasing air column diameter and volumetric flow rate. The diameter of the column (see Figure 3-20) can be approximated by:

$$D_c = 0.5X_c^{0.88} \quad [3.8]$$

where:

D_c = column diameter at hood face

X_c = $y + z$ = the distance from the hypothetical point source to the hood face, ft

y = distance from the process surface to the hood face, ft

z = distance from the process surface to the hypothetical point source, ft

" z " can be calculated from:

$$z = (2 \Delta_s)^{1.138} \quad [3.9]$$

where:

Δ_s = diameter of hot source, ft.

The velocity of the rising hot air column can be calculated from:

$$V_f = 8(A_s)^{0.33} \frac{(\Delta t)^{0.42}}{X_c^{0.25}} \quad [3.10]$$

where:

V_f = velocity of hot air column at the hood face, fpm

A_s = area of the hot source, ft²

Δt = the temperature difference between the hot

source and the ambient air, F

X_c = $y + z$ = the distance from the hypothetical point source to the hood face, ft.

The diameter of the hood face must be larger than the diameter of the rising hot air column to assure complete capture. The hood diameter is calculated from:

$$D_f = D_c = 0.8y \quad [3.11]$$

where:

D_f = diameter of the hood face, ft

Total hood air flow rate is

$$Q_t = V_f A_c + V_r (A_f - A_c) \quad [3.12]$$

where:

Q_t = total volume entering hood, cfm

V_f = velocity of hot air column at the hood face, fpm

A_c = area of the hot air column at the hood face, ft²

V_r = the required air velocity through the remaining hood area, fpm

A_f = total area of hood face, ft²

EXAMPLE PROBLEM

Given: 4.0 ft diameter melting pot (D_a)

1000 F metal temperature

100 F ambient temperature

Circular canopy hood located 10 ft above pot (y)

Calculate x_c :

$$x_c = y + z = y + (2D_s)^{1.138}$$

$$x_c = 10 + (2 \times 4)^{1.138}$$

$$x_c = 10.7 \text{ ft}$$

Calculate the diameter of the hot air column at the hood face:

$$D_c = 0.5 x_c^{0.88}$$

$$D_c = 0.5(20.7)^{0.88}$$

$$D_c = 7.2 \text{ ft}$$

Calculate the velocity of the hot air column at the hood face:

$$V_f = 8(A_s)^{0.33} \frac{(\Delta t)^{0.42}}{(x)^{0.25}}$$

$$A_s = 0.25\pi D_c^2$$

$$A_s = 0.25\pi(4.2)^2$$

$$A_s = 12.6 \text{ ft}^2$$

$$\Delta t = 1000 - 100 = 900 \text{ F}$$

$$V_f = 8(1.26)^{0.33} \frac{(900)^{0.42}}{(20.7)^{0.25}}$$

$$V_f = (8)(2.31) \frac{(17.4)}{(2.13)}$$

$$V_f = 151 \text{ fpm}$$

Calculate diameter of hood face:

$$D_f = D_c + 0.8y$$

$$D_f = 7.2 = 0.8^{(10)}$$

$$D_f = 15.2 \text{ ft}$$

Calculate total hood airflow rate

$$Q_f = V_f A_c + V_r (A_f - A_c)$$

$$A_c = 0.25\pi D_c^2$$

$$A_c = 0.25\pi(7.2)^2$$

$$A_c = 41 \text{ ft}^2$$

$$A_f = 0.25\pi D_f^2$$

$$A_f = 0.25\pi(15.2)^2$$

$$A_f = 181 \text{ ft}^2$$

$$Q_f = 151(41) + 100(181 - 41)$$

$$Q_f = 10,290 \text{ cfm}$$

3.9.2 Rectangular High Canopy Hoods: Hot air columns from sources which are not circular may be better controlled by a rectangular canopy hood. Hood air flow calculations are performed in the same manner as for circular hoods except the dimensions of the hot air column at the hood (and the hood dimensions) are determined by considering both the length and width of the source. Equations 3.8, 3.9, and 3.11 are used individually to determine length and width of the hot air column and the hood. The remaining values are calculated in the same manner as for the circular hood.

EXAMPLE PROBLEM

Given: 2.5 ft x 4 ft rectangular melting furnace

700 F metal temperature

80 F ambient temperature

Rectangular canopy hood located 8 ft above furnace (y)

Calculate X_c for each furnace dimension.

$$\begin{aligned} X_{c2.5} &= y + Z_{2.5} = y + (2D_{s2.5})^{1.138} \\ &= 8 + (2 \times 2.5)^{1.138} \\ &= 14.2 \text{ ft} \end{aligned}$$

$$\begin{aligned} X_{c4} &= 8 + (2 \times 4)^{1.138} \\ &= 18.7 \text{ ft} \end{aligned}$$

Calculate the width of the hot air column at the hood face.

$$\begin{aligned} D_{c2.5} &= 0.5 X_{c2.5}^{0.88} \\ &= 0.5(14.2)^{0.88} \\ &= 5.2 \text{ ft} \end{aligned}$$

$$\begin{aligned} D_{c4.0} &= 0.5(18.7)^{0.88} \\ &= 6.6 \text{ ft} \end{aligned}$$

Calculate the velocity of the hot air column at the hood face.

$$V_f = 8(A_s)^{0.33} \frac{(\Delta t)^{0.42}}{(X_c)^{0.25}}$$

$$A_s = 2.5 \times 4 = 10 \text{ ft}^2$$

$$\Delta t = 700 - 80 = 620 \text{ F}$$

$$X_c = x_{c2.5} = 14.2 \text{ ft}$$

Note: $X_{c2.5}$ is used rather than $x_{c4.0}$ as it is smaller and as such will yield a slightly larger V_f which results in a margin of safety.

$$\begin{aligned} V &= 8(10)^{0.33} \frac{(620)^{0.42}}{(14.2)^{0.25}} \\ &= 8(2.1) \frac{(14.9)}{1.9} \end{aligned}$$

$$= 132 \text{ fpm}$$

Calculate hood face dimensions.

$$\begin{aligned} \text{Hood width} &= D_{c2.5} + 0.8y \\ &= (5.2) + 0.8(8) \\ &= 11.6 \text{ ft} \end{aligned}$$

$$\begin{aligned} \text{Hood length} &= D_{c4.0} + 0.8y \\ &= 6.6 + 0.8(8) \\ &= 13.0 \text{ ft} \end{aligned}$$

Calculate the total hood air flow rate.

$$Q_f = V_f A_c + V_r (A_f - A_c)$$

$$\begin{aligned} A_c &= (D_{c2.5})(D_{c4.0}) \\ &= (5.2)(6.6) \\ &= 34 \text{ ft}^2 \end{aligned}$$

$$\begin{aligned} A_f &= (\text{hood length})(\text{hood width}) \\ &= (11.6)(13.0) \\ &= 151 \text{ ft}^2 \end{aligned}$$

$$\begin{aligned}
 Q_f &= (151)(34) + 100(151 - 34) \\
 &= 5134 + 11,700 \\
 &= 16,834 \text{ cfm}
 \end{aligned}$$

3.9.3 Low Canopy Hoods: If the distance between the hood and the hot source does not exceed approximately the diameter of the source or 3 ft, whichever is smaller, the hood may be considered a low canopy hood. Under such conditions, the diameter or cross-section of the hot air column will be approximately the same as the source. The diameter or side dimensions of the hood therefore need only be 1 ft larger than the source.

The total flow rate for a circular low canopy hood is

$$Q_t = 4.7 (D_f)^{2.33} (\Delta t)^{0.41} \quad [3.13]$$

where:

- Q_t = total hood air flow, cfm
- D_f = diameter of hood, ft
- Δt = difference between temperature of the hot source, and the ambient, F.

The total flow rate for a rectangular low hood is

$$\frac{Q_t}{L} = 6.2b^{1.33} \Delta t^{0.42}$$

where:

- Q_t = total hood air flow, cfm
- L = length of the rectangular hood, ft
- b = width of the rectangular hood, ft
- Δt = difference between temperature of the hot source and the ambient, F.

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Chapter 4

AIR CLEANING DEVICES

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4.1 INTRODUCTION

Air cleaning devices remove contaminants from an air or gas stream. They are available in a wide range of designs to meet variations in air cleaning requirements. Degree of removal required, quantity and characteristics of the contaminant to be removed, and conditions of the air or gas stream will all have a bearing on the device selected for any given application. In addition, fire safety and explosion control must be considered in all selections. (See NFPA publications.)

For particulate contaminants, air cleaning devices are divided into two basic groups: AIR FILTERS and DUST COLLECTORS. Air filters are designed to remove low dust concentrations of the magnitude found in atmospheric air. They are typically used in ventilation, air-conditioning, and heating systems where dust concentrations seldom exceed 1.0 grains per thousand cubic feet of air and are usually well below 0.1 grains per thousand cubic feet of air. (One pound equals 7000 grains. A typical atmospheric dust concentration in an urban area is 87 micrograms per cubic meter or 0.038 grains per thousand cubic feet of air.)

Dust collectors are usually designed for the much heavier loads from industrial processes where the air or gas to be cleaned originates in local exhaust systems or process stack gas effluents. Contaminant concentrations will vary from less than 0.1 to 100 grains or more for each cubic foot of air or gas. Therefore, dust collectors are, and must be, capable of handling concentrations 100 to 20,000 times greater than those for which air filters are designed.

Small, inexpensive versions of all categories of air cleaning devices are available. The principles of selection, application, and operation are the same as for larger equipment. However, due to the structure of the market that focuses on small, quickly available, and inexpensive equipment, much of the available equipment is of light duty design and construction. One of the major economies of unit collectors implies recirculation, for which such equipment may or may not be suitable. For adequate prevention of health hazards, fires, and explosions, application engineering is just as essential for unit collectors as it is for major systems.

4.2 SELECTION OF DUST COLLECTION EQUIPMENT

Dust collection equipment is available in numerous designs utilizing many different principles and featuring wide variations in effectiveness, first cost, operating and maintenance cost, space, arrangement, and materials of construction. Consultation with the equipment manufacturer is the recommended procedure in selecting a collector for any problem where extensive previous plant experience on the specific dust problem is not available. Factors influencing equipment selection include the following:

4.2.1 Contaminant Characteristics: Contaminants in exhaust systems cover an extreme range in concentration and

particle size. Concentrations can range from less than 0.1 to much more than 100,000 grains of dust per cubic foot of air. In low pressure conveying systems, the dust ranges from 0.5 to 100 or more microns in size. Deviation from mean size (the range over and under the mean) will also vary with the material.

4.2.2 Efficiency Required: Currently, there is no accepted standard for testing and/or expressing the "efficiency" of a dust collector. It is virtually impossible to accurately compare the performance of two collectors by comparing efficiency claims. The only true measure of performance is the actual mass emission rate, expressed in terms such as mg/m³ or grains/ft³. Evaluation will consider the need for high efficiency—high cost equipment requiring minimum energy such as high voltage electrostatic precipitators, high efficiency—moderate cost equipment such as fabric or wet collectors, or the lower cost primary units such as the dry centrifugal group. If either of the first two groups is selected, the combination with primary collectors should be considered.

When the cleaned air is to be discharged outdoors, the required degree of collection can depend on plant location; nature of contaminant (its salvage value and its potential as a health hazard, public nuisance, or ability to damage property); and the regulations of governmental agencies. In remote locations, damage to farms or contribution to air pollution problems of distant cities can influence the need for and importance of effective collection equipment. Many industries, originally located away from residential areas, failed to anticipate the residential building construction which frequently develops around a plant. Such lack of foresight has required installation of air cleaning equipment at greater expense than initially would have been necessary. Today, the remotely located plant must comply, in most cases, with the same regulations as the plant located in an urban area. With the present emphasis on public nuisance, public health, and preservation and improvement of community air quality, management can continue to expect criticism for excessive emissions of air contaminants whether located in a heavy industry section of a city or in an area closer to residential zones.

The mass rate of emission will also influence equipment selection. For a given concentration, the larger the exhaust volumetric flow rate, the greater the need for better equipment. Large central steam-generating stations might select high efficiency electrostatic precipitators or fabric collectors for their pulverized coal boiler stacks while a smaller industrial pulverized fuel boiler might be able to use slightly less efficient collectors.

A safe recommendation in equipment selection is to select the collector that will allow the least possible amount of contaminant to escape and is reasonable in first cost and maintenance while meeting all prevailing air pollution regulations. For some applications even the question of reasonable

cost and maintenance must be sacrificed to meet established standards for air pollution control or to prevent damage to health or property.

It must be remembered that visibility of an effluent will be a function of the light reflecting surface area of the escaping material. Surface area per pound increases inversely as the square of particle size. This means that the removal of 80% or more of the dust on a weight basis may remove only the coarse particles without altering the stack appearance.

4.2.3 Gas Stream Characteristics: The characteristics of the carrier gas stream can have a marked bearing on equipment selection. Temperature of the gas stream may limit the material choices in fabric collectors. Condensation of water vapor will cause packing and plugging of air or dust passages in dry collectors. Corrosive chemicals can attack fabric or metal in dry collectors and when mixed with water in wet collectors can cause extreme damage.

4.2.4 Contaminant Characteristics: The contaminant characteristics will also affect equipment selection. Chemicals emitted may attack collector elements or corrode wet type collectors. Sticky materials, such as metallic buffing dust impregnated with buffing compounds, can adhere to collector elements, plugging collector passages. Linty materials will adhere to certain types of collector surfaces or elements. Abrasive materials in moderate to heavy concentrations will cause rapid wear on dry metal surfaces. Particle size, shape, and density will rule out certain designs. For example, the parachute shape of particles like the "bees wings" from grain will float through centrifugal collectors because their velocity of fall is less than the velocity of much smaller particles having the same specific gravity but a spherical shape. The combustible nature of many finely divided materials will require specific collector designs to assure safe operation.

4.2.5 Energy Considerations: The cost and availability of energy makes essential the careful consideration of the total energy requirement for each collector type which can achieve the desired performance. An electrostatic precipitator, for example, might be a better selection at a significant initial cost penalty because of the energy savings through its inherently lower pressure drop.

4.2.6 Dust Disposal: Methods of removal and disposal of collected materials will vary with the material, plant process, quantity involved, and collector design. Dry collectors can be unloaded continuously or in batches through dump gates, trickle valves, and rotary locks to conveyors or containers. Dry materials can create a secondary dust problem if careful thought is not given to dust-free material disposal or to collector dust bin locations suited to convenient material removal. See Figures 4-1, 4-2, and 4-3 for some typical discharge arrangements and valves.

Wet collectors can be arranged for batch removal or con-

tinual ejection of dewatered material. Secondary dust problems are eliminated, although disposal of wet sludge can be a material handling problem. Solids carry-over in waste water can create a sewer or stream pollution problem if waste water is not properly clarified.

Material characteristics can influence disposal problems. Packing and bridging of dry materials in dust hoppers, floating or slurry forming characteristics in wet collectors are examples of problems that can be encountered.

4.3 DUST COLLECTOR TYPES

The four major types of dust collectors for particulate contaminants are electrostatic precipitators, fabric collectors, wet collectors, and dry centrifugal collectors.

4.3.1 Electrostatic Precipitators: In electrostatic precipitation, a high potential electric field is established between discharge and collecting electrodes of opposite electrical charge. The discharge electrode is of small cross-sectional area, such as a wire or a piece of flat stock, and the collection electrode is large in surface area such as a plate.

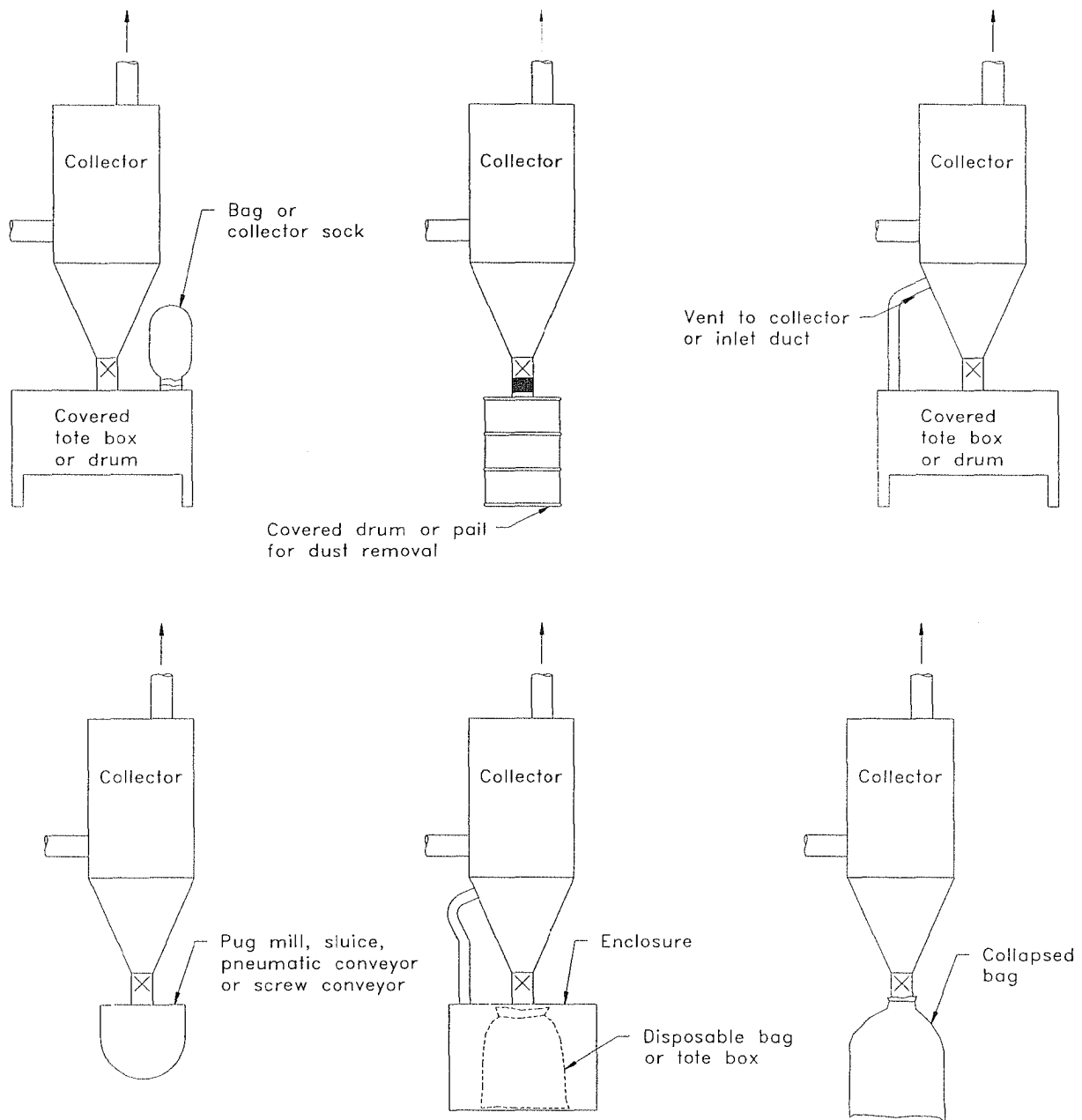
The gas to be cleaned passes through an electrical field that develops between the electrodes. At a critical voltage, the gas molecules are separated into positive and negative ions. This is called "ionization" and takes place at, or near, the surface of the discharge electrode. Ions having the same polarity as the discharge electrode attach themselves to neutral particles in the gas stream as they flow through the precipitator. These charged particles are then attracted to a collecting plate of opposite polarity. Upon contact with the collecting surface, dust particles lose their charge and then can be easily removed by washing, vibration, or gravity.

The electrostatic process consists of:

1. Ionizing the gas.
2. Charging the dust particles.
3. Transporting the particles to the collecting surface.
4. Neutralizing, or removing the charge from the dust particles.
5. Removing the dust from the collecting surface.

The two basic types of electrostatic precipitators are "Cottrell," or single-stage, and "Penny," or two-stage (see Figures 4-4 and 4-5).

The "Cottrell," single-stage, precipitator (Figure 4-4) combines ionization and collection in a single stage. Because it operates at ionization voltages from 40,000 to 70,000 volts DC, it may also be called a high voltage precipitator and is used extensively for heavy-duty applications such as utility boilers, larger industrial boilers, and cement kilns. Some precipitator designs use sophisticated voltage control systems and rigid electrodes instead of wires to minimize maintenance problems.

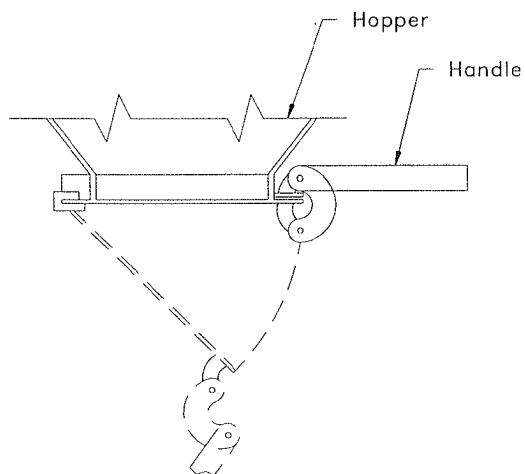


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*DRY TYPE DUST COLLECTORS
DUST DISPOSAL*

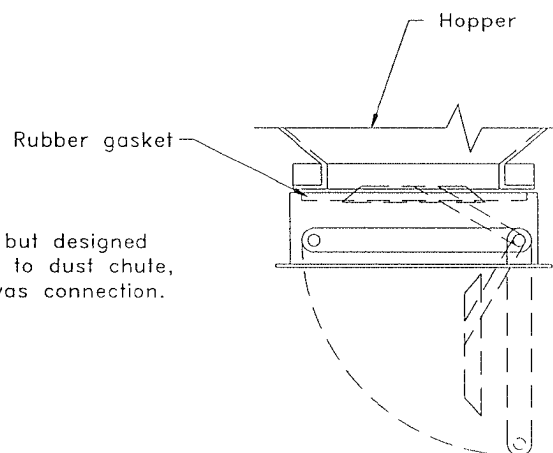
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FIGURE 4-1



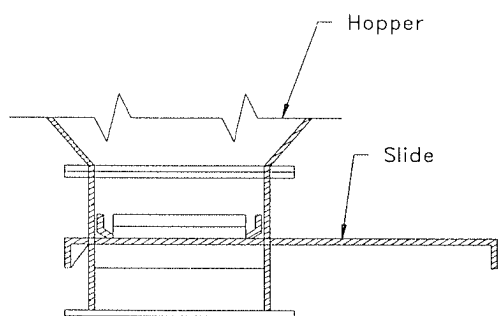
DUST DOOR

For intermittent manual dumping
where dust loads are light.



DUST GATE

Similar to dust door but designed
for direct attachment to dust chute,
external pipe or canvas connection.



SLIDE GATE

For intermittent, manual dumping where
dust loads are light. Flange for connection
to dust disposal chute.

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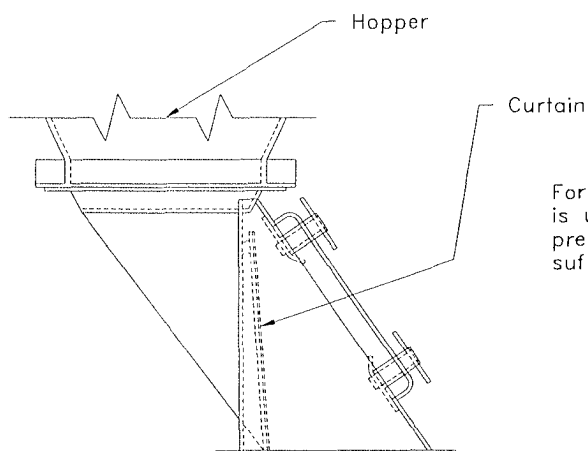
*DRY TYPE DUST COLLECTORS
DISCHARGE VALVES*

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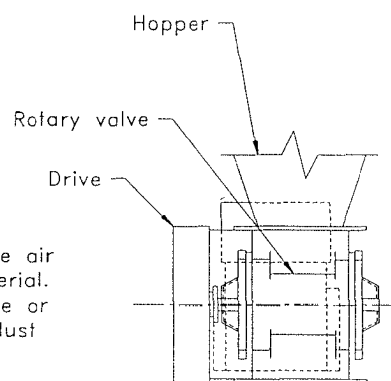
FIGURE

4-2



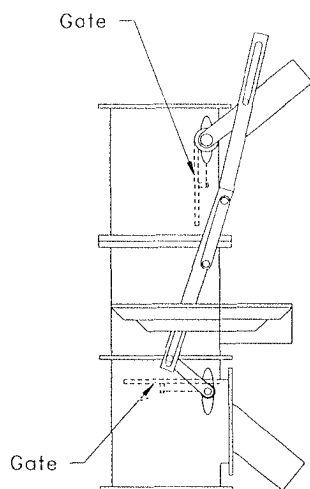
TRICKLE VALVE

For continuous removal of collected dust where hopper is under negative pressure. Curtain is kept closed by pressure differential until collected material builds up sufficient height to overcome pressure.



ROTARY LOCK

Motor driven multiple blade rotary valve provide air lock while continuously dumping collected material. Can be used with hoppers under either positive or negative pressure. Flanged for connection to dust disposal chute.



DOUBLE DUMP VALVE

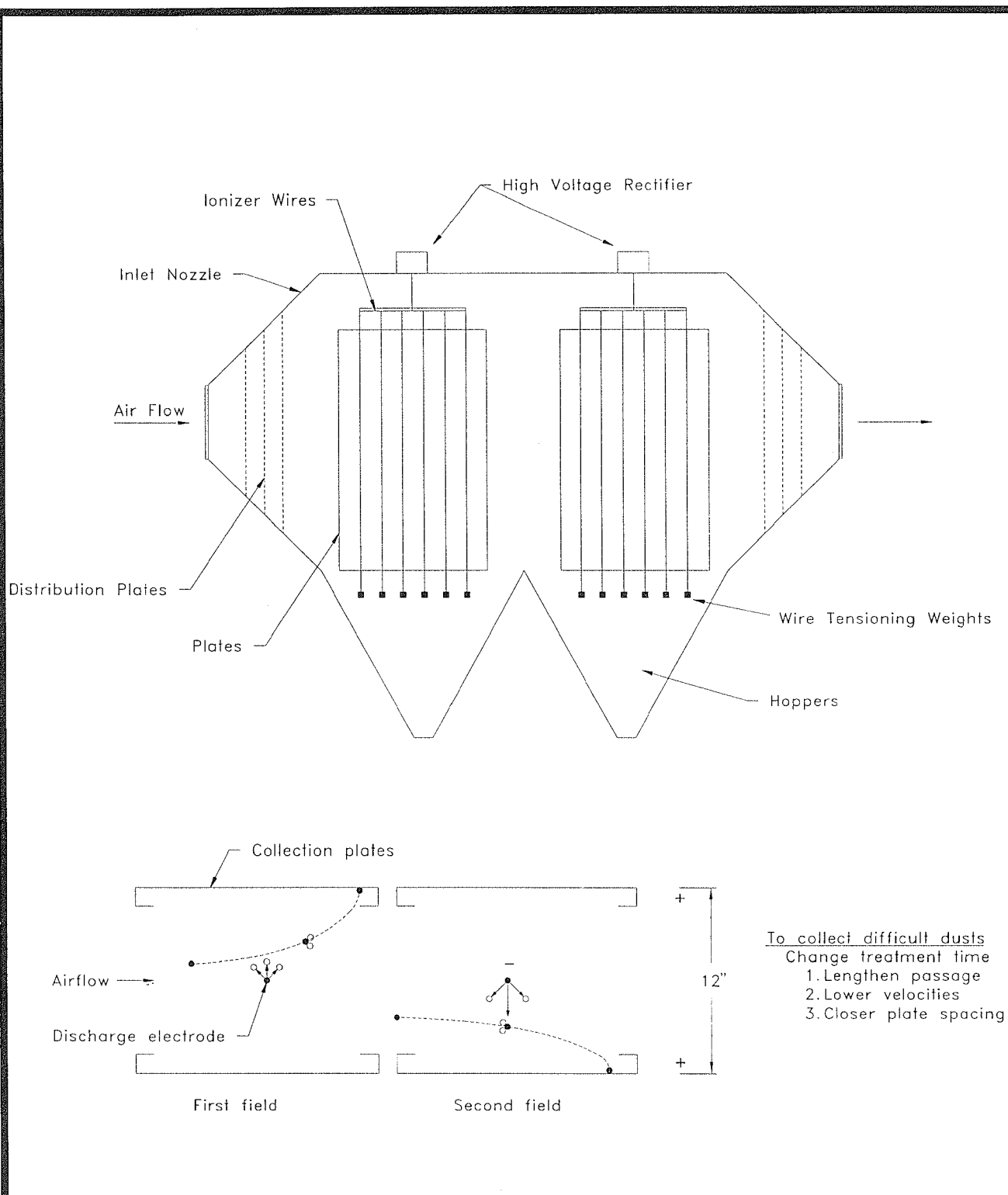
Motor driven, double gate valve for continuous removal of collected dust. Gates are sequenced so only one is open at a time in order to provide air seal. Flanged for connection to dust disposal chute.

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*DRY TYPE DUST COLLECTORS
DISCHARGE VALVES*

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FIGURE 4-3



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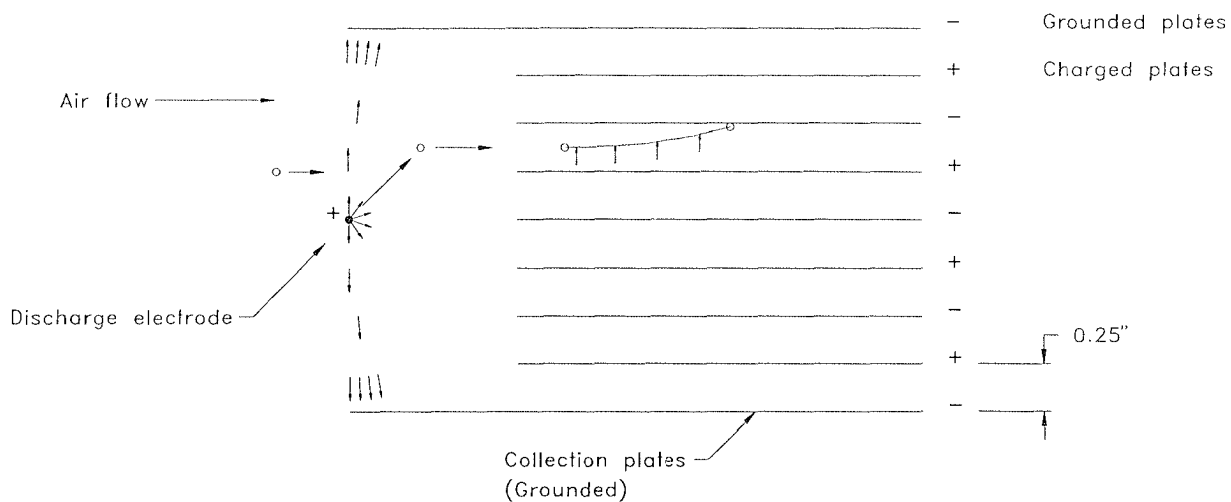
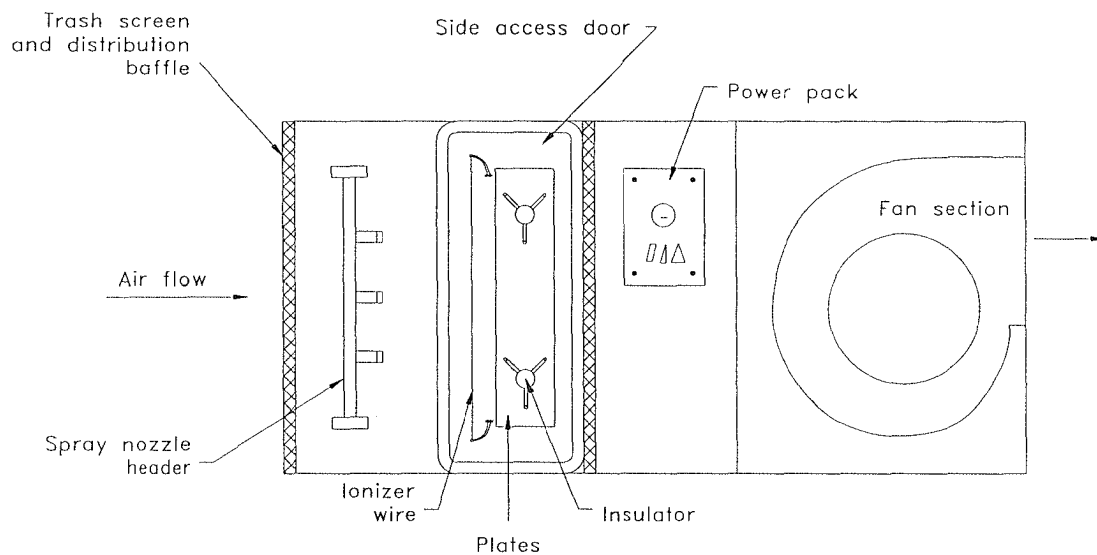
*ELECTROSTATIC PRECIPITATOR
HIGH VOLTAGE DESIGN
(40,000 TO 75,000 VOLTS)*

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FIGURE

4-4



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*ELECTROSTATIC PRECIPITATOR
LOW VOLTAGE DESIGN
(11,000 TO 15,000 VOLTS)*

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FIGURE 4-5

The "Penny," or two-stage, precipitator (Figure 4-5) uses DC voltages from 11,000 to 14,000 volts for ionization and is frequently referred to as a low voltage precipitator. Its use is limited to low concentrations, normally not exceeding 0.025 grains per cubic foot. It is the most practical collection technique for the many hydrocarbon applications where an initially clear exhaust stack turns into a visible emission as vapor condenses. Some applications include plasticizer ovens, forge presses, die-casting machines, and various welding operations. Care must be taken to keep the precipitator inlet temperature low enough to insure that condensation has already occurred.

For proper results, the inlet gas stream should be evaluated and treated where necessary to provide proper conditions for ionization. For high voltage units, a cooling tower is sometimes necessary. Low voltage units may use wet scrubbers, evaporative coolers, heat exchangers or other devices to condition the gas stream for best precipitator performance.

The pressure drop of an electrostatic precipitator is extremely low, usually less than 1 "wg; therefore, the energy requirement is significantly less than for other techniques.

4.3.2 Fabric Collectors: Fabric collectors remove particulate by straining, impingement, interception, diffusion, and electrostatic charge. The "fabric" may be constructed of any fibrous material, either natural or man-made, and may be spun into a yarn and woven or felted by needling, impacting, or bonding. Woven fabrics are identified by thread count and weight of fabric per unit area. Non-woven (felts) are identified by thickness and weight per unit area. Regardless of construction, the fabric represents a porous mass through which the gas is passed unidirectionally such that dust particles are retained on the dirty side and the cleaned gas passes on through.

The ability of the fabric to pass air is stated as "permeability" and is defined as the cubic feet of air passed through one square foot of fabric each minute at a pressure drop of 0.5 "wg. Typical permeability values for commonly used fabrics range from 25 to 40 cfm.

A non-woven (felted) fabric is more efficient than a woven fabric of identical weight because the void areas or pores in the non-woven fabric are smaller. A specific type of fabric can be made more efficient by using smaller fiber diameters, a greater weight of fiber per unit area, and by packing the fibers more tightly. For non-woven construction, the use of finer needles for felting also improves efficiency. While any fabric is made more efficient by these methods, the cleanability and permeability are reduced. A highly efficient fabric that cannot be cleaned represents an excessive resistance to air flow and is not an economical engineering solution. Final fabric selection is generally a compromise between efficiency and permeability.

Choosing a fabric with better cleanability or greater perme-

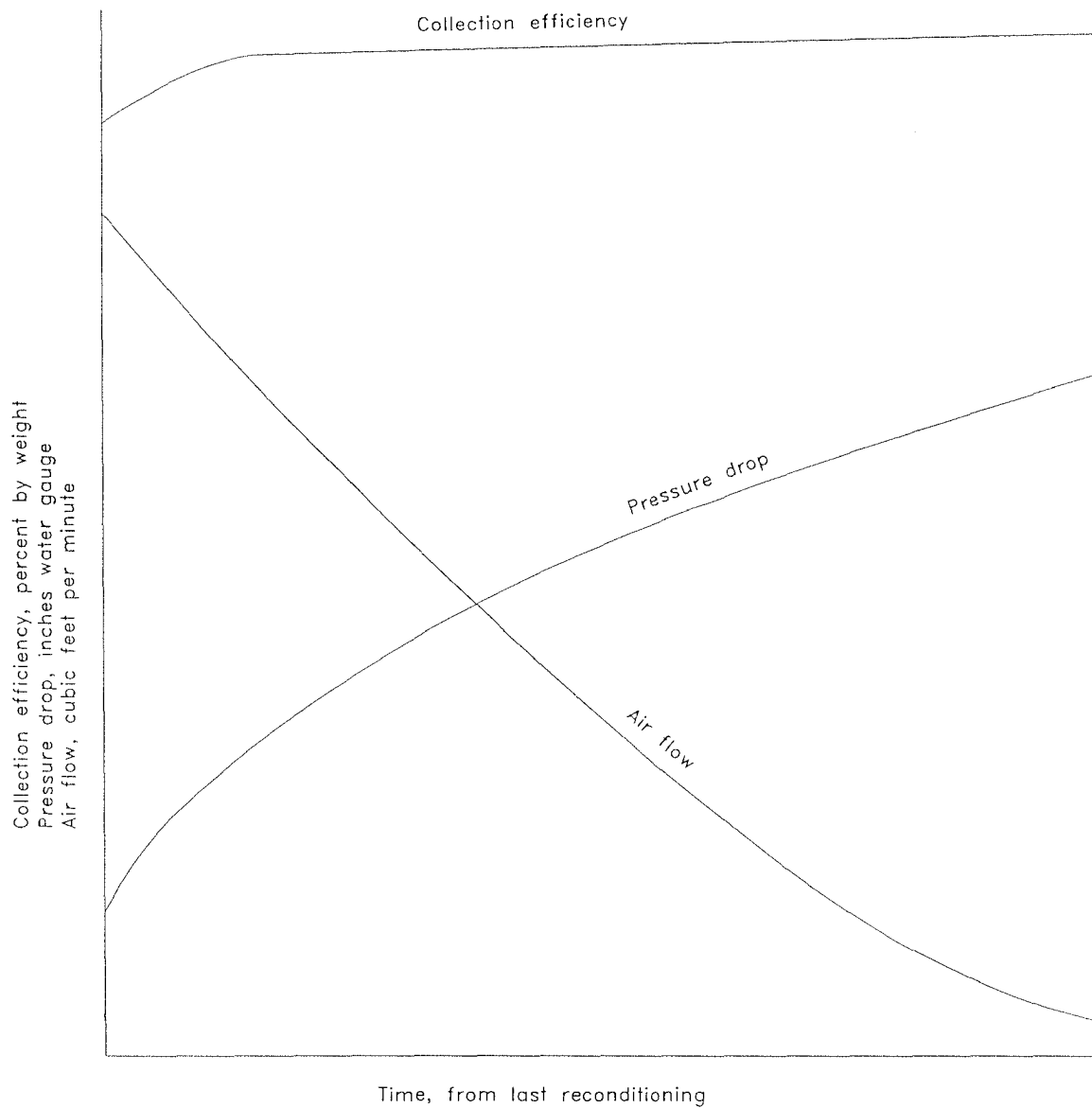
ability but lower inherent efficiency is not as detrimental as it may seem. The efficiency of the fabric as a filter is meaningful only when new fabric is first put into service. Once the fabric has been in service any length of time, collected particulate in contact with the fabric acts as a filter aid, improving collection efficiency. Depending on the amount of particulate and the time interval between fabric reconditioning, it may well be that virtually all filtration is accomplished by the previously collected particulate—or dust cake—as opposed to the fabric itself. Even immediately after cleaning, a residual and/or redeposited dust cake provides additional filtration surface and higher collection efficiency than obtainable with new fabric. While the collection efficiency of new, clean fabric is easily determined by laboratory test and the information is often published, it is not representative of operating conditions and therefore is of little importance in selecting the proper collector.

Fabric collectors are not 100% efficient, but well-designed, adequately sized, and properly operated fabric collectors can be expected to operate at efficiencies in excess of 99%, and often as high as 99.9% or more on a mass basis. The inefficiency, or penetration, that does occur is greatest during or immediately after reconditioning. Fabric collector inefficiency is frequently a result of by-pass due to damaged fabric, faulty seals, or sheet metal leaks rather than penetration of the fabric. Where extremely high collection efficiency is essential, the fabric collector should be leak tested for mechanical leaks.

The combination of fabric and collected dust becomes increasingly efficient as the dust cake accumulates on the fabric surface. At the same time, the resistance to air flow increases. Unless the air moving device is adjusted to compensate for the increased resistance, the gas flow rate will be reduced. Figure 4-6 shows how efficiency, resistance to flow, and flow rate change with time as dust accumulates on the fabric. Fabric collectors are suitable for service on relatively heavy dust concentrations. The amount of dust collected on a single square yard of fabric may exceed five pounds per hour. In virtually all applications, the amount of dust cake accumulated in just a few hours will represent sufficient resistance to flow to cause an unacceptable reduction in air flow.

In a well-designed fabric collector system, the fabric or filter mat is cleaned or reconditioned before the reduction in air flow is critical. The cleaning is accomplished by mechanical agitation or air motion, which frees the excess accumulation of dust from the fabric surface and leaves a residual or base cake. The residual dust cake does not have the same characteristics of efficiency or resistance to air flow as new fabric.

Commercially available fabric collectors employ fabric configured as bags or tubes, envelopes (flat bags), rigid elements, or pleated cartridges. Most of the available fabrics, whether woven or non-woven, are employed in either bag or



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*PERFORMANCE VS TIME
BETWEEN RECONDITIONINGS
FABRIC COLLECTORS*

DATE 1-88

FIGURE 4-6

envelope configuration. The pleated cartridge arrangement uses a paper-like fiber in either a cylindrical or panel configuration. It features extremely high efficiency on light concentrations. Earlier designs employed cellulose-based media. Today, more conventional media, such as polypropylene or spun-bonded polyester, are frequently used.

The variable design features of the many fabric collectors available are:

1. Type of fabric (woven or non-woven).
2. Fabric configuration (bags or tubes, envelopes, cartridges).
3. Intermittent or continuous service.
4. Type of reconditioning (shaker, pulse-jet, reverse-air).
5. Housing configuration (single compartment, multiple compartment).

At least two of these features will be interdependent. For example, non-woven fabrics are more difficult to recondition and therefore require high-pressure cleaning.

A fabric collector is selected for its mechanical, chemical, and thermal characteristics. Table 4-1 lists those characteristics for some common filter fabrics.

Fabric collectors are sized to provide a sufficient area of filter media to allow operation without excessive pressure drop. The amount of filter area required depends on many factors, including:

1. Release characteristics of dust.
2. Porosity of dust cake.
3. Concentration of dust in carrier gas stream.
4. Type of fabric and surface finish, if any.
5. Type of reconditioning.
6. Reconditioning interval.
7. Air flow pattern within the collector.
8. Temperature and humidity of gas stream.

Because of the many variables and their range of variation, fabric collector sizing is a judgment based on experience. The sizing is usually made by the equipment manufacturer, but at times may be specified by the user or a third party. Where no experience exists, a pilot installation is the only reliable way to determine proper size.

The sizing or rating of a fabric collector is expressed in terms of air flow rate versus fabric media area. The resultant ratio is called "air to cloth ratio" with units of cfm per square foot of fabric. This ratio represents the average velocity of the gas stream through the filter media. The expression "filtration velocity" is used synonymously with air to cloth ratio for rating fabric collectors. For example, an air to cloth ratio of 7:1 (7 cfm/sq ft) is equivalent to a filtration velocity of 7 fpm.

Table 4-2 compares the various characteristics of fabric

collectors. The different types will be described in detail later. Inspection of Table 4-2 now may make the subsequent discussion more meaningful. The first major classification of fabric collectors is intermittent or continuous duty. Intermittent-duty fabric collectors cannot be reconditioned while in operation. By design, they require that the gas flow be interrupted while the fabric is agitated to free accumulated dust cake. Continuous-duty collectors do not require shut down for reconditioning.

Intermittent-duty fabric collectors may use a tube, cartridge, or envelope configuration of woven fabric and will generally employ shaking or vibration for reconditioning. Figure 4-7 shows both tube and envelope shaker collector designs. For the tube type, dirty air enters the open bottom of the tube and dust is collected on the inside of the fabric. The bottoms of the tubes are attached to a tube sheet and the tops are connected to a shaker mechanism. Since the gas flow is from inside to outside, the tubes tend to inflate during operation and no other support of the fabric is required.

Gas flow for envelope-type collectors is from outside to inside; therefore, the envelopes must be supported during operation to prevent collapsing. This is normally done by inserting wire mesh or fabricated wire cages into the envelopes. The opening of the envelope from which the cleaned air exits is attached to a tube sheet and, depending on design, the other end may be attached to a support member or cantilevered without support. The shaker mechanism may be located in either the dirty air or cleaned air compartments.

Periodically (usually at 3- to 6-hour intervals) the air flow must be stopped to recondition the fabric. Figure 4-8 illustrates the system air flow characteristics of an intermittent-duty fabric collector. As dust accumulates on the fabric, resistance to flow increases and air flow decreases until the fan is turned off and the fabric reconditioned. Variations in air flow due to changing pressure losses is sometimes a disadvantage and, when coupled with the requirement to periodically stop the air flow, may preclude the use of intermittent collectors. Reconditioning seldom requires more than two minutes but must be done without air flow through the fabric. If reconditioning is attempted with air flowing, it will be less effective and the flexing of the woven fabric will allow a substantial amount of dust to escape to the clean air side.

The filtration velocity for large intermittent-duty fabric collectors seldom exceeds 6 fpm and normal selections are in the 2-4 fpm range. Lighter dust concentrations and the ability to recondition more often allow the use of higher filtration velocities. Ratings are usually selected so that the pressure drop across the fabric will be in the 2-5 "wg range between start and end of operating cycle.

With multiple-section, continuous-duty, automatic fabric collectors, the disadvantage of stopping the air flow to permit fabric reconditioning and the variations in air flow with dust cake build-up can be overcome. The use of sections or com-

TABLE 4-1. Characteristics of Filter Fabrics*

Generic Names	Example Trade Name Fabrics**	Max. Temp. F		Resistance to Physical Action					Resistance to Chemicals				
		Continuous	Intermittent	Dry Heat	Moist Heat	Abrasion	Shaking	Flexing	Mineral Acid	Organic Acid	Alkalies	Oxidizing	Solvents
Cotton	Cotton	180	—	G	G	F	G	G	P	G	F	F	E
Polyester	Dacron ⁽¹⁾ Fortre ⁽²⁾ Vycron ⁽³⁾ Kodel ⁽⁴⁾ Enka												
	Polyester ⁽³⁾	275	—	G	F	G	E	E	G	G	F	G	E
Acrylic	Orlon ⁽¹⁾ Acrilan ⁽⁶⁾ Creslan ⁽⁷⁾ Dralon T ⁽⁸⁾ Zefran												
	Zefran	275	285	G	G	G	G	E	G	G	F	G	E
Modacrylic	Dynel ⁽¹⁰⁾ Verel ⁽⁴⁾												
	Verel ⁽⁴⁾	160	—	F	F	F	P-F	G	G	G	G	G	G
Nylon (Polyamide)	Nylon 6,6 ^(1,2,6) Nylon 6 ^(5,11,12) Nomex ⁽¹¹⁾												
	Nylon 6 ^(5,11,12)	225	—	G	G	E	E	E	P	F	G	F	E
	Nomex ⁽¹¹⁾	400	450	E	E	E	E	E	P-F	E	G	G	E
Polymide	P-84 ⁽¹⁸⁾	500	580	E	P	G	G	E	P-F	G	F	G	E
Polypropylene	Herculon ⁽¹³⁾ Reevon ⁽¹⁴⁾ Vectra ⁽¹⁵⁾												
	Vectra ⁽¹⁵⁾	200	250	G	F	E	E	G	E	E	E	G	G
Teflon (fluorocarbon)	Teflon TFE ⁽¹⁾ Teflon FEP ⁽¹⁾												
	TFE ⁽¹⁾	500	550	E	E	P-F	G	G	E	E	E	E	E
	FEP ⁽¹⁾	450	—	E	E	P-F	G	G	E	E	E	E	E
Expanded PFTE	Rastex	500	550	E	E	P-F	G	G	E	E	E	E	E
Vinyon	Vinyon ⁽¹⁶⁾ Clevylt ⁽¹⁷⁾												
	Clevylt ⁽¹⁷⁾	350	—	F	F	F	G	G	E	E	G	G	P
Glass	Glass	500	600	E	E	P	P	F	E	E	F	E	E
Fiberglass	Fiberglass ⁽¹⁹⁾	550	550	E	E	P	P	G	G	G	G	E	G

*E - excellent; G = good; F = fair; P = poor

** Registered Trademarks

(1) Du Pont; (2) Celanese; (3) Beaunit; (4) Eastman; (5) American Enka; (6) Chemstrand; (7) American Cyanamid; (8) Farbenfabriken Bayer AG; (9) Dow Chemical; (10) Union Carbide; (11) Allied Chemical; (12) Firestone; (13) Hercules; (14) Alamo Polymer; (15) National Plastic; (16) FMC; (17) Societe Rhovyl; (18) Lenzing; (19) Huyglas

partments, as indicated in Figure 4-7, allows continuous operation of the exhaust system because automatic dampers periodically remove one section from service for fabric reconditioning while the remaining compartments handle the total gas flow. The larger the number of compartments, the more constant the pressure loss and air flow. Either tubes or envelopes may be used and fabric reconditioning is usually accomplished by shaking or vibrating.

Figure 4-8 shows air flow versus time for a multiple-section collector. Each individual section or compartment has an air flow versus time characteristic like that of the intermittent collector, but the total variation is reduced because of the multiple compartments. Note the more constant air flow characteristic of the five-compartment unit as opposed to the three-compartment design. Since an individual section is out of service only a few minutes for reconditioning and remaining sections handle the total gas flow during that time, it is possible to clean the fabric more frequently than with the intermittent type. This permits the multiple-section unit to handle higher dust concentrations. Compartments are reconditioned in fixed sequence with the ability to adjust the time interval between cleaning of individual compartments.

One variation of this design is the low-pressure, reverse-air collector which does not use shaking for fabric reconditioning. Instead, a compartment is isolated for cleaning and the tubes collapsed by means of a low pressure secondary blower, which draws air from the compartment in a direction opposite to the primary air flow. This is a "gentle" method of fabric reconditioning and was developed primarily for the fragile glass cloth used for high-temperature operation. The reversal of air flow and tube deflation is accomplished very gently to avoid damage to the glass fibers. The control sequence usually allows the deflation and re-inflation of tubes several times for complete removal of excess dust. Tubes are 6–11 inches in diameter and can be as long as 30 feet. For long tubes, stainless steel rings may be sewn on the inside to help break up the dust

cake during deflation. A combination of shaking and reverse air flow has also been utilized.

When shaking is used for fabric reconditioning, the filtration velocity usually is in the 1–4 fpm range. Reverse-air collapse-type reconditioning generally necessitates lower filtration velocities since reconditioning is not as complete. They are seldom rated higher than 3 fpm. The air to cloth ratio or filtration velocity is based on *net* cloth area available when a compartment is out of service for reconditioning.

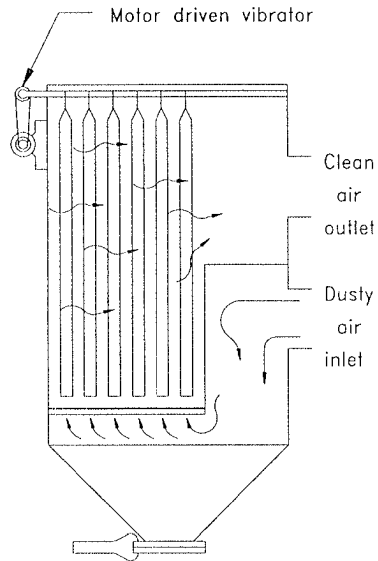
Reverse-jet, continuous-duty fabric collectors may use envelopes or tubes of non-woven (felted) fabric, pleated cartridges of non-woven mat (paper-like) in cylindrical or panel configuration, or rigid elements such as sintered polyethylene. They differ from the low-pressure reverse-air type in that they employ a brief burst of high-pressure air to recondition the fabric. Woven fabric is not used because it allows excessive dust penetration during reconditioning. The most common designs use compressed air at 80–100 psig, while others use an integral pressure blower at a lower pressure but higher secondary flow rate. Those using compressed air are generally called pulse-jet collectors and those using pressure blowers are called fan-pulse collectors.

All designs collect dust on the outside and have air flow from outside to inside the fabric. All recondition the media by introducing the pulse of cleaning air into the opening where cleaned air exits from the tube, envelope, or cartridge. In many cases, a venturi shaped fitting is used at this opening to provide additional cleaning by inducing additional air flow. The venturi also directs or focuses the cleaning pulse for maximum efficiency.

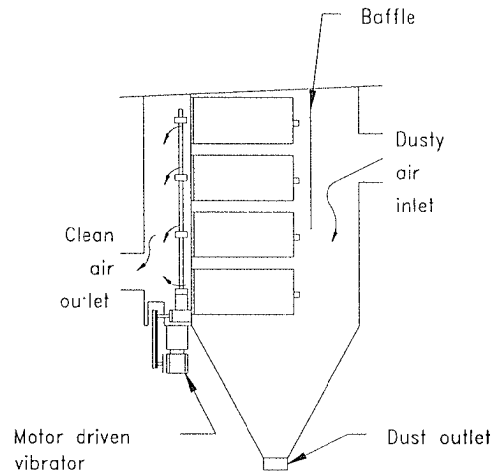
Figure 4-9 shows a typical pulse-jet collector. Under normal operation (air flow from outside to inside), the fabric shape will tend to collapse; therefore, a support cage is required. The injection of a short pulse of high-pressure air induces a secondary flow from the clean air compartment in

Table 4-2. Summary of Fabric-Type Collectors and Their Characteristics

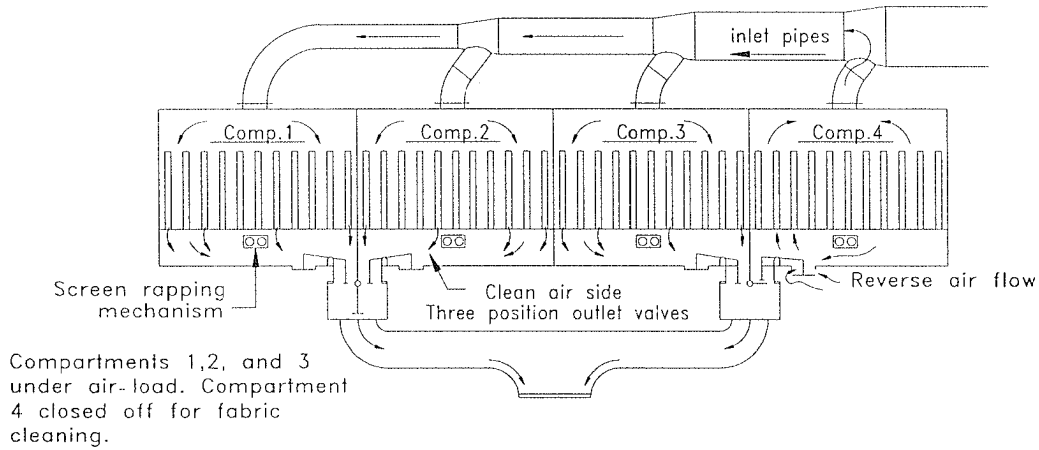
	INTERRUPTABLE OPERATION Light to Moderate Loading	INTERRUPTABLE OPERATION Heavy Loading		CONTINUOUS OPERATIONS Any Loading	
Fabric Reconditioning Requirement	Intermittent	Continuous			
Type of Reconditioning	Shaker	Shaker	Reverse Air (Low Pressure)	Reverse Pulse – (High Pressure) Pulse Jet or Fan Pulse	
Collector Configuration	Single Compartment	Multiple Compartment with inlet or outlet dampers for each		Single Compartment	
Fabric Configuration	Tube, Cartridge or Envelope	Tube or Envelope	Tube	Tube or Envelope	Pleated Cartridge
Type of Fabric	Woven	Woven		Non-Woven (Felt)	Non-Woven
Air Flow	Highly Variable	Slightly Variable		Virtually Constant	Virtually Constant
Normal Rating (filtration velocity, fpm)	1 to 6 fpm	1 to 3 fpm	1 to 3 fpm	5 to 12 fpm	<1 to 7 fpm



TUBE TYPE



ENVELOPE TYPE



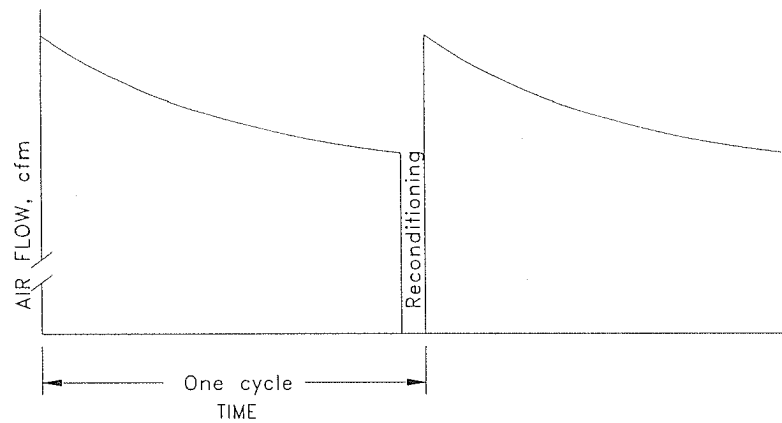
MULTIPLE SECTION CONTINUOUS AUTOMATIC

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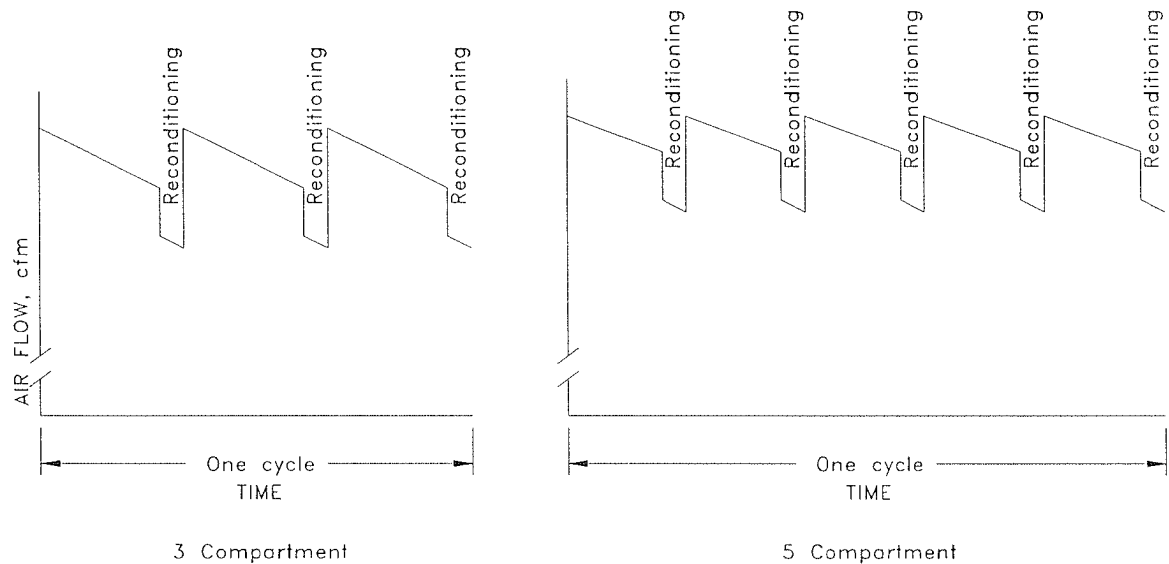
FABRIC COLLECTORS

DATE 1-88

FIGURE 4-7



INTERMITTENT DUTY FABRIC COLLECTOR



MULTIPLE SECTION, CONTINUOUS DUTY FABRIC COLLECTOR

Note:
The flow variation has been exaggerated.

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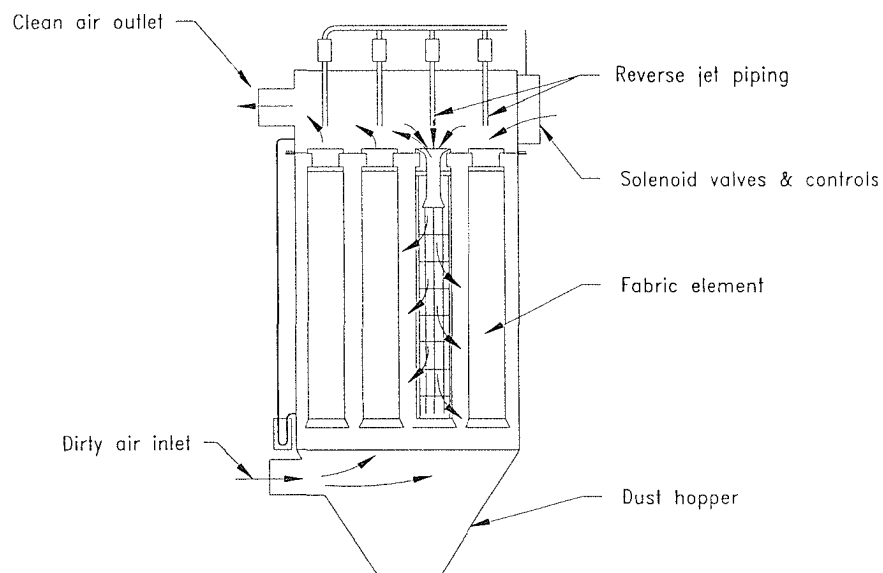
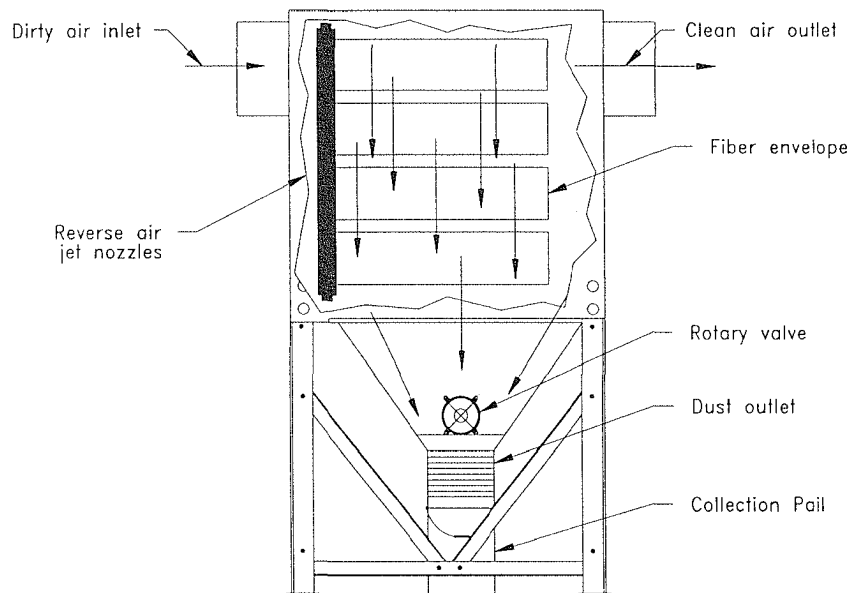
*AIR FLOW THROUGH
FABRIC COLLECTORS*

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1-88

FIGURE

4-8



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*FABRIC COLLECTORS
PULSE JET TYPE*

DATE 1-88

FIGURE 4-9

a direction opposite to the normal air flow. Reconditioning is accomplished by the pulse of high-pressure air which stops forward air flow, then rapidly pressurizes the media, breaking up the dust cake and freeing accumulated dust from the fabric. The secondary or induced air acts as a damper, preventing flow in the normal direction during reconditioning. The entire process, from injection of the high-pressure pulse and initiation of secondary flow until the secondary flow ends, takes place in approximately one second. Solenoid valves which control the pulses of compressed air may be open for a tenth of a second or less. An adequate flow rate of clean and dry compressed air of sufficient pressure must be supplied to ensure effective reconditioning.

Reverse-jet collectors normally clean no more than 10% of the fabric at any one time. Because such a small percentage is cleaned at any one time and because the induced secondary flow blocks normal flow during that time, reconditioning can take place while the collector is in service and without the need for compartmentation and dampers. The cleaning intervals are adjustable and are considerably more frequent than the intervals for shaker or reverse-air collectors. An individual element may be pulsed and reconditioned as often as once a minute to every six minutes.

Due to this very short reconditioning cycle, higher filtration velocities are possible with reverse-jet collectors. However, with all reverse-jet collectors, accumulated dust that is freed from one fabric surface may become reentrained and redeposited on an adjacent surface, or even on the original surface. This phenomenon of redeposition tends to limit filtration velocity to something less than might be anticipated with cleaning intervals of just a few minutes.

Laboratory tests^(4.1) have shown that for a given collector design redeposition increases with filtration velocity. Other test work^(4.2) indicates clearly that redeposition varies with collector design and especially with flow patterns in the dirty air compartment. EPA-sponsored research^(4.3) has shown that superior performance results from downward flow of the dirty air stream. This downward air flow reduces redeposition since it aids gravity in moving dust particles toward the hopper.

Filtration velocities of 5–12 fpm are normal for reverse-jet collectors. The pleated cartridge type of reverse-jet collector is limited to filtration velocities in the 7 fpm range. The pleat configuration may produce very high approach velocities and greater redeposition.

4.3.3 Wet Collectors: Wet collectors, or scrubbers, are commercially available in many different designs, with pressure drops from 1.5 "wg to as much as 100 "wg. There is a corresponding variation in collector performance. It is generally accepted that, for well-designed equipment, efficiency depends on the energy utilized in air to water contact and is independent of operating principle. Efficiency is a function of total energy input per cfm whether the energy is supplied

to the air or to the water. This means that well-designed collectors by different manufacturers will provide similar efficiency if equivalent power is utilized.

Wet collectors have the ability to handle high-temperature and moisture-laden gases. The collection of dust in a wetted form minimizes a secondary dust problem in disposal of collected material. Some dusts represent explosion or fire hazards when dry. Wet collection minimizes the hazard; however, the use of water may introduce corrosive conditions within the collector and freeze protection may be necessary if collectors are located outdoors in cold climates. Space requirements are nominal. Pressure losses and collection efficiency vary widely for different designs.

Wet collectors, especially the high-energy types, are frequently the solution to air pollution problems. It should be recognized that disposal of collected material in water without clarification or treatment may create water pollution problems.

Wet collectors have one characteristic not found in other collectors — the inherent ability to humidify. Humidification, the process of adding water vapor to the air stream through evaporation, may be either advantageous or disadvantageous depending on the situation. Where the initial air stream is at an elevated temperature and not saturated, the process of evaporation reduces the temperature and the volumetric flow rate of the gas stream leaving the collector. Assuming the fan is to be selected for operation on the clean air side of the collector, it may be smaller and will definitely require less power than if there had been no cooling through the collector. This is one of the obvious advantages of humidification; however, there are other applications where the addition of moisture to the gas stream is undesirable. For example, the exhaust of humid air to an air-conditioned space normally places an unacceptable load on the air conditioning system. High humidity can also result in corrosion of finished goods. Therefore, humidification effects should be considered before designs are finalized. While all wet collectors humidify, the amount of humidification varies for different designs. Most manufacturers publish the humidifying efficiency for their equipment and will assist in evaluating the results.

Chamber or Spray Tower: Chamber or spray tower collectors consist of a round or rectangular chamber into which water is introduced by spray nozzles. There are many variations of design, but the principal mechanism is impaction of dust particles on the liquid droplets created by the nozzles. These droplets are separated from the air stream by centrifugal force or impingement on water eliminators.

The pressure drop is relatively low (on the order of 0.5–1.5 "wg), but water pressures range from 10–400 psig. The high pressure devices are the exception rather than the rule. In general, this type of collector utilizes low-pressure supply water and operates in the lower efficiency range for wet

collectors. Where water is supplied under high pressure, as with fog towers, collection efficiency can reach the upper range of wet collector performance.

For conventional equipment, water requirements are reasonable, with a maximum of about 5 gpm per thousand scfm of gas. Fogging types using high water pressure may require as much as 10 gpm per thousand scfm of gas.

Packed Towers: Packed towers (see Figure 4-10) are essentially contact beds through which gases and liquid pass concurrently, counter-currently, or in cross-flow. They are used primarily for applications involving gas, vapor, and mist removal. These collectors can capture solid particulate matter, but they are not used for that purpose because dust plugs the packing and requires unreasonable maintenance.

Water rates of 5–10 gpm per thousand scfm are typical for packed towers. Water is distributed over V-notched ceramic or plastic weirs. High temperature deterioration is avoided by using brick linings, allowing gas temperatures as high as 1600 F to be handled direct from furnace flues.

The air flow pressure loss for a four foot bed of packing, such as ceramic saddles, will range from 1.5–3.5 "wg. The face velocity (velocity at which the gas enters the bed) will typically be 200–300 fpm.

Wet Centrifugal Collectors: Wet centrifugal collectors (see Figure 4-11) comprise a large portion of the commercially available wet collector designs. This type utilizes centrifugal force to accelerate the dust particle and impinge it upon a wetted collector surface. Water rates are usually 2–5 gpm per thousand scfm of gas cleaned. Water distribution can be from nozzles, gravity flow or induced water pickup. Pressure drop is in the 2–6 "wg range.

As a group, these collectors are more efficient than the chamber type. Some are available with a variable number of impingement sections. A reduction in the number of sections results in lower efficiency, lower cost, less pressure drop, and smaller space. Other designs contain multiple collecting tubes. For a given air flow rate, a decrease in the tube size provides higher efficiency because the centrifugal force is greater.

Wet Dynamic Precipitator: The wet dynamic precipitator (see Figure 4-12) is a combination fan and dust collector. Dust particles in the dirty air stream impinge upon rotating fan blades wetted with spray nozzles. The dust particles impinge into water droplets and are trapped along with the water by a metal cone while the cleaned air makes a turn of 180 degrees and escapes from the front of the specially shaped impeller blades. Dirty water from the water cone goes to the water and sludge outlet and the cleaned air goes to an outlet section containing a water elimination device.

Orifice Type: In this group of wet collector designs (see

Figure 4-12), the air flow through the collector is brought in contact with a sheet of water in a restricted passage. Water flow may be induced by the velocity of the air stream or maintained by pumps and weirs. Pressure losses vary from 1 "wg or less for a water wash paint booth to a range of 3–6 "wg for most of the industrial designs. Pressure drops as high as 20 "wg are used with some designs intended to collect very small particles.

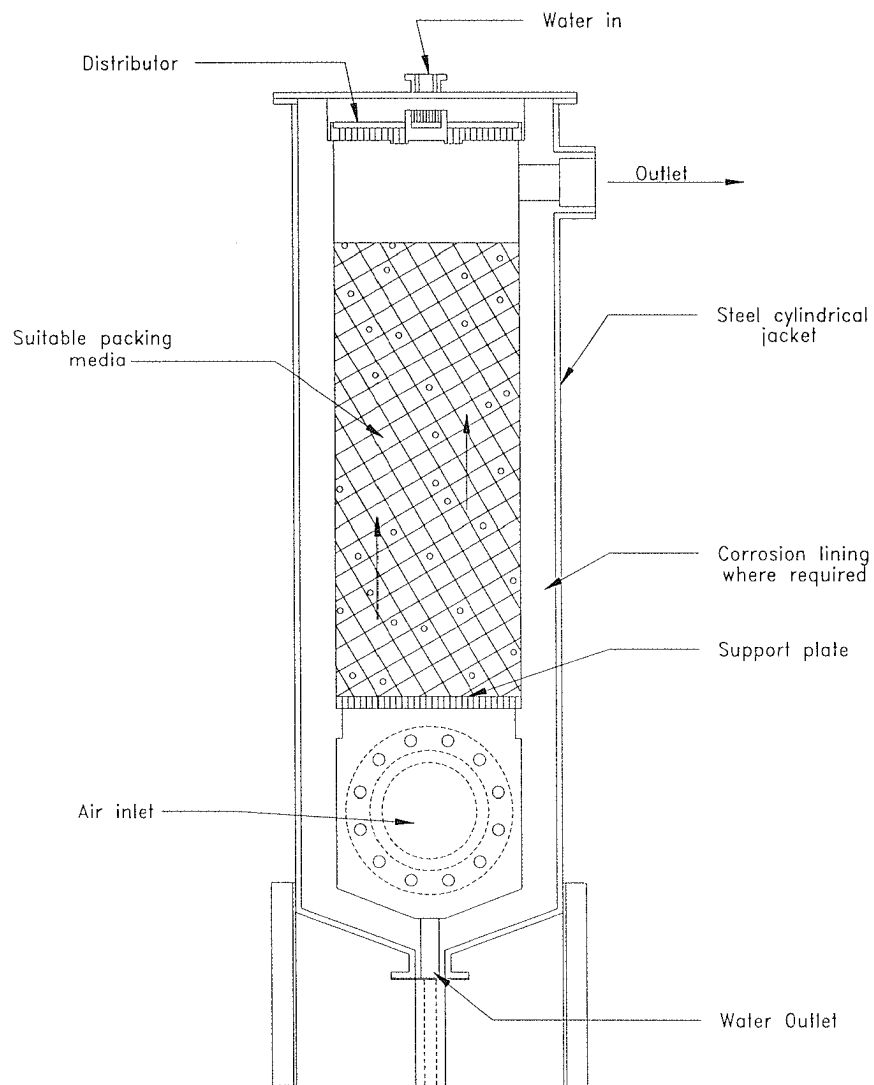
Venturi: The venturi collector (see Figure 4-11) uses a venturi-shaped constriction to establish throat velocities considerably higher than those used by the orifice type. Gas velocities through venturi throats may range from 12,000–24,000 fpm. Water is supplied by piping or jets at or ahead of the throat at rates from 5–15 gpm per thousand scfm of gas.

The collection mechanism of the venturi is impaction. As is true for all well-designed wet collectors, collection efficiency increases with higher pressure drops. Specific pressure drops are obtained by designing for selected velocities in the throat. Some venturi collectors are made with adjustable throats allowing operation over a range of pressure drops for a given flow rate or over a range of flow rates with a constant pressure drop. Systems are available with pressure drops as low as 5 "wg for moderate collection efficiency and as high as 100 "wg for collection of extremely fine particles.

The venturi itself is a gas conditioner causing intimate contact between the particulates in the gas and the multiple jet streams of scrubbing water. The resulting mixture of gases, fume-dust agglomerates, and dirty water must be channeled through a separation section for the elimination of entrained droplets as shown in Figure 4-11.

4.3.4 Dry Centrifugal Collectors: Dry centrifugal collectors separate entrained particulate from an air stream by the use or combination of centrifugal, inertial, and gravitational force. Collection efficiency is influenced by:

1. Particle size, weight and shape. Performance is improved as size and weight become larger and as the shape becomes more spherical.
2. Collector size and design. The collection of fine dust with a mechanical device requires equipment designed to best utilize mechanical forces and fit specific application needs.
3. Velocity. Pressure drop through a cyclone collector increases approximately as the square of the inlet velocity. There is, however, an optimum velocity that is a function of collector design, dust characteristics, gas temperature and density.
4. Dust concentration. Generally, the performance of a mechanical collector increases as the concentration of dust becomes greater.



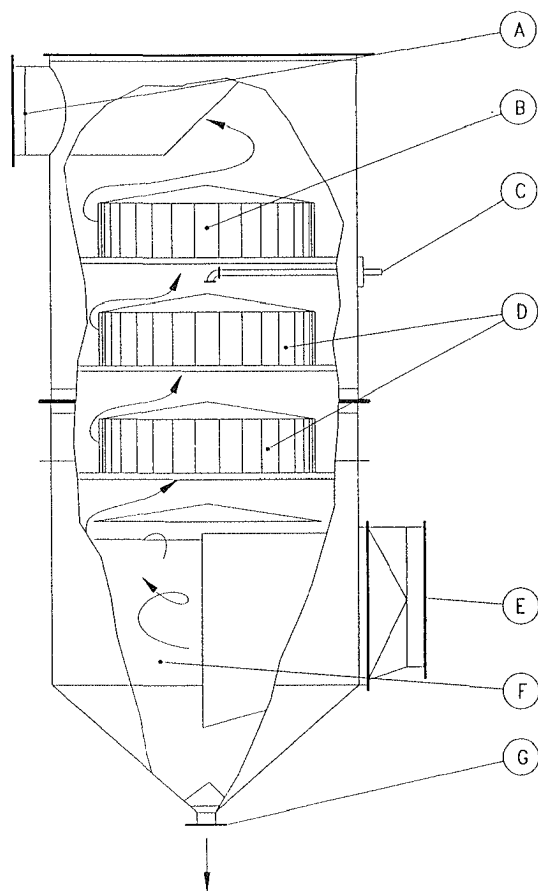
PACKED TOWER

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*WET TYPE COLLECTOR
(FOR GASEOUS CONTAMINANT)*

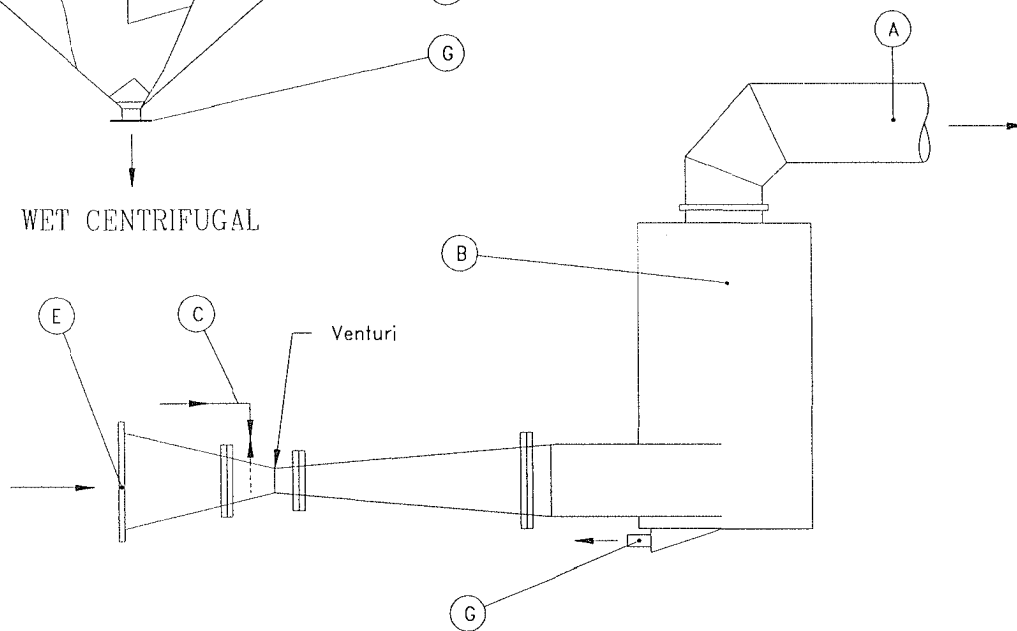
DATE 1-88

FIGURE 4-10



WET CENTRIFUGAL

Symbols	Parts
A	Clean air outlet.
B	Entrainment separator.
C	Water inlet.
D	Impingement plates.
E	Dirty air inlet.
F	Wet cyclone for collecting heavy material.
G	Water and sludge drain.

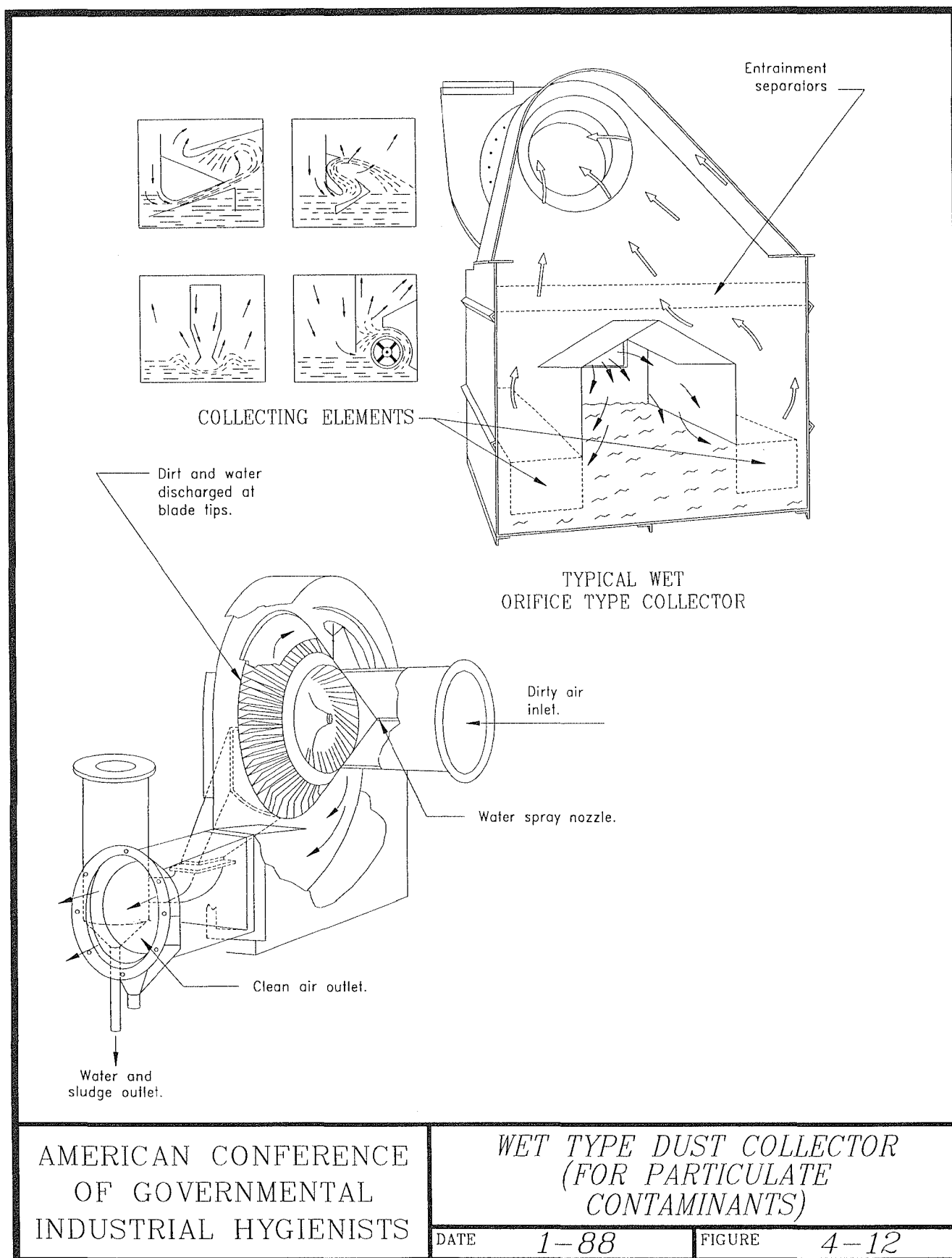


VENTURI SCRUBBER

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**WET TYPE DUST COLLECTORS
(FOR PARTICULATE
CONTAMINANTS)**

DATE 1-88 FIGURE 4-11



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WET TYPE DUST COLLECTOR
(FOR PARTICULATE
CONTAMINANTS)

DATE

1-88

FIGURE

4-12

Gravity Separators: Gravity separators consist of a chamber or housing in which the velocity of the gas stream is made to drop rapidly so that dust particles settle out by gravity. Extreme space requirements and the usual presence of eddy currents nullify this method for removal of anything but extremely coarse particles.

Inertial Separators: Inertial separators depend on the inability of dust to make a sharp turn because its inertia is much higher than that of the carrier gas stream. Blades or louvers in a variety of shapes are used to require abrupt turns of 120° or more. Well-designed inertial separators can separate particles in the 10–20 micron range with about 90% efficiency.

Cyclone Collector: The cyclone collector (see Figure 4-13) is commonly used for the removal of coarse dust from an air stream, as a precleaner to more efficient dust collectors, and/or as a product separator in air conveying systems. Principal advantages are low cost, low maintenance, and relatively low pressure drops (in the 0.75–1.5 "wg range). It is not suitable for the collection of fine particles.

High Efficiency Centrifugals: High-efficiency centrifugals (see Figure 4-13) exert higher centrifugal forces on the dust particles in a gas stream. Because centrifugal force is a function of peripheral velocity and angular acceleration, improved dust separation efficiency has been obtained by:

1. Increasing the inlet velocity.
2. Making the cyclone body and cone longer.
3. Using a number of small diameter cyclones in parallel.
4. Placing units in series.

While high-efficiency centrifugals are not as efficient on small particles as electrostatic, fabric, and wet collectors, their effective collection range is appreciably extended beyond that of other mechanical devices. Pressure losses of collectors in this group range from 3–8 "wg.

4.4 ADDITIONAL AIDS IN DUST COLLECTOR SELECTION

The collection efficiencies of the five basic groups of air cleaning devices have been plotted against mass mean particle size (Figure 4-14). The graphs were found through laboratory and field testing and were not compiled mathematically. The number of lines for each group indicates the range that can be expected for the different collectors operating under the same principle. Variables, such as type of dust, velocity of air, water rate, etc., will also influence the range for a particular application.

Deviation lines shown in the upper right hand corner of the chart allow the estimation of mass mean material size in the effluent of a collector when the inlet mean size is known. Space does not permit a detailed explanation of how the slopes of these lines were determined, but the following example illustrates how they are used. The deviation lines should not

be used for electrostatic precipitators but can be used for the other groups shown at the bottom of the figure.

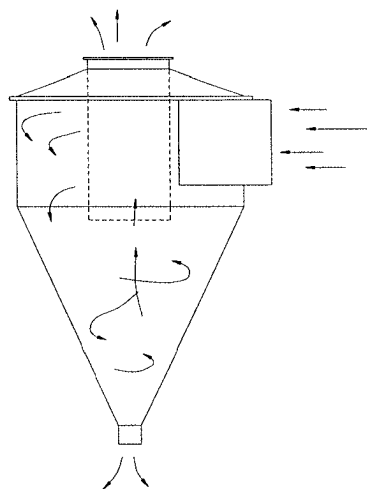
Example: A suitable collector will be selected for a lime kiln to illustrate the use of the chart. Referring to Figure 4-14, the concentration and mean particle size of the material leaving the kiln can vary between 3 and 10 grains per cubic foot, with 5–10 microns the range for mass mean particle size. Assume an inlet concentration of 7.5 grains per cubic foot and a mean inlet size of 9 microns. Projection of this point vertically downwardly to the collection efficiency portion of the chart will indicate that a low-resistance cyclone will be less than 50% efficient; a high-efficiency centrifugal will be 60–80% efficient and a wet collector, fabric arrester and electrostatic precipitator will be 97% efficient or more. A precleaner is usually feasible for dust concentrations over 5 grains per cubic foot unless it is undesirable to have the collected dust separated by size. For this example a high-efficiency centrifugal will be selected as the precleaner. The average efficiency is 70% for this group, therefore the effluent from this collector will have a concentration of $7.5 (1.00 - 0.70) = 2.25$ grains per cubic foot. Draw a line through the initial point with a slope parallel to the deviation lines marked "industrial dust." Where deviation is not known, the average of this group of lines normally will be sufficiently accurate to predict the mean particle size in the collector effluent. A vertical line from the point of intersection between the 2.25 grains per cubic foot horizontal and the deviation line to the base of the chart will indicate a mean effluent particle size of 6.0 microns.

A second high-efficiency centrifugal in series would be less than 50% efficient on this effluent. A wet collector, fabric arrester, or electrostatic would have an efficiency of 94% or better. Assume that a good wet collector will be 98% efficient. The effluent would then be $2.25 (1.00 - 0.98) = 0.045$ grains per cubic foot. Using the previous deviation line and its horizontal intersection of 0.045 grains per cubic foot yields a vertical line intersecting the mean particle size chart at 1.6 microns, the mean particle size of the wet collector effluent.

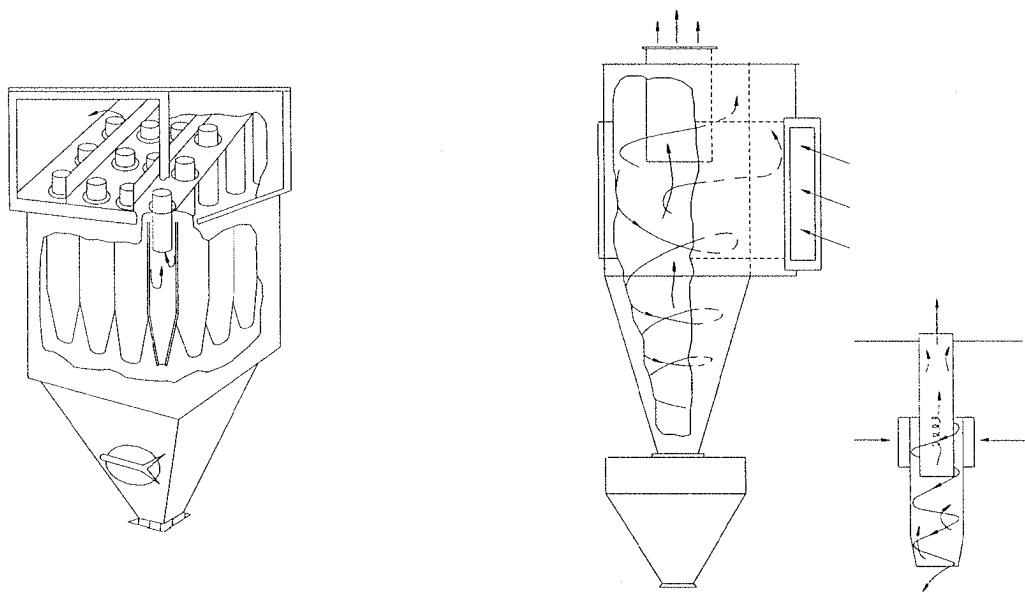
In Table 4-3, an effort has been made to report types of dust collectors used for a wide range of industrial processes. While many of the listings are purely arbitrary, they may serve as a guide in selecting the type of dust collector most frequently used.

4.5 CONTROL OF MIST, GAS, AND VAPOR CONTAMINANTS

Previous discussion has centered on the collection of dust and fume or particulate existing in the solid state. Only the packed tower was singled out as being used primarily to collect mist, gas, or vapor. The character of a mist aerosol is very similar, aerodynamically, to that of a dust or fume aerosol, and the mist can be removed from an air stream by applying the principles that are used to remove solid particulate.



LOW PRESSURE CYCLONE



HIGH EFFICIENCY CENTRIFUGALS

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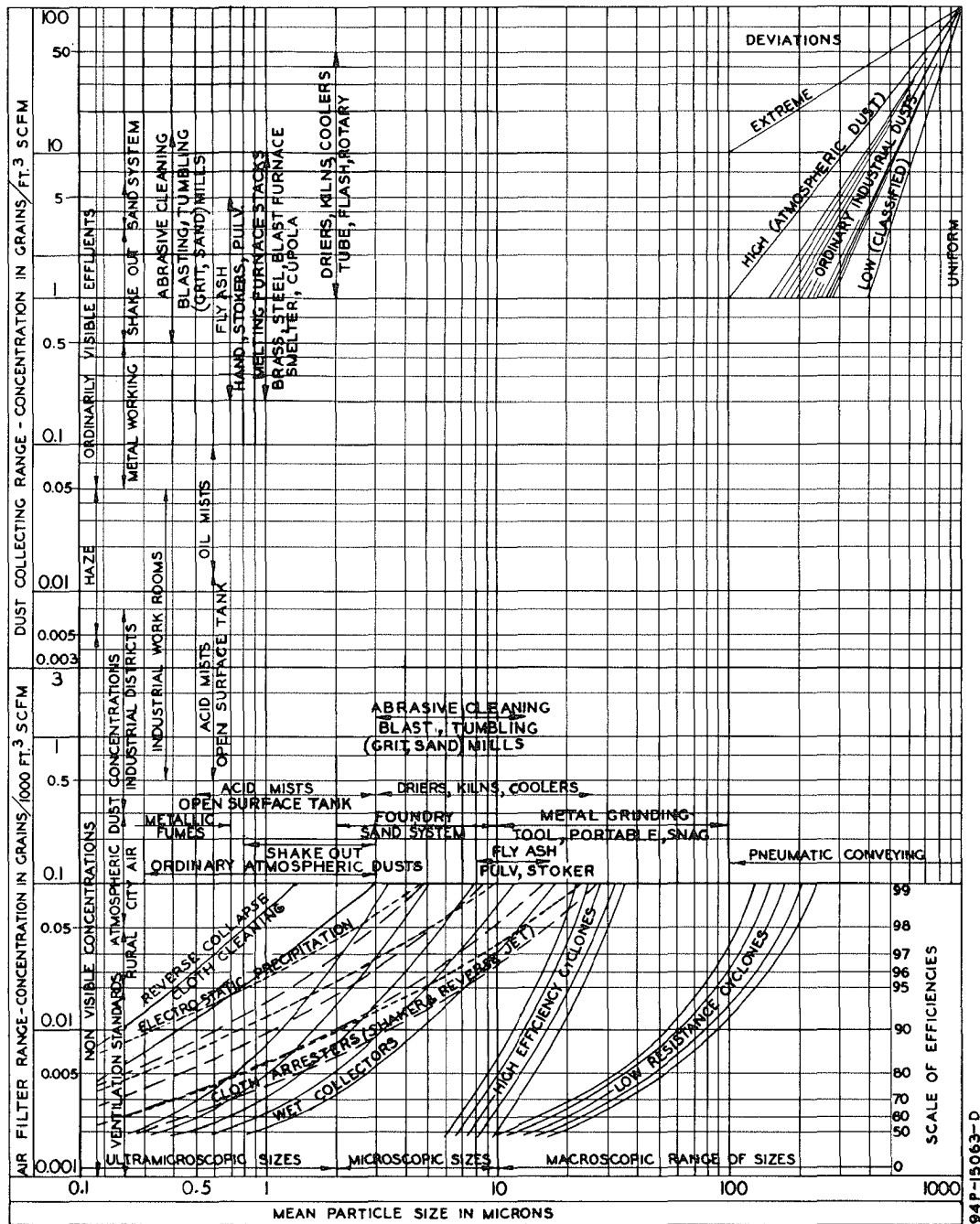
*DRY TYPE
CENTRIFUGAL COLLECTORS*

DATE

3-97

FIGURE

4-13



RANGE OF PARTICLE SIZE, CONCENTRATION, & COLLECTOR PERFORMANCE

COMPILED BY S. SYLVAN APRIL 1952 : COPYRIGHT 1952 AMERICAN AIR FILTER CO. INC.

ACKNOWLEDGEMENTS OF PARTIAL SOURCES OF DATA REPORTED:

- 1 FRANK W.G. - AMERICAN AIR FILTER - SIZE AND CHARACTERISTICS OF AIR BORNE SOLIDS - 1931
- 2 FIRST AND DRINKER - ARCHIVES OF INDUSTRIAL HYGIENE AND OCCUPATIONAL MEDICINE - APRIL 1952
- 3 TAFT INSTITUTE AND AAF LABORATORY TEST DATA - 1961 - '63
- 4 REVERSE COLLAPSE CLOTH CLEANING ADDED 1964

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RANGE OF PARTICLE SIZE

DATE

1-88

FIGURE

4-14

Standard wet collectors are used to collect many types of mists. Specially designed electrostatic precipitators are frequently employed to collect sulfuric acid or oil mist. Even fabric and centrifugal collectors, although not the types previously mentioned, are widely used to collect oil mist generated by high speed machining.

4.6 GASEOUS CONTAMINANT COLLECTORS

Equipment designed specifically to control gas or vapor contaminants can be classified as:

1. Adsorbers
2. Thermal oxidizers
3. Direct combustors
4. Catalytic oxidizers

4.6.1 Absorbers: Absorbers remove soluble or chemically reactive gases from an air stream by contact with a suitable liquid. While all designs utilize intimate contact between the gaseous contaminant and the absorbent, different brands vary widely in configuration and performance. Removal may be by absorption if the gas solubility and vapor pressure promote absorption or chemical reaction. Water is the most frequently used absorbent, but additives are frequently required. Occasionally other chemical solutions must be used. Packed towers (Figure 4-14) are typical absorbers.

4.6.2 Adsorbers: Adsorbers remove contaminants by collection on a solid. No chemical reaction is involved as adsorption is a physical process where molecules of a gas adhere to surfaces of the solid adsorbent. Activated carbon or molecular sieves are popular adsorbents.

4.6.3 Thermal Oxidizers: Thermal oxidizers, or afterburners, may be used where the contaminant is combustible. The contaminated air stream is introduced to an open flame or heating device followed by a residence chamber where combustibles are oxidized producing carbon dioxide and water vapor. Most combustible contaminants can be oxidized at temperatures between 1000 and 1500 F. The residence chamber must provide sufficient dwell time and turbulence to allow complete oxidation.

4.6.4 Direct Combustors: Direct combustors differ from thermal oxidizers by introducing the contaminated gases and auxiliary air directly into the burner as fuel. Auxiliary fuel, usually natural gas or oil, is generally required for ignition and may or may not be required to sustain burning.

4.6.5 Catalytic Oxidizers: Catalytic oxidizers may be used where the contaminant is combustible. The contaminated gas stream is preheated and then passed through a catalyst bed which promotes oxidation of the combustibles to carbon dioxide and water vapor. Metals of the platinum family are commonly used catalysts which will promote

oxidation at temperatures between 700 and 900 F.

To use either thermal or catalytic oxidation, the combustible contaminant concentration must be below the lower explosive limit. Equipment specifically designed for control of gaseous or vapor contaminants should be applied with caution when the air stream also contains solid particles. Solid particulates can plug absorbers, adsorbers, and catalysts and, if noncombustible, will not be converted in thermal oxidizers and direct combustors.

Air streams containing both solid particles and gaseous contaminants may require appropriate control devices in series.

4.7 UNIT COLLECTORS

Unit collector is a term usually applied to small fabric collectors having capacities in the 200–2000 cfm range. They have integral air movers, feature small space requirements and simplicity of installation. In most applications cleaned air is recirculated, although discharge ducts may be used if the added resistance is within the capability of the air mover. One of the primary advantages of unit collectors is a reduction in the amount of duct required, as opposed to central systems, and the addition of discharge ducts to unit collectors negates that advantage.

When cleaned air is to be recirculated, a number of precautions are required (see Chapter 7).

Unit collectors are used extensively to fill the need for dust collection from isolated, portable, intermittently used or frequently relocated dust producing operations. Typically, a single collector serves a single dust source with the energy saving advantage that the collector must operate only when that particular dust producing machine is in operation.

Figure 4-15 shows a typical unit collector. Usually they are the intermittent-duty, shaker-type in envelope configuration. Woven fabric is nearly always used. Automatic fabric cleaning is preferred. Manual methods without careful scheduling and supervision are unreliable.

4.8 DUST COLLECTING EQUIPMENT COST

The variations in equipment cost, especially on an installed basis, are difficult to estimate. Comparisons can be misleading if these factors are not carefully evaluated.

4.8.1 Price Versus Capacity: All dust collector prices per cfm of gas will vary with the gas flow rate. The smaller the flow rate, the higher the cost per cfm. The break point, where price per cfm cleaned tends to level off, will vary with the design. See the typical curves shown on Figure 4-16.

4.8.2 Accessories Included: Careful analysis of components of equipment included is very important. Some collector designs include exhaust fan, motor, drive, and starter. In other designs, these items and their supporting structure must be obtained by the purchaser from other sources. Likewise, while

Table 4-3. Dust Collector Selection Guide

Collector Types Used in Industry								
Operation	Concen- tration Note 1	Particle Sizes Note w	Dry Cen- trifugal Collector	Wet Collector	Fabric Collector	Low-Volt Electro- static	Hi-Volt Electro- static	See Remark No.
CERAMICS								
a. Raw product handling	light	fine	S	O	O	N	N	1
b. Fettling	light	fine- medium	S	S	O	N	N	2
c. Refractory sizing	heavy	coarse	N	S	O	N	N	3
d. Glaze & vitr. enamel spray	moderate	medium	N	O	O	N	N	
CHEMICALS								
a. Material handling	light- moderate	fine- medium	S	O	O	N	N	49
b. Crushing, grinding	moderate- heavy	fine- coarse	O	S	O	N	N	4
c. Pneumatic conveying	very heavy	fine- coarse	O	S	O	N	N	5
d. Roasters, kilns, coolers	heavy	mid- coarse	O	O	O	N	N	6
COAL, MINING AND POWER PLANT								
a. Material handling	moderate	medium	O	S	O	N	N	7
b. Bunker ventilation	light	fine	S	S	O	N	N	8
c. Dedusting, air cleaning	heavy	medium- coarse	S	O	O	N	N	9
d. Drying	moderate	fine	N	O	O	N	N	10
FLY ASH								
a. Coal burning--chain grate	light	fine	S	S	O	N	O	11
b. Coal burning--stoker fired	moderate	fine- coarse	S	S	O	N	O	12
c. Coal burning--pulverized fuel	moderate	fine	S	S	O	N	O	13
d. Wood burning	varies	coarse	S	S	O	N	S	14
FOUNDRY								
a. Shakeout	light- moderate	fine	N	O	O	N	N	15
b. Sand handling	moderate	fine- medium	N	O	O	N	N	16
c. Tumbling mills	heavy	medium- coarse	N	S	O	N	N	17
d. Abrasive cleaning	moderate- heavy	fine- medium	N	S	O	N	N	18
GRAIN ELEVATOR, FLOUR AND FEED MILLS								
a. Grain handling	light	medium	O	S	O	N	N	49
b. Grain dryers	light	coarse	S	S	O	N	N	19
c. Flour dust	moderate	medium	O	S	O	N	N	20
d. Feed mill	moderate	medium	O	S	O	N	N	21
METAL MELTING								
a. Steel blast furnace	heavy	varied	N	O	S	N	S	22
b. Steel open hearth	moderate	fine- coarse	N	O	S	N	S	23
c. Steel electric furnace	light	fine	N	S	O	N	S	24
d. Ferrous cupola	moderate	varied	N	O	O	N		25
e. Non-ferrous reverberatory	varied	fine	N	S	O	N	N	26
f. Non-ferrous crucible	light	fine	N	S	O	N	N	27
METAL MINING AND ROCK PRODUCTS								
a. Material handling	moderate	fine- medium	N	O	O	N	N	28
b. Dryers, kilns	moderate	medium- coarse	O	O	O	N	O	29
c. Rock dryer	moderate	fine- medium	N	S	S	N	S	30
d. Cement kiln	heavy	fine- medium	N	N	O	N	S	31
								32

Collector Types Used in Industry								
Operation	Concentration Note 1	Particle Sizes Note w	Dry Cen- trifugal Collector	Wet Collector	Fabric Collector	Low-Volt Electro- static	Hi-Volt Electro- static	See Remark No.
e. Cement grinding	moderate	fine	N	N	O	N	N	33
f. Cement clinker cooler	moderate	coarse	O	N	O	N	N	34
METAL WORKING								49
a. Production grinding, scratch brushing, abrasive cut off	light	coarse	O	O	O	N	N	35
b. Portable and swing frame	light	medium	S	O	O	N	N	
c. Buffing	light	varied	S	O	O	N	N	36
d. Tool room	light	fine	S	S	S	N	N	37
e. Cast iron machining	moderate	varied	O	O	O	S	N	38
PHARMACEUTICAL AND FOOD PRODUCTS								
a. Mixers, grinders, weighing, blending, bagging, packaging	light	medium	O	O	O	N	N	39
b. Coating pans	varied	fine-medium	N	O	O	N	N	40
PLASTICS								49
a. Raw material processing	(See comments under Chemicals)		O	S	O	N	N	41
b. Plastic finishing	light-moderate	varied	S	S	O	N	N	42
c. Extrusion	light	fine	N	S	N	O	N	
RUBBER PRODUCTS								49
a. Mixers	moderate	fine	S	O	S	N	N	43
b. Batchout rolls	light	fine	S	O	S	S	N	
c. Talc dusting and dedusting	moderate	medium	S	S	O	N	N	44
d. Grinding	moderate	coarse	O	O	O	N	N	45
WOODWORKING								49
a. Woodworking machines	moderate	varied	O	S	O	N	N	46
b. Sanding	moderate	fine	S	S	O	N	N	47
c. Waste conveying, hogs	heavy	varied	O	S	S	N	N	48

Note 1: Light: less than 2 gr/ft³; Moderate: 2 to 5 gr/ft³; Heavy: 5 gr/ft³ and up.

Note 2: Fine: 50% less than 5 microns; Medium: 50% 5 to 15 microns; Coarse: 50% 15 microns and larger.

Note 3: O = often; S = seldom; N = never.

Remarks Referred to in Table 4-3

- Dust released from bin filling, conveying, weighing, mixing, pressing forming. Refractory products, dry pan and screen operations more severe.
- Operations found in vitreous enameling, wall and floor tile, pottery.
- Grinding wheel or abrasive cut-off operation. Dust abrasive.
- Operations include conveying, elevating, mixing, screening, weighing, packaging. Category covers so many different materials that recommendation will vary widely.
- Cyclone and high efficiency centrifugals often act as primary collectors followed by fabric or wet type.
- Cyclones used as product collector followed by fabric arrester for high over-all collection efficiency.
- Dust concentration determines need for dry centrifugal; plant location, product value determines need for final collectors. High temperatures are usual and corrosive gases not unusual.
- Conveying, screening, crushing, unloading.
- Remove from other dust producing points. Separate collector usually.
- Heavy loading suggests final high efficiency collector for all except very remote locations.
- Difficult problem but collectors will be used more frequently with air pollution emphasis.
- Public nuisance from boiler blow-down indicates collectors are needed.
- Large installations in residential areas require electrostatic in addition to dry centrifugal.
- Cyclones used as spark arresters in front of fabric collectors.
- Hot gases and steam usually involved.
- Steam from hot sand, adhesive clay bond involved.
- Concentration very heavy at start of cycle.
- Heaviest load from airless blasting due to higher cleaning speed. Abrasive shattering greater with sand than with grit or shot. Amounts removed greater with sand castings, less with forging scale removal, least when welding scale is removed.
- Operations such as car unloading, conveying, weighing, storing.
- Collection equipment expensive but public nuisance complaints becoming more frequent.
- Operations include conveyors, cleaning rolls, sifters, purifiers, bins and packaging.

Remarks Referred to in Table 4-3 (continued)

- | | |
|---|--|
| <p>22. Operations include conveyors, bins, hammer mills, mixers, feeders and baggers.</p> <p>23. Primary dry trap and wet scrubbing usual. Electrostatic is added where maximum cleaning required.</p> <p>24. Use of this technique declining.</p> <p>25. Air pollution standards will probably require increased usage of fabric arresters.</p> <p>26. CAUTION! Recent design improvements such as coke-less, plasma-fired type, have altered emission characteristics.</p> <p>27. Zinc oxide loading heavy during zinc additions. Stack temperatures high.</p> <p>28. Zinc oxide plume can be troublesome in certain plant locations.</p> <p>29. Crushing, screening, conveying involved. Wet ores often introduce water vapor in exhaust air.</p> <p>30. Dry centrifugals used as primary collectors, followed by final cleaner.</p> <p>31. Industry is aggressively seeking commercial uses for fines.</p> <p>32. Collectors usually permit salvage of material and also reduce nuisance from settled dust in plant area.</p> <p>33. Salvage value of collected material high. Same equipment used on raw grinding before calcining.</p> <p>34. Coarse abrasive particles readily removed in primary collector types.</p> <p>35. Roof discoloration, deposition on autos can occur with cyclones and less frequently with high efficiency dry centrifugal. Heavy duty air filters sometimes used as final cleaners.</p> <p>36. Linty particles and sticky buffing compounds can cause pluggage and fire hazard in dry collectors.</p> | <p>37. Unit collectors extensively used, especially for isolated machine tools.</p> <p>38. Dust ranges from chips to fine floats including graphitic carbon. Low voltage ESP applicable only when a coolant is used.</p> <p>39. Materials vary widely. Collector selection depends on salvage value, toxicity, sanitation yardsticks.</p> <p>40. Controlled temperature and humidity of supply air to coating pans makes recirculation desirable.</p> <p>41. Plastic manufacture allied to chemical industry and varies with operations involved.</p> <p>42. Operations and collector selection similar to woodworking. See Item 13.</p> <p>43. Concentration is heavy during feed operation. Carbon black and other fine additions make collection and dust-free disposal difficult.</p> <p>44. Salvage of collected material often dictates type of high efficiency collector.</p> <p>45. Fire hazard from some operations must be considered.</p> <p>46. Bulking material. Collected material storage and bridging from splinters and chips can be a problem.</p> <p>47. Dry centrifugals not effective on heavy concentration of fine particles from production sanding.</p> <p>48. Dry centrifugal collectors required. Wet or fabric collectors may be used for final collectors.</p> <p>49. See NFPA publications for fire hazards, e.g., zirconium, magnesium, aluminum, woodworking, plastics, etc.</p> |
|---|--|

dust storage hoppers are integral parts of some dust collector designs, they are not provided in other types. Duct connections between elements may be included or omitted. Recirculating water pumps and/or settling tanks may be required but not included in the equipment price.

4.8.3 Installation Cost: The cost of installation can equal or exceed the cost of the collector. Actual cost will depend on the method of shipment (completely assembled, sub-assembled or completely knocked down), the location (which may require expensive rigging), and the need for expensive supporting steel and access platforms. Factory installed media will reduce installation cost. The cost can also be measurably influenced by the need for water and drain connections, special or extensive electrical work, and expensive material handling equipment for collection material disposal. Items in the latter group will often also be variable, decreasing in cost per cfm as the flow rate of gas to be cleaned increases.

4.8.4 Special Construction: Prices shown in any tabulation must necessarily assume standard or basic construction. The increase in cost for corrosion resisting material, special high-temperature fabrics, insulation, and/or weather protection for outdoor installations can introduce a multiplier of one to four times the standard cost.

A general idea of relative dust collector cost is provided in Figure 4-16. The additional notes and explanations included

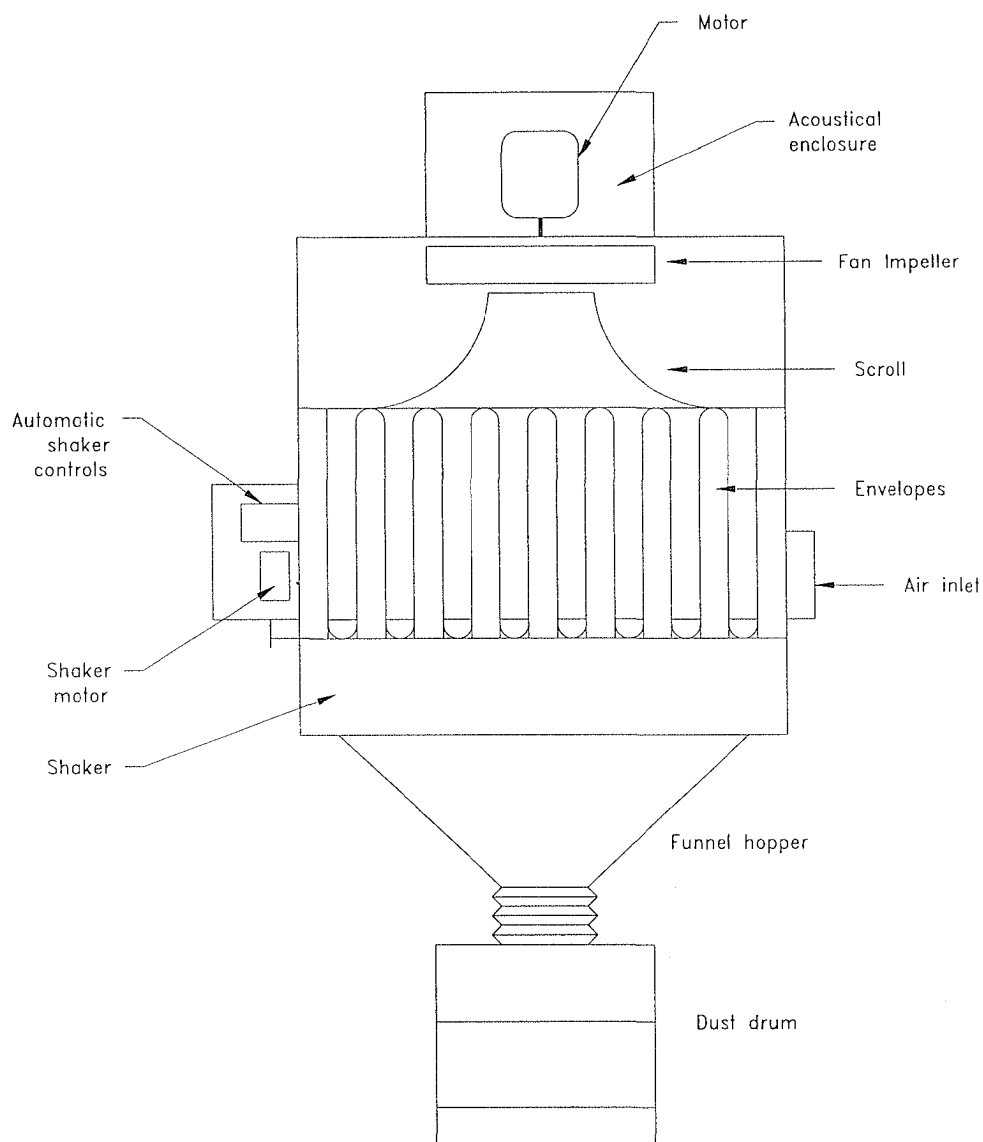
in these data should be carefully examined before they are used for estimating the cost of specific installations. For more accurate data, the equipment manufacturer or installer should be asked to provide estimates or a past history record for similar control problems utilized. Table 4-4 lists other characteristics that must be evaluated along with equipment cost.

Price estimates included in Figure 4-16 are for equipment of standard construction in normal arrangement. Estimates for exhausters and dust storage hoppers have been included, as indicated in Notes 1 and 2, where they are normally furnished by others.

4.9 SELECTION OF AIR FILTRATION EQUIPMENT

Air filtration equipment is available in a wide variety of designs and capability. Performance ranges from a simple throwaway filter for the home furnace to the "clean room" in the electronics industry, where the air must be a thousand times as clean as in a hospital surgical suite. Selection is based on efficiency, dust holding capacity, and pressure drop. There are five basic methods of air filtration.

4.9.1 Straining: Straining occurs when a particle is larger than the opening between fibers and cannot pass through. It is a very ineffective method of filtration because the vast majority of particles are far smaller than the spaces between fibers. Straining will remove lint, hair, and other large parti-



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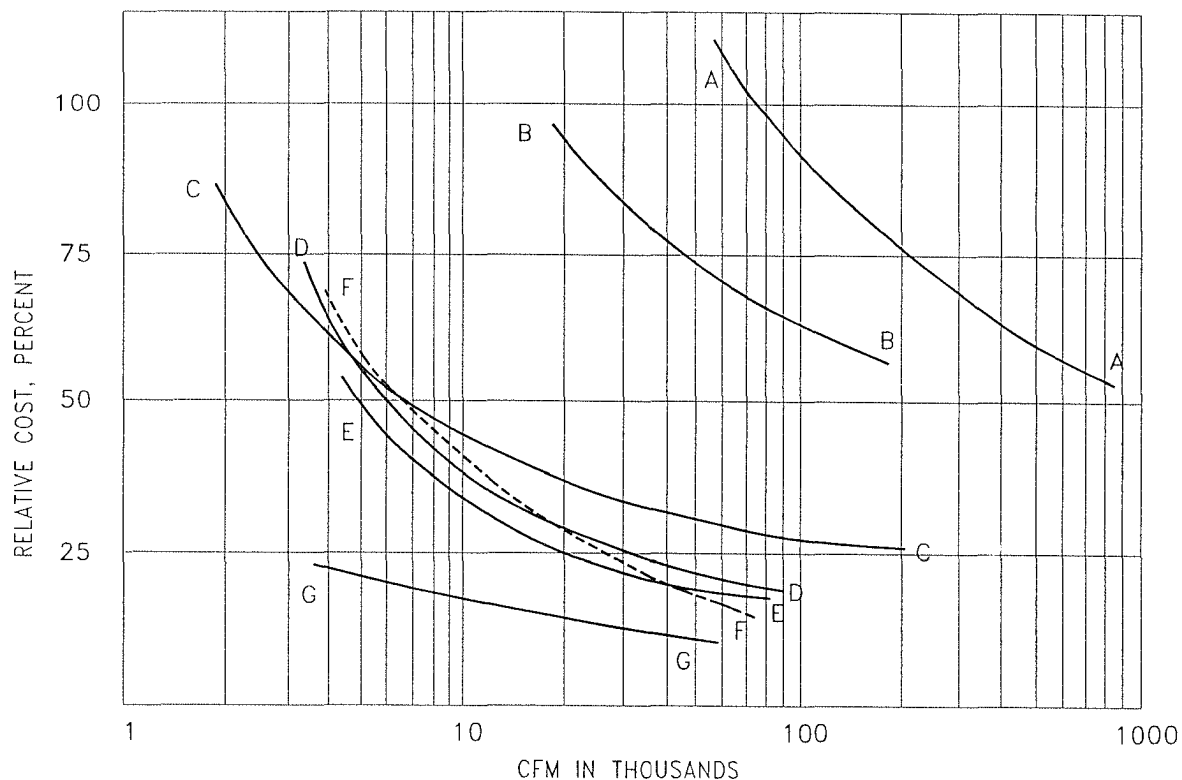
*UNIT COLLECTOR
(FABRIC-SHAKER TYPE)*

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FIGURE

4-15



- A. High voltage precipitator (minimum cost range)
- B. Continuous duty high temperature fabric collector (2.0:1)
- C. Continuous duty reverse pulse (8:1)
- D. Wet collector
- E. Intermittent duty fabric collector (2.0:1)
- F. Low voltage precipitator
- G. Cyclone

Note 1: Cost based on collector section only. Does not include ducts, dust disposal devices, pumps, exhausters or other accessories not an integral part of the collector.

Note 2: Price of high voltage precipitator will vary substantially with applications and efficiency requirements. Costs shown are for fly ash applications where velocities of 200 to 300 fpm are normal.

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*COST ESTIMATES OF
DUST COLLECTING EQUIPMENT*

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FIGURE 4-16

TABLE 4-4. Comparison of Some Important Dust Collector Characteristics

Type	Higher efficiency Range on Particles Greater than Mean Size in Microns	Pressure Loss Inches	H ₂ O Gal per 1,000 cfm	Space	Sensitivity to cfm Change		Humid Air Influence	Max. Temp. F Standard Construction Note 4
					Pressure	Efficiency		
Electrostatic	0.25	½	—	Large	Negligible	Yes	Improves Efficiency	500
Fabric								
Intermittent—Shaker	0.25	3-6	—	Large	As cfm	Negligible	May Make Reconditioning Difficult	See Table 4-1
Continuous—Shaker	0.25	3-6	—	Large	As cfm	Negligible		
Continuous—Reverse Air	0.25	3-6	—	Large	As cfm	Negligible		
Continuous—Reverse Pulse	0.25	3-6	—	Moderate	As cfm	Negligible		
Glass, Reverse flow	0.25	3-8	—	Large	As cfm	Negligible		550
Wet:								
Packed Tower	1-5	1.5-3.5	5-10	Large	As cfm	Yes	None	Unlimited
Wet Centrifugal	1-5	2.5-6	3-5	Moderate	As (cfm) ²	Yes		
Wet Dynamic	1-2	Note 2	½ to 1	Small	Note 2	No		
Orifice Types	1-5	2½-6	10-40	Small	As cfm or less	Varies with design		
Higher Efficiency:								
Fog Tower	0.5-5	2-4	5-10	Moderate	As (cfm) ²	Slightly	None	Note 3
Venturi	0.5-2	10-100	5-15	Moderate	As (cfm) ²	Yes		Unlimited
Dry Centrifugal:								
Low Pressure Cyclone	20-40	0.75-1.5	—	Large	As (cfm) ²	Yes	May Cause condensation and plugging	750
High Eff. Centrifugal	10-30	3-6	—	Moderate	As (cfm) ²	Yes		750
Dry Dynamic	10-20	Note 2	—	Small	Note 2	No		

Note 1: Pressure loss is that for fabric and dust cake. Pressure losses associated with outlet connections to be added by system designer.

Note 2: A function of the mechanical efficiency of these combined exhausters and dust collectors.

Note 3: Precooling of high temperature gases will be necessary to prevent rapid evaporation of fine droplets.

Note 4: See NFPA requirements for fire hazards, e.g., zirconium, magnesium, aluminum, woodworking, etc.

cles.

4.9.2 Impingement: When air flows through a filter, it changes direction as it passes around each fiber. Larger dust particles, however, cannot follow the abrupt changes in direction because of their inertia. As a result, they do not follow the air stream and collide with a fiber. Filters using this method are often coated with an adhesive to help fibers retain the dust particles that impinge on them.

4.9.3 Interception: Interception is a special case of impingement where a particle is small enough to move with the air stream but, because its size is very small in relation to the fiber, makes contact with a fiber while following the tortuous air flow path of the filter. The contact is not dependent on inertia and the particle is retained on the fiber because of the inherent adhesive forces that exist between the particle and fiber. These forces, called van der Waals (J. D. van der Waals, 1837–1923) forces, enable a fiber to trap a particle without the use of inertia.

4.9.4 Diffusion: Diffusion takes place on particles so small that their direction and velocity are influenced by molecular collisions. These particles do not follow the air stream, but behave more like gases than particulate. They move across the direction of air flow in a random fashion. When a particle does strike a fiber, it is retained by the van der Waals forces existing between the particle and the fiber. Diffusion is the primary mechanism used by most extremely efficient filters.

4.9.5 Electrostatic: A charged dust particle will be attracted to a surface of opposite electrical polarity. Most dust particles are not electrically neutral; therefore, electrostatic attraction between dust particle and filter fiber aids the collection efficiency of all barrier-type air filters. Electrostatic filters establish an ionization field to charge dust particles so that they can be collected on a surface that is grounded or of opposite polarity. This concept was previously discussed in Section 4.3.1.

Table 4-5 shows performance versus filter fiber size for several filters. Note that efficiency increases as fiber diameter decreases because more small fibers are used per unit volume.

Note also that low velocities are used for high-efficiency filtration by diffusion.

The wide range in performance of air filters makes it necessary to use more than one method of efficiency testing. The industry-accepted methods in the United States are ASHRAE Arrestance, ASHRAE Efficiency, and DOP. For ASHRAE Arrestance, a measured quantity of 72% standardized air cleaner test dust, 23% carbon black, and 5% cotton lint is fed to the filter. The efficiency by weight on this specific test dust is the ASHRAE Arrestance. ASHRAE Efficiency is a measure of the ability of a filter to prevent staining or discoloration. It is determined by light reflectance readings taken before and after the filter in a specified test apparatus. Atmospheric dust is used for the test. Both ASHRAE tests are described in ASHRAE Publication 52-76.^(4,4)

In a DOP Test, 0.3 micron particles of dioctylphthalate (DOP) are drawn through a HEPA (High Efficiency Particulate Air) filter. Efficiency is determined by comparing the downstream and upstream particle counts. To be designated as a HEPA filter, the filter must be at least 99.97% efficient, i.e., only three particles of 0.3 micron size can pass for every ten thousand particles fed to the filter. Unlike both ASHRAE tests, the DOP test is not destructive, so it is possible to repair leaks and retest a filter that has failed.

The three tests are not directly comparable; however, Figure 4-17 shows the general relationship. Table 4-6 compares several important characteristics of commonly used air filters. Considerable life extension of an expensive final filter can be obtained by the use of one or more cheaper, less efficient, prefilters. For example, the life of a HEPA filter can be increased 25% with a throwaway prefilter. If the throwaway filter is followed by a 90% efficient extended surface filter, the life of the HEPA filter can be extended nearly 900%. This concept of "progressive filtration" allows the final filters in clean rooms to remain in place for ten years or more.

The European Committee on the Construction of Air Handling Equipment has developed a method for testing air filters in general ventilation. Although their method, called Eurovent 4/5, is based directly on ASHRAE Standard 52-76, some wording and definitions have been amended to suit the needs of Eurovent. Eurovent 4/5 aims to establish a uniform

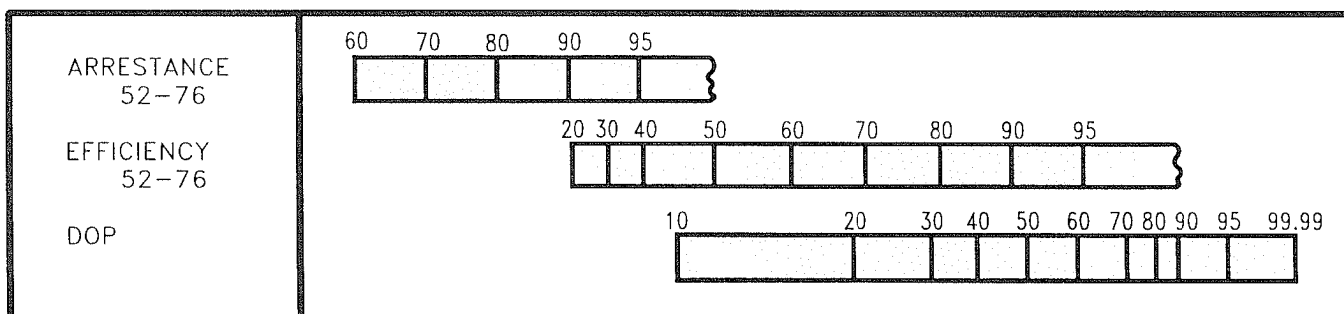


FIGURE 4-17. Comparison between various methods of measuring air cleaning capability.

comparative testing procedure for air filters having volumetric flow rates greater than 0.236 m³/s (500 cfm) and an average dust spot efficiency up to 98%.

The wide range of filter efficiency is segregated into 14 grades of filters from EU1 to EU14.

4.10 RADIOACTIVE AND HIGH TOXICITY OPERATIONS

There are three major requirements for air cleaning equipment to be utilized for radioactive or high toxicity applications:

1. High efficiency
2. Low maintenance
3. Safe disposal

High efficiency is essential because of extremely low tolerances for the quantity and concentration of stack effluent and the high cost of the materials handled. Not only must the efficiency be high, it must also be verifiable because of the legal requirement to account for all radioactive material.

The need for low maintenance is of special importance when exhausting any hazardous material. For many radioactive processes, the changing of bags in a conventional fabric collector may expend the daily radiation tolerances of 20 or more persons. Infrequent, simple, and rapid maintenance requirements are vital. Another important factor is the desirability of low residual buildup of material in the collector since dose rates increase with the amount of material and reduce the allowable working time.

Disposal of radioactive or toxic materials is a serious and very difficult problem. For example, scalping filters loaded with radioactive dust are usually incinerated to reduce the quantity of material that must be disposed of in special burial grounds. The incinerator will require an air cleaning device, such as a wet collector of very special design, to avoid unacceptable pollution of air and water.

With these factors involved, it is necessary to select an air cleaning device that will meet efficiency requirements without causing too much difficulty in handling and disposal.

Filter units especially designed for high efficiency and low maintenance are available. These units feature quick changeout through a plastic barrier which is intended to encapsulate spent filters, thereby eliminating the exposure of personnel to radioactive or toxic material. A filtration efficiency of 99.97% by particle count on 0.3 micron particles is standard for this type of unit.

For further information on this subject, see Reference 4.5.

4.11 EXPLOSION VENTING

There is a wide range of dusts which are combustible and capable of producing an explosion. Explosions occur when the right concentration of finely divided dust is suspended in air and exposed to a sufficient source of ignition. A dust collector, by its very operation, maintains a cloud of finely

TABLE 4-5. Media Velocity vs. Fiber Size

Filter Type	Filter Size (microns)	Media Velocity (fpm)	Filtration Mechanism
Panel Filters	25-50	250-625	Impingement
Automatic Roll Filters	25-50	500	Impingement
Extended Surface Filters	0.75-2.5	20-25	Interception
HEPA Filters	0.5-6.3	5	Diffusion

divided particles suspended in air. If a source of ignition initiates the combustion of the dust cloud, the gases in the cloud will rapidly expand due to heat developed during the combustion. If a dust collector vessel constructs this expansion, a rapid pressure buildup inside the collector casing will cause a violent rupture. When dust particles are known to be combustible, precautions for an explosion must be taken and suitable protection provided to reduce the risk of property damage and personal injury.

To begin taking precautions, sources of possible ignition must be identified and controlled to minimize the risk of a dust cloud explosion. Usual causes of explosions include static by minimizing the ignition sources such as static discharge, hot surfaces on machinery and sparks and flames from processes. After identifying possible sources of ignition, preventive measures should be taken. Static grounding of the equipment and spark traps are typical preventive measures. The addition of an inert gas to replace oxygen in a dust collector can prevent an explosion by ensuring the minimum oxygen content required for ignition is never reached. Inerting can be very effective in closed loop systems but is not economical in typical local exhaust systems because of the constant loss of expensive inerting gas. Should ignition occur, protective measures must be taken to limit the damage. Typical protective measures include explosion suppression, explosion containment, and explosion venting.

Explosion suppression requires the early detection of an explosion, usually within the first 20 milliseconds. Once ignition is detected, an explosion suppression device injects a pressurized chemical suppressant into the collector to displace the oxygen and impede combustion. These are typically used in conjunction with fast acting isolation valves on the inlet and outlet ducts. These systems can be very useful when toxic dusts are being handled.

Explosion containment uses specialized dust collectors designed to withstand the maximum pressure generated and contain the explosion. Most pressure capabilities of commercially available dust collectors are not sufficient to contain an explosion in progress.

Explosion venting, the most common protection, is afforded by fitting pressure relief vents to the collector housing.

TABLE 4-6. Comparison of Some Important Air Filter Characteristics*

Type	Pressure Drops "wg (Notes 1 & 2)		ASHRAE Performance (Note 4)		Face Velocity fpm	Maintenance (Note 5)	
	Initial	Final	Arrestance	Efficiency		Labor	Material
Low/Medium Efficiency							
1. Glass Throwaway (2" deep)	0.1	0.5	77%	NA Note 6	300	High	High
2. High Velocity (permanent units) (2" deep)	0.1	0.5	73%	NA Note 6	500	High	Low
3. Automatic (viscous)	0.4	0.4	80%	NA Note 6	500	Low	Low
Medium/High Efficiency							
1. Extended Surface (dry)	0.15–0.60	0.5–1.25	90–99%	25–95%	300–625	Medium	Medium
2. Electrostatic:							
a. Dry Agglomerator/ Roll Media	0.35	0.35	NA Note 7	90%	500	Medium	Low
b. Dry Agglomerator/ Extended Surface Media	0.55	1.25	NA Note 7	95%+	530	Medium	Medium
c. Automatic Wash Type	0.25	0.25	NA Note 7	85–95%	400–600	Low	Low
Ultra High Efficiency							
1. HEPA	0.5–1.0	1.0–3.0	Note 3	Note 3	250–500	High	High

Note 1: Pressure drop values shown constitute a range or average, whichever is applicable.

Note 2: Final pressure drop indicates point at which filter or filter media is removed and the media is either cleaned or replaced. All others are cleaned in place, automatically, manually or media renewed automatically. Therefore, pressure drop remains approximately constant.

Note 3: 95-99.97% by particle count, DOP test.

Note 4: ASHRAE Standard 52-76 defines (a) Arrestance as a measure of the ability to remove injected synthetic dust, calculated as a percentage on a weight basis and (b) Efficiency as a measure of the ability to remove atmospheric dust determined on a light-transmission (dust spot) basis.

Note 5: Compared to other types within efficiency category.

Note 6: Too low to be meaningful.

Note 7: Too high to be meaningful.

As pressure increases quickly leading up to an explosion, a relief vent opens to allow the rapidly expanding gases to escape. This effectively limits the maximum pressure build-up to less than the bursting pressure of the vessel. The necessary area for such a relief vent is a function of the vessel volume, vessel strength, the opening pressure of the relief vent and the rate of pressure rise characteristic of the dust in question. Most standard dust collectors will require reinforcing to withstand the reduced maximum pressure experienced during an explosion.

To choose the most reliable, economical and effective means of explosion control, an evaluation of the specifics of the exhaust system and the degree of protection required is necessary.

NAPA 68-1994, Guide for Explosion Venting,(4.6) is the most commonly recognized standard and should be studied

and thoroughly familiar to anyone responsible for the design or evaluation of dust collectors applied to potentially explosive dusts.

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Chapter 5

EXHAUST SYSTEM DESIGN PROCEDURE

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5.1 INTRODUCTION

The duct system that connects the hoods, air cleaning device(s), and fan must be properly designed. This process is much more involved than merely connecting pieces of duct. If the system is not carefully designed in a manner which inherently ensures that the design flow rates will be realized, contaminant control may not be achieved.

The results of the following design procedure will determine the duct sizes, material thickness, and the fan operating point (system flow rate and required pressure) required by the system. Chapter 6 describes how to select a fan based on these results.

5.2 PRELIMINARY STEPS

Coordinate design efforts with all personnel involved, including the equipment or process operator as well as maintenance, health, safety, fire, and environmental personnel. The designer should have, at a minimum, the following data available at the start of the design calculations:

1. A layout of the operations, workroom, building (if necessary), etc. The available location(s) for the air cleaning device and fan should be determined. An important aspect that must be considered at this time is to locate the system exhaust point (where the air exits the system) so that the discharged air will not re-enter the work space, either through openings in the building perimeter or through replacement air unit intakes. (See Figures 5-28 and 5-29.)

2. A line sketch of the duct system layout, including plan and elevation dimensions, fan location, air cleaning device location, etc. Number, letter, or otherwise identify each branch and section of main duct on the line sketch for convenience. The examples show hoods numbered and other points lettered.

Locate the fan close to pieces of equipment with high losses. This will facilitate balancing and may result in lower operating costs.

Flexible duct is susceptible to sagging and excessive bending, which increases static pressure losses. Usually, these additional System Pressure (SP) losses cannot be predicted accurately. Use hard duct whenever possible and keep flexible duct lengths as short as possible.

3. A design or sketch of the desired hood for each operation with direction and elevation of outlet for duct connection.
4. Information about the details of the operation(s), specifically toxicity, ergonomics, physical and chemical characteristics, required flow rate, minimum required duct velocity, entry losses, and required capture velocities.
5. Consider the method and location of the replacement

air distribution devices on the hood's performance. The type and location of these fixtures can dramatically lower contaminant control by creating undesirable turbulence at the hood (see Chapter 7). Perforated plenums or perforated duct provide better replacement air distribution with fewer adverse effects on hood performance.

5.3 DESIGN PROCEDURE

All exhaust systems are comprised of hoods, duct segments, and special fittings leading to an exhaust fan. A complex system is merely an arrangement of several simple exhaust systems connected to a common duct. There are two general classes of duct system designs: tapered systems and plenum systems. The duct in a tapered system gradually gets larger as additional flows are merged together, thus keeping duct velocities nearly constant. If the system transports particulate (dust, mist, or condensable vapors), the tapered system maintains the minimum velocity required to prevent settling. The duct in a plenum system (see Section 5.7) is generally larger than that in a tapered system, and the velocity in it is usually low. Any particulate in the air stream can settle out in the large ducts. Figures 5-4 and 5-5 illustrate design alternatives. Regardless of which system is used, the following procedure will result in a workable system design.

1. Select or design each exhaust hood based on the toxicity, physical, and chemical characteristics of the material and the ergonomics of the process and determine its design flow rate, minimum duct velocity, and entry losses (see Chapters 3 and 10). Note that minimum duct velocity is only important for systems transporting particulate, condensing vapors, or mist and to prevent explosive concentrations building up in the duct (see Section 5.18 for a discussion on economic velocities for non-particulate systems).
2. Start with the duct segment that has the greatest number of duct segments between it and the fan. A duct segment is defined as the constant diameter round (or constant area rectangular) duct that separates points of interest such as hoods, entry points, fan inlet, etc.
3. Determine the duct area by dividing the design flow rate by the minimum duct velocity. Convert the resultant cross-sectional area into a tentative duct diameter. A commercially available duct size (see Table 5-8) should be selected. If solid particulates or condensable vapors are being transported through the system, a minimum velocity is required (see Chapters 3 and 10). If the tentative duct diameter is not a standard size, select the next smaller size to ensure that the actual duct velocity is equal to or greater than the minimum required.
4. Using the line sketch, determine the design length for each duct segment and the number and type of fittings

(elbows, entries, and other special fittings) needed. Design length is the centerline distance along the duct (the distance between the intersection of the centerlines of the straight duct components).

5. Calculate the pressure losses for the duct segments that merge at a common junction point. (See Section 5.4 for the details on how to calculate these losses.)
6. Directly after each junction point, there must be one and only one SP, *regardless of the path taken to reach that point*. If not ensured by the design process, the system will "self-balance" by reducing the flow rate in the higher-resistance duct segment(s) and increasing the flow rate in the lower-resistance duct segment(s) until there is a single SP in the duct downstream of each junction point.

SP balance at any junction point can be achieved by either one of two fundamental design methods: 1) Adjust the flow rate through the hood(s) until the SPs at each junction point are the same. 2) Increase the resistance in the low resistance duct segment(s) by means of some artificial device such as a blast gate, orifice plate, or other obstruction in the segment.

Section 5.5 discusses the details of these procedures.

7. Select both the air cleaning device and fan based upon final calculated system flow rate, temperature, moisture

condition, contaminant loading, physical and chemical characteristics, and overall system resistance.

8. Check the duct sizes designed against the available space and resolve any interference problems. (For example, will the elbow size desired actually fit in the available space?) This may cause a redesign of part of the system.
9. Determine the material type and thickness (gauge) for each duct segment based on the air stream characteristics.

5.4 DUCT SEGMENT CALCULATIONS

The Velocity Pressure (VP) Method is based on the fact that all frictional and dynamic (fitting) losses in ducts and hoods are functions of the velocity pressure and can be calculated by a loss coefficient multiplied by the velocity pressure. Loss coefficients for hoods, straight ducts, elbows, branch entries, contractions, and expansions are shown in Figures 5-13 through 5-16. Figure 5-1 shows the application of these coefficients. For convenience, loss coefficients for round elbows and entries are also presented on the calculation sheet (see Figure 5-3).

Friction data for this method are presented as Tables 5-5 and 5-6. These tables give the loss coefficients per foot of galvanized and commercial steel, aluminum, PVC, and stainless steel duct. The equations for these tables are listed on

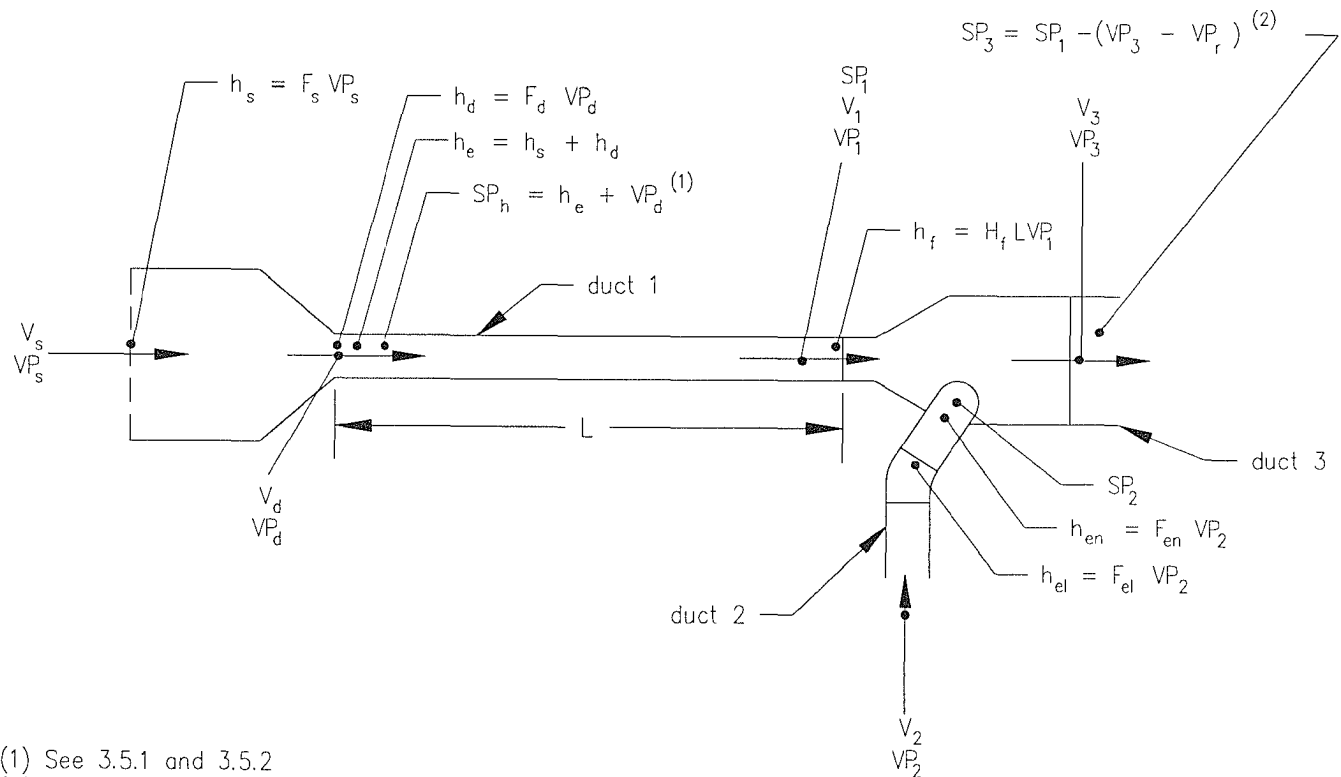


FIGURE 5-1. System duct calculation parameter location

these tables and also on the calculation sheet (see Figure 5-3).^(5.1) These equations and the resultant tables have been designed to be no more than 4% different from the "exact" values of the Colebrook-White equation and were designed to err on the high side of the normal velocity range of exhaust ventilation systems.

For convenience, two data sets determined from the same equations were used to generate the friction tables. These tables are possible because, for a specific diameter, the friction loss coefficient changes only slightly with velocity. Each table lists the friction coefficient as a function of diameter for six different velocities. The error in using these data with velocities plus or minus 1000 fpm is within 6%. If desired, a linear interpolation between velocity values can be performed.

In Chapter 1, an equation was presented for flexible duct with the wires covered. No data are presented here for this type of material due to the wide variability from manufacturer to manufacturer. Perhaps an even more important reason is that these data are for straight duct losses, and flexible duct, by its very nature, is seldom straight. Typically, bends in flexible duct can produce extremely large losses which cannot be predicted easily. Be very careful to keep the flexible duct as straight and as short as possible.

The following steps will establish the overall pressure loss of a duct segment that starts at a hood. Figure 5-2 shows a simple one-hood ventilation system. The use of a calculation sheet can be very beneficial when performing the calculations manually. Figure 5-3 shows the details of the calculations for each component of the system. There is also a profile through the system showing the magnitude and relationships of total, static, and velocity pressures on both the "suction" and the "pressure" sides of the fan on Figure 5-2. It should be noted that VP is always positive. Also, while total and static pressure may be either negative or positive with respect to atmospheric pressure, Total Pressure (TP) is always greater than SP ($TP = SP + VP$).

NOTE: The numbers in the problems presented in this chapter were generated using one of the available computer programs (see Section 5.6). The values presented in the calculation sheets may be different from those determined by other methods.

1. Determine the *actual* velocity by dividing the flow rate by the area of the commercial duct size chosen. Then determine the corresponding velocity pressure from Table 5-7 or the equations in Chapter 1. In the example, the diameter chosen was 4" (line 5), the actual velocity is given on line 7 and the VP corresponding to this actual velocity is given on line 8.
2. Determine the hood static pressure from the equations in Chapter 3. In this example, there are no slots, so the duct entry loss is as given on lines 17 through 22.

3. Multiply the design duct length by the loss coefficient from the tabulated data of Tables 5-5 or 5-6 (lines 23 through 25.) The use of galvanized sheet metal duct was assumed throughout this chapter.
4. Determine the number and type of fittings in the duct segment. For each fitting type (see Figures 5-13, 5-14, 5-15, and 5-16), determine the loss coefficient and multiply by the number of fittings (there were none in this example.)
5. Add the results of Steps 3 and 4 above and multiply by the duct VP. This is the actual loss in inches of water for the duct segment (given on line 34).
6. Add the result of Step 5 to the hood suction. If there are any additional losses (expressed in inches of water), such as for an air cleaning device, add them in also. This establishes the cumulative energy required, expressed as static pressure, to move the design flow rate through the duct segment (line 37). Note that the value on line 37 is negative.

The calculations listed in the last three columns of Figure 5-3 will be discussed in Section 5.8.3.

5.5 DISTRIBUTION OF AIR FLOW

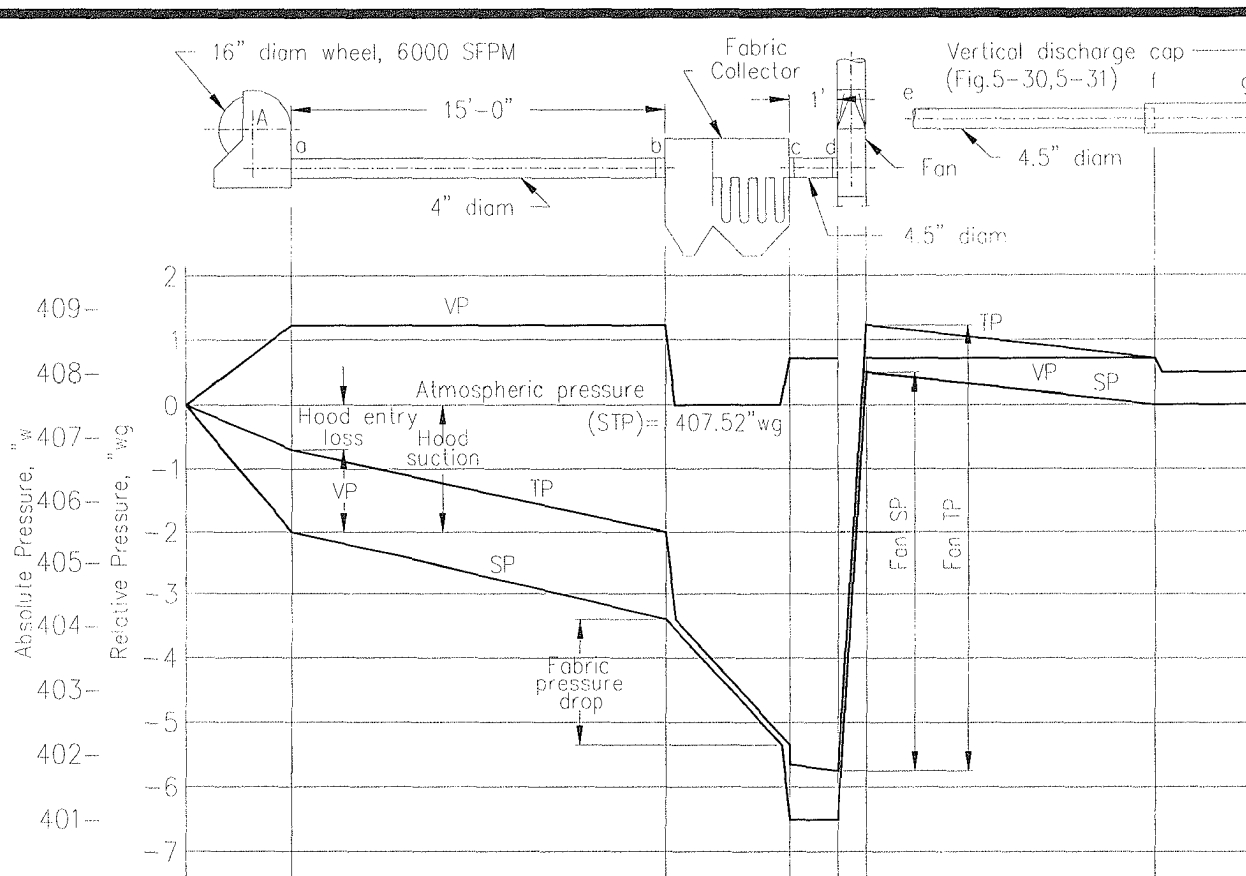
As discussed previously, a complex exhaust system is actually a group of simple exhaust systems connected to a common main duct. Therefore, when designing a system of multiple hoods and branches, the same rules apply. In a multiple branch system, however, it is necessary to provide a means of distributing air flow between the branches either by balanced design or by the use of blast gates.

Air will always take the path of least resistance. A natural balance at each junction will occur; that is, the exhaust flow rate will distribute itself automatically according to the pressure losses of the available flow paths. The designer must provide distribution such that the design air flow at each hood will never fall below the minimums listed in Chapter 3 and/or Chapter 10. To do so, the designer must make sure that all flow paths (ducts) entering a junction will have equal calculated static pressure requirements.

To accomplish this, the designer has a choice of two methods. The object of both methods is the same: to obtain the desired flow rate at each hood in the system while maintaining the desired velocity in each branch and main.

The two methods, labeled Balance by Design Method and Blast Gate Method, are outlined below. Their relative advantages and disadvantages can be found in Table 5-1.

5.5.1 Balance by Design Method: This procedure (see Section 5.10) provides for achievement of desired air flow (a "balanced" system) without the use of blast gates. It is often called the "Static Pressure Balance Method." In this type of design, the calculation usually begins at the hood farthest from



Details of Operation

NO.	HOOD NO.	VS-PRINT	REQUIRED AIR-FLOW, cfm
1	16" Diameter Grinding wheel, 2" Wide	A	80-11
			390

Dimensions

No. of Branch or Main	Straight Run, Ft	CFM Required	Elbows	Entries
ab	15	390	---	---
bc	Collector	390	---	---
cd	1	390	---	---
ef	10	390	---	---
fg	Stack Head	390	---	---

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FIGURE

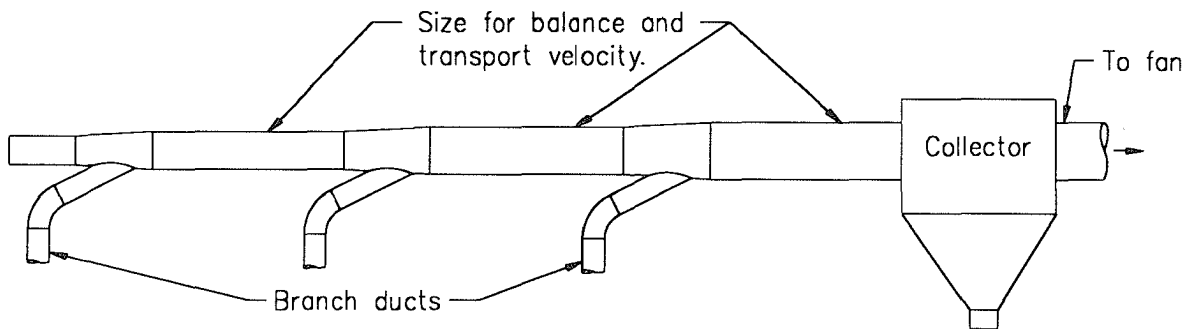
5-2

VELOCITY PRESSURE METHOD CALCULATION SHEET

 Problem #1 Class Designer Date

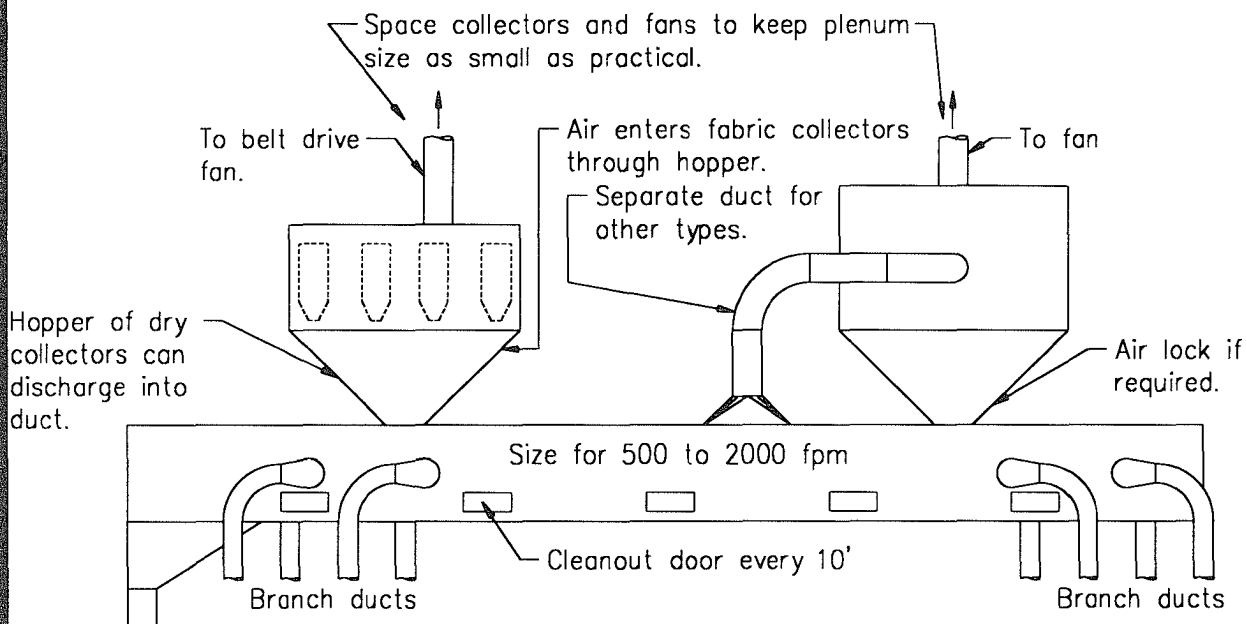
1	Duct Segment Identification		a-b	b-c	c-d	e-f	
2	Target Volumetric Flow Rate		cfm	390	390	390	390
3	Minimum Transport Velocity		fpm	4000			
4	Maximum Duct Diameter		inches	4.23		4.5	4.5
5	Selected Duct Diameter		inches	4		4.5	4.5
6	Duct Area		ft ²	0.0873		0.1104	0.1104
7	Actual Duct Velocity		fpm	4469		3531	3531
8	Duct Velocity Pressure		"wg	1.25		0.78	0.78
9	H O O D S U C T I O N	Maximum Slot Area	ft ²				
10		Slot Area Selected	ft ²				
11		Slot Velocity	fpm				
12		Slot Velocity Pressure	"wg				
13		Slot Loss Coefficient					
14		Acceleration Factor	(0 or 1)				
15		Slot Loss per VP	(13 + 14)				
16		Slot Static Pressure	(12 x 15)				
17		Duct Entry Loss Coefficient	(Fig. 5-13)	.65		.5	
18		Acceleration Factor	(1 or 0)	1		1	
19	Duct Entry Loss per VP	(17 + 18)	1.65		1.5		
20	Duct Entry Loss	(8 x 19)	"wg	2.05		1.17	
21	Other Losses	"wg		2.0			
22	Hood Static Pressure	(16 + 20 + 21)	"wg	2.05	2.0	1.17	
23	Straight Duct Length		feet	15		1	10
24	Friction Factor (H _f)			0.0703		0.0620	0.0620
25	Friction Loss per VP		(23 x 24)	1.05		0.06	0.62
26	Number of 90 deg. Elbows						
27	Elbow Loss Coefficient						
28	Elbow Loss per VP		(26 x 27)				
29	Number of Branch Entries		(1 or 0)				
30	Entry Loss Coefficient						
31	Branch Entry Loss per VP		(29 x 30)				
32	Special Fitting Loss Coefficients						
33	Duct Loss per VP		(25 + 28 + 31 + 32)	1.05		0.06	0.62
34	Duct Loss		(33 x 8)	"wg	1.31	0.05	0.48
35	Duct Segment Static Pressure Loss		(22 + 34)	"wg	3.36	2.0	1.22
36	Other Losses (VP-VP _n , etc.)		"wg				
37	Cumulative Static Pressure		"wg	-3.36	-5.36	-6.58	+0.48
38	Governing Static Pressure		"wg				
39	Corrected Volumetric Flow Rate		cfm				
40	Corrected Velocity		fpm				
41	Corrected Velocity Pressure		"wg				
42	Resultant Velocity Pressure		"wg				

Temperature _____ Elevation _____						Remarks:																			
						Pertinent Information From Chapter 5																			
						$Q_{\text{corr}} = Q_{\text{design}} \sqrt{\frac{SP_{\text{gov}}}{SP_{\text{duct}}}}$																			
						$VP_r = \frac{Q_1}{Q_3} VP_1 + \frac{Q_2}{Q_3} VP_2$																			
						Straight Duct Friction Loss $H_f = a \frac{V^b}{Q^c}$																			
						<table border="1" style="width: 100%; border-collapse: collapse;"> <tr> <th style="width: 60%;">Duct Material</th> <th style="width: 10%;">a</th> <th style="width: 10%;">b</th> <th style="width: 10%;">c</th> </tr> <tr> <td>Galvanized</td> <td>0.0307</td> <td>0.533</td> <td>0.612</td> </tr> <tr> <td>Black iron, Aluminum, PVC, Stainless steel</td> <td>0.0425</td> <td>0.465</td> <td>0.602</td> </tr> <tr> <td>Flexible (fabric covered wires)</td> <td>0.0311</td> <td>0.604</td> <td>0.639</td> </tr> </table>				Duct Material	a	b	c	Galvanized	0.0307	0.533	0.612	Black iron, Aluminum, PVC, Stainless steel	0.0425	0.465	0.602	Flexible (fabric covered wires)	0.0311	0.604	0.639
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Flexible (fabric covered wires)	0.0311	0.604	0.639																						
						Fan Static Pressure $\text{FAN SP} = SP_{\text{out}} - SP_{\text{in}} - VP_{\text{in}}$																			
						Branch Entry Loss Coefficients																			
						<table border="1" style="width: 100%; border-collapse: collapse;"> <tr> <th style="width: 30%;">Angle</th> <th style="width: 70%;">Loss Coefficients</th> </tr> <tr> <td>15°</td> <td>0.09</td> </tr> <tr> <td>30°</td> <td>0.18</td> </tr> <tr> <td>45°</td> <td>0.28</td> </tr> </table>				Angle	Loss Coefficients	15°	0.09	30°	0.18	45°	0.28								
						Angle	Loss Coefficients																		
						15°	0.09																		
30°	0.18																								
45°	0.28																								
						90° Round Elbow Loss Coefficients (5 piece)																			
						<table border="1" style="width: 100%; border-collapse: collapse;"> <tr> <th style="width: 30%;">R/D</th> <th style="width: 70%;">Loss Coefficients</th> </tr> <tr> <td>1.5</td> <td>0.24</td> </tr> <tr> <td>2.0</td> <td>0.19</td> </tr> <tr> <td>2.5</td> <td>0.17</td> </tr> </table>				R/D	Loss Coefficients	1.5	0.24	2.0	0.19	2.5	0.17								
						R/D	Loss Coefficients																		
						1.5	0.24																		
2.0	0.19																								
2.5	0.17																								
60° elbow = 2/3 loss 45° elbow = 1/2 loss 30° elbow = 1/3 loss																									
Adapted from Michigan Industrial Ventilation Conference (8/96)																									



TAPERED DUCT SYSTEM

Maintains transport velocity



EXTENDED PLENUM SYSTEM

Self cleaning type

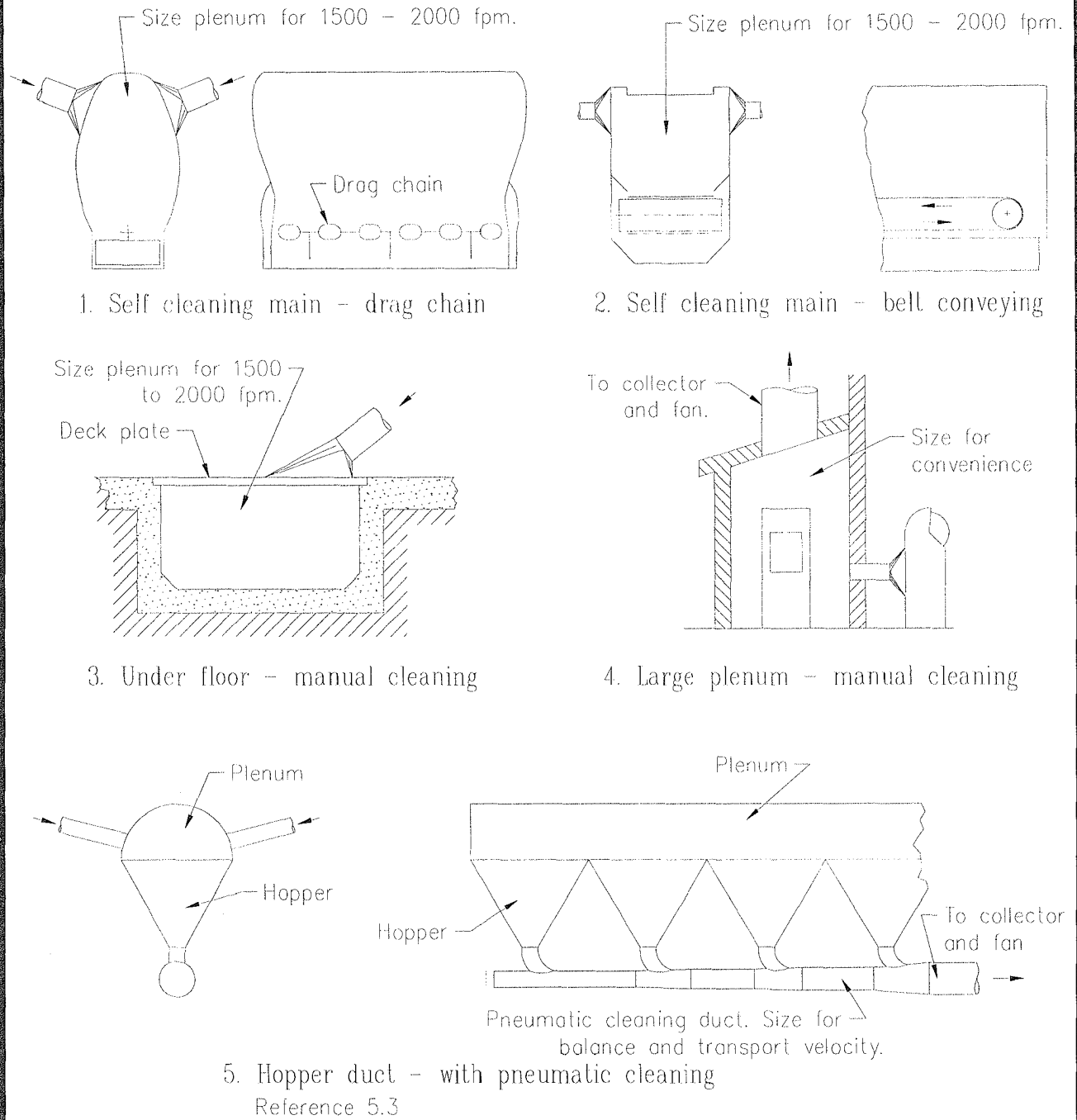
NOTE: Design plenum velocities are at the most 1/2 the branch duct design velocities and typically less than 2000 fpm.

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*PLENUM vs
CONVENTIONAL SYSTEM*

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FIGURE 5-4



NOTE: Design plenum velocities are at the most 1/2 the branch duct design velocities and typically less than 2000 fpm.

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TYPES OF PLENUMS

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FIGURE

5-5

the fan (in terms of number of duct segments) and proceeds, segment by segment, to the fan. At each junction, the static pressure necessary to achieve desired flow in one stream must equal the static pressure in the joining air stream. The static pressures are balanced by suitable choice of duct sizes, elbow radii, etc., as detailed below.

5.5.2 Blast Gate Method: The design procedure depends on the use of blast gates which must be adjusted after installation to achieve the desired flow at each hood. At each junction, the flow rates of two joining ducts are achieved by blast gate adjustment which results in the desired static pressure balance.

It is a common practice to design systems on the assumption that only a fraction of the total number of hoods will be used at a time and the flow to the branches not used will be shut off with dampers. For tapered system designs, where particulate is transported, this practice may lead to plugging in the main duct due to settled particulate.

5.5.3 Choice of Methods: The Balance by Design Method is normally selected where highly toxic materials are controlled to safeguard against tampering with blast gates and consequently subjecting personnel to potentially excessive exposures. This method is mandatory where explosives, radioactive dusts, and biologicals are exhausted because the possibility of accumulations in the system caused by a blast gate obstruction is eliminated.

5.5.4 Balance by Design Procedure: The pressure loss of each duct segment is calculated from an exhaust hood to the junction with the next branch based on hood design data, fittings, and total duct length. At each junction, the SP for each parallel path of air flow must be the same. Where the ratio of the higher SP to the lower SP is greater than 1.2, redesign of

the branch with the lower pressure loss should be considered. This may include a change of duct size, selection of different fittings, and/or modifications to the hood design. Where static pressures of parallel paths are unequal, balance can be obtained by increasing the air flow through the run with the lower resistance. This change in flow rate is calculated by noting that pressure losses vary with the velocity pressure and therefore as the square of the flow rate, so:

$$Q_{\text{Corrected}} = Q_{\text{Design}} \sqrt{\frac{SP_{\text{gov}}}{SP_{\text{duct}}}} \quad [5.1]$$

where the "governing" SP is the desired SP at the junction point and the "duct" SP is that calculated for the duct segment being designed.

5.5.5 Blast Gate Procedure: Data and calculations involved are the same as for the balanced design method except that the duct sizes, fittings, and flow rates are not adjusted; the blast gates are set after installation to provide the design flow rates. It should be noted that a change in any of the blast gate settings will change the flow rates in all of the other branches. Readjusting the blast gates during the system balancing process sometimes can result in increases to the actual fan static pressure and increased fan power requirements.

Recent work^(5,2) describes a method whereby blast gate settings can be made by means of pressure readings instead of by velocity readings. The biggest advantage of this method is that the process of resetting the insertion depths need not be a repetitive procedure.

5.5.6 System Redesign: Many ventilation systems are changed after installation (processes are changed; operations are relocated; additional equipment is added to the production floor; etc.). When such changes occur, the effect of the pro-

TABLE 5-1. Relative Advantages and Disadvantages of the Balance by Design Method and Blast Gate Method

Balance by Design Method	Blast Gate Method
1. Flow rates cannot be changed easily by workers or at the whim of the operator.	1. Flow rates may be changed relatively easily. Such changes are desirable where pickup of unnecessary quantities of material may affect the process.
2. There is little degree of flexibility for future equipment changes or additions. The duct is "tailor made" for the job.	2. Depending on the fan and motor selected, there is somewhat greater flexibility for future changes or additions.
3. The choice of exhaust flow rates for a new operation may be incorrect. In such cases, some duct revision may be necessary.	3. Correcting improperly estimated exhaust flow rates is relatively easy within certain ranges.
4. No unusual erosion or accumulation problems will occur.	4. Partially closed blast gates may cause erosion thereby changing resistance or causing particulate accumulation.
5. Duct will not plug if velocities are chosen correctly.	5. Duct may plug if blast gate insertion depth has been adjusted improperly.
6. Total flow rate may be greater than design due to higher air requirements.	6. Balance may be achieved with design flow rate; however, the net energy required may be greater than for the Balance by Design Method.
7. The system must be installed exactly as designed, with all obstructions cleared and length of runs accurately determined.	7. Moderate variations in duct layout are possible.

posed change(s) to the ventilation system should be calculated. Often, systems are altered without adequate design, resulting in catastrophic changes to some hood flow rates. The result is that worker safety and health are jeopardized.

5.6 AIDS TO CALCULATIONS

As an alternative to performing these calculations manually, programmable calculators and computers can be used to provide assistance with the design of systems. The ACGIH Industrial Ventilation Committee does not recommend any specific hardware or software. Many firms have developed their own software, and software packages are available commercially. Many of these software packages are available through ACGIH.

5.7 PLENUM EXHAUST SYSTEMS

Plenum systems differ from the designs illustrated earlier (see Figures 5-4 and 5-5). Minimum transport velocities are maintained only in the branch ducts to prevent settling of particulate matter; the main duct is oversized and velocities are allowed to decrease far below normal values, many times below 2000 fpm. The function of the main duct is to provide a low-pressure loss path for air flow from the various branches to the air cleaner or the fan. This helps to maintain balanced exhaust in all of the branches and often provides a minimum operating power.

Advantages of the plenum-type exhaust system include:

1. Branch ducts can be added, removed, or relocated at any convenient point along the main duct.
2. Branch ducts can be closed off and the flow rate in the entire system reduced, provided minimum transport velocities are maintained in the remaining branches.
3. The main duct can act as a primary separator (settling chamber) for large particulate matter and refuse material which might be undesirable in the air cleaner or fan.

Limitations of this design include:

1. Sticky, linty materials, such as buffing dust, tend to clog the main duct. It may be expected that greatest difficulty will be encountered with the drag chain type of cleaning, but the other types will be susceptible to buildup as well.
2. Materials that are subject to direct or spontaneous combustion must be handled with care. Wood dust has been handled successfully in systems of this type; buffing dust and lint are subject to this limitation and are not recommended. Explosive dusts such as magnesium, aluminum, titanium, or grain dusts should not be handled in systems of this type.

5.7.1 Choice of Systems: Various types of plenum exhaust systems are used in industry (see Figure 5-5). They

include both self-cleaning and manual-cleaning designs. Self-cleaning types include pear-shaped designs which incorporate a drag chain conveyor in the bottom of the duct to convey the dust to a chute, tote box, or enclosure for disposal. Another self-cleaning design uses a rectangular main with a belt conveyor. In these types, the conveyors may be run continuously or on periodic cycles to empty the main duct before considerable buildup and clogging occur. A third type^(5.3) of self-cleaning design utilizes a standard conveying main duct system to remove the collected material from a hopper-type of main duct above. Such a system is usually run continuously to avoid clogging of the pneumatic air circuit. Manual-cleaning designs may be built into the floor or may be large enclosures behind the equipment to be ventilated. Experience indicates that these should be generously oversized, particularly the underfloor designs, to permit added future exhaust capacity as well as convenient housekeeping intervals.

5.7.2 Design: Control flow rates, hoods, and duct sizes for all branches are calculated in the same manner as with tapered duct systems. The branch segment with the greatest pressure loss will govern the static pressure required in the main duct. Other branches will be designed to operate at this static pressure or locking dampers can be used to adjust their pressure loss to the same static pressure as the governing branch. Where the main duct is relatively short or where the air cleaners or fans can be spaced along the duct, static pressure losses due to air flow in the main duct can be ignored. For extremely long ducts, it is necessary to calculate the static pressure loss along the main in a manner similar to that used in the balanced and blast gate methods. Design plenum velocities are at most one-half the branch velocity design duct velocities and typically less than 2000 fpm. Duct connections to air cleaners, fans, and discharge to outdoors are handled in the normal manner.

5.8 FAN PRESSURE CALCULATIONS

Exhaust system calculations are based on static pressure; that is, all hood static pressures and balancing or governing pressures at the duct junctions are given as static pressures which can be measured directly as described in Chapter 9. Most fan rating tables are based on Fan Static Pressure. An additional calculation is required to determine Fan Static Pressure before selecting the fan.

5.8.1 Fan Total Pressure (FTP) is the increase in total pressure through or across the fan and can be expressed by the equation:

$$FTP = TP_{\text{outlet}} - TP_{\text{inlet}} \quad [5.2]$$

Some fan manufacturers base catalog ratings on Fan Total Pressure. To select a fan on this basis the Fan Total Pressure is calculated noting that $TP = SP + VP$:

$$FTP = (SP_{\text{outlet}} + VP_{\text{outlet}}) - (SP_{\text{inlet}} + VP_{\text{inlet}}) \quad [5.3]$$

5.8.2 Fan Static Pressure: The Air Movement and Control Association Test Code defines the Fan Static Pressure (FSP) as follows: "the static pressure of the fan is the total pressure diminished by the fan velocity pressure. The fan velocity pressure is defined as the pressure corresponding to the air velocity at the fan outlet."^(5.4) Fan Static Pressure can be expressed by the equation:

$$\text{FSP} = \text{FTP} - \text{VP}_{\text{inlet}} \quad [5.4]$$

or

$$\text{FSP} = \text{SP}_{\text{outlet}} - \text{SP}_{\text{inlet}} - \text{VP}_{\text{inlet}} \quad [5.5]$$

In selecting a fan from catalog ratings, the rating tables should be examined to determine whether they are based on Fan Static Pressure or Fan Total Pressure. Fan system effects (see Chapter 6) should also be considered when selecting a fan. The proper pressure rating can then be calculated keeping in mind the proper algebraic signs; i.e., VP is always positive (+), SP_{inlet} is usually negative (-), and $\text{SP}_{\text{outlet}}$ is usually positive (+).

5.8.3 Completion of the Example on Figure 5-3: To determine the Fan Static Pressure and Fan Total Pressure, note that the second column adds the fabric pressure drop through the bags in the collector. Column 3 adds the losses from the clean air plenum to the fan inlet, and the last column determines the pressure losses through the stack.

The FSP and FTP can be calculated from these values. At the outlet of the fan, the SP must be 0.48 "wg. At the inlet to the fan, the SP is -6.56 "wg. The VP at both locations is 0.78 "wg. From Equation 5.3, the system FTP = $(0.48 + 0.78) - (-6.56 + 0.78) = 7.04$ "wg. From Equation 5.5, the FSP = $0.48 - (-6.56) - 0.78 = 6.26$ "wg.

5.9 CORRECTIONS FOR VELOCITY CHANGES

Variations in duct velocity occur at many locations in exhaust systems because of necessary limitations of available standard duct sizes (area) or due to duct selections based on balanced system design. As noted earlier, small accelerations and decelerations are usually compensated automatically in the system where good design practices and proper fittings are used. There are times, however, when special circumstances require the designer to have a knowledge of the energy losses and regains which occur since these may work to his advantage or disadvantage in the final performance of the system.

5.9.1 Branch Entries to Main Ducts: Sometimes the final main duct velocity exceeds the higher of the two velocities in the branches entering the main. If the difference is significant, additional static pressure is required to produce the increased velocity. A difference of 0.10 "wg or greater between the main VP and the resultant VP of the two branches should be corrected.

At any junction point, energy must be conserved. The

energy entering each of the two air streams would be $Q(\text{TP}) = Q(\text{SP} + \text{VP})$. The first law of thermodynamics states that the sum of these must equal the energy leaving, or

$$Q_1(\text{VP}_1 + \text{SP}_1) + Q_2(\text{VP}_2 + \text{SP}_2) = Q_3(\text{VP}_3 + \text{SP}_3) + \text{Losses}$$

Note that the overall losses would be:

$$\text{Losses} = F_1 Q_1 \text{VP}_1 + F_2 Q_2 \text{VP}_2$$

where the subscripts refer to the ducts shown in Figure 5-6. In this manual, F_1 is considered to be zero and F_2 is given on Figure 5-15. Assuming we are balanced and the junction losses are included such that $\text{SP}_1 = \text{SP}_2$ and $Q_3 = Q_1 + Q_2$ (see Figure 5-6), there might be an additional change in static pressure due to the acceleration or deceleration of the gas stream. The following equation shows this effect:

$$\text{SP}_3 + \text{VP}_3 = \text{SP}_1 + \left(\frac{Q_1}{Q_3}\right)\text{VP}_1 + \left(\frac{Q_2}{Q_3}\right)\text{VP}_2$$

The last two terms on the right are defined as the resultant velocity pressure, VP_r ; this can be simplified to

$$\text{VP}_r = \left(\frac{Q_1}{Q_3}\right)\text{VP}_1 + \left(\frac{Q_2}{Q_3}\right)\text{VP}_2 \quad [5.6]$$

where:

VP_r = resultant velocity pressure of the combined branches

Q_1 = flow rate in branch #1

Q_2 = flow rate in branch #2

Q_3 = combined flow rate leaving the junction

Note that the above equation is valid for all conditions, including merging different density gas streams, as long as the velocity pressures include the density effects. Also note that, if the flow rate through one branch was changed to balance at the branch entry, the velocity pressure and corrected flow rates should be used in Equation 5.6.

The resultant velocity pressure (VP_r) is computed using Equation 5.6. If VP_3 is less than VP_r , a deceleration has occurred and SP has increased. If VP_3 is greater than VP_r , an acceleration has occurred, and the difference between VP_3 and VP_r is the necessary loss in SP required to produce the increase in kinetic energy between VP_3 and VP_r . The correction is made as follows:

$$\text{SP}_3 = \text{SP}_1 - (\text{VP}_3 - \text{VP}_r) \quad [5.7]$$

where:

SP_3 = SP in main #3

SP_1 = SP at branch #1 = SP at branch #2

VP_3 = velocity pressure in main #3

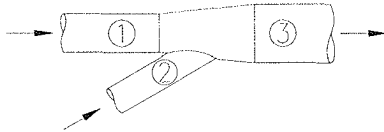
It should be noted that many designers believe a conservative approach to fan selection would be to ignore any correction if VP_r is larger than VP_3 .

A simpler equation for VP_r was used in prior editions of this manual:

$$VP_r = \left[\frac{Q_1 + Q_2}{4005(A_1 + A_2)} \right]^2$$

This equation gives acceptable results (less than a 4% error) *when the velocities of the two merging air streams are within 500 fpm of each other.*

EXAMPLE



Duct No.	Dia.	Area	Q	V	VP	SP
(1)	10	0.545	1935	3550	0.79	-2.11
(2)	4	0.087	340	3890	0.94	-2.11
Main (3)	10	0.545	2275	4170	1.08	--

FIGURE 5-6. Branch entry velocity correction

With the data shown,

$$VP_r = \frac{(1935)(0.79)}{2275} + \frac{(340)(0.94)}{2275} = 0.81 \text{ "wg}$$

$$\begin{aligned} SP_3 &= SP_1 - (VP_3 - VP_r) = -2.11 - (1.08 - 0.81) \\ &= -2.11 - 0.27 = -2.38 \text{ "wg} \end{aligned}$$

Therefore, in this situation, an additional -0.27 "wg should be added to the junction SP to account for losses in pressure due to acceleration of the air stream.

5.9.2 Contractions and Expansions: Contractions are used when the size of the duct must be reduced to fit into tight places, to fit equipment, or to provide a high discharge velocity at the end of the stack. Expansions are used to fit a particular piece of equipment or to reduce the energy consumed in the system by reducing velocity and friction. Expansions are not desirable in transport systems since the duct velocity may become less than the minimum transport velocity and material may settle in the ducts.

Regain of pressure in a duct system is possible because static pressure and velocity pressure are mutually convertible. This conversion is accompanied by some energy loss. The amount of this loss is a function of the geometry of the transition piece (the more abrupt the change in velocity, the greater the loss) and depends on whether air is accelerated or decelerated. Loss is expressed as a loss coefficient multiplied by the velocity pressure in the smaller area duct of the transition piece. One minus the loss factor is the efficiency of the energy conversion or regain.

A perfect (no loss) contraction or expansion would cause no change in the total pressure in the duct. There would be an increase or decrease in static pressure corresponding exactly to the decrease or increase in velocity pressure of the air. In practice, the contraction or expansion will not be perfect, and there will be a change in total pressure (see Figure 5-7). In each example, total pressure and static pressure are plotted to show their relationship at various points in each system. See Figure 5-16 for design data.

5.10 SAMPLE SYSTEM DESIGN

A discussion of the calculations for either tapered duct method can best be done by a typical example using the exhaust system shown in Figure 5-8. Calculation sheets illustrate the orderly and concise arrangement of data and calculations (see Figures 5-9 and 5-10). The procedure outlined in Section 5.3 was used to develop the design. Each column is for a constant diameter duct segment that starts at a hood, junction point, air cleaning device, fan, or transition point.

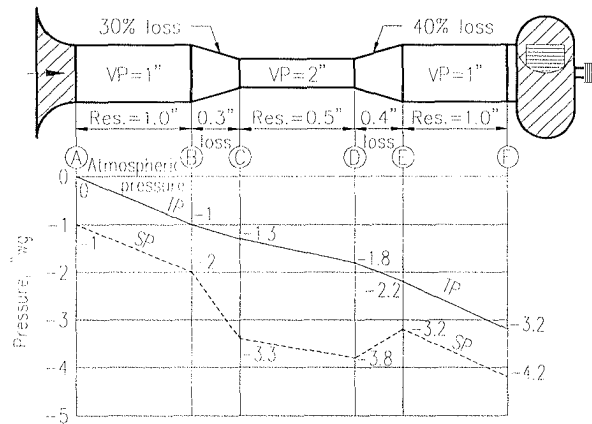
The problem considered is a foundry sand-handling and shake-out system. A minimum conveying velocity of 3500 fpm is used throughout the problem except in ducts where excess moisture or dust loading increases that value. The operations, hood designations on the diagram, VS-print references, and required flow rates are presented in Table 5-2.

5.11 DIFFERENT DUCT MATERIAL FRICTION LOSSES

The friction loss table, Table 5-5, provides average values for galvanized sheet metal duct material (0.0005 feet equivalent sand grain roughness, where the roughness height represents the average height of the roughness elements of the material). Table 5-6 provides the same information for black iron and other materials possessing a roughness height of 0.00015 feet. Recent research indicates that an equivalent sand grain roughness factor of 0.0003 feet more accurately reflects the losses incurred in new HVAC galvanized duct systems. However, past experiences in industrial ventilation applications successfully reinforce the application of the 0.0005 feet equivalent sand grain roughness factor. This may be due to inherently shorter duct runs and dustier environments which are common within industrial ventilation applications. The values in both tables can be used with no significant error for the majority of designs but special considerations may be desired if environmental conditions could significantly affect the duct design parameters. If the design requires special material, operates at a non-standard density, or is very hot, the duct material manufacturer should be consulted for the anticipated friction loss.

5.12 FRICTION LOSS FOR NON-CIRCULAR DUCTS

Round ducts are preferred for industrial exhaust systems because they provide a more uniform air velocity to resist settling of material and an ability to withstand higher static

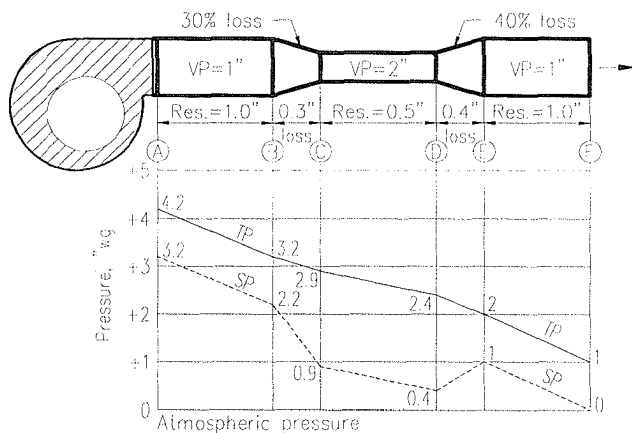


EXAMPLE 1 — DUCT LOCATED ON SUCTION SIDE OF FAN

Velocity changes as indicated. Since all the duct is on the suction side of the fan, TP at the fan inlet (point F) is equal to VP at the fan inlet plus the total duct resistance up to that point. This equals -4.2 SP since static pressure on the suction side of the fan is always negative. The duct system is the same as was used in Example 2 and therefore has the same overall resistance of 3.2 . If it is again assumed that the inlet and discharge of the fan are equal areas, the total pressure across the fan will be the same as in Example 2 and, in each case, the fan will deliver the same air horsepower when handling equal volumes of air.

Static pressure conversion between B and C follows contraction formula (Figure 5-16). There must be sufficient SP at B to furnish the additional VP required at C. In addition, the energy transfer between these two points is accompanied by a loss of 0.3 . Since SP at B = -2 , SP at C = $-2.0 + (-1.0) + (-0.3) = -3.3$ "wg.

Static pressure regain between D and E follows the regain formulae (Figure 5-16). If there were no loss in the transition piece, the difference of 1 "wg in velocity pressure would be regained as static pressure at E, and SP at that point would be -2.8 . However, the transition is only 60% efficient (0.4 loss) so the SP at E = $-2.8 + (-0.4) = -3.2$.



EXAMPLE 2 — DUCT LOCATED ON DISCHARGE SIDE OF THE FAN

Velocity changes as indicated. The duct is located on the discharge side of the fan. Total pressure at the fan discharge (point A) is equal to the velocity pressure at the discharge end of the duct (point F) plus the accumulated resistances. These add up to $1.0 + 1.0 + 0.4 + 0.5 + 0.3 + 1.0 = 4.2$.

Static pressure regain between D and E follows the regain formulae (Figure 5-16). If there were no energy loss in the transition piece, static pressure at D would be 0 because the difference in VP of 1 would show up as static pressure regain. However, the transition is only 60% efficient which means a loss of 0.4 , so SP at point D = $0 + 0.4 = 0.4$.

Conversion of static pressure into velocity pressure between B and C follows contraction formulae (Figure 5-16). There must be sufficient static pressure at B to furnish the additional velocity pressure required at C. In addition, transformation of energy between these two points is accompanied by a loss of 0.3 . Since SP at C = 0.9 , SP at B = $0.9 + 0.3 + 1.0 = 2.2$. Since there is no duct on the suction side of the fan, total pressure against which the fan is operating is 4.2 .

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FIGURE

5-7

TABLE 5-2. Details of Operation

No.	Hood No.	VS-Print	Minimum Exhaust, cfm
1. Vibrating Shakeout 4' x 6' grate	1	20-02	9600
2. Shakeout hopper	2	20-03	960
3. Vibrating pan feeder 24" wide	3	20-03	700
4. Incline sand belt 24" x 28" long	5		700
5. Magnetic pulley			
6. Tramp iron box			
7. Bucket elevator 24" x 30" casing	7a(lower) 7b(upper)	50-01	250 250
8. Vibrating screen 24 ft ²	8	99-01	1200
9. Sand bin 600 ft ³ 18" x 20" opening	9	50-10	500
10. Waster sand box 44" x 54", 6" clearance	10	99-03 (V = 150 fpm)	1225
11. Sand weigh hopper	11	60-02	900
12. Sand muller 6' dia.	12	60-02	
13. Wet dust collector (includes fan)			

DIMENSIONS

No. of Branch or Main	CFM Required, Minimum	Straight Run, ft	Elbows	Entries
1-A	9600	13	1-90°	
2-B	960	3	1-60°	1-30°
3-B	700	4	1-90°+1-60°	1-30°
B-A	1660	18	2-90°	1-30°
A-C	11,260	34		
5-D	700	7	1-30°+1-60°	1-30°
7a-D	250	5		
D-C	950	14	1-90°+1-60°	1-30°
C-E	12,210	6.5		
8-F	1200	11	2-90°	
9-F	500	4	1-90°+1-60°	1-30°
F-G	1700	5		
7b-G	250	15	1-60°	1-30°
G-E	1950	6	1-60°	1-30°
E-H	14,160	3.5		
10-J	1225	6	1-45°	
12-J	900	2.5	1-30°	1-30°
J-H	2125	8	1-90°+1-60°	1-30°
H-K	16,285	9	2-45°	
13	16,285			
14-L	16,285	20		

pressure. At times, however, the designer must use other duct shapes.

Rectangular duct friction can be calculated by using Table 5-5 or 5-6 in conjunction with Table 5-9 to obtain rectangular equivalents for circular ducts on the basis of equal friction loss. It should be noted that, on this basis, the area of the rectangular duct will be larger than the equivalent round duct; consequently, the actual air velocity in the duct will be reduced. Therefore, it is necessary to use care to maintain minimum transport velocities.

Occasionally, the designer will find it necessary to estimate the air handling ability of odd-shaped ducts. The following procedure^(5,5) will be helpful in determining the frictional pressure losses for such ducts. The wetted perimeter in the following discussion is the inside perimeter of the odd-shaped duct corresponding to the cross-sectional area.

1. Find duct cross-sectional area, ft² A
2. Find wetted perimeter, ft P
3. Calculate hydraulic radius, ft R (R = A/P)
4. Convert R to inches r (r = 12R)
5. Calculate equivalent diameter, in D (D = 4r)
6. Use the proper friction table based on the equivalent diameter and flow rate (or velocity).

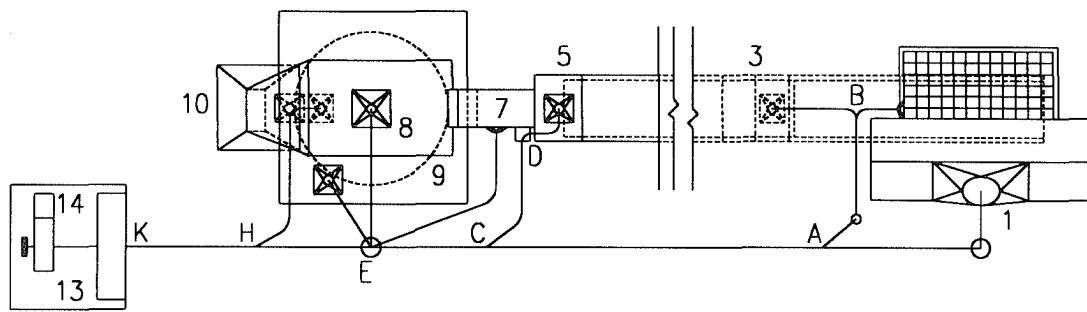
5.13 CORRECTIONS FOR NONSTANDARD DENSITY

Fan tables and exhaust flow rate requirements assume a *standard air density* of 0.075 lbm/ft³, which corresponds to sea level pressure, no moisture, and 70 F. Changes in air density can come from several factors including elevation, temperature, internal duct pressure, changes in apparent molecular weight (moisture content, gas stream constituents, etc.), and amount of suspended particulate. Where appreciable variation occurs, the change in air density must be considered.

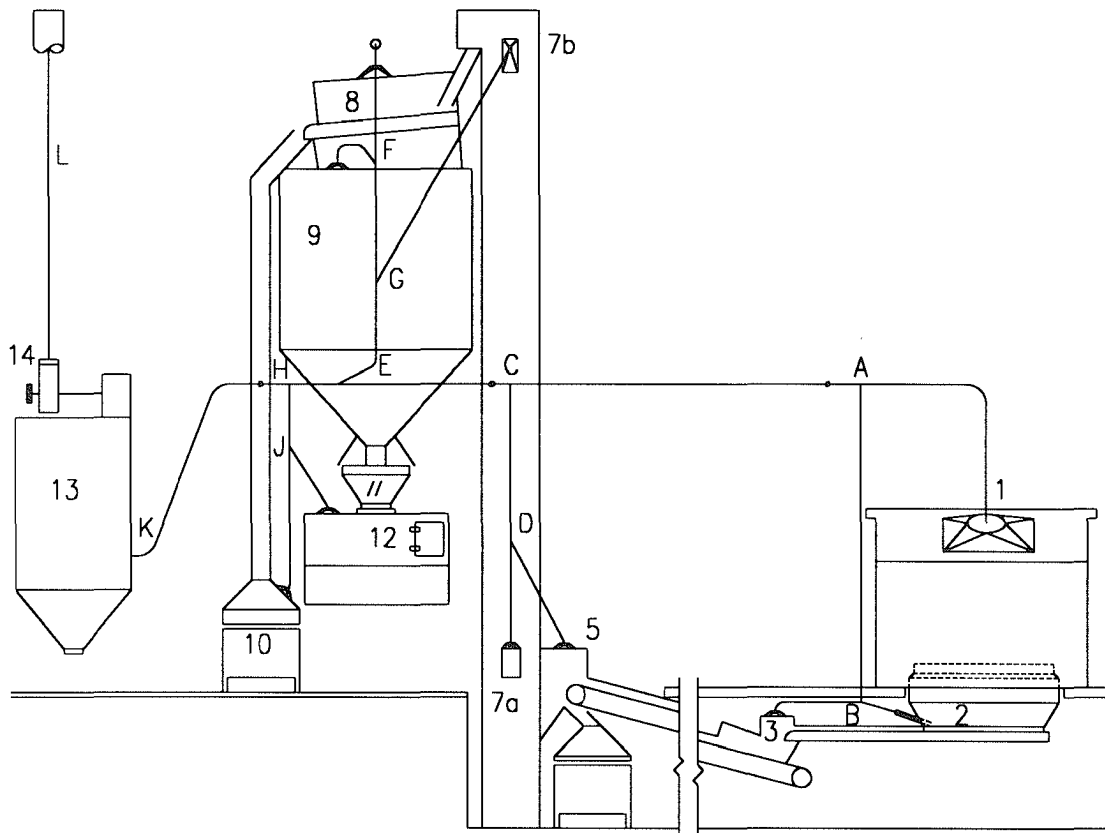
Factors for different temperatures and elevations are listed in Table 5-10. Correction for temperatures between 40 F and 100 F and/or elevations between -1,000 feet and +1,000 feet are seldom required with the permissible variations in usual exhaust system design.

Similarly, if internal duct pressures vary by more than 20 "wg from standard pressure, the density will change by over 5%. If there is excessive moisture in the airstream, the density will decrease. Suspended particulate is assumed to be only a trace impurity in industrial exhaust systems. If there are significant quantities of particulate in the duct system, this addition to the air stream density should be addressed. This field is called material conveying and is beyond the scope of this manual.

Many times, the system designer is confronted with a combination of these five means of changing density. If so, then the density of the air stream should be determined and, if the density is more than 5% different from standard density,



All elbows - $\frac{1}{2}$ radius = 2.0 D (5 piece)
Branch entries - 30°
All duct lengths are $\frac{1}{2}$ to $\frac{1}{2}$



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PROBLEM 2

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FIGURE 5-8

VELOCITY PRESSURE METHOD CALCULATION SHEET

Problem #2 Class Designer Date

1	Duct Segment Identification		2-B	3-B	B-A	1-A	1-A
2	Target Volumetric Flow Rate	cfm	960	700	1711	9600	9600
3	Minimum Transport Velocity	fpm	4000	4000	4000	4000	4000
4	Maximum Duct Diameter	inches	6.63	5.66	8.86	20.97	
5	Selected Duct Diameter	inches	6	5.5	8	20	18
6	Duct Area	ft ²	0.1963	0.1650	0.3491	2.1817	1.7671
7	Actual Duct Velocity	fpm	4889	4243	4902	4400	5432
8	Duct Velocity Pressure	"wg	1.49	1.12	1.50	1.21	1.84
9	H O O D S S U C T I O N	Maximum Slot Area	ft ²			4.8	4.8
10		Slot Area Selected	ft ²			4.8	4.8
11		Slot Velocity	fpm			2000	2000
12		Slot Velocity Pressure	"wg			0.25	0.25
13		Slot Loss Coefficient				1.78	1.78
14		Acceleration Factor	(0 or 1)			0	0
15		Slot Loss per VP	(13 + 14)			1.78	1.78
16		Slot Static Pressure	(12 x 15)			0.44	0.44
17	S U C T I O N	Duct Entry Loss Coefficient	(Fig. 5-13)	.25	.25	.25	.25
18		Acceleration Factor	(1 or 0)	1	1	1	1
19		Duct Entry Loss per VP	(17 + 18)	1.25	1.25	1.25	1.25
20		Duct Entry Loss	(8 x 19)	1.86	1.40	1.51	2.30
21		Other Losses	"wg				
22		Hood Static Pressure	(16 + 20 + 21)	1.86	1.40	1.95	2.74
23	Straight Duct Length		feet	3	4	18	13
24	Friction Factor (H_f)			.0425	.0478	.0299	.0098
25	Friction Loss per VP		(23 x 24)	.13	.19	.54	.13
26	Number of 90 deg. Elbows			.67	1.67	2	1
27	Elbow Loss Coefficient			.19	.19	.19	.33
28	Elbow Loss per VP		(26 x 27)	.13	.32	.38	.19
29	Number of Branch Entries		(1 or 0)	1	1	1	
30	Entry Loss Coefficient			.18	.18	.18	
31	Branch Entry Loss per VP		(29 x 30)	.18	.18	.18	
32	Special Fitting Loss Coefficients						
33	Duct Loss per VP		(25 + 28 + 31 + 32)	.43	.69	1.10	.32
34	Duct Loss		(33 x 8)	.65	.77	1.64	.38
35	Duct Segment Static Pressure Loss		(22 + 34)	-2.51	-2.18	-1.64	-2.34
36	Other Losses (VP-VP _r , etc.)		"wg			.10	
37	Cumulative Static Pressure		"wg	-2.51	-2.18	-4.25	-2.34
38	Governing Static Pressure		"wg		-2.51		-4.25
39	Corrected Volumetric Flow Rate		cfm		751		10416
40	Corrected Velocity		fpm		4551		5894
41	Corrected Velocity Pressure		"wg		1.29		2.16
42	Resultant Velocity Pressure		"wg	1.40	1.40	2.07	2.07

FIGURE 5-9. Balanced design method

Temperature _____ Elevation _____								Remarks:																			
A-C	7a-D	5-D	5-D	D-C	D-C	C-E	1	Pertinent Information From Chapter 5																			
12127	250	700	700	955	955	13130	2																				
4000	3500	3500	3500	3500	3500	4000	3																				
23.58	3.62	6.05		7.07		24.53	4	$Q_{\text{corr}} = Q_{\text{design}} \sqrt{\frac{SP_{\text{gov}}}{SP_{\text{duct}}}}$																			
22	3.5	6	5.5	7	6	24	5																				
2.6398	0.0668	0.1963	0.1650	0.2673	0.1963	3.1416	6																				
4594	3742	3565	4243	3573	4864	4179	7	$VP_r = \frac{Q_1}{Q_3} VP_1 + \frac{Q_2}{Q_3} VP_2$																			
1.32	0.87	0.79	1.12	0.80	1.47	1.09	8																				
							9																				
							10	Straight Duct Friction Loss $H_f = a \frac{V^b}{Q^c}$																			
							11																				
							12																				
							13																				
							14																				
							15	<table border="1" style="width: 100%; border-collapse: collapse;"> <tr> <th>Duct Material</th> <th>a</th> <th>b</th> <th>c</th> </tr> <tr> <td>Galvanized</td> <td>0.0155</td> <td>0.533</td> <td>0.612</td> </tr> <tr> <td>Black iron, Aluminum, PVC, Stainless steel</td> <td>0.662</td> <td>0.645</td> <td>0.602</td> </tr> <tr> <td>Flexible (fabric covered wires)</td> <td>0.0186</td> <td>0.604</td> <td>0.639</td> </tr> </table>				Duct Material	a	b	c	Galvanized	0.0155	0.533	0.612	Black iron, Aluminum, PVC, Stainless steel	0.662	0.645	0.602	Flexible (fabric covered wires)	0.0186	0.604	0.639
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							16																				
	1.0	.25	.25				17	Fan Static Pressure $\text{FAN SP} = SP_{\text{out}} - SP_{\text{in}} - VP_{\text{in}}$																			
	1	1	1				18																				
	2.0	1.25	1.25				19																				
	1.74	.99	1.40				20																				
							21	Branch Entry Loss Coefficients <table border="1" style="width: 100%; border-collapse: collapse;"> <tr> <th>Angle</th> <th>Loss Coefficients</th> </tr> <tr> <td>15°</td> <td>0.09</td> </tr> <tr> <td>30°</td> <td>0.18</td> </tr> <tr> <td>45°</td> <td>0.28</td> </tr> </table>				Angle	Loss Coefficients	15°	0.09	30°	0.18	45°	0.28								
Angle	Loss Coefficients																										
15°	0.09																										
30°	0.18																										
45°	0.28																										
	1.74	.99	1.40				22																				
34	5	7	7	14	14	6.5	23																				
.0087	.0840	.0436	.0478	.0361	.0425	.0079	24	90° Round Elbow Loss Coefficients (5 piece) <table border="1" style="width: 100%; border-collapse: collapse;"> <tr> <th>R/D</th> <th>Loss Coefficients</th> </tr> <tr> <td>1.5</td> <td>0.24</td> </tr> <tr> <td>2.0</td> <td>0.19</td> </tr> <tr> <td>2.5</td> <td>0.17</td> </tr> </table> <p>60° elbow = 2/3 loss 45° elbow = 1/2 loss 30° elbow = 1/3 loss</p> <p>Adapted from Michigan Industrial Ventilation Conference (8/96)</p>				R/D	Loss Coefficients	1.5	0.24	2.0	0.19	2.5	0.17								
R/D	Loss Coefficients																										
1.5	0.24																										
2.0	0.19																										
2.5	0.17																										
.30	.42	.30	.33	.51	.60	.05	25																				
		1	1	1.67	1.67		26																				
		.19	.19	.19	.19		27																				
		.19	.19	.32	.32		28																				
		1	1	1	1		29																				
		.18	.18	.18	.18		30	60° elbow = 2/3 loss 45° elbow = 1/2 loss 30° elbow = 1/3 loss Adapted from Michigan Industrial Ventilation Conference (8/96)																			
		.18	.18	.18	.18		31																				
							32																				
.30	.42	.67	.70	1.01	1.10	.05	33																				
.39	.37	.53	.79	.80	1.61	.06	34																				
.39	2.11	1.52	2.19	.80	1.61	.06	35																				
				.41			36	60° elbow = 2/3 loss 45° elbow = 1/2 loss 30° elbow = 1/3 loss Adapted from Michigan Industrial Ventilation Conference (8/96)																			
-4.64	-2.11	-1.52	-2.19	-2.99	-4.21	-4.70	37																				
	-2.19				-4.64		38																				
	255				1003		39																				
	3817				5109		40																				
	.91				1.63		41																				
1.34	1.06		1.06		1.34	1.15	42																				

FIGURE 5-9. Balanced design method (continued)

VELOCITY PRESSURE METHOD CALCULATION SHEET

Problem #2 (Continued) Class Designer Date

1	Duct Segment Identification		8-F	9-F	9-F	F-G	7b-G
2	Target Volumetric Flow Rate	cfm	1200	500	500	1730	250
3	Minimum Transport Velocity	fpm	3500	3500	3500	3500	3500
4	Maximum Duct Diameter	inches	7.93	5.12		9.52	3.62
5	Selected Duct Diameter	inches	7	5	4.5	9	3.5
6	Duct Area	ft ²	0.2673	0.1364	0.1104	0.4418	0.0668
7	Actual Duct Velocity	fpm	4490	3667	4527	3916	3742
8	Duct Velocity Pressure	"wg	1.26	0.84	1.28	0.96	0.87
9	H O O D S S U C T I O N	Maximum Slot Area	ft ²				
10		Slot Area Selected	ft ²				
11		Slot Velocity	fpm				
12		Slot Velocity Pressure	"wg				
13		Slot Loss Coefficient					
14		Acceleration Factor	(0 or 1)				
15		Slot Loss per VP	(13 + 14)				
16		Slot Static Pressure	(12 x 15)				
17	D U C T I O N	Duct Entry Loss Coefficient	(Fig. 5-13)	.5	.25	.25	1.0
18		Acceleration Factor	(1 or 0)	1	1	1	1
19		Duct Entry Loss per VP	(17 + 18)	1.5	1.25	1.25	2.0
20		Duct Entry Loss	(8 x 19)	"wg 1.89	1.05	1.60	1.74
21		Other Losses	"wg				
22		Hood Static Pressure	(16 + 20 + 21)	"wg 1.89	1.05	1.60	1.74
23	Straight Duct Length	feet	11	4	4	5	15
24	Friction Factor (H _f)		.0354	.0543	.0608	.0263	.0840
25	Friction Loss per VP	(23 x 24)	.39	.22	.24	.12	1.26
26	Number of 90 deg. Elbows		2	1.67	1.67		.67
27	Elbow Loss Coefficient		.19	.19	.19		.19
28	Elbow Loss per VP	(26 x 27)	.38	.32	.32		.13
29	Number of Branch Entries	(1 or 0)		1	1		1
30	Entry Loss Coefficient			.18	.18		.18
31	Branch Entry Loss per VP	(29 x 30)		.18	.18		.18
32	Special Fitting Loss Coefficients						
33	Duct Loss per VP	(25 + 28 + 31 + 32)	.77	.71	.74	.12	1.57
34	Duct Loss	(33 x 8)	"wg .97	.60	.95	.12	1.37
35	Duct Segment Static Pressure Loss	(22 + 34)	"wg 2.86	1.65	2.55	.12	3.11
36	Other Losses (VP-VP _n , etc.)	"wg					
37	Cumulative Static Pressure	"wg	-2.86	-1.65	-2.55	-2.98	-3.11
38	Governing Static Pressure	"wg			-2.86	-3.11	
39	Corrected Volumetric Flow Rate	cfm			530	1767	
40	Corrected Velocity	fpm			4801	3999	
41	Corrected Velocity Pressure	"wg			1.44	1.00	
42	Resultant Velocity Pressure	"wg	1.31		1.31	.98	.98

FIGURE 5-9. Balanced design method (continued)

Temperature _____ Elevation _____							Remarks:																	
G-E	G-E	E-H	12-J	10-J	10-J	J-H	1	Pertinent Information From Chapter 5																
2017	2017	15308	900	1225	1225	2274	2																	
3500	3500	4000	4500	3500	3500	4500	3																	
10.28		26.49	6.06	8.01		9.57	4																	
10	9	26	6	8	7	9	5	$Q_{\text{corr}} = Q_{\text{design}} \sqrt{\frac{SP_{\text{gov}}}{SP_{\text{duct}}}}$																
0.5454	0.4418	3.687	0.1963	0.3491	0.2673	0.4418	6																	
3698	4566	4152	4584	3509	4584	5147	7																	
0.85	1.32	1.07	1.31	0.77	1.31	1.65	8	$VP_r = \frac{Q_1}{Q_3} VP_1 + \frac{Q_2}{Q_3} VP_2$																
							9																	
							10																	
							11	Straight Duct Friction Loss $H_f = a \frac{V^b}{Q^c}$																
							12																	
							13																	
							14																	
							15	<table border="1" style="width: 100%; border-collapse: collapse;"> <tr> <th>Duct Material</th><th>a</th><th>b</th><th>c</th></tr> <tr> <td>Galvanized</td><td>0.0307</td><td>0.533</td><td>0.612</td></tr> <tr> <td>Black iron, Aluminum, PVC, Stainless steel</td><td>0.0425</td><td>0.465</td><td>0.602</td></tr> <tr> <td>Flexible (fabric covered wires)</td><td>0.0311</td><td>0.604</td><td>0.639</td></tr> </table>	Duct Material	a	b	c	Galvanized	0.0307	0.533	0.612	Black iron, Aluminum, PVC, Stainless steel	0.0425	0.465	0.602	Flexible (fabric covered wires)	0.0311	0.604	0.639
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			.25	.25	.25		17																	
			1	1	1		18																	
			1.25	1.25	1.25		19																	
			1.64	.96	1.64		20	Fan Static Pressure FAN SP = SP _{out} - SP _{in} - VP _{in}																
			1.64	.96	1.64		21																	
6	6	3.5	11	6	6	8	22																	
.0232	.0260	.0072	.0427	.0307	.0354	.0258	23																	
.14	.16	.03	.47	.18	.21	.21	24	Branch Entry Loss Coefficients <table border="1" style="width: 100%; border-collapse: collapse;"> <tr> <th>Angle</th><th>Loss Coefficients</th></tr> <tr> <td>15°</td><td>0.09</td></tr> <tr> <td>30°</td><td>0.18</td></tr> <tr> <td>45°</td><td>0.28</td></tr> </table>	Angle	Loss Coefficients	15°	0.09	30°	0.18	45°	0.28								
Angle	Loss Coefficients																							
15°	0.09																							
30°	0.18																							
45°	0.28																							
.67	.67		.33	.5	.5	1.67	25																	
.19	.19		.19	.19	.19	.19	26																	
.13	.13		.06	.10	.10	.32	27																	
1	1		1			1	28	90° Round Elbow Loss Coefficients (5 piece) <table border="1" style="width: 100%; border-collapse: collapse;"> <tr> <th>R/D</th><th>Loss Coefficients</th></tr> <tr> <td>1.5</td><td>0.24</td></tr> <tr> <td>2.0</td><td>0.19</td></tr> <tr> <td>2.5</td><td>0.17</td></tr> </table> <p>60° elbow = 2/3 loss 45° elbow = 1/2 loss 30° elbow = 1/3 loss</p>	R/D	Loss Coefficients	1.5	0.24	2.0	0.19	2.5	0.17								
R/D	Loss Coefficients																							
1.5	0.24																							
2.0	0.19																							
2.5	0.17																							
.18	.18		.18			.18	29																	
.18	.18		.18			.18	30																	
							31																	
.45	.47	.03	.71	.28	.31	.61	32	Adapted from Michigan Industrial Ventilation Conference (8/96)																
.38	.60	.03	.94	.21	.41	1.01	33																	
.38	.60	.03	2.58	1.17	2.05	1.01	34																	
	.34					.18	35																	
-3.49	-4.05	-4.73	-2.58	-1.17	-2.05	-3.77	36																	
	-4.70				-2.58	-4.73	37																	
	2178				1374	2492	38																	
	4930				5140	5641	39																	
	1.52				1.65	1.98	40																	
	1.15	1.20	1.47		1.47	1.20	41																	
							42																	

FIGURE 5-9. Balanced design method (continued)

VELOCITY PRESSURE METHOD CALCULATION SHEET							
Problem #2 (Continued)		Class		Designer		Date	
1	Duct Segment Identification		H-K	K-FAN	FAN-L		
2	Target Volumetric Flow Rate	cfm	17800	17800	17800		
3	Minimum Transport Velocity	fpm	4500	2600	2600		
4	Maximum Duct Diameter	inches	26.93	35.4			
5	Selected Duct Diameter	inches	26	35.5	34		
6	Duct Area	ft ²	3.6870	6.8736	6.3050		
7	Actual Duct Velocity	fpm	4828	2590	2823		
8	Duct Velocity Pressure	"wg	1.45	0.42	0.50		
9	S L O T S U C T I O N	Maximum Slot Area	ft ²				
10		Slot Area Selected	ft ²				
11		Slot Velocity	fpm				
12		Slot Velocity Pressure	"wg				
13		Slot Loss Coefficient					
14		Acceleration Factor (0 or 1)					
15		Slot Loss per VP (13 + 14)					
16		Slot Static Pressure (12 x 15)					
17		Duct Entry Loss Coefficient (Fig. 5-13)					
18		Acceleration Factor (1 or 0)					
19		Duct Entry Loss per VP (17 + 18)					
20		Duct Entry Loss (8 x 19)	"wg				
21	Other Losses	"wg		4.5	.08		
22	Hood Static Pressure (16 + 20 + 21)	"wg		4.5	.08		
23	Straight Duct Length	feet	9	2	20		
24	Friction Factor (H _f)		.0071	.0051	.0053		
25	Friction Loss per VP (23 x 24)		.06	.01	.11		
26	Number of 90 deg. Elbows		1				
27	Elbow Loss Coefficient		.19				
28	Elbow Loss per VP (26 x 27)		.19				
29	Number of Branch Entries (1 or 0)						
30	Entry Loss Coefficient						
31	Branch Entry Loss per VP (29 x 30)						
32	Special Fitting Loss Coefficients						
33	Duct Loss per VP (25 + 28 + 31 + 32)		.25	.01	.11		
34	Duct Loss (33 x 8)	"wg	.37	0	.05		
35	Duct Segment Static Pressure Loss (22 + 34)	"wg	.37	4.5	.13		
36	Other Losses (VP-VP _n , etc.)	"wg	.25				
37	Cumulative Static Pressure	"wg	-5.35	-9.85	.13		
38	Governing Static Pressure	"wg					
39	Corrected Volumetric Flow Rate	cfm					
40	Corrected Velocity	fpm					
41	Corrected Velocity Pressure	"wg					
42	Resultant Velocity Pressure	"wg					

FIGURE 5-9. Balanced design method (continued)

NOTES

1. Balancing at B: SP ratio = $-2.51/-2.18 = 1.15$, so the flow rate through the lower resistance run can be corrected. From Equation 5.1,

$$Q_{\text{corrected}} = 700 \sqrt{\frac{-2.51}{-2.18}} = 751 \text{ cfm}$$

From Equation 5.6,

$$VP_r = \frac{960}{1711} 1.49 + \frac{751}{1711} 1.29 = 1.40 \text{ "wg}$$

Note that the numbers for 3-B reflected the corrected values. The velocity pressure of 1.29 "wg corresponds to the new velocity after correcting the flow rate to 751 cfm ($751/0.165 = 4551 \text{ fpm}$, up from 4243 fpm).

The VP in duct B-A is 1.50, while the VP_r at A was 1.40, so there was an acceleration at B of 0.10. This is reflected in the overall pressure drop to point A ($2.51 + 0.10 + 1.64 = 4.25 \text{ "wg}$.) Column 19 is a convenient place to enter this additional SP drop.

2. Balancing at A: Initial SP ratio at A ($-4.25/-2.34 = 1.82$) is too high to allow flow rate increase, so 1-A was redesigned with an 18" dia. duct to result in an acceptable SP ratio ($-4.25 \div -3.61 = 1.18$). Then

$$Q_{\text{corrected}} = 9600 \sqrt{\frac{-4.25}{-3.61}} = 10416 \text{ cfm}$$

From Equation 5.6, $VP_r = 2.07 \text{ "wg}$. As the VP in the

segment from A to C was clearly less than VP_r , no correction to the SP was made.

3. This same procedure was followed at each of the other junction points until reaching the collector. At junction points J and H, the SP ratio was slightly over the 1.20 recommended. However, the flows were increased using Equation 5.1 anyway because reducing the diameter would have resulted in unacceptably high duct velocities and SP drops.
4. K is the inlet to that collector and the fan is labeled FAN. Assuming the collector loss of 4.5 "wg is the "flange-to-flange" loss, and 2 feet of straight duct separates the fan from the collector, it might be advisable to use a non-standard diameter duct equal to the fan's inlet to connect the two devices. That is why a 35.5" diameter was chosen here.
5. The duct loss calculations for the stack (FAN-L) show an overall loss of 0.05 "wg. However, there is an additional effect to consider. Most centrifugal fans have an exit area virtually the same as the inlet area. If so, then there would be an additional acceleration from approximately 0.42 "wg to the duct VP of 0.50 "wg, or 0.08 "wg. This acceleration "loss" is included in line 19 of the FAN-L column.
6. The above FSP calculation assumes that there are no losses due to the contraction from the fan exit to the 32" diameter duct. This is usually a small enough loss to ignore. For instance, if the contraction half-angle was 15 degrees with a loss factor of 0.08 (see Figure 5-16), the maximum error in ignoring this loss would be less than 0.02 "wg.

FIGURE 5-9. Balanced design method (continued)

VELOCITY PRESSURE METHOD CALCULATION SHEET							
Problem <u>#2 With Blast Gates</u>		Class _____	Designer _____	Date _____			
1	Duct Segment Identification		1-A	2-B	3-B	B-A	A-C
2	Target Volumetric Flow Rate	cfm	9600	960	700	1660	11260
3	Minimum Transport Velocity	fpm	4000	4000	4000	4000	4000
4	Maximum Duct Diameter	inches	20.97	6.63	5.66	8.72	22.72
5	Selected Duct Diameter	inches	20	6	5.5	8	22
6	Duct Area	ft ²	2.1917	0.1963	0.1650	0.3491	2.6398
7	Actual Duct Velocity	fpm	4400	4889	4243	4756	4265
8	Duct Velocity Pressure	"wg	1.21	1.49	1.12	1.41	1.13
9	H O O D S L O T S U C T I O N	Maximum Slot Area	ft ²	4.8			
10		Slot Area Selected	ft ²	4.8			
11		Slot Velocity	fpm	2000			
12		Slot Velocity Pressure	"wg	.25			
13		Slot Loss Coefficient		1.78			
14		Acceleration Factor (0 or 1)	0				
15		Slot Loss per VP (13 + 14)	1.78				
16		Slot Static Pressure (12 x 15)	.44				
17		Duct Entry Loss Coefficient (Fig. 5-13)	.25	.25	.25		
18		Acceleration Factor (1 or 0)	1	1	1		
19	Duct Entry Loss per VP (17 + 18)	1.25	1.25	1.25			
20	Duct Entry Loss (8 x 19)	"wg	1.51	1.86	1.40		
21	Other Losses	"wg					
22	Hood Static Pressure (16 + 20 + 21)	"wg	1.95	1.86	1.40		
23	Straight Duct Length	feet	13	3	4	18	34
24	Friction Factor (H _f)		0.0098	0.0425	0.0478	0.0299	0.0088
25	Friction Loss per VP (23 x 24)		.13	.13	.19	.54	.30
26	Number of 90 deg. Elbows		1	.67	1.67	2	
27	Elbow Loss Coefficient		.19	.19	.19	.19	
28	Elbow Loss per VP (26 x 27)		.19	.13	.32	.38	
29	Number of Branch Entries (1 or 0)			1	1		
30	Entry Loss Coefficient			.18	.18		
31	Branch Entry Loss per VP (29 x 30)			.18	.18		
32	Special Fitting Loss Coefficients						
33	Duct Loss per VP (25 + 28 + 31 + 32)		.32	.44	.69	.92	.30
34	Duct Loss (33 x 8)	"wg	.38	.65	.77	1.30	.34
35	Duct Segment Static Pressure Loss (22 + 34)	"wg	2.33	2.51	2.17	1.30	.34
36	Other Losses (VP-VP _n , etc.)	"wg				0.07	
37	Cumulative Static Pressure	"wg	-2.33	-2.51	-2.17	-3.88	-4.22
38	Governing Static Pressure	"wg	-3.88		-2.51		
39	Corrected Volumetric Flow Rate	cfm	GATE		GATE		
40	Corrected Velocity	fpm					
41	Corrected Velocity Pressure	"wg					
42	Resultant Velocity Pressure	"wg	1.24	1.34	1.34	1.24	1.14

FIGURE 5-10. Blast gate method

Temperature _____ Elevation _____								Remarks:																			
5-D	7a-D	D-C	C-E	8-F	9-F	F-G	1	Pertinent Information From Chapter 5																			
700	250	950	12210	1200	500	1700	2																				
3500	3500	3500	4000	3500	3500	3500	3																				
6.05	3.62	7.05	23.65	7.93	5.12	9.44	4	$Q_{corr} = Q_{design} \sqrt{\frac{SP_{gov}}{SP_{duct}}}$																			
6	3.5	7	22	7	5	9	5																				
0.1963	0.0668	0.2673	2.6398	0.2673	0.1364	0.4418	6																				
3565	3742	3555	4625	4490	3667	3848	7	$VP_r = \frac{Q_1}{Q_3} VP_1 + \frac{Q_2}{Q_3} VP_2$																			
0.79	0.87	0.79	1.33	1.26	0.84	0.92	8																				
							9																				
							10	Straight Duct Friction Loss $H_f = a \frac{V^b}{Q^c}$																			
							11																				
							12																				
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.81	.81	1.14	1.26	1.13	1.13	.92	42																				

FIGURE 5-10. Blast gate method (continued)

VELOCITY PRESSURE METHOD CALCULATION SHEET						
Problem #2 With Blast Gates		Class		Date		
1	Duct Segment Identification	7b-G	G-E	E-H	12-J	10-J
2	Target Volumetric Flow Rate	cfm	250	1950	14160	900
3	Minimum Transport Velocity	fpm	3500	3500	4000	4500
4	Maximum Duct Diameter	inches	3.62	10.1	25.48	6.06
5	Selected Duct Diameter	inches	3.5	10	24	6
6	Duct Area	ft ²	0.0668	0.5454	3.1416	0.1963
7	Actual Duct Velocity	fpm	3742	3575	4507	4584
8	Duct Velocity Pressure	"wg	0.87	0.80	1.27	1.31
9	HOOD SLOTS SECTION	Maximum Slot Area	ft ²			
10		Slot Area Selected	ft ²			
11		Slot Velocity	fpm			
12		Slot Velocity Pressure	"wg			
13		Slot Loss Coefficient				
14		Acceleration Factor	(0 or 1)			
15		Slot Loss per VP	(13 + 14)			
16		Slot Static Pressure	(12 x 15)			
17		Duct Entry Loss Coefficient	(Fig. 5-13)	1.0		.25
18		Acceleration Factor	(1 or 0)	1		1
19	Duct Entry Loss per VP	(17 + 18)	2.00		1.25	
20	Duct Entry Loss	(8 x 19)	"wg	1.74	1.64	
21	Other Losses	"wg				
22	Hood Static Pressure	(16 + 20 + 21)	"wg	1.74	1.64	
23	Straight Duct Length	feet	15	6	3.5	11
24	Friction Factor (H _f)		0.0840	0.0233	0.0078	0.0427
25	Friction Loss per VP	(23 x 24)	1.26	.14	.03	.47
26	Number of 90 deg. Elbows		.67	.67		.33
27	Elbow Loss Coefficient		.19	.19		.19
28	Elbow Loss per VP	(26 x 27)	.13	.13		.06
29	Number of Branch Entries	(1 or 0)	1			1
30	Entry Loss Coefficient		.18			.18
31	Branch Entry Loss per VP	(29 x 30)	.18			.18
32	Special Fitting Loss Coefficients					
33	Duct Loss per VP	(25 + 28 + 31 + 32)	1.57	.94	.03	.71
34	Duct Loss	(33 x 8)	"wg	1.37	.75	.94
35	Duct Segment Static Pressure Loss	(22 + 34)	"wg	3.11	.75	.03
36	Other Losses (VP-VP _r , etc.)	"wg				
37	Cumulative Static Pressure	"wg	-3.11	-3.86	-4.52	-2.58
38	Governing Static Pressure	"wg		-4.49		-2.58
39	Corrected Volumetric Flow Rate	cfm		GATE		GATE
40	Corrected Velocity	fpm				
41	Corrected Velocity Pressure	"wg				
42	Resultant Velocity Pressure	"wg	.92	1.26	1.29	1.00

FIGURE 5-10. Blast gate method (continued)

Temperature _____ Elevation _____								Remarks:																	
J-H	H-K	K-FAN	FAN-L				1	Pertinent Information From Chapter 5																	
2125	16285	16285	16285				2																		
4500	4500	2600	2600				3																		
9.31	25.76	33.9					4																		
9	26	35.5	34				5	$Q_{corr} = Q_{design} \frac{\sqrt{SP_{gov}}}{\sqrt{SP_{duct}}}$																	
0.4418	3.6870	6.8736	6.305				6																		
4810	4417	2369	2583				7																		
1.44	1.22	0.35	0.42				8																		
							9	$VP_r = \frac{Q_1}{Q_3} VP_1 + \frac{Q_2}{Q_3} VP_2$																	
							10																		
							11																		
							12																		
							13	Straight Duct Friction Loss $H_f = a \frac{V^b}{Q^c}$																	
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1.02	.31	4.50	.12				35																		
.44							36																		
-4.04	-4.83	-9.33	.12				37																		
-4.52							38																		
GATE							39																		
							40																		
							41																		
1.29							42																		

FIGURE 5-10. Blast gate method (continued)

corrections to the system design by the following means should be made.

The density variation equations of Chapter 1 (Section 1.4) demonstrate that if temperature increases or absolute pressure decreases, the density will decrease. For the *mass* flow rate at the hood(s) to remain the same, the flow rate must change if density changes. It is helpful to remember that a fan connected to a given system will exhaust the same volume flow rate regardless of air density. The mass of air moved, however, will be a function of the density.

5.13.1 Variable Temperature and/or Different Altitude:

Consider an exhaust system at sea level where 5000 cfm of air at 70 F is drawn into a hood. The air is then heated to 600 F and the density of the air leaving the heater becomes 0.0375 lbm/ft³. The flow rate downstream of the heater would be 10,000 actual cubic feet per minute (acfm) at the new density of 0.0375 lbm/ft³. This is true because the 50% decrease in density *must* correspond to a twofold increase in the volume flow rate since the mass flow rate has remained constant.

If this temperature effect is ignored and a fan selected for 5000 cfm is placed in the system, the hood flow rate will be well below that required to maintain contaminant control. The exact operating point of such a system would have to be recalculated based upon the operating point of the incorrectly sized fan.

5.13.2 Elevated Moisture: When air temperature is under 100 F, no correction for humidity is necessary. When air temperature exceeds 100 F and moisture content is greater than 0.02 lbs H₂O per pound of dry air, correction is required to determine fan operating RPM and power. Correction coefficients may be read from the psychrometric charts such as those illustrated in Figures 5-17 through 5-20.

5.13.3 Psychrometric Principles: The properties of moist air are presented on the psychrometric chart at a single pressure. These parameters define the physical properties of an air/water vapor mixture. *The actual gas flow rate and the density of the gas stream at the inlet of the fan must be known in order to select the fan.* The psychrometric chart provides the information required to calculate changes in the flow rate and density of the gas as it passes through the various exhaust system components. These properties are:

- **Dry-Bulb Temperature** is the temperature observed with an ordinary thermometer. Expressed in degrees Fahrenheit, it may be read directly on the chart and is indicated on the bottom horizontal scale.
- **Wet-Bulb Temperature** is the temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature. Expressed in degrees Fahrenheit, it is read directly at the intersection of the constant enthalpy line

with the 100% saturation curve.

- **Dew Point Temperature** is that temperature at which the air in an air/vapor mixture becomes saturated with water vapor and any further reduction of dry bulb temperature causes the water vapor to condense or deposit as drops of water. Expressed in degrees Fahrenheit, it is read directly at the intersection of the saturation curve with a horizontal line representing constant moisture content.
- **Percent Saturation** curves reflect the mass of moisture actually in the air as a percentage of the total amount possible at the various dry bulb and moisture content combinations. Expressed in percent, it may be read directly from the curved lines on the chart.
- **Density Factor** is a dimensionless quantity which expresses the ratio of the actual density of the mixture to the density of standard air (0.075 lbm/ft³). The lines representing density factor typically do not appear on low-temperature psychrometric charts when relative humidity or percent saturation curves are presented. A method of calculating the density of the gas defined by a point on the chart (when density factor curves are not presented) is discussed in Section 5.13.4.
- **Moisture Content**, or weight of water vapor, is the amount of water that has been evaporated into the air. In ordinary air, it is very low pressure steam and has been evaporated into the air at a temperature corresponding to the boiling point of water at that low pressure. Moisture content is expressed in grains of water vapor per pound of dry air (7000 grains = one pound) or pounds of water vapor per pound of dry air and is read directly from a vertical axis.
- **Enthalpy (Total Heat)** as shown on the psychrometric chart is the sum of the heat required to raise the temperature of a pound of air from 0 F to the dry-bulb temperature, plus the heat required to raise the temperature of the water contained in that pound of air from 32 F to the dew point temperature, plus the latent heat of vaporization, plus the heat required to superheat the vapor in a pound of air from the dew point temperature to the dry-bulb temperature. Expressed in British Thermal Units per pound of dry air, it is shown by following the diagonal wet-bulb temperature lines.
- **Humid Volume** is the volume occupied by the air/vapor mixture per pound of dry air and is expressed in cubic feet of mixture per pound of dry air. It is most important to understand the dimensions of this parameter and realize that *the reciprocal of humid volume is not density*. Humid volume is the parameter used most frequently in determining flow rate changes within a system as a result of mixing gases of different properties

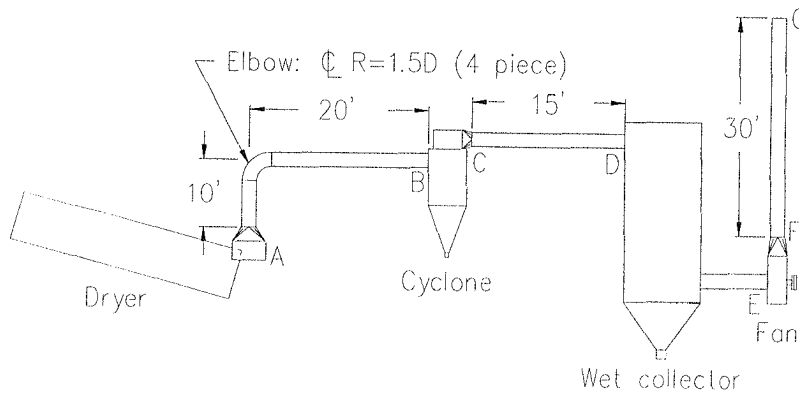


FIGURE 5-11. System layout

or when evaporative cooling occurs within the system.

5.13.4 Density Determination: When the quality of an air/vapor mixture is determined by a point on a psychrometric chart having a family of density factor curves, all that must be done to determine the actual density of the gas at the pressure reference for which the chart is drawn is to multiply the density factor taken from the chart by the density of standard air (0.075 lbm/ft³). Should relative humidity curves be presented on the chart in lieu of density factor curves, information available through dimensional analysis must be used to determine the actual density of the mixture. This can be done quite easily as follows: The summation of one pound of dry air plus the mass of the moisture contained within that pound of dry air divided by the humid volume will result in the actual density of the mixture.

$$\rho = \frac{1+W}{HV} \quad [5.8]$$

where:

ρ = density of the mix (lbm/ft³)

W = moisture content (lbm H₂O/lbm dry air)

HV = humid volume (ft³ mix/lbm dry air)

5.13.5 Hood Flow Rate Changes with Density: If the density of the air entering a hood is different from standard density due to changes in elevation, ambient pressure, temperature, or moisture, the flow rate through the hood should be changed to keep the mass flow rate the same as for standard air. This can be accomplished by multiplying the hood flow rate required for standard air by the ratio of the density of standard air to the actual ambient density.

The example shown in Figure 5-11 illustrates the effect of elevated moisture and temperature and a method of calculation:

EXAMPLE

GIVEN: The exit flow rate from a 60" x 24' dryer is 16,000 scfm plus removed moisture. The exhaust air temperature is 500 F. The drier delivers 60 tons/hr of dried material with

capacity to remove 5% moisture. Required suction at the dryer hood is -2.0 "wg; minimum conveying velocity must be 4000 fpm (see Figure 5-11).

It has been determined that the air pollution control system should include a cyclone for dry product recovery and a high-energy wet collector. These devices have the following operating characteristics:

- **Cyclone:** Pressure loss is 4.5 "wg at rated flow rate of 35,000 scfm. The pressure loss across any cyclone varies directly with any change in density and as the square of any change in flow rate from the rated conditions.
- **High-Energy Wet Scrubber:** The manufacturer has determined that a pressure loss of 20 "wg is required in order to meet existing air pollution regulations and has sized the collector accordingly. The humidifying efficiency of the wet collector is 90%.

NOTE: As a practical matter, a high energy scrubber as described in this example would have essentially 100% humidifying efficiency. The assumption of 90% humidifying efficiency along with a high pressure drop allows discussion of multiple design considerations in one example and was therefore adopted for instructional purposes.

- **Fan:** A size #34 "XYZ" fan with the performance shown in Table 5-3 has been recommended.

REQUIRED:

Size the duct and select fan RPM and motor size.

SOLUTION:

Step 1

Find the actual gas flow rate that must be exhausted from the dryer. This flow rate must include both the air used for drying and the water, as vapor, which has been removed from

the product. Since it is actual flow rate, it must be corrected from standard air conditions to reflect the actual moisture, temperature, and pressures which exist in the duct.

Step 1A:

Find the amount (weight) of water vapor exhausted.

Dryer discharge = 60 tons/hr of dried material (given)

Since the dryer has capacity to remove 5% moisture, the dryer discharge is $95\% \times$ dryer feed rate.

60 tons/hr dried material = (0.95) (dryer feed)

$$\text{dryer feed} = \frac{60 \text{ tons/hr}}{0.95} = 63.2 \text{ tons/hr}$$

$$\begin{aligned} \text{Moisture removed} &= (\text{feed rate}) - (\text{discharge rate}) \\ &= 63.2 \text{ tons/hr} - 60 \text{ tons/hr} \\ &= 6400 \text{ lbs/hr or } 106.7 \text{ lbm/min} \end{aligned}$$

Step 1B

Find the amount (weight) of dry air exhausted.

Dry air exhausted = 16,000 scfm at 70 F and 29.92 "Hg (0.075 lbs/ft³ density)

$$\begin{aligned} \text{Exhaust rate, lbs/min} &= (16,000 \text{ scfm})(0.075 \text{ lbs/ft}^3) \\ &= 1200 \text{ lbs/min dry air} \end{aligned}$$

Step 1C

Knowing the water-to-dry air ratio and the temperature of the mixture, it is possible to determine other quantities of the air-to-water mixture. This can be accomplished by the use of psychrometric charts (see Figures 5-17 to 5-20) which are most useful tools when working with humid air.

$W = 0.089 \text{ lbs H}_2\text{O/lb dry air}$

Dry bulb temperature = 500 F (given)

The intersection of the 500 F dry-bulb temperature line and the 0.089 lbs H₂O/lb dry air line can be located on the psychrometric chart (see Figure 5-12). Point #1 completely defines the quality of the air and water mixture. Other data relative to this specific mixture can be read as follows:

Dew Point Temperature:	122 F
Wet-Bulb Temperature:	145 F
Humid volume, ft ³ of mix/lb of dry air:	27.5 ft ³ /lb dry air
Enthalpy, BTU/lb of dry air:	235 BTU/lb dry air
Density factor, df:	0.53

Step 1D

Find actual gas flow rate, (acfm).

Exhaust flow rate, acfm = (humid volume)(weight of dry air/min). Humid volume, HV, was found in Step 1C as 27.5 ft³/lb. Weight of dry air/min was found in Step 1B as 1200 lb/min. Exhaust flow rate = (27.5 ft³/lb)(1200 lb/min) = 33,000 acfm.

Step 2

Size the duct. Minimum conveying velocity of 4000 fpm was given. Suction at the dryer exit of -2.0 "wg corresponds to hood suction.

The duct area equals the actual flow rate divided by the minimum duct velocity, or $A = 33,000 \div 4,000 = 8.25 \text{ ft}^2$. A 38" diameter duct with a cross-sectional area of 7.876 ft² should be chosen as this is the largest size available with an area smaller than calculated. Then the actual duct velocity would be $33,000 \text{ actual ft}^3/\text{min} \div 7.876 \text{ ft}^2 = 4,190 \text{ fpm}$.

Step 2A

The velocity pressure in the duct cannot be found using the equation $VP = (V \div 4005)^2$, as this equation is for standard air

TABLE 5-3. Fan Rating Table																						
Fan size No. 34						Inlet diameter = 34"												Max. safe rpm = 1700				
CFM	20" SP		22" SP		24" SP		26" SP		28" SP		30" SP		32" SP		34" SP		36" SP		38" SP		40" SP	
	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
14688	1171	73.3	1225	81.4	1277	89.8	1326	98.3	1374	107	1421	116	1466	125	1510	134	1552	143	1594	153	1634	162
16524	1181	81.8	1234	90.2	1286	98.8	1335	107	1382	116	1428	126	1472	135	1516	145	1557	155	1600	165	1639	175
18360	1191	90.2	1244	99.5	1294	108	1344	118	1391	127	1437	137	1481	146	1524	157	1565	167	1606	178	1645	188
20196	1204	99.9	1256	109	1306	119	1354	129	1400	139	1446	149	1490	160	1532	170	1574	181	1615	191	1654	202
22032	1217	110	1268	120	1318	130	1366	141	1412	151	1456	162	1499	173	1542	184	1584	196	1624	207	1663	218
23868	1230	120	1282	131	1331	142	1378	154	1424	165	1468	176	1511	187	1553	199	1594	211	1633	223	1672	235
25704	1245	131	1296	143	1345	155	1391	167	1437	179	1481	191	1524	203	1565	215	1606	227	1645	239	1683	252
27540	1261	143	1311	156	1359	168	1406	181	1450	193	1494	206	1537	219	1578	232	1618	245	1658	258	1695	271
29376	1277	156	1327	169	1374	182	1421	196	1465	209	1508	222	1550	236	1591	249	1631	263	1670	277		
31212	1295	170	1344	184	1391	197	1436	211	1480	225	1523	239	1564	253	1605	268	1644	282	1683	297		
33048	1313	184	1361	198	1407	213	1453	228	1496	242	1538	257	1580	272	1620	287	1659	302	1697	317		
34884	1331	198	1379	214	1425	229	1469	245	1513	260	1555	276	1595	291	1635	307	1674	323				

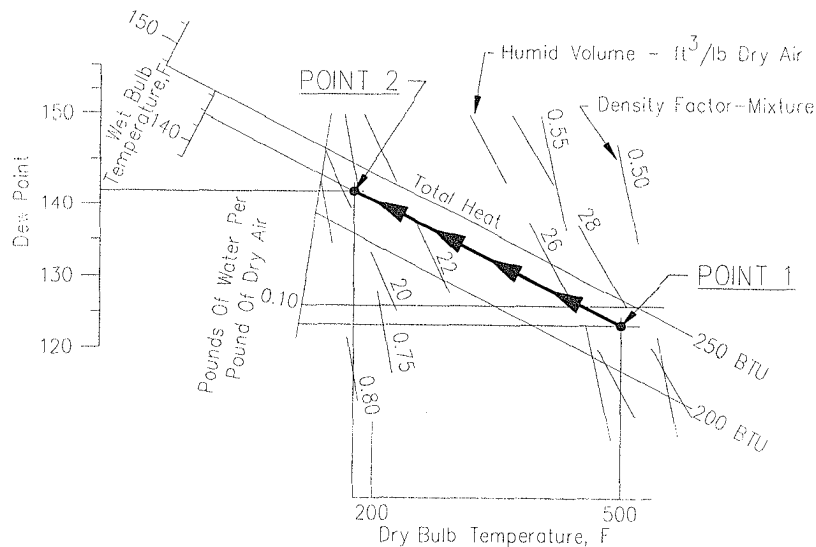


FIGURE 5-12. Psychrometric chart for humid air (see Figures 5-17 through 5-20)

only. The actual velocity pressure in the duct is given by

$$VP_{\text{actual}} = (df)(VP_{\text{std}})$$

where: df = density factor

As the density factor was determined in Step 1C, the actual velocity pressure in the duct will be

$$VP = (df)(VP_{\text{std}}) = (0.53)(1.09 \text{ "wg}) = 0.58 \text{ "wg}$$

Step 3

Calculate the pressure loss from A to B and determine static pressure at Point B.

The data from Figure 5-14 and Table 5-5 can be used directly. The static pressure loss through the duct can be found by multiplying the length of duct by the friction coefficient, adding the elbow loss coefficient, and multiplying the result by the duct velocity pressure:

$$\begin{aligned} \text{SP loss} &= [(0.0045)(30) + 0.27][0.58] = (0.405)(0.58) \\ &= 0.23 \text{ "wg} \end{aligned}$$

Then the static pressure at the inlet to the cyclone should be -2.23 "wg (hood suction plus friction and fitting losses).

Step 4

The pressure loss of the cyclone is provided by the manufacturer. In this example, the cyclone pressure loss is 4.5 "wg at a rated flow of $35,000 \text{ scfm}$. The pressure loss through a cyclone, as with duct, varies as the square of the change in flow rate and directly with change in density.

Therefore, the actual loss through the cyclone would be

$$(4.5) \left[\frac{33,000}{35,000} \right]^2 (0.53) = -2.12 \text{ "wg}$$

and the static pressure at the cyclone outlet would be -4.35 "wg .

Step 5

The calculation from Point C to D is the same as from A to B in Step 3. Thus, the static pressure at the wet collector inlet would be

$$-4.35 - (0.0045)(15)(0.58) = -4.39 \text{ "wg}$$

NOTE: Information for Steps 6 and 7 which involve calculation of changes in flow rate, density, etc., across the wet collector should be provided by the equipment manufacturer.

Step 6

An important characteristic of wet collectors is their ability to humidify a gas stream. The humidification process is generally assumed to be adiabatic (without gain or loss of heat to the surroundings). Therefore, water vapor is added to the mixture, but the enthalpy, expressed in BTU/lb dry air, remains unchanged. During the process of humidification, the point on the psychrometric chart that defines the quality of the mixture moves to the left, along a line of constant enthalpy, toward saturation.

All wet collectors do not have the same ability to humidify. If a collector is capable of taking an air stream to complete adiabatic saturation, it is said to have a humidifying efficiency of 100%. The humidifying efficiency of a given device may be expressed by either of the following equations:

$$\eta_h = \frac{T_i - T_o}{T_i - T_s} \times 100$$

where:

η_n = humidifying efficiency, %
 T_i = dry-bulb temperature at collector inlet, F
 T_o = dry-bulb temperature at collector outlet, F
 T_s = adiabatic saturation temperature, F

or

$$\eta_n = \frac{W_i - W_o}{W_i - W_s} \times 100$$

where:

W_i = moisture content in lb H₂O/lb dry air at inlet
 W_o = moisture content in lb H₂O/lb dry air at outlet
 W_s = moisture content in lb H₂O/lb dry air at adiabatic saturation conditions

Step 6A

Find the quality of the air to water mixture at Point 2, the collector outlet.

Humidifying Efficiency = 90% (given). Dry-bulb Temperature at Collector Inlet = 500 F (given). Adiabatic saturation temperature = 145 F from inspection of Psychrometric Chart.

$$90 = \frac{(500 - t_o)}{(500 - 145)} \times 100$$

where:

$$t_o = 180 \text{ F}$$

Then the air leaving the collector will have a dry-bulb temperature of 180 F and an enthalpy of 235 BTU/lb of dry air as the humidifying process does not change the total heat or enthalpy.

The point of intersection of 180 F dry bulb and 235 BTU/lb dry air on the psychrometric chart defines the quality of the air leaving the collector and allows other data to be read from the chart as follows:

Dew Point Temperature	143 F
Wet-Bulb Temperature	145 F
Humid Volume, ft ³ /lb dry air	20.5 ft ³ /lb dry air
Enthalpy, BTU/lb dry air	235 BTU/lb dry air
Density factor, df	0.76

Step 7

What is the exhaust flow rate in acfm and the density factor at the collector outlet?

Step 7A

Exhaust flow rate = (humid volume)(weight of dry air/min). Humid Volume from Step 6 is 20.5 ft³/lb dry air. Weight of dry air/min from Step 1B is 1200 lbs/min. Flow rate = (20.5 ft³/lb)(1200 lbs/min) = 24,600 acfm.

As the wet collector loss was stated to be 20 "wg, the static pressure at the wet collector outlet would be -24.39 "wg.

Step 7B

On low-pressure exhaust systems, where the negative pressure at the fan inlet is less than 20 "wg, the effect of the negative pressure is usually ignored. However, as the pressures decrease, or the magnitude of negative pressures increases, it is understood that gases expand to occupy a larger volume. Unless this larger volume is anticipated and the fan sized to handle the larger flow rate, it will have the effect of reducing the amount of air that is pulled into the hood at the beginning of the system. From the characteristic equation for the ideal gas laws, $PQ = wRT$ (where w = the mass flow rate in lbm/min), the pressure flow rate relationship is

$$P_1 Q_1 = P_2 Q_2$$

or

$$\frac{P_1}{P_2} = \frac{Q_2}{Q_1}$$

Up to this point, the air has been considered to be at standard atmospheric pressure which is 14.7 psia, 29.92 "Hg or 407 "wg. The pressure within the duct at Point E is -24.4 "wg and minus or negative only in relation to the pressure outside the duct which is 407 "wg. Therefore, the absolute pressure within the duct is 407 "wg - 24.4 "wg = 382.6 "wg.

$$\frac{407}{382.6} = \frac{Q_2}{24,600 \text{ cfm}}$$

$$Q_2 = 26,170 \text{ acfm}$$

Step 7C

Pressure also affects the density of the air. From $PQ = wRT$ the relationship

$$\frac{(w_1 + Q_1)RT_1}{(w_2 + Q_2)RT_2} = \frac{P_1}{P_2}$$

can be derived. Density factor is directly proportional to the density and the equation can be rewritten

$$\frac{P_1}{P_2} = \frac{df_1}{df_2}$$

Substitute

$$\frac{407}{382.6} = \frac{0.76}{df_2}$$

(df_1 was determined to be 0.76 in Step 6.)

$$df_2 = 0.71$$

Step 7D

The duct from the wet collector to the fan can now be sized.

The flow rate leaving the wet collector was 26,170 acfm. As the fan selected has a 34-in. diameter inlet (area = 6.305 ft²), it is logical to make the duct from the wet collector to the fan a 34-in. diameter. Thus, the velocity through the duct would be $26,170 \div 6.305 = 4,151$ fpm. The VP would be $(0.71)(4151 \div 4005)^2 = (0.71)(1.07) = 0.76$ "wg.

Step 7E

The duct pressure loss, based on 26,170 cfm and a 34-in. diameter duct, would be $(0.0052)(5)(0.76) = 0.02$ "wg. Therefore, the SP at the fan inlet would be -24.41 "wg.

Step 8

Calculate the pressure loss from fan discharge F to stack discharge G. Since the air is now on the discharge side of the fan, the pressure is very near atmospheric. No pressure correction is needed. The flow rate and density factor are 24,600 acfm and 0.76, respectively.

Assuming that the fan discharge area is nearly the same as at the fan inlet, the same 34-in. diameter duct would result in a velocity of 3902 fpm. The velocity pressure would be $(0.76)(3902 \div 4005)^2 = 0.72$ "wg.

From Table 5-5, the friction coefficient is 0.0052 and the frictional pressure loss for the 30 ft. high stack would be $(0.0052)(30)(0.72) = 0.11$ "wg. As the static pressure at the exit of the stack must be atmospheric, the static pressure at the fan exit will be positive.

Step 9

Determine actual fan static pressure.

$$\begin{aligned}\text{Actual FSP} &= \text{SP}_{\text{out}} - \text{SP}_{\text{in}} - \text{VP}_{\text{in}} \\ &= +0.11 - (-24.41) - 0.76 \\ &= 23.76 \text{ "wg}\end{aligned}$$

Step 10

Determine equivalent fan static pressure in order to enter fan rating table. Equivalent fan static pressure is determined by dividing the actual fan static pressure by the density factor at the fan inlet. This is necessary since fan rating tables are based on standard air.

$$\text{Equivalent FSP} = \frac{23.76}{0.71} = 33.46 \text{ "wg}$$

Step 11

Select fan from rating table using the equivalent fan SP and the fan inlet flow rate. Interpolating the fan rating table (Table 5-3) for 26,200 cfm at 33.5 "wg yields a fan speed of 1559 RPM at 217 BHP.

Step 12

Determine the actual required fan power. Since actual

density is less than standard air density, the actual required power is determined by multiplying by the density factor, or $(217 \text{ BHP})(0.71) = 154 \text{ BHP}$. If a damper is installed in the duct to prevent overloading of the motor, at cold start the motor need only be a 200 HP (see Chapter 6).

5.14 AIR CLEANING EQUIPMENT

Dusts, fumes, and toxic or corrosive gases should not be discharged to the atmosphere. Each exhaust system handling such materials should be provided with an adequate air cleaner as outlined in Chapter 4. As a rule, the exhaust fan should be located on the clean air side of such equipment. An exception is in the use of cyclone cleaners where the hopper discharge is not tightly sealed and better performance is obtained by putting the fan ahead of the collector.

5.15 EVASÉ DISCHARGE

An evasé discharge is a gradual enlargement at the outlet of the exhaust system (see Figure 5-16). The purpose of the evasé is to reduce the air discharge velocity efficiently; thus, the available velocity pressure can be regained and credited to the exhaust system instead of being wasted. Practical considerations usually limit the construction of an evasé to approximately a 10° angle (5° side angle) and a discharge velocity of about 2000 fpm (0.25 "wg velocity pressure) for normal exhaust systems. Further streamlining or lengthening the evasé yields diminishing returns.

It should be noted, however, that for optimum vertical dispersion of contaminated air, many designers feel that the discharge velocity from the stack should not be less than 3000–3500 fpm. When these considerations prevail, the use of an evasé is questionable.

The following example indicates the application of the evasé fitting. It is not necessary to locate the evasé directly after the outlet of the fan. It should be noted that, depending upon the evasé location, the static pressure at the fan discharge may be below atmospheric pressure, i.e., negative (–), as shown in this example.

EXAMPLE

Duct No.	Dia.	Q	V	VP	SP
1 Fan Inlet	20	8300	3800	0.90	-7.27
2 Fan Discharge = 16.5 x 19.5		8300	3715	0.86	
3 Round Duct Connection	20		3800	0.90	
4 Evasé Outlet	28		1940	0.23	0

To calculate the effect of the evasé, see Figure 5-16 for expansion at the end of the duct where the Diameter Ratio, $D_4 \div D_3 = 28 \div 20 = 1.4$ and Taper length $L/D = 40 \div 20 = 2$.

$$R = 0.52 \times 70\% \text{ (since the evasé is within 5 diameters of the fan outlet)}$$

$$VP_3 = 0.9 \text{ as given}$$

$$SP_4 = 0 \text{ (since the end of the duct is at atmospheric pressure)}$$

$$\begin{aligned} SP_3 &= SP_4 - R(VP_3) \\ &= 0 - (0.52)(0.70)(0.90) \\ &= -0.33 \text{ "wg} \end{aligned}$$

$$\begin{aligned} FSP &= SP_{\text{outlet}} - SP_{\text{inlet}} - VP_{\text{inlet}} \\ &= -0.33 - (-7.27) - 0.9 = 6.04 \text{ "wg} \end{aligned}$$

5.16 EXHAUST STACK OUTLETS

The final component of the ventilation system is the exhaust stack, an extension of the exhaust duct above the roof. There are two reasons for the placement of an exhaust stack on a ventilation system. First, the air exhausted by a local exhaust system should escape the building envelope. Second, once it has escaped the building envelope, the stack should provide sufficient dispersion so that the plume does not cause an unacceptable situation when it reaches the ground. This brief description of stack design will address only the first concern.

When placing an exhaust stack on the roof of a building, the designer must consider several factors. The most important is the pattern of the air as it passes the building. Even in the case of a simple building design with a perpendicular wind, the air flow patterns over the building can be complex to analyze. Figure 5-28a shows the complex interaction between the building and the wind at height H . A stagnation zone is formed on the upwind wall. Air flows away from the stagnation zone resulting in a down draft near the ground. Vortices are formed by the wind action resulting in a recirculation zone along the front of the roof or roof obstructions, down flow along the downwind side, and forward flow along the upwind side of the building.

Figure 5-28b shows a schematic of the critical zones formed within the building cavity. A recirculation zone is formed at the leading edge of the building. A recirculation zone is an area where a relatively fixed amount of air moves in a circular fashion with little air movement through the boundary. A stack discharging into the recirculation zone can contaminate the zone. Consequently, all stacks should penetrate the recirculation zone boundary.

The high turbulence region is one through which the air passes; however, the flow is highly erratic with significant downward flow. A stack that discharges into this region will contaminate anything downwind of the stack. Consequently, all stacks should extend high enough that the resulting plume does not enter the high turbulence region upwind of an air intake.

Because of the complex flow patterns around simple buildings, it is almost impossible to locate a stack that is not influenced by vortices formed by the wind. Tall stacks are often used to reduce the influence of the turbulent flow, to

release the exhaust air above the influence of the building and to prevent contamination of the air intakes. Selection of the proper location is made more difficult when the facility has several supply and exhaust systems and when adjacent buildings or terrain cause turbulence around the facility itself.

When locating the stack and outdoor air inlets for the air handling systems, it is often desirable to locate the intakes upwind of the source. However, often there is no true upwind position. The wind in all locations is variable. Even when there is a natural prevailing wind, the direction and speed are constantly changing. If stack design and location rely on the direction of the wind, the system will clearly fail.

The effect of wind on stack height varies with speed:

- At very low wind speeds, the exhaust jet from a vertical stack will rise above the roof level resulting in significant dilution at the air intakes.
- Increasing wind speed will decrease plume rise and consequently decrease dilution.
- Increasing wind speed will increase turbulence and consequently increase dilution.

The prediction of the location and form of the recirculation cavity, high turbulence region and roof wake is difficult. However, for wind perpendicular to a rectangular building, the height (H) and the width (W) of the upwind building face determine the airflow patterns. The critical dimensions are shown in Figure 5-28b. According to Wilson,^(5,6) the critical dimensions depend on a scaling coefficient (R) which is given by:

$$R = B_s^{0.67} \times B_l^{0.33} \quad [5.9]$$

where B_s is the smaller and B_l is the larger of the dimensions H and W . When B_l is larger than $8B_s$, use $B_l = 8B_s$ to calculate the scaling coefficient. For a building with a flat roof, Wilson^(5,7) estimated the maximum height (H_c), center (X_c), and lengths (L_c) of the recirculation region as follows:

$$H_c = 0.22 R \quad [5.10]$$

$$X_c = 0.5 R \quad [5.11]$$

$$L_c = 0.9 R \quad [5.12]$$

In addition, Wilson estimated the length of the building wake recirculation region by:

$$L_r = 1.0 R \quad [5.13]$$

The exhaust air from a stack often has not only an upward momentum due to the exit velocity of the exhaust air but buoyancy due to its density as well. For the evaluation of the stack height, the effective height is used (see Figure 5-29a). The effective height is the sum of the actual stack height (H_s), the rise due to the vertical momentum of the air, and any wake downwash effect that may exist. A wake downwash occurs when air passing a stack forms a downwind vortex. The vortex

will draw the plume down, reducing the effective stack height (see Figure 5-29b). This vortex effect is eliminated when the exit velocity is greater than 1.5 times the wind velocity. If the exit velocity exceeds 3000 fpm, the momentum of the exhaust air reduces the potential downwash effect.

The ideal design extends the stack high enough that the expanding plume does not meet the wake region boundary. More realistically, the stack is extended so that the expanding plume does not intersect the high turbulence region or any recirculation cavity. According to Wilson,^(5,6) the high turbulence region boundary (Z_2) follows a 1:10 downward slope from the top of the recirculation cavity.

To avoid entrainment of exhaust gas into the wake, stacks must terminate above the recirculation cavity. The effective stack height to avoid excessive reentry can be calculated by assuming that the exhaust plume spreads from the effective stack height with a slope of 1:5 (see Figure 5-28b). The first step is to raise the effective stack height until the lower edge of the 1:5 sloping plume avoids contact with all recirculation zone boundaries. The zones can be generated by rooftop obstacles such as air handling units, penthouses or architectural screens. The heights of the cavities are determined by Equations 5.10, 5.11 and 5.12 using the scaling coefficient for the obstacle. Equation 5.13 can be used to determine the length of the wake recirculation zone downwind of the obstacle.

If the air intakes, including windows and other openings, are located on the downwind wall, the lower edge of the plume with a downward slope of 1:5 should not intersect with the recirculation cavity downwind of the building. The length of the recirculation cavity (L_r) is given by Equation 5.13. If the air intakes are on the roof, the downward plume should not intersect the high turbulence region above the air intakes. When the intake is above the high turbulence boundary, extend a line from the top of the intake to the stack with a slope of 1:5. When the intake is below the high turbulence region boundary, extend a vertical line to the boundary, then extend back to the stack with a slope of 1:5. This allows the calculation of the necessary stack height. The minimum stack height can be determined for each air intake. The maximum of these heights would be the required stack height.

In large buildings with many air intakes, the above procedure will result in very tall stacks. An alternate approach is to estimate the amount of dilution that is afforded by stack height, distance between the stack and the air intake and internal dilution that occurs within the system itself. This approach is presented in the "Airflow Around Buildings" chapter in the *Fundamentals* volume of the 1993 ASHRAE Handbook.^(5,8)

5.16.1 Stack Considerations:

1. Discharge velocity and gas temperature influence the effective stack height.

2. Wind can cause a downwash into the wake of the stack reducing the effective stack height. Stack velocity should be at least 1.5 times the wind velocity to prevent downwash.
3. A good stack velocity is 3000 fpm because it:
 - Prevents downwash for winds up to 2000 fpm (22 mph). Higher wind speeds have significant dilution effects.
 - Increases effective stack height.
 - Allows selection of a smaller centrifugal exhaust fan to provide a more stable operation point on the fan curve (see Chapter 6).
 - Provides conveying velocity if there is dust in the exhaust or there is a failure of the air cleaning device.
4. High exit velocity is a poor substitute for stack height. For example, a flush stack requires a velocity over 8000 fpm to penetrate the recirculation cavity boundary.
5. The terminal velocity of rain is about 2000 fpm. A stack velocity above 2600 fpm will prevent rain from entering the stack when the fan is operating.
6. Locate stacks on the highest roof of the building when possible. If not possible, a much higher stack is required to extend beyond the wake of the high bay, penthouse, or other obstacle.
7. The use of an architectural screen should be avoided. The screen becomes an obstacle and the stack must be raised to avoid the wake effect of the screen.
8. The best stack shape is a straight cylinder. If a drain is required, a vertical stack head is preferred (see Figure 5-30). In addition, the fan should be provided with a drain hole and the duct should be slightly sloped toward the fan.
9. Rain caps should not be used. The rain cap directs the air toward the roof, increases the possibility of reentry, and causes exposures to maintenance personnel on the roof. Moreover, rain caps are not effective. A field study^(5,9) with a properly installed standard rain cap showed poor performance. A 12-inch diameter stack passed 16% of all rain and as high as 45% during individual storms.
10. Separating the exhaust points from the air intakes can reduce the effect of reentry by increasing dilution.
11. In some circumstances, several small exhaust systems can be manifolded to a single exhaust duct to provide internal dilution thereby reducing reentry.
12. A combined approach of vertical discharge, stack height, remote air intakes, proper air cleaning device, and internal dilution can be effective in reducing the

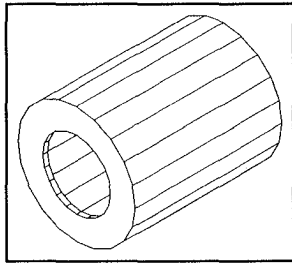
consequences of reentry.

13. A tall stack is not an adequate substitute for good emission control. The reduction achieved by properly designed air cleaning devices can have a significant impact on the potential for reentry.

5.17 AIR BLEED-INS

Bleed-ins are used at the ends of branch ducts to provide additional air flow rates to transport heavy material loads as in woodworking at saws and jointers or at the ends of a main duct to maintain minimum transport velocity when the system has been oversized deliberately to provide for future expansion. Some designers use bleed-ins also to introduce additional air to an exhaust system to reduce air temperature and to assist in balancing the system.

EXAMPLE



End cap bleed-in (see sketch). Consider it to be an orifice or slot. From Figure 5-13, $h_e = 1.78$ VP.

1. Calculate SP for branch duct to junction (X).
2. Determine flow rate in main duct according to design or future capacity or determine Q bleed-in directly from temperature or moisture considerations.
3. $Q_{\text{bleed-in}} = (Q_{\text{main duct}}) - (Q_{\text{branch}})$
4. $SP_{\text{bleed-in}} = SP_{\text{branch as calculated}} = (h_e + 1 \text{ VP}) = (1.78 + 1.0) \text{ VP}$
5. $VP_{\text{bleed-in}} = \frac{SP}{(1.78 + 1.0)} = \frac{SP}{2.78}$
6. Velocity, bleed-in from VP and Table 5-7a.
7. $\text{Area bleed-in} = \frac{Q_{\text{bleed-in}}}{V_{\text{bleed-in}}}$

5.18 OPTIMUM ECONOMIC VELOCITY

In systems which are intended to carry dust, a minimum conveying velocity is necessary to ensure that the dust will not settle in the duct. Also, when a system is installed in a quiet area, it may be necessary to keep velocities below some maximum to avoid excessive duct noise. When axial flow fans are used, duct velocities of 1000 to 1500 fpm are preferred. In a gas or vapor exhaust system installed in a typical factory

environment where none of these restrictions apply, the velocity may be selected to yield the lowest annual operating cost.

To determine the optimum economic velocity, the system must first be designed at any assumed velocity and the total initial costs of duct material, fabrication, and installation estimated.^(5,10)

This optimum economic velocity may range from under 2000 fpm to over 4000 fpm. Lengthy expected service periods and system operating times tend to lower the optimum while high interest rates and duct costs tend to raise the optimum. In general, a velocity of 2500 to 3000 fpm will not result in equivalent total annual costs much in excess of the true optimum.

5.19 CONSTRUCTION GUIDELINES FOR LOCAL EXHAUST SYSTEMS

Ducts are specified most often for use in the low static pressure range (-10 "wg to $+10$ "wg), but higher static pressures are occasionally encountered. The duct conveys air or gas which is sometimes at high temperatures and often contaminated with abrasive particulate or corrosive aerosols. Whether conditions are mild or severe, correct design and competent installation of ducts and hoods are necessary for proper functioning of any ventilation system. The following minimum specifications are recommended.

Exhaust systems should be constructed with materials suitable for the conditions of service and installed in a permanent and workman-like manner. To minimize friction loss and turbulence, the interior of all ducts should be smooth and free from obstructions — especially at joints.

5.19.1 Materials: Ducts are constructed of black iron, which has been welded, flanged, and gasketed; or of welded galvanized sheet steel unless the presence of corrosive gases, vapors, and mists or other conditions make such material impractical. Arc welding of black iron lighter than 18 gauge is not recommended. Galvanized construction is not recommended for temperatures exceeding 400 F. The presence of corrosive gases, vapor, and mist may require the selection of corrosive resistant metals, plastics, or coatings. It is recommended that a specialist be consulted for the selection of materials best suited for applications when corrosive atmospheres are anticipated. Table 5-4 provides a guide for selection of materials for corrosive conditions.

5.19.2 Construction:

1. There are four classifications for exhaust systems on noncorrosive applications:

Class 1 — Light Duty: Includes nonabrasive applications (e.g., replacement air, general ventilation, gaseous emissions control).

Class 2 — Medium Duty: Includes applications with

moderately abrasive particulate in light concentrations (e.g., buffing and polishing, woodworking, grain dust).

Class 3 — Heavy Duty: Includes applications with high abrasive in low concentrations (e.g., abrasive cleaning operations, dryers and kilns, boiler breeching, sand handling).

Class 4 — Extra Heavy Duty: Includes applications with highly abrasive particles in high concentrations (e.g., materials conveying high concentrations of particulate in all examples listed under Class 3 — usually used in heavy industrial plants such as steel mills, foundries, mining, and smelting).

2. For most conditions, round duct is recommended for industrial ventilation, air pollution control, and dust collecting systems. Compared to non-round duct, it provides for lower friction loss, and its higher structural integrity allows lighter gauge materials and fewer reinforcing members. Round duct should be constructed in accordance with the Reference 5.11. Metal thickness required for round industrial duct varies with classification, static pressure, reinforcement, and span between supports. Metal thicknesses required for the four classes are based on design and use experience.
3. Rectangular ducts should only be used when space requirements preclude the use of round construction. Rectangular ducts should be as nearly square as possible to minimize resistance, and they should be constructed in accordance with Reference 5.12.
4. For many applications, spiral wound duct is adequate and less expensive than custom construction. However, spiral wound duct should not be used for Classes 3 and 4 because it does not withstand abrasion well. Elbows, branch entries, and similar fittings should be fabricated, if necessary, to achieve good design. Special considerations concerning use of spiral duct are as follows:
 - A. Unless flanges are used for joints, the duct should be supported close to each joint, usually within 2 in. Additional supports may be needed. See Reference 5.11.
 - B. Joints should be sealed by methods shown to be adequate for the service.
 - C. Systems may be leak tested after installation at the maximum expected static pressure. The acceptable leakage criteria, often referred to as leakage class, should be carefully selected based on the hazards associated with the contaminant.
5. The following formula^(5,11) can be used for specifying ducts to be constructed of metals other than steel. For a duct of infinite length, the required thickness may be determined from:

$$\frac{t}{D} = \sqrt[3]{0.035714 p \frac{(1-v^2)}{E} (52+D)}$$

where:

t = the thickness of the duct in inches

D = the diameter of the duct in inches

p = the intensity of the negative pressure on the duct in psi

E = modulus of elasticity in psi

v = Poisson's ratio

The above equation for Class 1 ducts incorporates a safety factor which varies linearly with the diameter (D), beginning at 4 for small ducts and increasing to 8 for duct diameters of 60 in. This safety factor has been adopted by the sheet metal industry to provide for lack of roundness; excesses in negative pressure due to particle accumulation in the duct and other manufacturing or assembly imperfections unaccounted for by quality control; and tolerances provided by design specifications.

Additional metal thickness must be considered for Classes 2, 3 and 4. The designer is urged to consult the Sheet Metal and Air Conditioning Contractors National Association (SMACNA) standards for complete engineering design procedures.

6. Hoods should be a minimum of two gauges heavier than straight sections of connecting branches, free of sharp edges or burrs, and reinforced to provide necessary stiffness.
7. Longitudinal joints or seams should be welded. All welding should conform to the standards established by the American Welding Society (AWS) structural code.^(5,13) Double lock seams are limited to Class 1 applications.
8. Duct systems subject to wide temperature fluctuations should be provided with expansion joints. Flexible materials used in the construction of expansion joints should be selected with temperature and corrosion conditions considered.
9. Elbows and bends should be a minimum of two gauges heavier than straight lengths of equal diameter and have a centerline radius of at least two and preferably two and one-half times the pipe diameter (see Figure 5-21). Large centerline radius elbows are recommended where highly abrasive dusts are being conveyed.
10. Elbows of 90° should be of a five piece construction for round ducts up to 6 in. and of a seven piece construction for larger diameters. Bends less than 90° should have a proportional number of pieces. Prefabricated elbows of smooth construction may be used

TABLE 5-4. Typical Physical and Chemical Properties of Fabricated Plastics and Other Materials

Chemical Type	Trade Names	Max. Opr. Temp., F	Flam-mability	Resistance to							
				Gasoline	Mineral Oil	Strong Alk.	Weak Alk.	Strong Acid	Weak Acid	Salt Solution	Solvents
Urea Formaldehyde	Beetle Plaskon Sylplast	170	Self Ext.	Good	Good	Unac.	Fair	Poor	Poor	—	Good
Melamine Formaldehyde	Cymel Plaskon Resimene	210-300	Self Ext.	Good	Good	Poor	Good	Poor	Good	—	Good
Phenolic	Bakelite Durite Durez G.E. Resinox	250-450	Self Ext.	Fair	—	Poor	Fair	Poor	Fair	—	Fair
Alkyd	Plaskon	—	Self Ext.	Good	—	Unac.	Poor	—	Good	—	Good
Silicone	Bakelite G.E.	550	—	Good	Good	—	—	Good	Good	—	Unac.
Epoxy	Epiphem Araldite Maraset Renite Tool Plastik Epon Resin	50-200	Self Ext.	Good	—	Good	Good	Good	Good	—	Good
Cast Phenolic	Marblette	—	Self Ext.	—	—	Unac.	Fair	Good	Good	—	Good to Unac.
Allyl & Polyester	Laminac Bakelite Plaskon Glykon Paraplex	300-450	Self Ext.	—	—	Poor	Fair	Poor	Fair	—	Fair
Acrylic	Lucite Plexiglas Wascoline	140-200	0.5-2.0 in/min	—	—	—	Good	Unac.	Good	—	Good to Unac.
Polyethylene	Tenite Irrathene	140-200	Slow Burning	—	—	—	—	—	—	—	Unac.
Tetrafluoroethylene	Teflon	500	Non-Fl.	Good	—	Good	Good	Good	Good	—	Good
Chlorotrifluoroethylene	Kel F	—	—	—	—	—	—	—	—	—	—
Polyvinyl Formal & Butyral	Vinylite Butacite Saflex Butvar Formuaré	—	Slow Burning	Good	Good	Good	Good	Unac.	Unac.	—	Unac.
Vinyl Chloride Polymer & Copolymer	Krene Bakelite Vinyl Dow pvc Vygen	130-175	Slow Burning	—	—	Good	Good	Good	Good	—	Unac.
Vinylidene Chloride	Saran	160-200	Self Ext.	Good	Good	Good	Good	Good	Good	—	Fair
Styrene	Bakelite Catalin Styron Dylene Luxtrex	150-165	0.5-2.0 in/min	Unac.	Fair	Good	Good	—	—	Good	Poor
Polystyrene Reinforced with Fibrous Glass				Unac.	Fair	Good	Good	—	—	Good	Poor
Cellulose Acetate	Celanese Acetate Tenite	Thermo Plastic	0.5-2.0 in/min	Good	Good	Unac.	Unac.	Unac.	Fair	—	Poor
Nylon	Plaskon Zytel Tynex	250	Self Ext.	Good	Good	Good	Good	Unac.	Good	—	Good
Glass	Pyrex	450	Non-Fl.	Good	Good	Good	Good	Good	Good	Good	Good

NOTE: Each situation must be thoroughly checked for compatibility of materials during the design phase or if usage is changed.

(see Figure 5-22 for heavy duty elbows).

11. Where the air contaminant includes particulate that may settle in the ducts, clean-out doors should be provided in horizontal runs, near elbows, junctions, and vertical runs. The spacing of clean-out doors should not exceed 12 ft for ducts of 12 in. diameter and less but may be greater for larger duct sizes (see Figure 5-23). Removable caps should be installed at all terminal ends, and the last branch connection should not be more than 6 in. from the capped end.
12. Transitions in mains and sub-mains should be tapered. The taper should be at least five units long for each one unit change in diameter or 30° included angle (see Figure 5-25).
13. All branches should enter the main at the center of the transition at an angle not to exceed 45° with 30° preferred. To minimize turbulence and possible particulate fall out, connections should be to the top or side of the main with no two branches entering at opposite sides (see Figure 5-26).
14. Where condensation may occur, the duct system should be liquid tight and provisions should be made for proper sloping and drainage.
15. A straight duct section of at least six equivalent duct diameters should be used when connecting to a fan (see Figure 5-27). Elbows or other fittings at the fan inlet will seriously reduce the volume discharge (see Figures 6-23, 6-24 and AMCA 201^(5,14)). The diameter of the duct should be approximately equal to the fan inlet diameter.
16. Discharge stacks should be vertical and terminate at a point where height or air velocity limit re-entry into supply air inlets or other plant openings (see Figures 5-28 and 5-29).

5.19.3 System Details:

1. Provide duct supports of sufficient capacity to carry the weight of the system plus the weight of the duct half filled with material and with no load placed on connecting equipment. [See SMACNA standards.^(5,11, 5,12)]
2. Provide adequate clearance between ducts and ceilings, walls and floors for installation and maintenance.
3. Install fire dampers, explosion vents, etc., in accordance with the National Fire Protection Association Codes and other applicable codes and standards.
4. Avoid using blast gates or other dampers. However, if blast gates are used for system adjustment, place each in a vertical section midway between the hood and the junction. To reduce tampering, provide a means of locking dampers in place after the adjustments have been made. (See Figure 5-24 for types.)

5. Allow for vibration and expansion. If no other considerations make it inadvisable, provide a flexible connection between the duct and the fan. The fan housing and drive motor should be mounted on a common base of sufficient weight to dampen vibration or on a properly designed vibration isolator.
6. Exhaust fans handling explosive or flammable atmospheres require special construction (see Section 6.3.9).
7. Do not allow hoods and duct to be added to an existing exhaust system unless specifically provided for in the original design or unless the system is modified.
8. Locate fans and filtration equipment such that maintenance access is easy. Provide adequate lighting in penthouses and mechanical rooms.

5.19.4 Codes: Where federal, state, or local laws conflict with the preceding, the more stringent requirement should be followed. Deviation from existing regulations may require approval.

5.19.5 Other Types of Duct Materials:

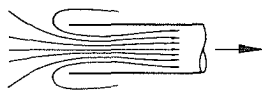
1. Avoid use of flexible ducts. Where required, use a noncollapsible type that is no longer than necessary. Refer to the manufacturer's data for friction and bend losses.
2. Commercially available seamless tubing for small duct sizes (i.e., up to 6 in.) may be more economical on an installed cost basis than other types.
3. Plastic pipe may be the best choice for some applications (e.g., corrosive conditions at low temperature; see Table 5-4.) For higher temperatures, consider fiberglass or a coated duct.
4. Friction losses for non-fabricated duct will probably be different than shown in Tables 5-5 and 5-6. For specific information, consult manufacturer's data.

5.19.6 Testing: The exhaust system should be tested and evaluated (see Chapter 9). Openings for sampling should be provided in the discharge stack or duct to test for compliance with air pollution codes or ordinances.

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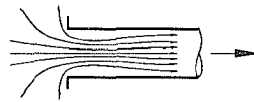
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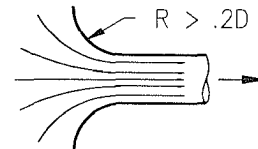
$$h_e = 0.93 VP_d$$

PLAIN DUCT END



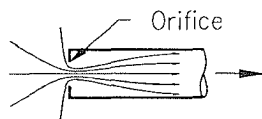
$$h_e = 0.49 VP_d$$

FLANGED DUCT END



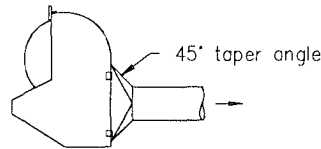
$$h_e = 0.04 VP_d$$

BELLMOUTH ENTRY



$$h_e = 1.78 VP_{\text{Orifice}}$$

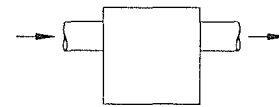
SHARP-EDGED ORIFICE



$$h_e = 0.4 VP_d \text{ (tapered take-off)}$$

$$h_e = 0.65 VP_d \text{ (no taper)}$$

STANDARD GRINDER HOOD

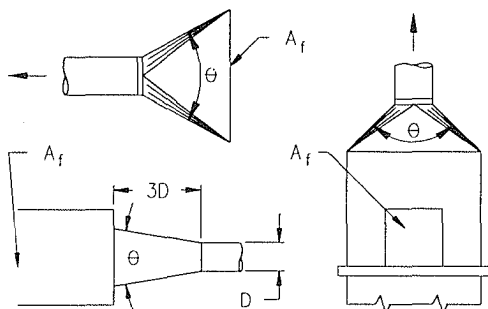


$$h_e = 1.5 VP_d$$

TRAP OR SETTLING CHAMBER

TAPERED HOODS

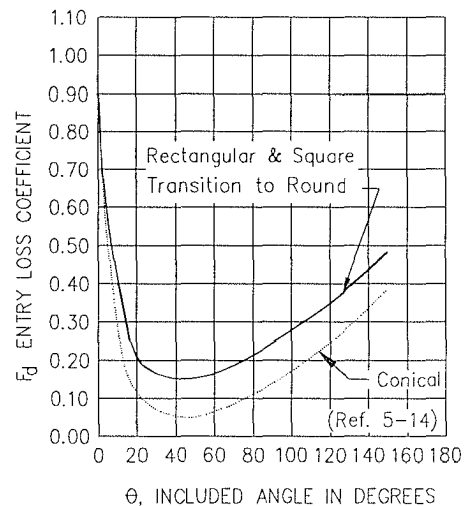
Flanged or unflanged; round, square or rectangular. θ is the major angle on rectangular hoods.



Face area (A_f) at least 2 times the duct area.

θ	ENTRY LOSS (h_d)	
	ROUND	RECTANGULAR
15°	0.15 VP	0.25 VP
30°	0.08 VP	0.16 VP
45°	0.06 VP	0.15 VP
60°	0.08 VP	0.17 VP
90°	0.15 VP	0.25 VP
120°	0.26 VP	0.35 VP
150°	0.40 VP	0.48 VP
180°	0.50 VP	0.50 VP

VP = Duct VP = VP_d
 Note: 180° values represent round ducts built into back of booth or hood without a rectangular to round transition.

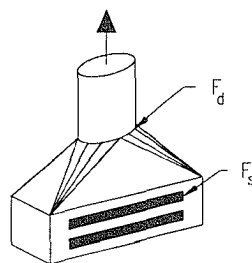


COMPOUND HOODS

A compound hood, such as the slot/plenum shown to the right, would have 2 losses, one through the slot and the other through the transition into the duct.

The slot entry loss coefficient, F_s , would have a value typically in the range of 1.00 to 1.78 (see Chapters 3 and 10).

The duct entry loss coefficient is given by the above data for tapered hoods.



$$h_e = F_s VP_s + F_d VP_d$$

MISCELLANEOUS VALUES

HOOD	ENTRY LOSS COEFFICIENT F_d
Abrasive blast chamber	1.0
Abrasive blast elevator	2.3
Abrasive separator	2.3
Elevators (enclosures)	0.69
Flanged pipe plus close elbow	0.8
Plain pipe plus close elbow	1.60

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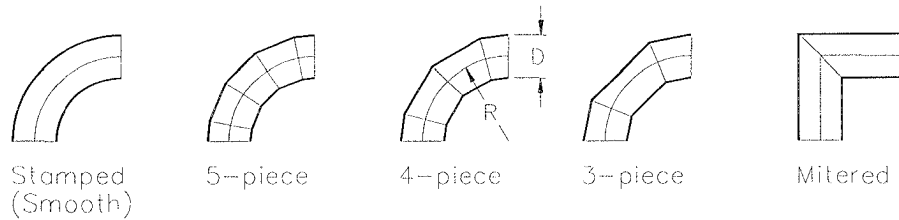
HOOD ENTRY LOSS COEFFICIENTS

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FIGURE

5-13



	R/D					
	0.5	0.75	1.00	1.50	2.00	2.50
Stamped	0.71	0.33	0.22	0.15	0.13	0.12
5-piece	--	0.46	0.33	0.24	0.19	0.17*
4-piece	--	0.50	0.37	0.27	0.24	0.23*
3-piece	0.90	0.54	0.42	0.34	0.33	0.33*

* extrapolated from published data

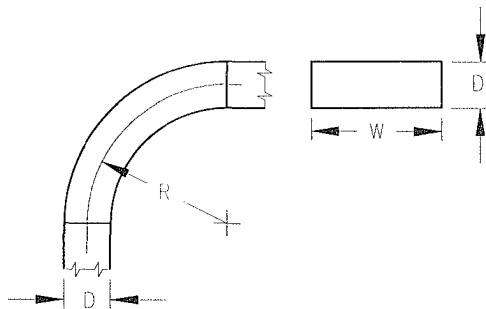
OTHER ELBOW LOSS COEFFICIENTS

Mitered, no vanes	1.2
Mitered, turning vanes	0.6
Flatback (R/D = 2.5)	0.05 (see Figure 5-23)

NOTE: Loss factors are assumed to be for elbows of "zero length." Friction losses should be included to the intersection of centerlines.

ROUND ELBOW LOSS COEFFICIENTS

(Ref. 5.13)



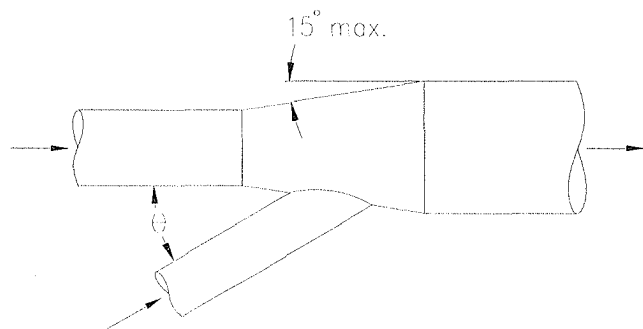
R/D	Aspect Ratio, W/D					
	0.25	0.5	1.0	2.0	3.0	4.0
0.0 (Mitered)	1.50	1.32	1.15	1.04	0.92	0.86
0.5	1.36	1.21	1.05	0.95	0.84	0.79
1.0	0.45	0.28	0.21	0.21	0.20	0.19
1.5	0.28	0.18	0.13	0.13	0.12	0.12
2.0	0.24	0.15	0.11	0.11	0.10	0.10
3.0	0.24	0.15	0.11	0.11	0.10	0.10

SQUARE & RECTANGULAR ELBOW LOSS COEFFICIENTS

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DUCT DESIGN DATA ELBOW LOSSES

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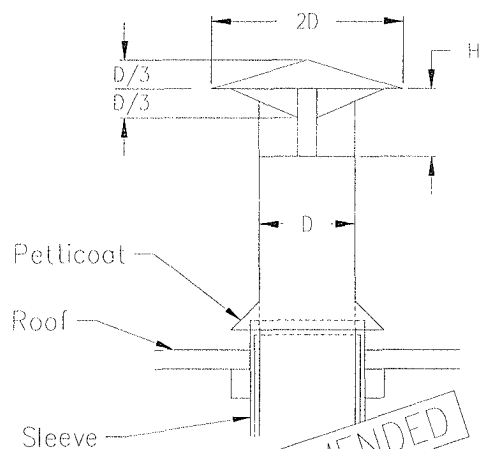


Note: Branch entry loss assumed to occur in branch and is so calculated.

Do not include an enlargement regain calculation for branch entry enlargements.

Angle θ Degrees	Loss Fraction of VP in Branch
10	0.06
15	0.09
20	0.12
25	0.15
30	0.18
35	0.21
40	0.25
45	0.28
50	0.32
60	0.44
90	1.00

BRANCH ENTRY LOSSES



H, No. of Diameters	Loss Fraction of VP
1.0 D	0.10
0.75 D	0.18
0.70 D	0.22
0.65 D	0.30
0.60 D	0.41
0.55 D	0.56
0.50 D	0.73
0.45 D	1.0

WEATHER CAP LOSSES

See Fig. 5-29

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DUCT DESIGN DATA

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FIGURE

5-15

Table 5-5 Tabulated Friction Loss Factors

Diameter inches	Galvanized Sheet Metal Duct					
	Friction Loss, No. VP per foot					
	1000 fpm	2000 fpm	3000 fpm	4000 fpm	5000 fpm	6000 fpm
0.5	1.0086	0.9549	0.9248	0.9040	0.8882	0.8755
1	0.4318	0.4088	0.3959	0.3870	0.3802	0.3748
1.5	0.2629	0.2489	0.2410	0.2356	0.2315	0.2282
2	0.1848	0.1750	0.1695	0.1657	0.1628	0.1605
2.5	0.1407	0.1332	0.1290	0.1261	0.1239	0.1221
3	0.1125	0.1065	0.1032	0.1009	0.0991	0.0977
3.5	0.0932	0.0882	0.0854	0.0835	0.0821	0.0809
4	0.0791	0.0749	0.0726	0.0709	0.0697	0.0687
4.5	0.0685	0.0649	0.0628	0.0614	0.0603	0.0595
5	0.0602	0.0570	0.0552	0.0540	0.0530	0.0523
5.5	0.0536	0.0507	0.0491	0.0480	0.0472	0.0465
6	0.0482	0.0456	0.0442	0.0432	0.0424	0.0418
7	0.0399	0.0378	0.0366	0.0358	0.0351	0.0346
8	0.0339	0.0321	0.0311	0.0304	0.0298	0.0294
9	0.0293	0.0278	0.0269	0.0263	0.0258	0.0255
10	0.0258	0.0244	0.0236	0.0231	0.0227	0.0224
11	0.0229	0.0217	0.0210	0.0206	0.0202	0.0199
12	0.0206	0.0195	0.0189	0.0185	0.0182	0.0179
13	0.0187	0.0177	0.0171	0.0168	0.0165	0.0162
14	0.0171	0.0162	0.0157	0.0153	0.0150	0.0148
15	0.0157	0.0149	0.0144	0.0141	0.0138	0.0136
16	0.0145	0.0137	0.0133	0.0130	0.0128	0.0126
17	0.0135	0.0127	0.0123	0.0121	0.0119	0.0117
18	0.0126	0.0119	0.0115	0.0113	0.0111	0.0109
19	0.0118	0.0111	0.0108	0.0105	0.0103	0.0102
20	0.0110	0.0104	0.0101	0.0099	0.0097	0.0096
21	0.0104	0.0098	0.0095	0.0093	0.0092	0.0090
22	0.0098	0.0093	0.0090	0.0088	0.0086	0.0085
23	0.0093	0.0088	0.0085	0.0083	0.0082	0.0081
24	0.0088	0.0084	0.0081	0.0079	0.0078	0.0077
25	0.0084	0.0080	0.0077	0.0075	0.0074	0.0073
26	0.0080	0.0076	0.0073	0.0072	0.0070	0.0069
27	0.0076	0.0072	0.0070	0.0069	0.0067	0.0066
28	0.0073	0.0069	0.0067	0.0066	0.0064	0.0063
29	0.0070	0.0066	0.0064	0.0063	0.0062	0.0061
30	0.0067	0.0064	0.0062	0.0060	0.0059	0.0058
31	0.0065	0.0061	0.0059	0.0058	0.0057	0.0056
32	0.0062	0.0059	0.0057	0.0056	0.0055	0.0054

$$H_f = 0.0307 \frac{V^{0.533}}{Q^{0.612}}$$

Table 5-5 Tabulated Friction Loss Factors (cont'd)

Diameter inches	Galvanized Sheet Metal Duct					
	Friction Loss, No. VP per foot					
	1000 fpm	2000 fpm	3000 fpm	4000 fpm	5000 fpm	6000 fpm
33	0.0060	0.0057	0.0055	0.0054	0.0053	0.0052
34	0.0058	0.0055	0.0053	0.0052	0.0051	0.0050
35	0.0056	0.0053	0.0051	0.0050	0.0049	0.0048
36	0.0054	0.0051	0.0049	0.0048	0.0047	0.0047
37	0.0052	0.0049	0.0048	0.0047	0.0046	0.0045
38	0.0050	0.0048	0.0046	0.0045	0.0044	0.0044
39	0.0049	0.0046	0.0045	0.0044	0.0043	0.0042
40	0.0047	0.0045	0.0043	0.0042	0.0042	0.0041
41	0.0046	0.0043	0.0042	0.0041	0.0040	0.0040
42	0.0045	0.0042	0.0041	0.0040	0.0039	0.0039
43	0.0043	0.0041	0.0040	0.0039	0.0038	0.0038
44	0.0042	0.0040	0.0039	0.0038	0.0037	0.0036
45	0.0041	0.0039	0.0038	0.0037	0.0036	0.0036
46	0.0040	0.0038	0.0037	0.0036	0.0035	0.0035
47	0.0039	0.0037	0.0036	0.0035	0.0034	0.0034
48	0.0038	0.0036	0.0035	0.0034	0.0033	0.0033
49	0.0037	0.0035	0.0034	0.0033	0.0032	0.0032
50	0.0036	0.0034	0.0033	0.0032	0.0032	0.0031
52	0.0034	0.0032	0.0031	0.0031	0.0030	0.0030
54	0.0033	0.0031	0.0030	0.0029	0.0029	0.0028
56	0.0031	0.0030	0.0029	0.0028	0.0028	0.0027
58	0.0030	0.0028	0.0027	0.0027	0.0026	0.0026
60	0.0029	0.0027	0.0026	0.0026	0.0025	0.0025
62	0.0028	0.0026	0.0025	0.0025	0.0024	0.0024
64	0.0027	0.0025	0.0024	0.0024	0.0023	0.0023
66	0.0026	0.0024	0.0023	0.0023	0.0023	0.0022
68	0.0025	0.0023	0.0023	0.0022	0.0022	0.0021
70	0.0024	0.0023	0.0022	0.0021	0.0021	0.0021
72	0.0023	0.0022	0.0021	0.0021	0.0020	0.0020
74	0.0022	0.0021	0.0020	0.0020	0.0020	0.0019
76	0.0022	0.0020	0.0020	0.0019	0.0019	0.0019
78	0.0021	0.0020	0.0019	0.0019	0.0018	0.0018
80	0.0020	0.0019	0.0019	0.0018	0.0018	0.0018
82	0.0020	0.0019	0.0018	0.0018	0.0017	0.0017
84	0.0019	0.0018	0.0017	0.0017	0.0017	0.0017
86	0.0019	0.0018	0.0017	0.0017	0.0016	0.0016
88	0.0018	0.0017	0.0017	0.0016	0.0016	0.0016
90	0.0018	0.0017	0.0016	0.0016	0.0015	0.0015

$$H_f = 0.0307 \frac{V^{0.533}}{Q^{0.612}}$$

Table 5-6 Tabulated Friction Loss Factors

Black Iron, Aluminum, Stainless Steel, PVC Ducts						
Diameter inches	Friction Loss, No. VP per foot					
	1000 fpm	2000 fpm	3000 fpm	4000 fpm	5000 fpm	6000 fpm
0.5	0.8757	0.7963	0.7533	0.7242	0.7024	0.6851
1	0.3801	0.3457	0.3270	0.3143	0.3049	0.2974
1.5	0.2333	0.2121	0.2007	0.1929	0.1871	0.1825
2	0.1650	0.1500	0.1419	0.1364	0.1323	0.1291
2.5	0.1261	0.1147	0.1085	0.1043	0.1012	0.0987
3	0.1013	0.0921	0.0871	0.0837	0.0812	0.0792
3.5	0.0841	0.0765	0.0724	0.0696	0.0675	0.0658
4	0.0716	0.0651	0.0616	0.0592	0.0574	0.0560
4.5	0.0621	0.0565	0.0535	0.0514	0.0499	0.0486
5	0.0547	0.0498	0.0471	0.0453	0.0439	0.0428
5.5	0.0488	0.0444	0.0420	0.0404	0.0392	0.0382
6	0.0440	0.0400	0.0378	0.0364	0.0353	0.0344
7	0.0365	0.0332	0.0314	0.0302	0.0293	0.0286
8	0.0311	0.0283	0.0267	0.0257	0.0249	0.0243
9	0.0270	0.0245	0.0232	0.0223	0.0216	0.0211
10	0.0238	0.0216	0.0204	0.0197	0.0191	0.0186
11	0.0212	0.0193	0.0182	0.0175	0.0170	0.0166
12	0.0191	0.0174	0.0164	0.0158	0.0153	0.0149
13	0.0173	0.0158	0.0149	0.0143	0.0139	0.0136
14	0.0158	0.0144	0.0136	0.0131	0.0127	0.0124
15	0.0146	0.0133	0.0125	0.0121	0.0117	0.0114
16	0.0135	0.0123	0.0116	0.0112	0.0108	0.0106
17	0.0125	0.0114	0.0108	0.0104	0.0101	0.0098
18	0.0117	0.0106	0.0101	0.0097	0.0094	0.0092
19	0.0110	0.0100	0.0094	0.0091	0.0088	0.0086
20	0.0103	0.0094	0.0089	0.0085	0.0083	0.0081
21	0.0097	0.0088	0.0084	0.0080	0.0078	0.0076
22	0.0092	0.0084	0.0079	0.0076	0.0074	0.0072
23	0.0087	0.0079	0.0075	0.0072	0.0070	0.0068
24	0.0083	0.0075	0.0071	0.0068	0.0066	0.0065
25	0.0079	0.0072	0.0068	0.0065	0.0063	0.0062
26	0.0075	0.0068	0.0065	0.0062	0.0060	0.0059
27	0.0072	0.0065	0.0062	0.0059	0.0058	0.0056
28	0.0069	0.0063	0.0059	0.0057	0.0055	0.0054
29	0.0066	0.0060	0.0057	0.0055	0.0053	0.0052
30	0.0063	0.0058	0.0054	0.0052	0.0051	0.0050
31	0.0061	0.0055	0.0052	0.0050	0.0049	0.0048
32	0.0059	0.0053	0.0050	0.0048	0.0047	0.0046

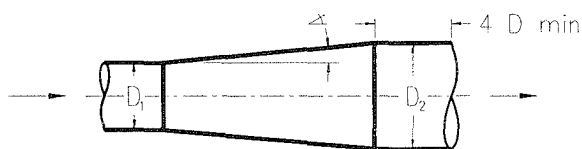
$$H_f = 0.0425 \frac{V^{0.465}}{Q^{0.602}}$$

Table 5-6 Tabulated Friction Loss Factors (cont'd)

Black Iron, Aluminum, Stainless Steel, PVC Ducts						
Diameter inches	Friction Loss, No. VP per foot					
	1000 fpm	2000 fpm	3000 fpm	4000 fpm	5000 fpm	6000 fpm
33	0.0056	0.0051	0.0049	0.0047	0.0045	0.0044
34	0.0054	0.0050	0.0047	0.0045	0.0044	0.0043
35	0.0053	0.0048	0.0045	0.0043	0.0042	0.0041
36	0.0051	0.0046	0.0044	0.0042	0.0041	0.0040
37	0.0049	0.0045	0.0042	0.0041	0.0039	0.0038
38	0.0048	0.0043	0.0041	0.0039	0.0038	0.0037
39	0.0046	0.0042	0.0040	0.0038	0.0037	0.0036
40	0.0045	0.0041	0.0039	0.0037	0.0036	0.0035
41	0.0043	0.0040	0.0037	0.0036	0.0035	0.0034
42	0.0042	0.0038	0.0036	0.0035	0.0034	0.0033
43	0.0041	0.0037	0.0035	0.0034	0.0033	0.0032
44	0.0040	0.0036	0.0034	0.0033	0.0032	0.0031
45	0.0039	0.0035	0.0033	0.0032	0.0031	0.0030
46	0.0038	0.0034	0.0033	0.0031	0.0030	0.0030
47	0.0037	0.0034	0.0032	0.0030	0.0030	0.0029
48	0.0036	0.0033	0.0031	0.0030	0.0029	0.0028
49	0.0035	0.0032	0.0030	0.0029	0.0028	0.0027
50	0.0034	0.0031	0.0029	0.0028	0.0027	0.0027
52	0.0033	0.0030	0.0028	0.0027	0.0026	0.0026
54	0.0031	0.0028	0.0027	0.0026	0.0025	0.0024
56	0.0030	0.0027	0.0026	0.0025	0.0024	0.0023
58	0.0029	0.0026	0.0025	0.0024	0.0023	0.0022
60	0.0027	0.0025	0.0024	0.0023	0.0022	0.0021
62	0.0026	0.0024	0.0023	0.0022	0.0021	0.0021
64	0.0025	0.0023	0.0022	0.0021	0.0020	0.0020
66	0.0024	0.0022	0.0021	0.0020	0.0020	0.0019
68	0.0024	0.0021	0.0020	0.0020	0.0019	0.0018
70	0.0023	0.0021	0.0020	0.0019	0.0018	0.0018
72	0.0022	0.0020	0.0019	0.0018	0.0018	0.0017
74	0.0021	0.0019	0.0018	0.0018	0.0017	0.0017
76	0.0021	0.0019	0.0018	0.0017	0.0017	0.0016
78	0.0020	0.0018	0.0017	0.0017	0.0016	0.0016
80	0.0019	0.0018	0.0017	0.0016	0.0016	0.0015
82	0.0019	0.0017	0.0016	0.0016	0.0015	0.0015
84	0.0018	0.0017	0.0016	0.0015	0.0015	0.0014
86	0.0018	0.0016	0.0015	0.0015	0.0014	0.0014
88	0.0017	0.0016	0.0015	0.0014	0.0014	0.0014
90	0.0017	0.0015	0.0015	0.0014	0.0014	0.0013

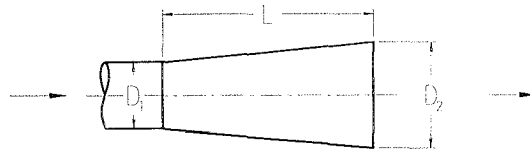
$$H_f = 0.0425 \frac{V^{0.465}}{Q^{0.602}}$$

STATIC PRESSURE REGAINS FOR EXPANSIONS



Within duct

Regain (R), fraction of VP difference					
Taper angle degrees	Diameter ratios D_2/D_1				
	1.25:1	1.5:1	1.75:1	2:1	2.5:1
3 1/2	0.92	0.88	0.84	0.81	0.75
5	0.88	0.84	0.80	0.76	0.68
10	0.85	0.76	0.70	0.63	0.53
15	0.83	0.70	0.62	0.55	0.43
20	0.81	0.67	0.57	0.48	0.43
25	0.80	0.65	0.53	0.44	0.28
30	0.79	0.63	0.51	0.41	0.25
Abrupt 90	0.77	0.62	0.50	0.40	0.25

Where: $SP_2 = SP_1 + R(VP_1 - VP_2)$ 

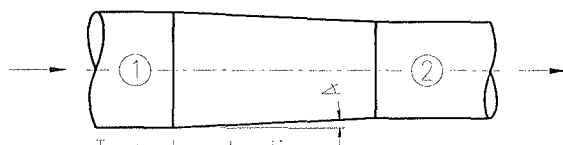
At end of duct

Regain (R), fraction of inlet VP						
Taper length to inlet diam L/D	Diameter ratios D_2/D_1					
	1.2:1	1.3:1	1.4:1	1.5:1	1.6:1	1.7:1
1.0:1	0.37	0.39	0.38	0.35	0.31	0.27
1.5:1	0.39	0.46	0.47	0.46	0.44	0.41
2.0:1	0.42	0.49	0.52	0.52	0.51	0.49
3.0:1	0.44	0.52	0.57	0.59	0.60	0.59
4.0:1	0.45	0.55	0.60	0.63	0.63	0.64
5.0:1	0.47	0.56	0.62	0.65	0.66	0.68
7.5:1	0.48	0.58	0.64	0.68	0.70	0.72

Where: $SP_1 = SP_2 - R(VP_1)$ **When $SP_2 = 0$ (atmosphere) SP_1 will be (-)

The regain (R) will only be 70% of value shown above when expansion follows a disturbance or elbow (including a fan) by less than 5 duct diameters.

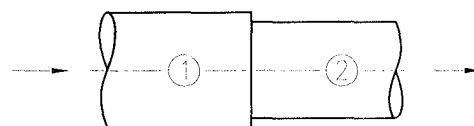
STATIC PRESSURE LOSSES FOR CONTRACTIONS



Tapered contraction

$$SP_2 = SP_1 - (VP_2 - VP_1) - L(VP_2 - VP_1)$$

Taper angle degrees	L(loss)
5	0.05
10	0.06
15	0.08
20	0.10
25	0.11
30	0.13
45	0.20
60	0.30
over 60	Abrupt contraction



Abrupt contraction

$$SP_2 = SP_1 - (VP_2 - VP_1) - K(VP_2)$$

Ratio A_2/A_1	K
0.1	0.48
0.2	0.46
0.3	0.42
0.4	0.37
0.4	0.32
0.6	0.26
0.7	0.20

A = duct area, ft^2

Note:

In calculating SP for expansion or contraction use algebraic signs: VP is (+), and usually SP is (+) in discharge duct from fan, and SP is (-) in inlet duct to fan.

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1-95

FIGURE

5-16

TABLE 5-7A. Velocity Pressure to Velocity Conversion — Standard Air

FROM: $V = 4005 \sqrt{VP}$

V = Velocity, fpm

VP = Velocity Pressure, "wg

VP	V	VP	V	VP	V	VP	V	VP	V	VP	V
0.01	401	0.51	2860	1.01	4025	1.51	4921	2.01	5678	2.60	6458
0.02	566	0.52	2888	1.02	4045	1.52	4938	2.02	5692	2.70	6581
0.03	694	0.53	2916	1.03	4065	1.53	4954	2.03	5706	2.80	6702
0.04	801	0.54	2943	1.04	4084	1.54	4970	2.04	5720	2.90	6820
0.05	896	0.55	2970	1.05	4104	1.55	4986	2.05	5734	3.00	6937
0.06	981	0.56	2997	1.06	4123	1.56	5002	2.06	5748	3.10	7052
0.07	1060	0.57	3024	1.07	4143	1.57	5018	2.07	5762	3.20	7164
0.08	1133	0.58	3050	1.08	4162	1.58	5034	2.08	5776	3.30	7275
0.09	1201	0.59	3076	1.09	4181	1.59	5050	2.09	5790	3.40	7385
0.10	1266	0.60	3102	1.10	4200	1.60	5066	2.10	5804	3.50	7493
0.11	1328	0.61	3128	1.11	4220	1.61	5082	2.11	5818	3.60	7599
0.12	1387	0.62	3154	1.12	4238	1.62	5098	2.12	5831	3.70	7704
0.13	1444	0.63	3179	1.13	4257	1.63	5113	2.13	5845	3.80	7807
0.14	1499	0.64	3204	1.14	4276	1.64	5129	2.14	5859	3.90	7909
0.15	1551	0.65	3229	1.15	4295	1.65	5145	2.15	5872	4.00	8010
0.16	1602	0.66	3254	1.16	4314	1.66	5160	2.16	5886	4.10	8110
0.17	1651	0.67	3278	1.17	4332	1.67	5176	2.17	5900	4.20	8208
0.18	1699	0.68	3303	1.18	4351	1.68	5191	2.18	5913	4.30	8305
0.19	1746	0.69	3327	1.19	4369	1.69	5206	2.19	5927	4.40	8401
0.20	1791	0.70	3351	1.20	4387	1.70	5222	2.20	5940	4.50	8496
0.21	1835	0.71	3375	1.21	4405	1.71	5237	2.21	5954	4.60	8590
0.22	1879	0.72	3398	1.22	4424	1.72	5253	2.22	5967	4.70	8683
0.23	1921	0.73	3422	1.23	4442	1.73	5268	2.23	5981	4.80	8775
0.24	1962	0.74	3445	1.24	4460	1.74	5283	2.24	5994	4.90	8865
0.25	2003	0.75	3468	1.25	4478	1.75	5298	2.25	6007	5.00	8955
0.26	2042	0.76	3491	1.26	4496	1.76	5313	2.26	6021	5.50	9393
0.27	2081	0.77	3514	1.27	4513	1.77	5328	2.27	6034	6.00	9810
0.28	2119	0.78	3537	1.28	4531	1.78	5343	2.28	6047	6.50	10211
0.29	2157	0.79	3560	1.29	4549	1.79	5358	2.29	6061	7.00	10596
0.30	2194	0.80	3582	1.30	4566	1.80	5373	2.30	6074	7.50	10968
0.31	2230	0.81	3604	1.31	4584	1.81	5388	2.31	6087	8.00	11328
0.32	2266	0.82	3627	1.32	4601	1.82	5403	2.32	6100	8.50	11676
0.33	2301	0.83	3649	1.33	4619	1.83	5418	2.33	6113	9.00	12015
0.34	2335	0.84	3671	1.34	4636	1.84	5433	2.34	6126	9.50	12344
0.35	2369	0.85	3692	1.35	4653	1.85	5447	2.35	6140	10.00	12665
0.36	2403	0.86	3714	1.36	4671	1.86	5462	2.36	6153	10.50	12978
0.37	2436	0.87	3736	1.37	4688	1.87	5477	2.37	6166	11.00	13283
0.38	2469	0.88	3757	1.38	4705	1.88	5491	2.38	6179	11.50	13582
0.39	2501	0.89	3778	1.39	4722	1.89	5506	2.39	6192	12.00	13874
0.40	2533	0.90	3799	1.40	4739	1.90	5521	2.40	6205	12.50	14160
0.41	2564	0.91	3821	1.41	4756	1.91	5535	2.41	6217	13.00	14440
0.42	2596	0.92	3841	1.42	4773	1.92	5549	2.42	6230	13.50	14715
0.43	2626	0.93	3862	1.43	4789	1.93	5564	2.43	6243	14.00	14985
0.44	2657	0.94	3883	1.44	4806	1.94	5578	2.44	6256	14.50	15251
0.45	2687	0.95	3904	1.45	4823	1.95	5593	2.45	6269	15.00	15511
0.46	2716	0.96	3924	1.46	4839	1.96	5607	2.46	6282	15.50	15768
0.47	2746	0.97	3944	1.47	4856	1.97	5621	2.47	6294	16.00	16020
0.48	2775	0.98	3965	1.48	4872	1.98	5636	2.48	6307	16.50	16268
0.49	2803	0.99	3985	1.49	4889	1.99	5650	2.49	6320	17.00	16513
0.50	2832	1.00	4005	1.50	4905	2.00	5664	2.50	6332	17.50	16754

TABLE 5-7B. Velocity to Velocity Pressure Conversion — Standard Air

FROM: $V = 4005 \sqrt{VP}$

V = Velocity, fpm

VP = Velocity Pressure, "wg

V	VP	V	VP	V	VP	V	VP	V	VP	V	VP
400	0.01	2600	0.42	3850	0.92	4880	1.48	5690	2.02	6190	2.39
500	0.02	2625	0.43	3875	0.94	4900	1.50	5700	2.03	6200	2.40
600	0.02	2650	0.44	3900	0.95	4920	1.51	5710	2.03	6210	2.40
700	0.03	2675	0.45	3925	0.96	4940	1.52	5720	2.04	6220	2.41
800	0.04	2700	0.45	3950	0.97	4960	1.53	5730	2.05	6230	2.42
900	0.05	2725	0.46	3975	0.99	4980	1.55	5740	2.05	6240	2.43
1000	0.06	2750	0.47	4000	1.00	5000	1.56	5750	2.06	6250	2.44
1100	0.08	2775	0.48	4020	1.01	5020	1.57	5760	2.07	6260	2.44
1200	0.09	2800	0.49	4040	1.02	5040	1.58	5770	2.08	6270	2.45
1300	0.11	2825	0.50	4060	1.03	5060	1.60	5780	2.08	6280	2.46
1400	0.12	2850	0.51	4080	1.04	5080	1.61	5790	2.09	6290	2.47
1450	0.13	2875	0.52	4100	1.05	5100	1.62	5800	2.10	6300	2.47
1500	0.14	2900	0.52	4120	1.06	5120	1.63	5810	2.10	6310	2.48
1550	0.15	2925	0.53	4140	1.07	5140	1.65	5820	2.11	6320	2.49
1600	0.16	2950	0.54	4160	1.08	5160	1.66	5830	2.12	6330	2.50
1650	0.17	2975	0.55	4180	1.09	5180	1.67	5840	2.13	6340	2.51
1700	0.18	3000	0.56	4200	1.10	5200	1.69	5850	2.13	6350	2.51
1750	0.19	3025	0.57	4220	1.11	5220	1.70	5860	2.14	6360	2.52
1800	0.20	3050	0.58	4240	1.12	5240	1.71	5870	2.15	6370	2.53
1825	0.21	3075	0.59	4260	1.13	5260	1.72	5880	2.16	6380	2.54
1850	0.21	3100	0.60	4280	1.14	5280	1.74	5890	2.16	6390	2.55
1875	0.22	3125	0.61	4300	1.15	5300	1.75	5900	2.17	6400	2.55
1900	0.23	3150	0.62	4320	1.16	5320	1.76	5910	2.18	6410	2.56
1925	0.23	3175	0.63	4340	1.17	5340	1.78	5920	2.18	6420	2.57
1950	0.24	3200	0.64	4360	1.19	5360	1.79	5930	2.19	6430	2.58
1975	0.24	3225	0.65	4380	1.20	5380	1.80	5940	2.20	6440	2.59
2000	0.25	3250	0.66	4400	1.21	5400	1.82	5950	2.21	6450	2.59
2025	0.26	3275	0.67	4420	1.22	5420	1.83	5960	2.21	6460	2.60
2050	0.26	3300	0.68	4440	1.23	5440	1.84	5970	2.22	6470	2.61
2075	0.27	3325	0.69	4460	1.24	5460	1.86	5980	2.23	6480	2.62
2100	0.27	3350	0.70	4480	1.25	5480	1.87	5990	2.24	6490	2.63
2125	0.28	3375	0.71	4500	1.26	5500	1.89	6000	2.24	6500	2.63
2150	0.29	3400	0.72	4520	1.27	5510	1.89	6010	2.25	6550	2.67
2175	0.29	3425	0.73	4540	1.29	5520	1.90	6020	2.26	6600	2.72
2200	0.30	3450	0.74	4560	1.30	5530	1.91	6030	2.27	6650	2.76
2225	0.31	3475	0.75	4580	1.31	5540	1.91	6040	2.27	6700	2.80
2250	0.32	3500	0.76	4600	1.32	5550	1.92	6050	2.28	6750	2.84
2275	0.32	3525	0.77	4620	1.33	5560	1.93	6060	2.29	6800	2.88
2300	0.33	3550	0.79	4640	1.34	5570	1.93	6070	2.30	6900	2.97
2325	0.34	3575	0.80	4660	1.35	5580	1.94	6080	2.30	7000	3.05
2350	0.34	3600	0.81	4680	1.37	5590	1.95	6090	2.31	7100	3.14
2375	0.35	3625	0.82	4700	1.38	5600	1.96	6100	2.32	7200	3.23
2400	0.36	3650	0.83	4720	1.39	5610	1.96	6110	2.33	7300	3.32
2425	0.37	3675	0.84	4740	1.40	5620	1.97	6120	2.34	7400	3.41
2450	0.37	3700	0.85	4760	1.41	5630	1.98	6130	2.34	7500	3.51
2475	0.38	3725	0.87	4780	1.42	5640	1.98	6140	2.35	7600	3.60
2500	0.39	3750	0.88	4800	1.44	5650	1.99	6150	2.36	7700	3.70
2525	0.40	3775	0.89	4820	1.45	5660	2.00	6160	2.37	7800	3.79
2550	0.41	3800	0.90	4840	1.46	5670	2.00	6170	2.37	7900	3.89
2575	0.41	3825	0.91	4860	1.47	5680	2.01	6180	2.38	8000	3.99

TABLE 5-8. Area and Circumference of Circles

Diam. in Inches	AREA		CIRCUMFERENCE		Diam. in Inches	AREA		CIRCUMFERENCE	
	Square Inches	Square Feet	Inches	Feet		Square Inches	Square Feet	Inches	Feet
1	0.79	0.0055	3.14	0.2618	30	706.9	4.909	94.2	7.854
1.5	1.77	0.0123	4.71	0.3927	31	754.8	5.241	97.4	8.116
2	3.14	0.0218	6.28	0.5236	32	804.2	5.585	100.5	8.378
2.5	4.91	0.0341	7.85	0.6545	33	855.3	5.940	103.7	8.639
3	7.07	0.0491	9.42	0.7854	34	907.9	6.305	106.8	8.901
3.5	9.62	0.0668	11.00	0.9163	35	962.1	6.681	110.0	9.163
4	12.57	0.0873	12.57	1.0472	36	1017.9	7.069	113.1	9.425
4.5	15.90	0.1104	14.14	1.1781	37	1075.2	7.467	116.2	9.687
5	19.63	0.1364	15.71	1.3090	38	1134.1	7.876	119.4	9.948
5.5	23.76	0.1650	17.28	1.4399	39	1194.6	8.296	122.5	10.210
6	28.27	0.1963	18.85	1.5708	40	1256.6	8.727	125.7	10.472
6.5	33.18	0.2304	20.42	1.7017	41	1320.3	9.168	128.8	10.734
7	38.48	0.2673	21.99	1.8326	42	1385.4	9.621	131.9	10.996
7.5	44.18	0.3068	23.56	1.9635	43	1452.2	10.085	135.1	11.257
8	50.27	0.3491	25.13	2.0944	44	1520.5	10.559	138.2	11.519
8.5	56.75	0.3941	26.70	2.2253	45	1590.4	11.045	141.4	11.781
9	63.62	0.4418	28.27	2.3562	46	1661.9	11.541	144.5	12.043
9.5	70.88	0.4922	29.85	2.4871	47	1734.9	12.048	147.7	12.305
10	78.54	0.5454	31.42	2.6180	48	1809.6	12.566	150.8	12.566
10.5	86.59	0.6013	32.99	2.7489	49	1885.7	13.095	153.9	12.828
11	95.03	0.6600	34.56	2.8798	50	1963.5	13.635	157.1	13.090
11.5	103.87	0.7213	36.13	3.0107	52	2123.7	14.748	163.4	13.614
12	113.10	0.7854	37.70	3.1416	54	2290.2	15.904	169.6	14.137
13	132.73	0.9218	40.84	3.4034	56	2463.0	17.104	175.9	14.661
14	153.94	1.0690	43.98	3.6652	58	2642.1	18.348	182.2	15.184
15	176.71	1.2272	47.12	3.9270	60	2827.4	19.635	188.5	15.708
16	201.06	1.3963	50.27	4.1888	62	3019.1	20.966	194.8	16.232
17	226.98	1.5763	53.41	4.4506	64	3217.0	22.340	201.1	16.755
18	254.47	1.7671	56.55	4.7124	66	3421.2	23.758	207.3	17.279
19	283.53	1.9689	59.69	4.9742	68	3631.7	25.220	213.6	17.802
20	314.16	2.1817	62.83	5.2360	70	3848.5	26.725	219.9	18.326
21	346.36	2.4053	65.97	5.4978	72	4071.5	28.274	226.2	18.850
22	380.13	2.6398	69.12	5.7596	74	4300.8	29.867	232.5	19.373
23	415.48	2.8852	72.26	6.0214	76	4536.5	31.503	238.8	19.897
24	452.39	3.1416	75.40	6.2832	78	4778.4	33.183	245.0	20.420
25	490.87	3.4088	78.54	6.5450	80	5026.5	34.907	251.3	20.944
26	530.93	3.6870	81.68	6.8068	82	5281.0	36.674	257.6	21.468
27	572.56	3.9761	84.82	7.0686	84	5541.8	38.485	263.9	21.991
28	615.75	4.2761	87.96	7.3304	86	5808.8	40.339	270.2	22.515
29	660.52	4.5869	91.11	7.5922	88	6082.1	42.237	276.5	23.038

The usual sheet metal fabricator will have patterns for ducts in 0.5-inch steps through 5.5-inch diameter; 1 inch steps 6 inches through 20 inches and 2-inch steps 22 inches and larger diameters.

TABLE 5-9. Circular Equivalents of Rectangular Duct Sizes

A \ B	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5	9.0	9.5	10.0	10.5	11.0	11.5	12.0	12.5	13.0	13.5	14.0	14.5	15.0	15.5	16.0
3.0	3.8	4.0	4.2	4.4	4.6	4.7	4.9	5.1	5.2	5.3	5.5	5.6	5.7	5.9	6.0	6.1	6.2	6.3	6.4	6.5	6.6	6.7	6.8	6.9	7.0
3.5	4.1	4.3	4.6	4.8	5.0	5.2	5.3	5.5	5.7	5.8	6.0	6.1	6.3	6.4	6.5	6.7	6.8	6.9	7.0	7.1	7.2	7.3	7.5	7.6	7.7
4.0	4.4	4.6	4.9	5.1	5.3	5.5	5.7	5.9	6.1	6.3	6.4	6.6	6.7	6.9	7.0	7.2	7.3	7.4	7.6	7.7	7.8	7.9	8.0	8.2	8.3
4.5	4.6	4.9	5.2	5.4	5.7	5.9	6.1	6.3	6.5	6.7	6.9	7.0	7.2	7.4	7.5	7.7	7.8	7.9	8.1	8.2	8.4	8.5	8.6	8.7	8.8
5.0	4.9	5.2	5.5	5.7	6.0	6.2	6.4	6.7	6.9	7.1	7.3	7.4	7.6	7.8	8.0	8.1	8.3	8.4	8.6	8.7	8.9	9.0	9.1	9.3	9.4
5.5	5.1	5.4	5.7	6.0	6.3	6.5	6.8	7.0	7.2	7.4	7.6	7.8	8.0	8.2	8.4	8.6	8.7	8.9	9.0	9.2	9.3	9.5	9.6	9.8	9.9
A \ B	6.0	7.0	8.0	9.0	10.0	11.0	12.0	13.0	14.0	15.0	16.0	17.0	18.0	19.0	20.0	22.0	24.0	26.0	28.0	30.0	32.0	34.0	36.0	38.0	40.0
6.0	6.6																								
7.0	7.1	7.7																							
8.0	7.6	8.2	8.7																						
9.0	8.0	8.7	9.3	9.8																					
10.0	8.4	9.1	9.8	10.4	10.9																				
11.0	8.8	9.5	10.2	10.9	11.5	12.0																			
12.0	9.1	9.9	10.7	11.3	12.0	12.6	13.1																		
13.0	9.5	10.3	11.1	11.8	12.4	13.1	13.7	14.2																	
14.0	9.8	10.7	11.5	12.2	12.9	13.5	14.2	14.7	15.3																
15.0	10.1	11.0	11.8	12.6	13.3	14.0	14.6	15.3	15.8	16.4															
16.0	10.4	11.3	12.2	13.0	13.7	14.4	15.1	15.7	16.4	16.9	17.5														
17.0	10.7	11.6	12.5	13.4	14.1	14.9	15.6	16.2	16.8	17.4	18.0	18.6													
18.0	11.0	11.9	12.9	13.7	14.5	15.3	16.0	16.7	17.3	17.9	18.5	19.1	19.7												
19.0	11.2	12.2	13.2	14.1	14.9	15.7	16.4	17.1	17.8	18.4	19.0	19.6	20.2	20.8											
20.0	11.5	12.5	13.5	14.4	15.2	16.0	16.8	17.5	18.2	18.9	19.5	20.1	20.7	21.3	21.9										
22.0	12.0	13.0	14.1	15.0	15.9	16.8	17.6	18.3	19.1	19.8	20.4	21.1	21.7	22.3	22.9	24.0									
24.0	12.4	13.5	14.6	15.6	16.5	17.4	18.3	19.1	19.9	20.6	21.3	22.0	22.7	23.3	23.9	25.1	26.2								
26.0	12.8	14.0	15.1	16.2	17.1	18.1	19.0	19.8	20.6	21.4	22.1	22.9	23.5	24.2	24.9	26.1	27.3	28.4							
28.0	13.2	14.5	15.6	16.7	17.7	18.7	19.6	20.5	21.3	22.1	22.9	23.7	24.4	25.1	25.8	27.1	28.3	29.5	30.6						
30.0	13.6	14.9	16.1	17.2	18.3	19.3	20.2	21.1	22.0	22.9	23.7	24.4	25.2	25.9	26.6	28.0	29.3	30.5	31.7	32.8					
32.0	14.0	15.3	16.5	17.7	18.8	19.8	20.8	21.8	22.7	23.5	24.4	25.2	26.0	26.7	27.5	28.9	30.2	31.5	32.7	33.9	35.0				
34.0	14.4	15.7	17.0	18.2	19.3	20.4	21.4	22.4	23.3	24.2	25.1	25.9	26.7	27.5	28.3	29.7	31.1	32.4	33.7	34.9	36.1	37.2			
36.0	14.7	16.1	17.4	18.6	19.8	20.9	21.9	22.9	23.9	24.8	25.7	26.6	27.4	28.2	29.0	30.5	32.0	33.3	34.6	35.9	37.1	38.2	39.4		
38.0	15.0	16.5	17.8	19.0	20.2	21.4	22.4	23.5	24.5	25.4	26.4	27.2	28.1	28.9	29.8	31.3	32.8	34.2	35.6	36.8	38.1	39.3	40.4	41.5	
40.0	15.3	16.8	18.2	19.5	20.7	21.8	22.9	24.0	25.0	26.0	27.0	27.9	28.8	29.6	30.5	32.1	33.6	35.1	36.4	37.8	39.0	40.3	41.5	42.6	43.7
42.0	15.6	17.1	18.5	19.9	21.1	22.3	23.4	24.5	25.6	26.6	27.6	28.5	29.4	30.3	31.2	32.8	34.4	35.9	37.3	38.7	40.0	41.3	42.5	43.7	44.8
44.0	15.9	17.5	18.9	20.3	21.5	22.7	23.9	25.0	26.1	27.1	28.1	29.1	30.0	30.9	31.8	33.5	35.1	36.7	38.1	39.5	40.9	42.2	43.5	44.7	45.8
46.0	16.2	17.8	19.3	20.6	21.9	23.2	24.4	25.5	26.6	27.7	28.7	29.7	30.6	31.6	32.5	34.2	35.9	37.4	38.9	40.4	41.8	43.1	44.4	45.7	46.9
48.0	16.5	18.1	19.6	21.0	22.3	23.6	24.8	26.0	27.1	28.2	29.2	30.2	31.2	32.2	33.1	34.9	36.6	38.2	39.7	41.2	42.6	44.0	45.3	46.6	47.9
50.0	16.8	18.4	19.9	21.4	22.7	24.0	25.2	26.4	27.6	28.7	29.8	30.8	31.8	32.8	33.7	35.5	37.2	38.9	40.5	42.0	43.5	44.9	46.2	47.5	48.8
54.0	17.3	19.0	20.6	22.0	23.5	24.8	26.1	27.3	28.5	29.7	30.8	31.8	32.9	33.9	34.9	36.8	38.6	40.3	41.9	43.5	45.1	46.5	48.0	49.3	50.7
58.0	17.8	19.5	21.2	22.7	24.2	25.5	26.9	28.2	29.4	30.6	31.7	32.8	33.9	35.0	36.0	38.0	39.8	41.6	43.3	45.0	46.6	48.1	49.6	51.0	52.4
62.0	18.3	20.1	21.7	23.3	24.8	26.3	27.6	28.9	30.2	31.5	32.6	33.8	34.9	36.0	37.1	39.1	41.0	42.9	44.7	46.4	48.0	49.6	51.2	52.7	54.1
66.0	18.8	20.6	22.3	23.9	25.5	26.9	28.4	29.7	31.0	32.3	33.5	34.7	35.9	37.0	38.1	40.2	42.2	44.1	46.0	47.7	49.4	51.1	52.7	54.2	55.7
70.0	19.2	21.1	22.8	24.5	26.1	27.6	29.1	30.4	31.8	33.1	34.4	35.6	36.8	37.9	39.1	41.2	43.3	45.3	47.2	49.0	50.8	52.5	54.1	55.7	57.3
74.0	19.6	21.5	23.3	25.1	26.7	28.2	29.7	31.2	32.5	33.9	35.2	36.4	37.7	38.8	40.0	42.2	44.4	46.4	48.4	50.3	52.1	53.8	55.5	57.2	58.8
78.0	20.0	22.0	23.8	25.6	27.3	28.8	30.4	31.8	33.3	34.6	36.0	37.2	38.5	39.7	40.9	43.2	45.4	47.5	49.5	51.4	53.3	55.1	56.9	58.6	60.2
82.0	20.4	22.4	24.3	26.1	27.8	29.4	31.0	32.5	33.9	35.4	36.7	38.0	39.3	40.6	41.8	44.1	46.4	48.5	50.6	52.6	54.5	56.4	58.2	59.9	61.6
86.0	20.8	22.9	24.8	26.6	28.3	30.0	31.6	33.1	34.6	36.1	37.4	38.8	40.1	41.4	42.6	45.0	47.3	49.6	51.7	53.7	55.7	57.6	59.4	61.2	63.0
90.0	21.2	23.3	25.2	27.1	28.9	30.6	32.2	33.8	35.3	36.7	38.2	39.5	40.9	42.2	43.5	45.9	48.3	50.5	52.7	54.8	56.8	58.8	60.7	62.5	64.3

TABLE 5-9. Circular Equivalents of Rectangular Duct Sizes (cont.)

A \ B	42.0	44.0	46.0	48.0	50.0	54.0	58.0	62.0	66.0	70.0	74.0	78.0	82.0	86.0	90.0
6.0															
7.0															
8.0															
9.0															
10.0															
11.0															
12.0															
13.0															
14.0															
15.0															
16.0															
17.0															
18.0															
19.0															
20.0															
22.0															
24.0															
26.0															
28.0															
30.0															
32.0															
34.0															
36.0															
38.0															
40.0															
42.0	45.9														
44.0	47.0	48.1													
46.0	48.0	49.2	50.3												
48.0	49.1	50.2	51.4	52.5											
50.0	50.0	51.2	52.4	53.6	54.7										
54.0	52.0	53.2	54.4	55.6	56.8	59.0									
58.0	53.8	55.1	56.4	57.6	58.8	61.2	63.4								
62.0	55.5	56.9	58.2	59.5	60.8	63.2	65.5	67.8							
66.0	57.2	58.6	60.0	61.3	62.6	65.2	67.6	69.9	72.1						
70.0	58.8	60.3	61.7	63.1	64.4	67.1	69.6	72.0	74.3	76.5					
74.0	60.3	61.9	63.3	64.8	66.2	68.9	71.5	74.0	76.4	78.7	80.9				
78.0	61.8	63.4	64.9	66.4	67.9	70.6	73.3	75.9	78.4	80.7	83.0	85.3			
82.0	63.3	64.9	66.5	68.0	69.5	72.3	75.1	77.8	80.3	82.8	85.1	87.4	89.6		
86.0	64.7	66.3	67.9	69.5	71.0	74.0	76.8	79.6	82.2	84.7	87.1	89.5	91.8	94.0	
90.0	66.0	67.7	69.4	71.0	72.6	75.6	78.5	81.3	84.0	86.6	89.1	91.5	93.9	96.2	98.4

$$D_{\text{equiv}} = 1.3 \frac{(A \times B)^{0.625}}{(A + B)^{0.25}}$$

where:

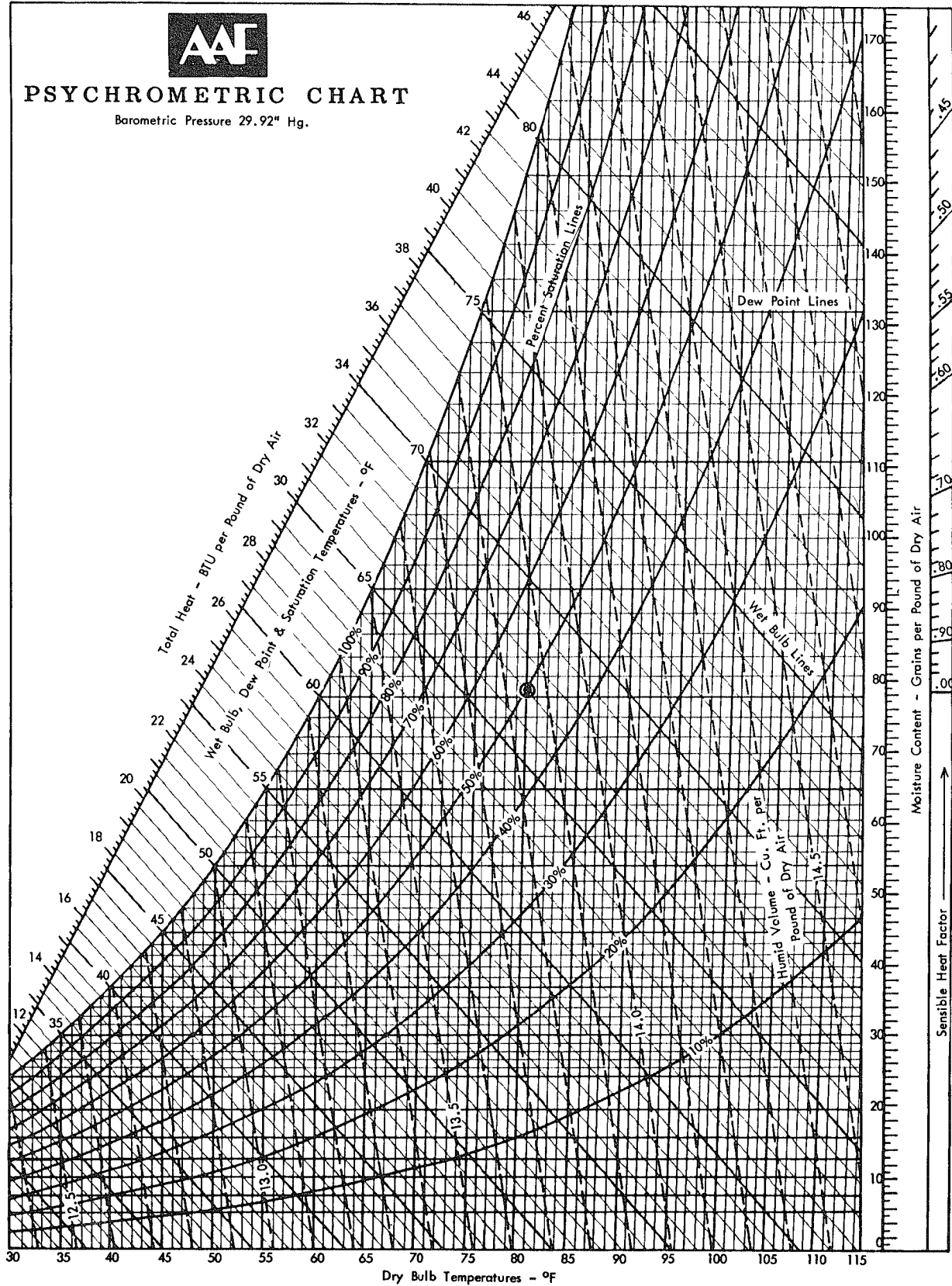
D_{equiv} = equivalent round duct size for rectangular duct, in.

A = one side of rectangular duct, in.

B = adjacent side of rectangular duct, in.

TABLE 5-10. Air Density Correction Factor, df

ALTITUDE RELATIVE TO SEA LEVEL, ft																	
	-5000	-4000	-3000	-2000	-1000	0	1000	2000	3000	4000	5000	6000	7000	8000	9000	10000	
Barometric Pressure																	
"Hg	35.74	34.51	33.31	32.15	31.02	29.92	28.86	27.82	26.82	25.84	24.89	23.98	23.09	22.22	21.39	20.57	
"w	486.74	469.97	453.67	437.84	422.45	407.50	392.98	378.89	365.21	351.93	339.04	326.54	314.42	302.66	291.26	280.21	
Temp., F	Density Factor, df																
-40	1.51	1.46	1.40	1.36	1.31	1.26	1.22	1.17	1.13	1.09	1.05	1.01	0.97	0.94	0.90	0.87	
0	1.38	1.33	1.28	1.24	1.19	1.15	1.11	1.07	1.03	1.00	0.96	0.92	0.89	0.86	0.82	0.79	
40	1.27	1.22	1.18	1.14	1.10	1.06	1.02	0.99	0.95	0.92	0.88	0.85	0.82	0.79	0.76	0.73	
70	1.19	1.15	1.11	1.07	1.04	1.00	0.96	0.93	0.90	0.86	0.83	0.80	0.77	0.74	0.71	0.69	
100	1.13	1.09	1.05	1.02	0.98	0.95	0.91	0.88	0.85	0.82	0.79	0.76	0.73	0.70	0.68	0.65	
150	1.04	1.00	0.97	0.93	0.90	0.87	0.84	0.81	0.78	0.75	0.72	0.70	0.67	0.65	0.62	0.60	
200	0.96	0.93	0.89	0.86	0.83	0.80	0.77	0.75	0.72	0.69	0.67	0.64	0.62	0.60	0.57	0.55	
250	0.89	0.86	0.83	0.80	0.77	0.75	0.72	0.69	0.67	0.64	0.62	0.60	0.58	0.55	0.53	0.51	
300	0.83	0.80	0.78	0.75	0.72	0.70	0.67	0.65	0.62	0.60	0.58	0.56	0.54	0.52	0.50	0.48	
350	0.78	0.75	0.73	0.70	0.68	0.65	0.63	0.61	0.59	0.57	0.54	0.52	0.50	0.49	0.47	0.45	
400	0.74	0.71	0.69	0.66	0.64	0.62	0.59	0.57	0.55	0.53	0.51	0.49	0.48	0.46	0.44	0.42	
450	0.70	0.67	0.65	0.63	0.60	0.58	0.56	0.54	0.52	0.50	0.48	0.47	0.45	0.43	0.42	0.40	
500	0.66	0.64	0.61	0.59	0.57	0.55	0.53	0.51	0.49	0.48	0.46	0.44	0.43	0.41	0.39	0.38	
550	0.63	0.61	0.58	0.56	0.54	0.52	0.51	0.49	0.47	0.45	0.44	0.42	0.40	0.39	0.38	0.36	
600	0.60	0.58	0.56	0.54	0.52	0.50	0.48	0.46	0.45	0.43	0.42	0.40	0.39	0.37	0.36	0.34	
700	0.55	0.53	0.51	0.49	0.47	0.46	0.44	0.42	0.41	0.39	0.38	0.37	0.35	0.34	0.33	0.31	
800	0.50	0.49	0.47	0.45	0.44	0.42	0.41	0.39	0.38	0.36	0.35	0.34	0.32	0.31	0.30	0.29	
900	0.47	0.45	0.43	0.42	0.40	0.39	0.38	0.36	0.35	0.34	0.32	0.31	0.30	0.29	0.28	0.27	
1000	0.43	0.42	0.40	0.39	0.38	0.36	0.35	0.34	0.33	0.31	0.30	0.29	0.28	0.27	0.26	0.25	



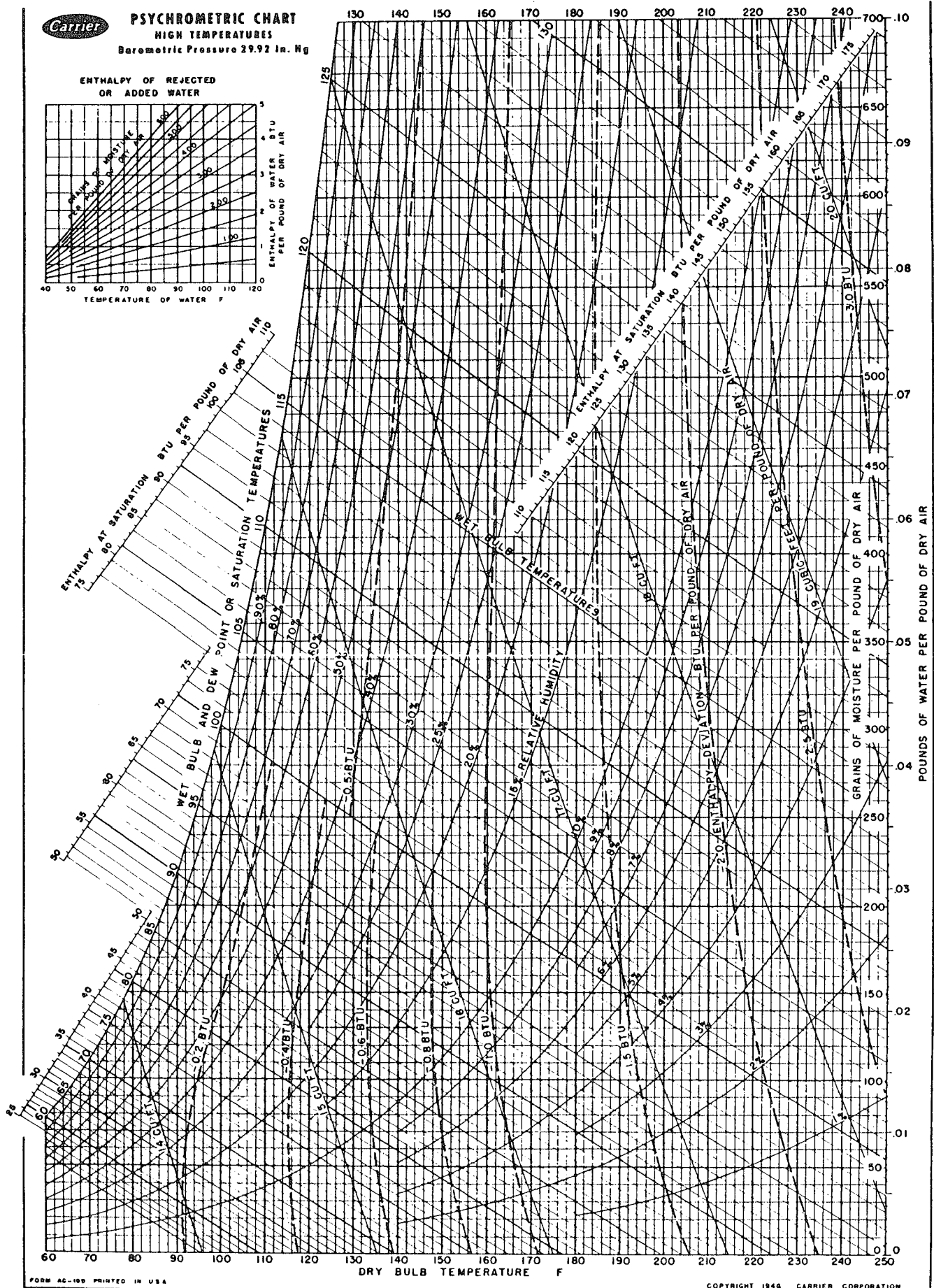


FIGURE 5-18

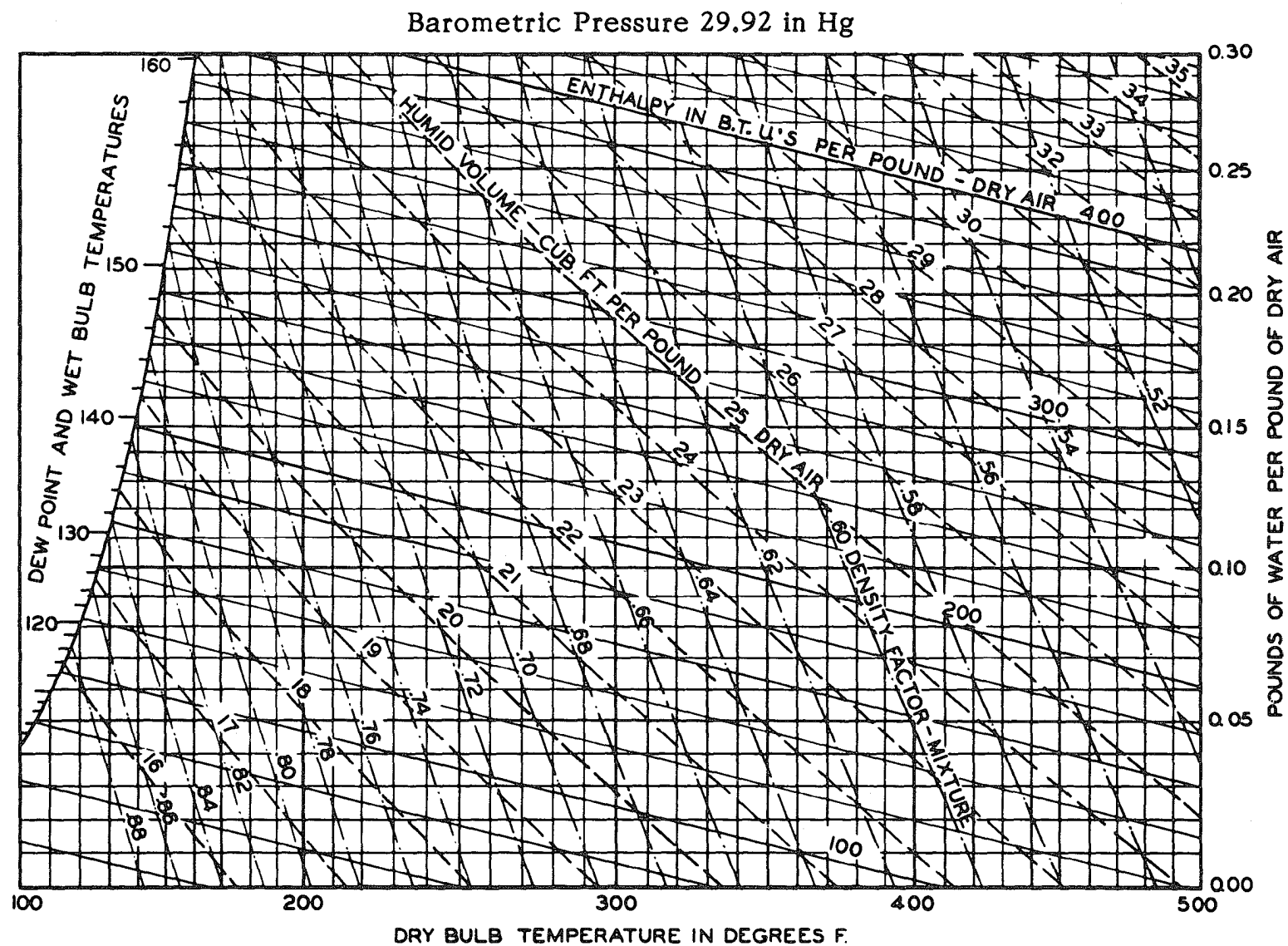


FIGURE 5-19. Psychrometric chart for humid air based on one pound dry weight (© 1951 American Air Filter Co., Inc., Louisville, KY)

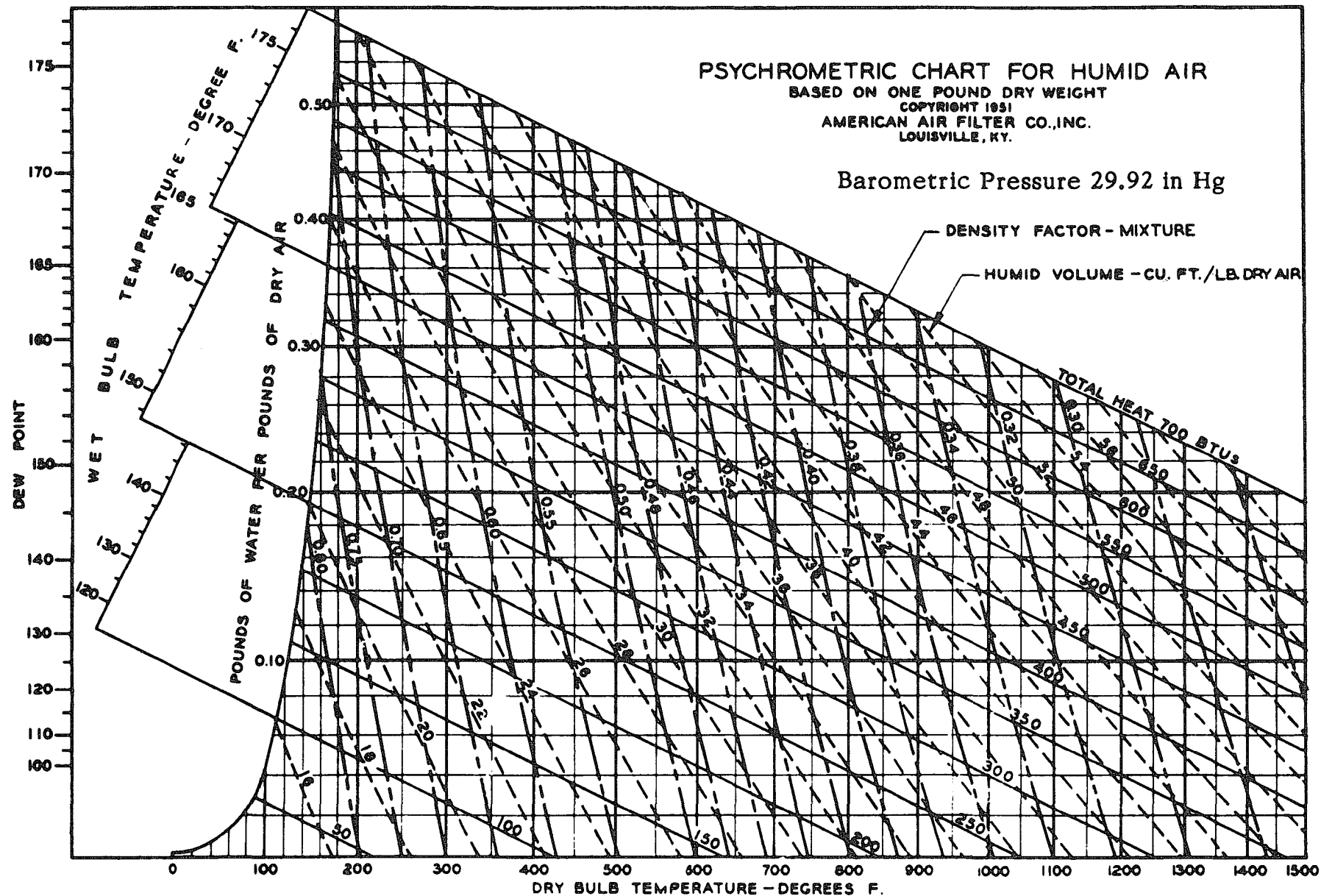
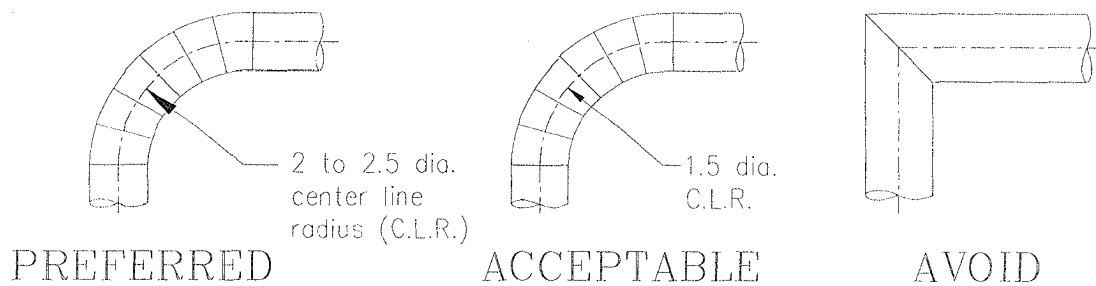
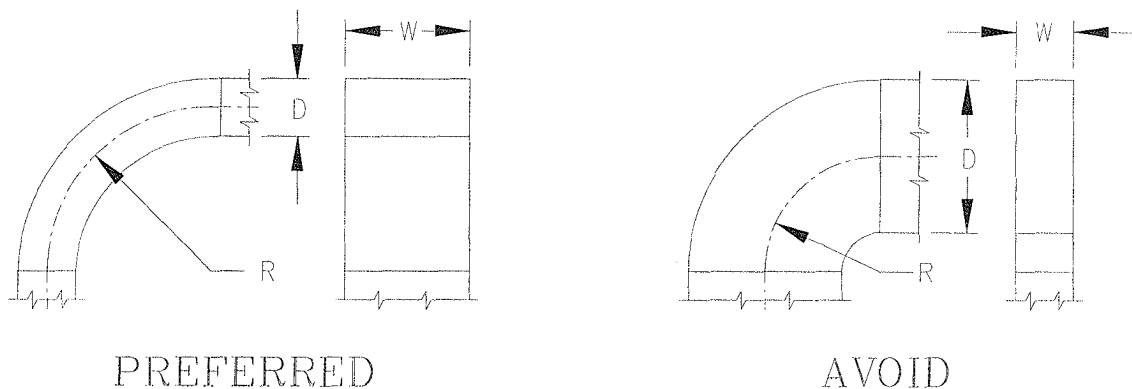


FIGURE 5-20



ELBOW RADIUS

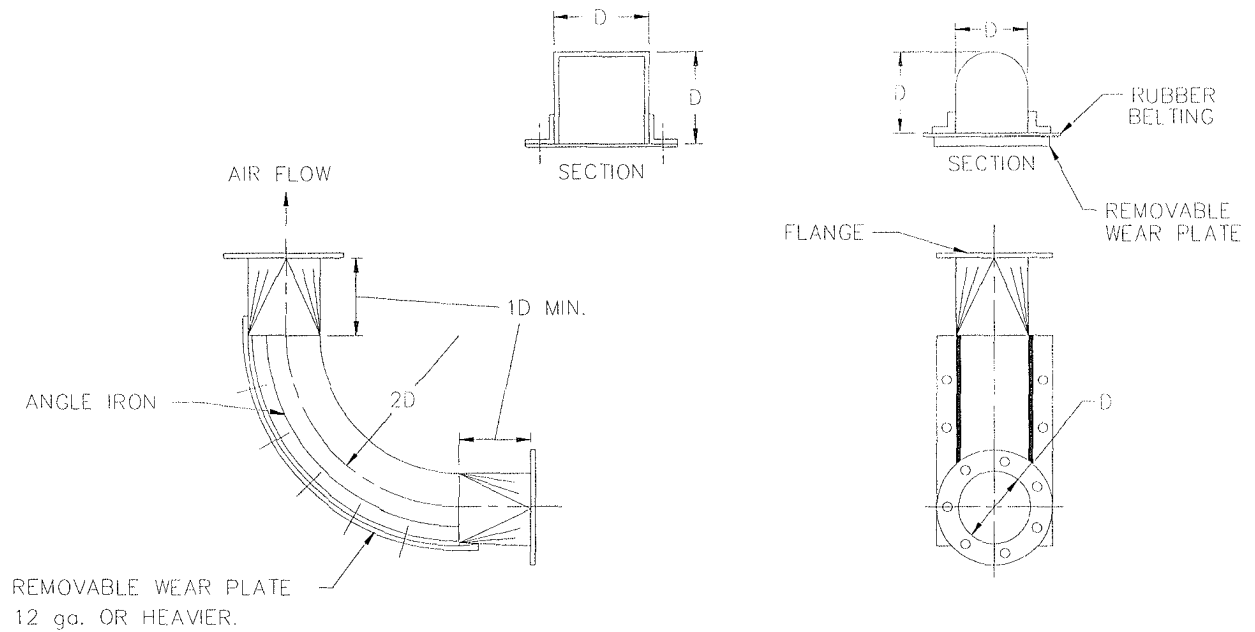
Elbows should be 2 to 2.5 diameter centerline radius except where space does not permit. See Fig. 5-13 for loss factor.



ASPECT RATIO ($\frac{W}{D}$)

Elbows should have ($\frac{W}{D}$) and ($\frac{R}{D}$) equal to or greater than (1). See Fig. 5-13 for loss factor.

Note: Avoid mitered elbows. If necessary, use only with clean air and provide turning vanes. Consult mfg. for turning vane loss factor.



FLAT BACK ELBOW



CONCRETE REINFORCED ELBOW

NOTE: PROVIDE SOLID MOUNTING FOR CONCRETE REINFORCED ELBOWS.

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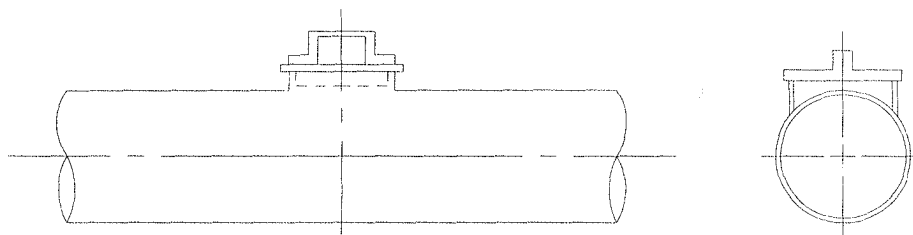
HEAVY DUTY ELBOWS

DATE

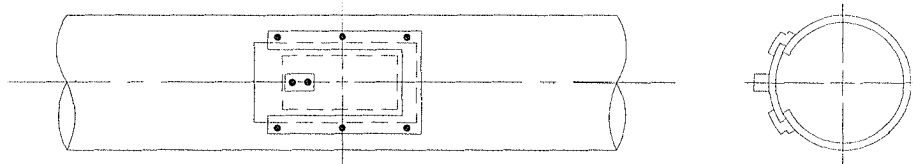
1-95

FIGURE

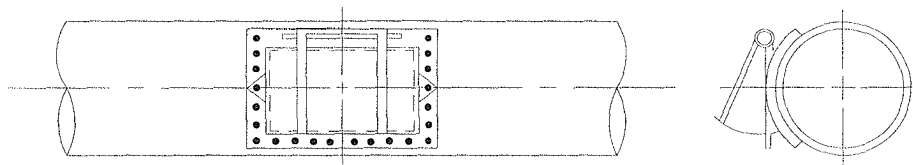
5-22



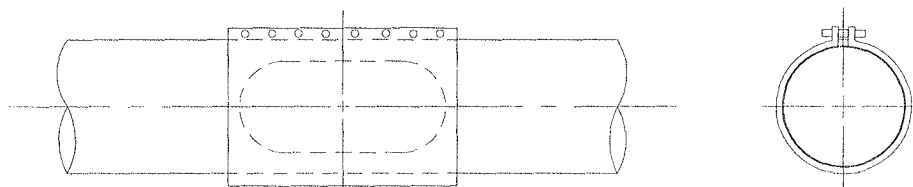
PULLOUT CAP



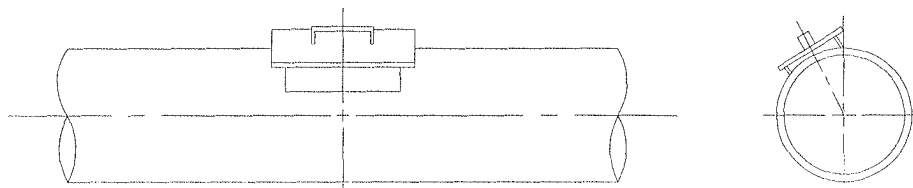
SLIDE



C.I. HINGED DOOR



SPLIT SLEEVE
(ALSO FAN CONNECTION)



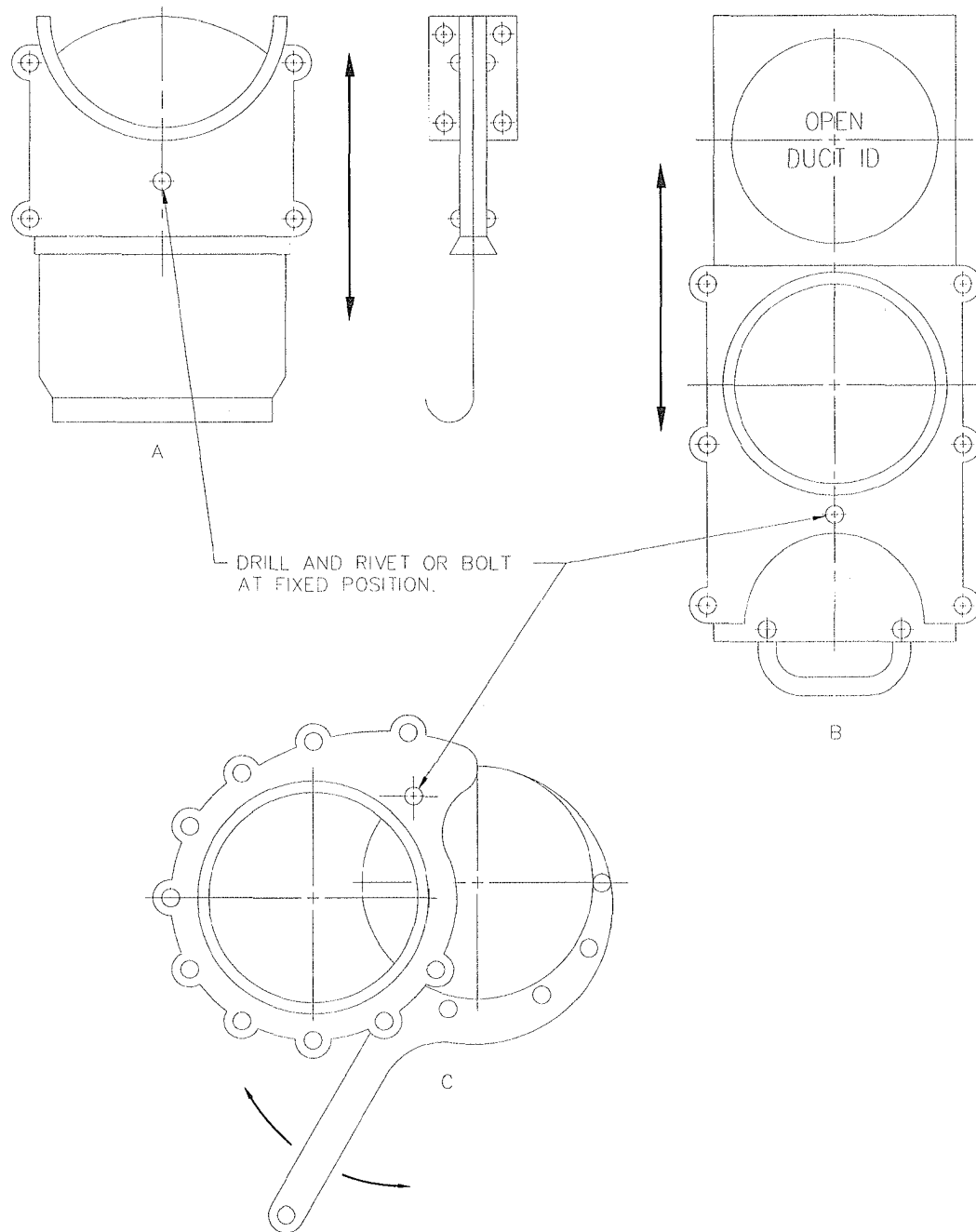
PULLOUT CAP

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CLEANOUT OPENINGS

DATE 1-95

FIGURE 5-23



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BLAST GATES

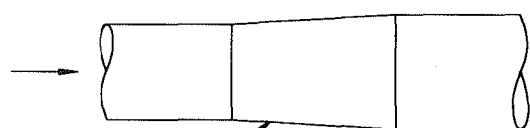
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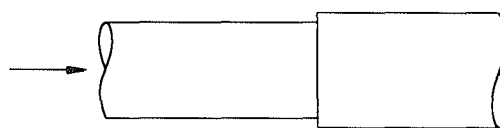
FIGURE

5-24

DUCT ENLARGEMENTS

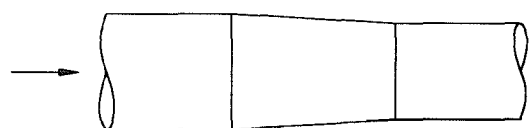


See Fig. 5-16
PREFERRED

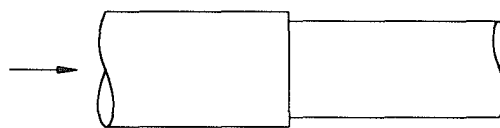


AVOID

DUCT CONTRACTIONS

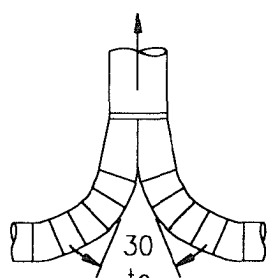


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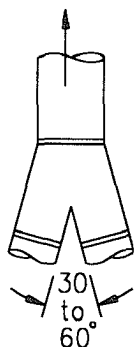


AVOID

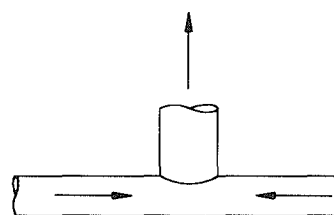
SYMMETRICAL WYES



PREFERRED



PREFERRED



AVOID

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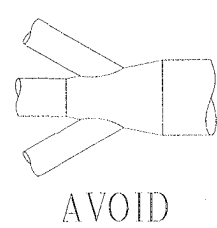
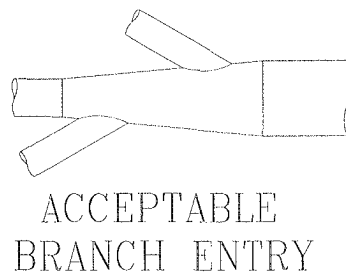
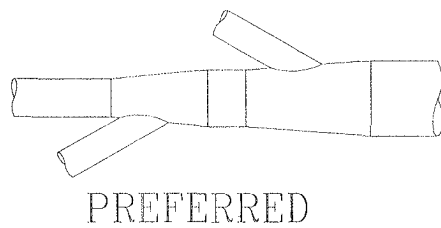
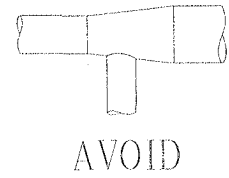
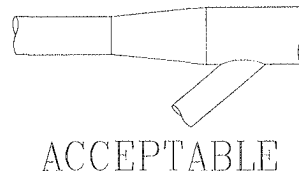
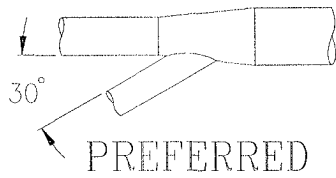
PRINCIPLES OF DUCT DESIGN

DATE

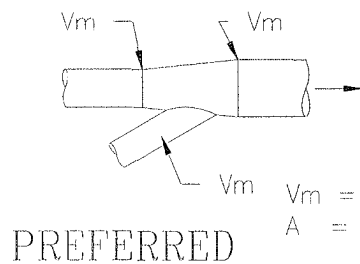
1-95

FIGURE

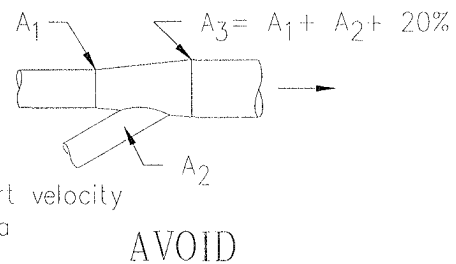
5-25



Branches should enter at gradual expansions and at an angle of 30° or less (preferred) to 45° if necessary. Expansion should be 15° maximum. See Fig. 5-15 for loss coefficients.



V_m = Minimum transport velocity
A = Cross section area



PROPER DUCT SIZE

Size the duct to maintain the selected or higher transport velocity.

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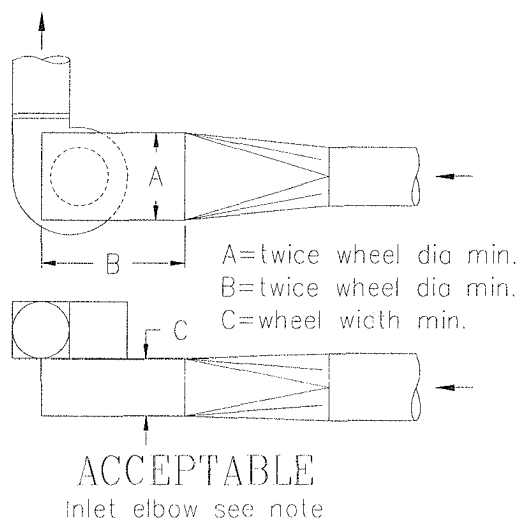
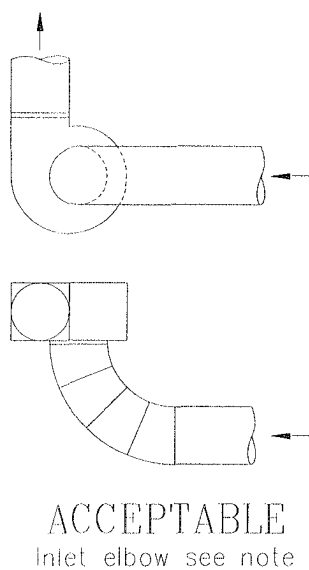
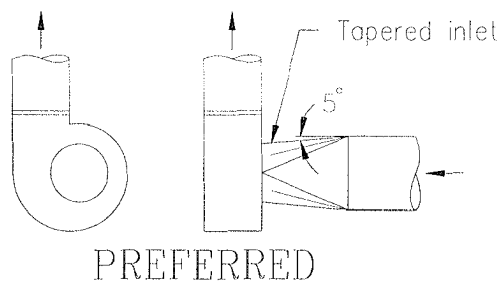
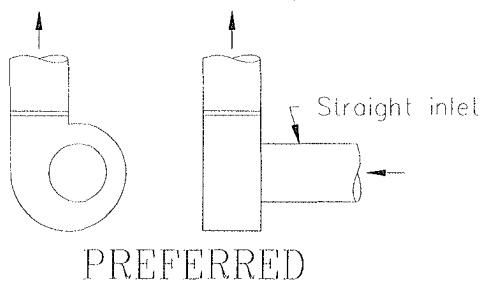
PRINCIPLES OF DUCT DESIGN
BRANCH ENTRY

DATE

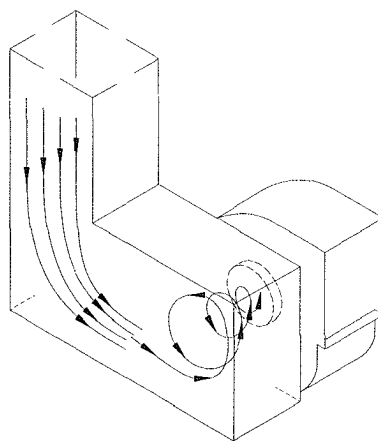
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FIGURE

5-26



Note:
See Chapter 6 for system effect
factors based on inlet and
outlet duct arrangements.



Use duct turn vanes to eliminate air
spin or uneven loading of fan wheel.

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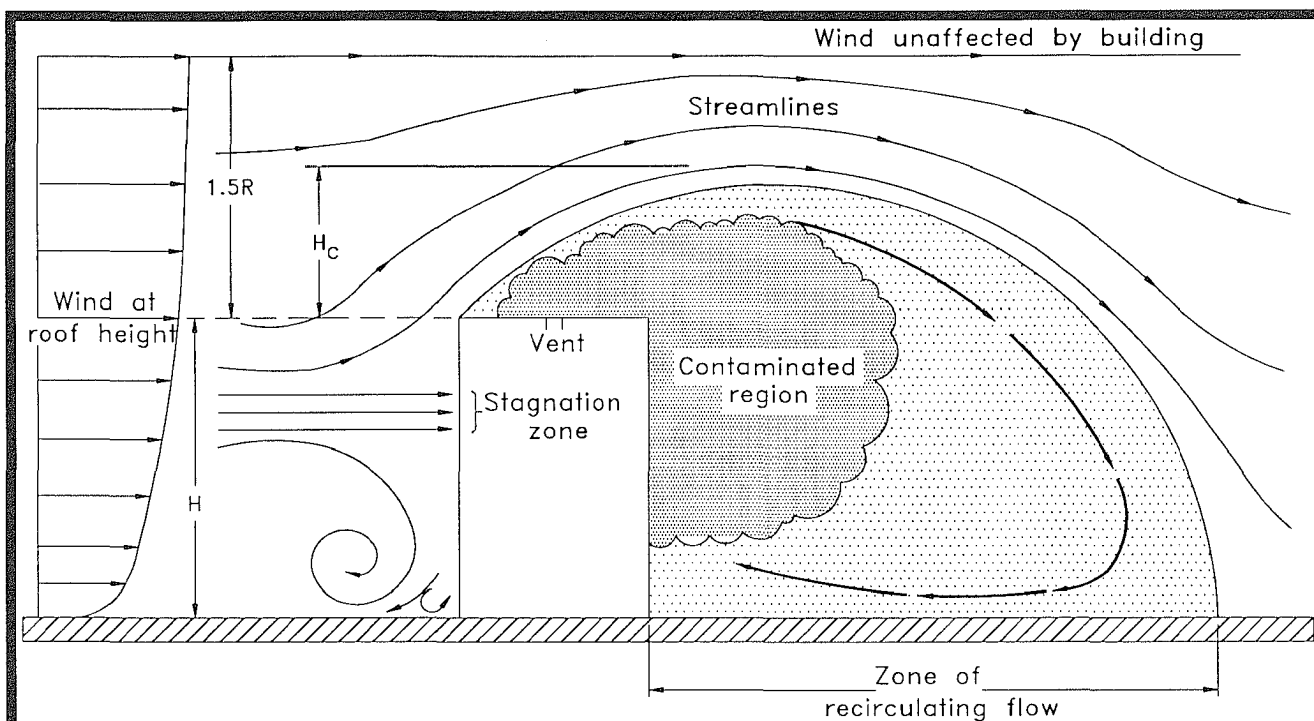
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FAN INLETS*

DATE

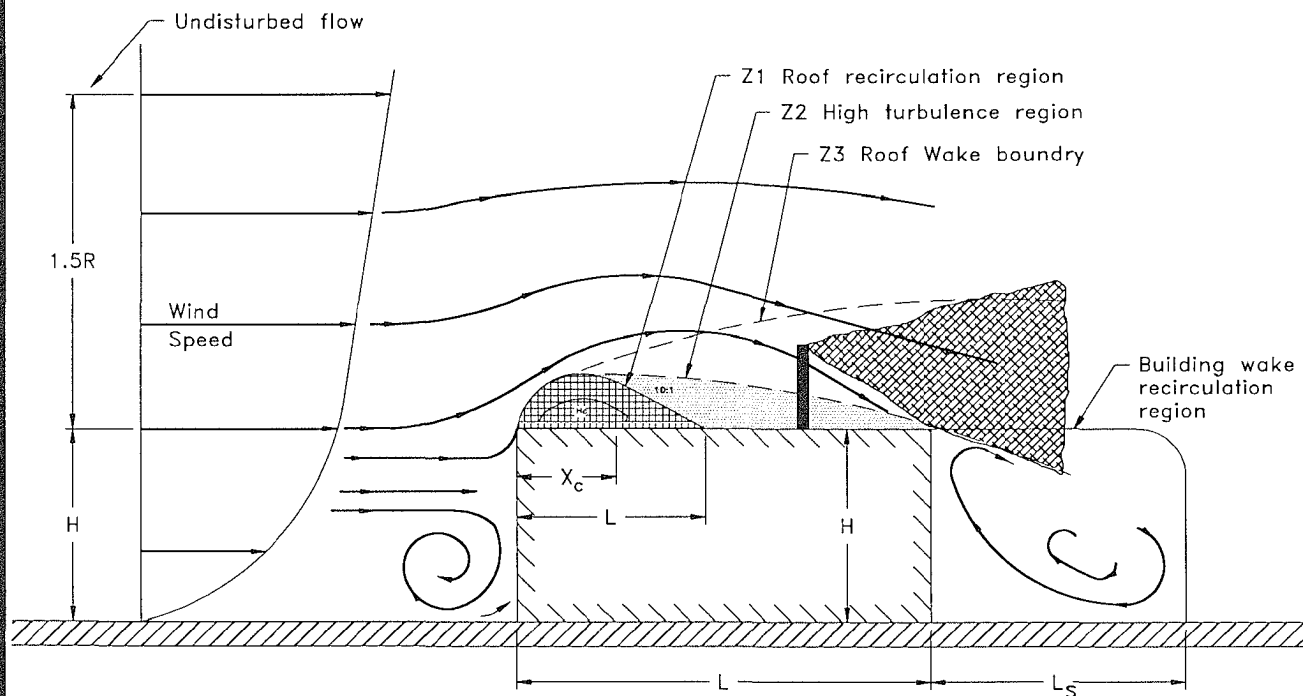
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FIGURE

5-27



A: Centerline flow patterns around a rectangular building



B: Building Recirculation Cavities

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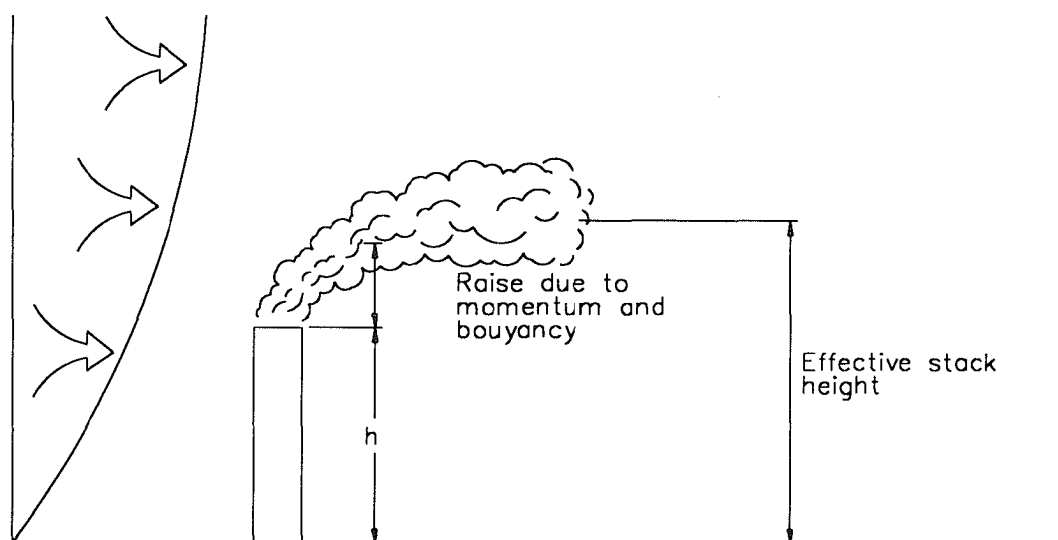
AIRFLOW AROUND BUILDINGS

DATE

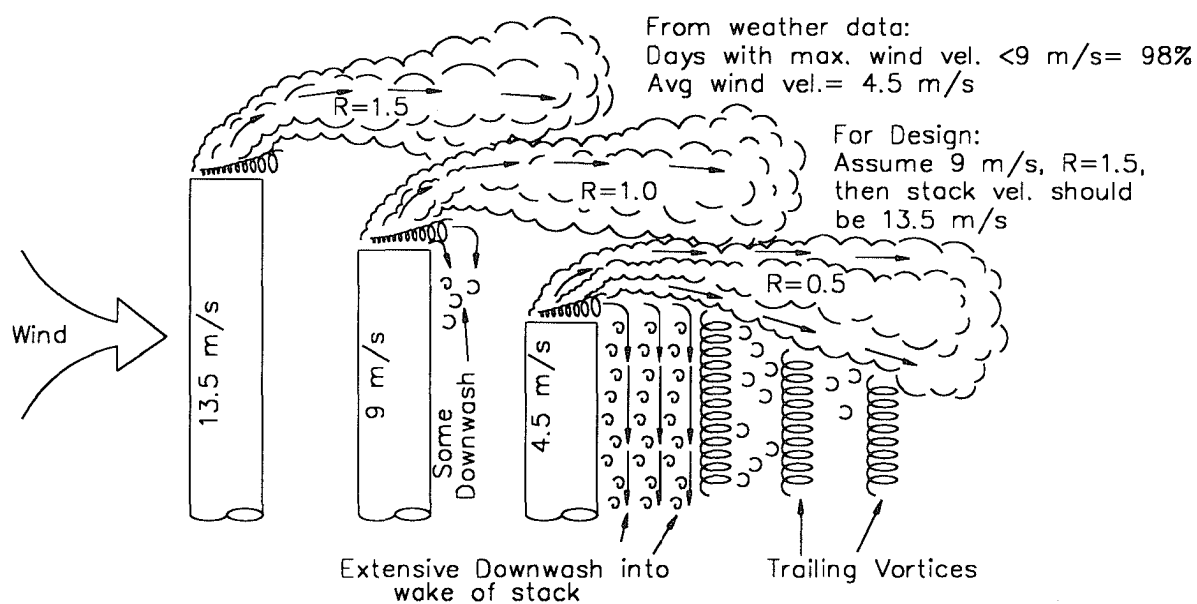
5-94

FIGURE

5-28



A: Effective stack height



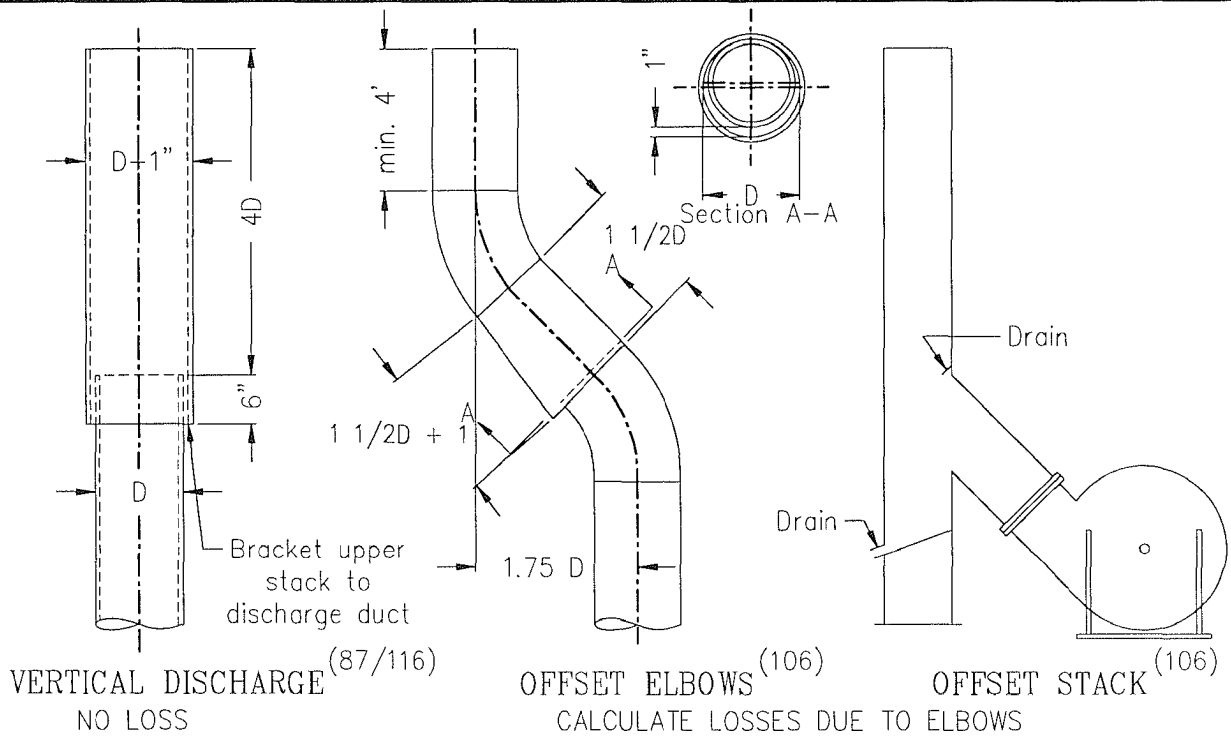
B: Wake Downwash

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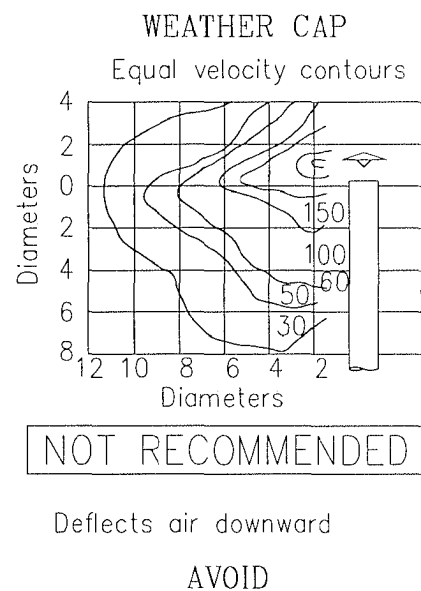
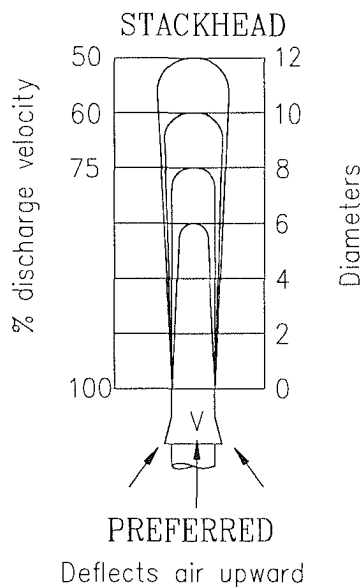
*EFFECTIVE STACK HEIGHT
AND WAKE DOWNWASH*

DATE 4-94

FIGURE 5-29



1. Rain protection characteristics of these caps are superior to a deflecting cap located $0.75D$ from top of a stack.
2. The length of upper stack is related to rain protection. Excessive additional distance may "blowout" of effluent at the gap between upper and lower sections. (86)



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STACKHEAD DESIGNS

DATE

1-95

FIGURE

5-30

Chapter 6

FANS

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6.1 INTRODUCTION

To move air in a ventilation or exhaust system, energy is required to overcome the system losses. This energy can be in the form of natural convection or buoyancy. Most systems, however, require some powered air moving device such as a fan or an ejector.

This chapter will describe the various air moving devices that are used in industrial applications, provide guidelines for the selection of the air moving device for a given situation, and discuss the proper installation of the air moving device in the system to achieve desired performance.

Selection of an air moving device can be a complex task, and the specifier is encouraged to take advantage of all available information from applicable trade associations as well as from individual manufacturers.

6.2 BASIC DEFINITIONS

Air moving devices can be divided into two basic classifications: ejectors and fans. Ejectors have low operating efficiencies and are used only for special material handling applications. Fans are the primary air moving devices used in industrial applications.

Fans can be divided into three basic groups: axial, centrifugal, and special types. As a general rule, axial fans are used for higher flow rates at lower resistances and centrifugal fans are used for lower flow rates at higher resistances.

6.2.1 Ejectors: (see Figure 6-1) Are used sometimes when it is not desirable to have contaminated air pass directly through the air moving device. Ejectors are utilized for air streams containing corrosive, flammable, explosive, hot, or sticky materials that might damage a fan; present a dangerous operating situation; or quickly degrade fan performance. Ejectors also are used in pneumatic conveying systems.

6.2.2 Axial Fans: There are three basic types of axial fans: propeller, tubeaxial, and vaneaxial (see Figures 6-2 and 6-3).

Propeller Fans are used for moving air against low static pressures and are used commonly for general ventilation. Two types of blades are available: disc blade types when there is no duct present; narrow or propeller blade types for moving air against low resistances (less than 1"wg). Performance is very sensitive to added resistance, and a small increase will cause a marked reduction in flow rate.

Tubeaxial Fans (Duct Fans) contain narrow or propeller-type blades in a short, cylindrical housing normally without any type of straightening vanes. Tubeaxial fans will move air against moderate pressures (less than 2"wg).

Vaneaxial Fans have propeller configuration with a hub and airfoil blades mounted in cylindrical housings which normally incorporate straightening vanes on the discharge side of the impeller. Compared to other axial flow fans,

vaneaxial fans are more efficient and generally will develop higher pressures (up to 8"wg). They are limited normally to clean air applications.

6.2.3 Centrifugal Fans: (see Figures 6-4 and 6-5): These fans have three basic impeller designs: forward curved, radial, and backward inclined/backward curved.

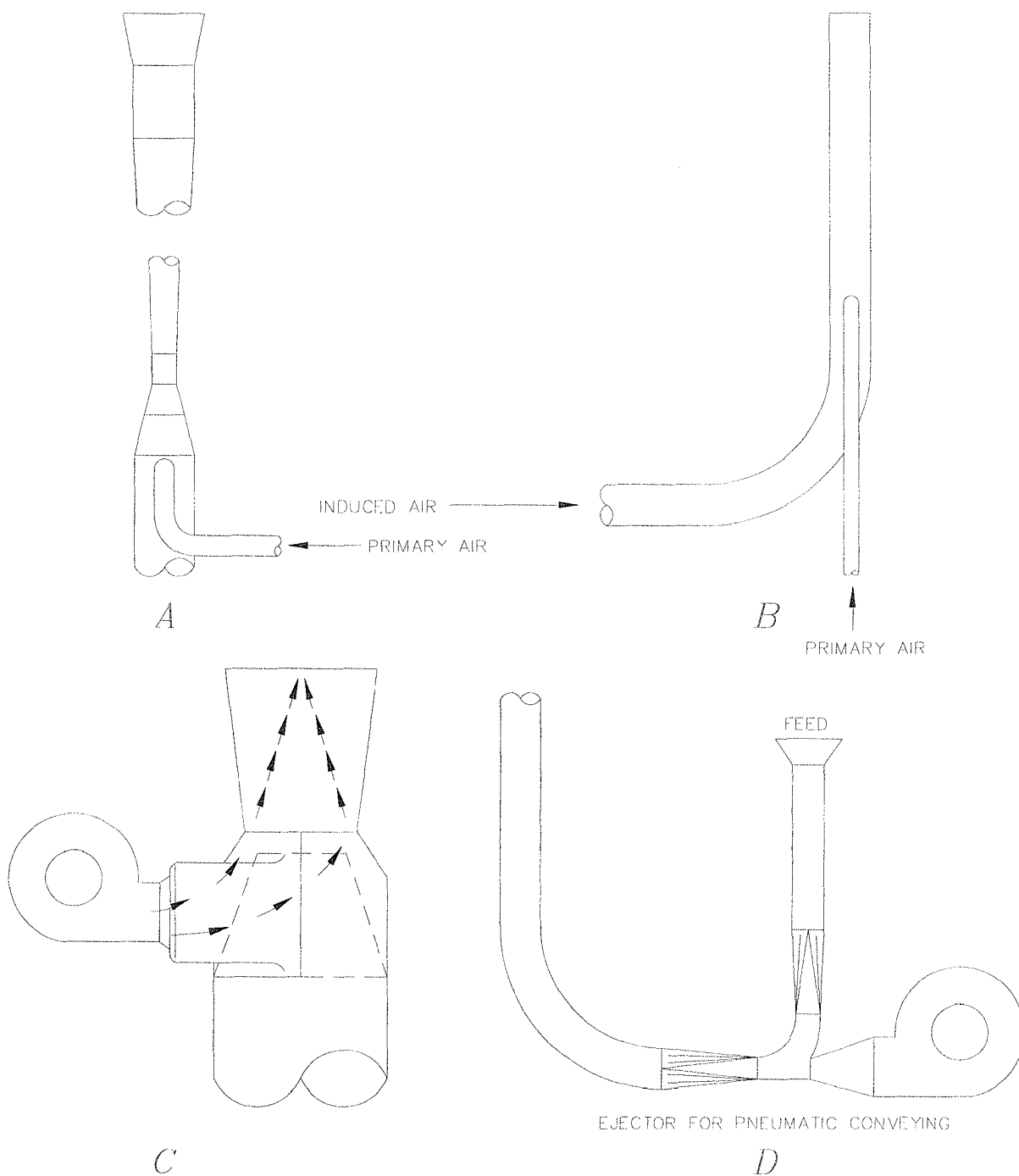
Forward curved (commonly called "squirrel cages") impellers have blades which curve toward the direction of rotation. These fans have low space requirements, low tip speeds, and are quiet in operation. They usually are used against low to moderate static pressures such as those encountered in heating and air conditioning work and replacement air systems. This type of fan is not recommended for dusts or particulates that would adhere to the short curved blades and cause unbalance.

Radial Impellers have blades which are straight or radial from the hub. The housings are designed with their inlets and outlets sized to produce material conveying velocities. There is a variety of impeller types available ranging from "high efficiency, minimum material" to "heavy impact resistance" designs. The radial blade shape will resist material buildup. This fan design is used for most exhaust system applications when particulates will pass through the fan. These fans usually have medium tip speeds and are used for a variety of exhaust systems which handle either clean or dirty air.

Backward Inclined/Backward Curved impeller blades are inclined opposite to the direction of fan rotation. This type usually has higher tip speeds and provides high fan efficiency and relatively low noise levels with "non-overloading" horsepower characteristics. In a non-overloading fan, the maximum horsepower occurs near the optimum operating point so any variation from that point due to a change in system resistance will result in a reduction in operating horsepower. The blade shape is conducive to material buildup so fans in this group should be limited as follows:

- *Single-Thickness Blade:* Solid blades allow the unit to handle light dust loading or moisture. It should not be used with particulates that would build up on the underside of the blade surfaces.
- *Airfoil Blade:* Airfoil blades offer higher efficiencies and lower noise characteristics. Hollow blades erode more quickly with material and can fill with liquid in high humidity applications. These should be limited to clean air service.

6.2.4 Special Type Fans (see Figure 6-4): *In-line Centrifugal Fans* have backward inclined blades with special housings which permit a straight line duct installation. Pressure versus flow rate versus horsepower performance curves are similar to a scroll-type centrifugal fan of the same blade type. Space requirements are similar to vaneaxial fans.

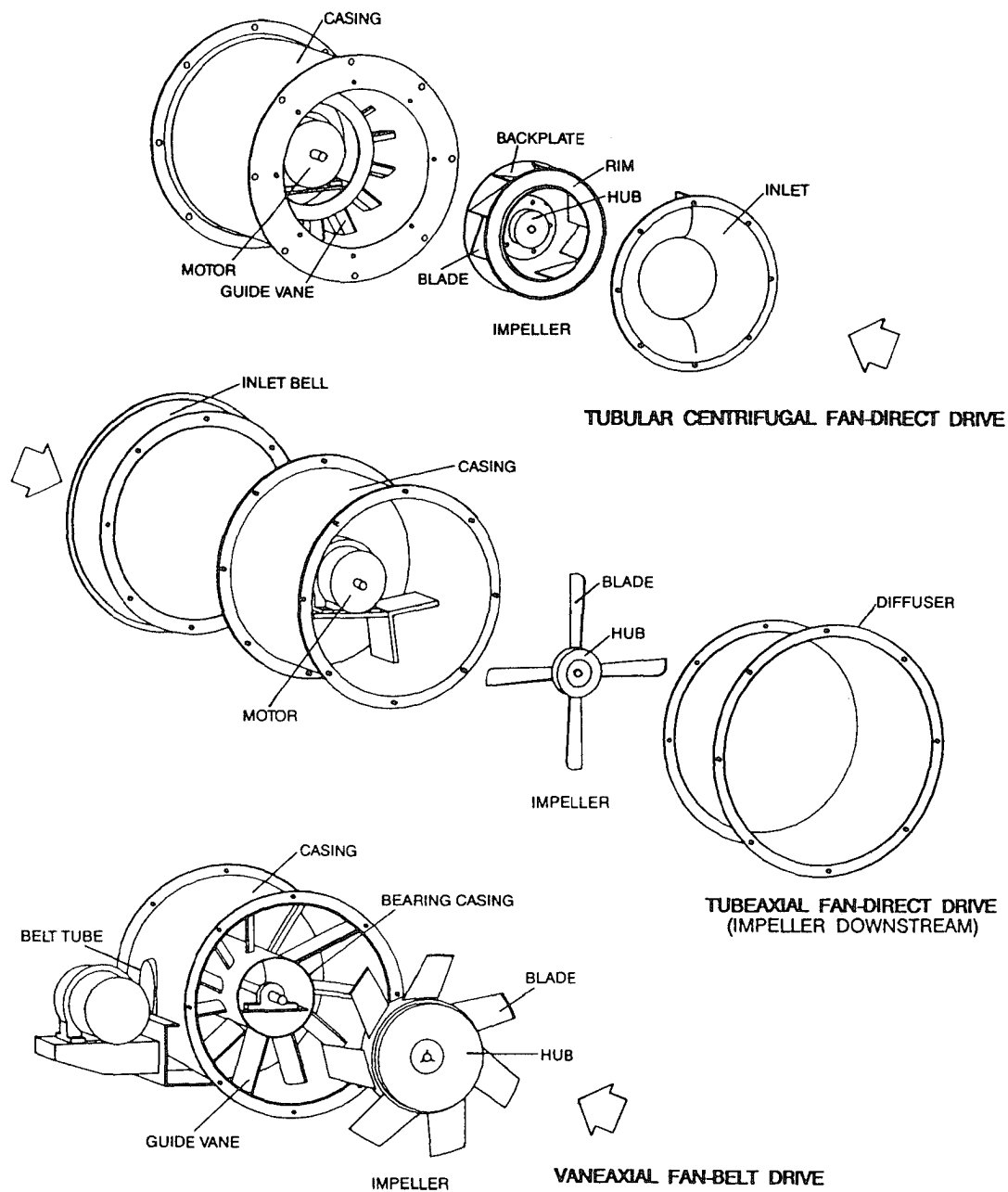


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AIR EJECTORS

DATE 1-88

FIGURE 6-1



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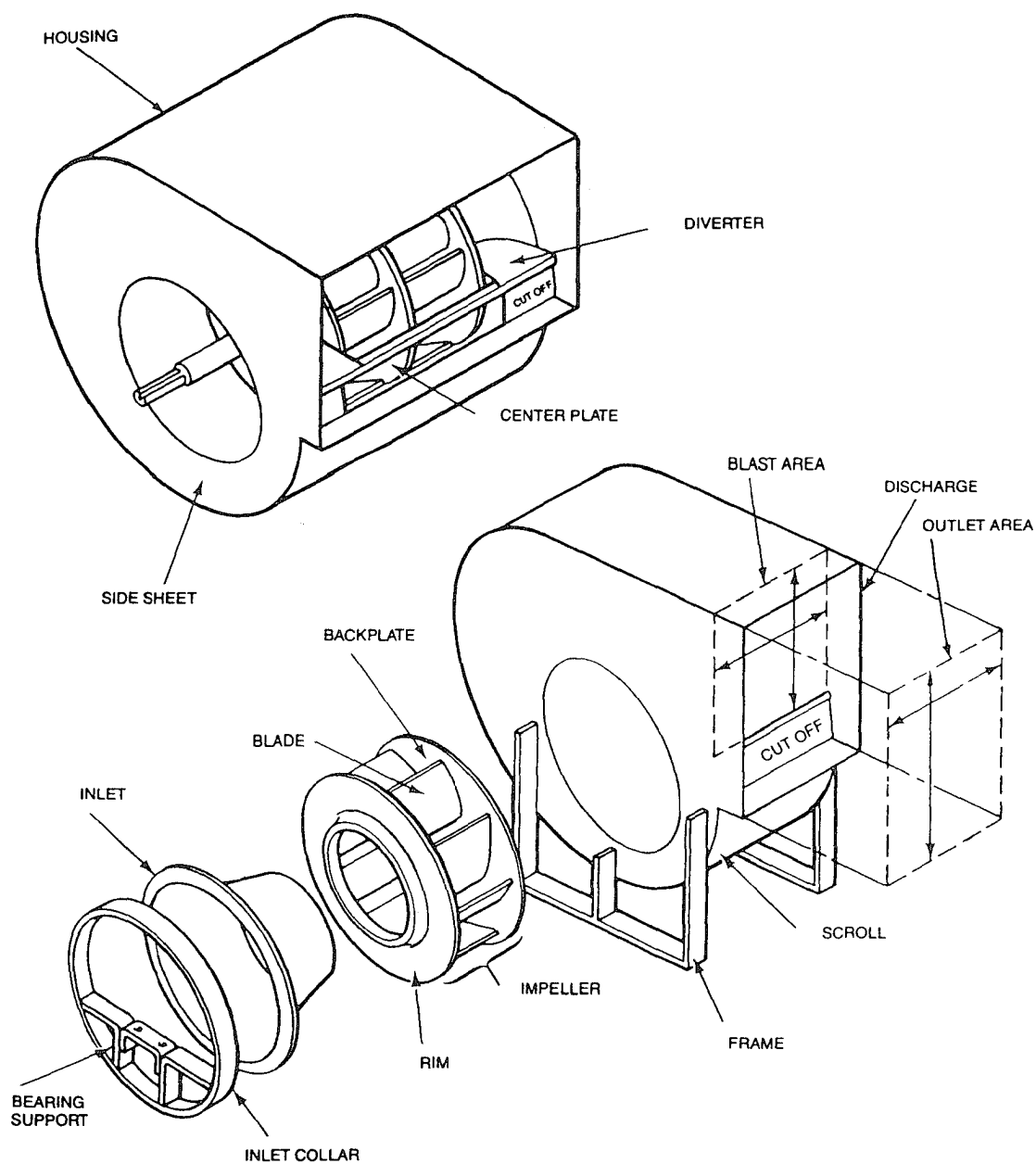
TERMINOLOGY FOR AXIAL AND TUBULAR CENTRIFUGAL FANS

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FIGURE

6-2



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*TERMINOLOGY FOR
CENTRIFUGAL FAN
COMPONENTS*

DATE

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FIGURE

6-3

Power Exhausters, Power Roof Ventilators are packaged units that can be either axial flow or centrifugal type. The centrifugal type does not use a scroll housing but discharges around the periphery of the ventilator to the atmosphere. These units can be obtained with either downward deflecting or upblast discharges.

Fan and Dust Collector Combination: There are several designs in which fans and dust collectors are packaged in a unit. If use of such equipment is contemplated, the manufacturer should be consulted for proper application and performance characteristics.

6.3 FAN SELECTION

Fan selection involves not only finding a fan to match the required flow and pressure considerations but all aspects of

an installation including the air stream characteristics, operating temperature, drive arrangement, and mounting. Section 6.2 discussed the various fan types and why they might be selected. This section offers guidelines to fan selection; however, the exact performance and operating limitations of a particular fan should be obtained from the original equipment manufacturer.

6.3.1 Considerations for Fan Selection:

CAPACITY

Flow Rate (Q): Based on system requirements and expressed as actual cubic feet per minute (acfm) at the fan inlet.

Pressure Requirements: Based on system pressure requirements which normally are expressed as Fan Static Pressure (FSP) or Fan Total Pressure (FTP) in inches of water gauge

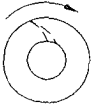
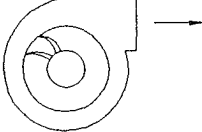

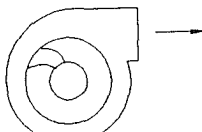

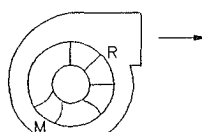

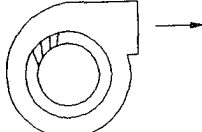
TYPE		IMPELLER DESIGN	HOUSING DESIGN
CENTRIFUGAL FANS	AIRFOIL	 <p>Highest efficiency of all centrifugal fan designs. 9 to 16 blades of airfoil contour curved away from the direction of rotation. Air leaves the impeller at a velocity less than its tip speed and relatively deep blades provide for efficient expansion within the blade passages. For given duty, this will be the highest speed of the centrifugal fan designs.</p>	 <p>Scroll-type, usually designed to permit efficient conversion of velocity pressure to static pressure, thus permitting a high static efficiency; essential that clearance and alignment between wheel and inlet bell be very close in order to reach the maximum efficiency capability. Concentric housings can also be used in power roof ventilators, since there is efficient pressure conversion in the wheel.</p>
	BACKWARD-INCLINED BACKWARD-CURVED	 <p>Efficiency is only slightly less than that of airfoil fans. Backward-inclined or backward-curved blades are single thickness. 9 to 16 blades curved or inclined away from the direction of rotation. Efficient for the same reasons given for the airfoil fan above.</p>	 <p>Utilizes the same housing configuration as the airfoil design.</p>
	RADIAL	 <p>Simplest of all centrifugal fans and least efficient. Has high mechanical strength and the wheel is easily repaired. For a given point of rating, this fan requires medium speed. This classification includes radial blades (R) and modified radial blades (M), usually 6 to 10 in number.</p>	 <p>Scroll-type, usually the narrowest design of all centrifugal fan designs described here because of required high velocity discharge. Dimensional requirements of this housing are more critical than for airfoil and backward-inclined blades.</p>
	FORWARD-CURVED	 <p>Efficiency is less than airfoil and backward-curved bladed fans. Usually fabricated of lightweight and low cost construction. Has 24 to 64 shallow blades with both the heel and tip curved forward. Air leaves wheel at velocity greater than wheel. Tip speed and primary energy transferred to the air is by use of high velocity in the wheel. For given duty, wheel is the smallest of all centrifugal types and operates at lowest speed.</p>	 <p>Scroll is similar to other centrifugal-fan designs. The fit between the wheel and inlet is not as critical as on airfoil and backward-inclined bladed fans. Uses large cut-off sheet in housing.</p>

FIGURE 6-4. Types of fans: impeller and housing designs (see facing page)

at standard conditions (0.075 lbm/ft^3). If the required pressure is known only at non standard conditions, a density correction (see Section 6.3.8) must be made.

AIR STREAM

Material handled through the fan: When the exhaust air contains a small amount of smoke or dust, a backward inclined centrifugal or axial fan should be selected. With light dust, fume or moisture, a backward inclined or radial centrifugal fan would be the preferred selection. If the particulate loading is high, or when material is handled, the normal selection would be a radial centrifugal fan.

Explosive or Flammable Material: Use spark resistant construction (explosion-proof motor if the motor is in the air

stream). Conform to the standards of the National Board of Fire Underwriters, the National Fire Protection Association and governmental regulations (see Section 6.3.9).

Corrosive Applications: May require a protective coating or special materials of construction (stainless, fiberglass, etc.)

Elevated Air Stream Temperatures: Maximum operating temperature affects strength of materials and therefore must be known for selection of correct materials of construction, arrangement, and bearing types.

PHYSICAL LIMITATIONS

Fan size should be determined by performance requirements. Inlet size and location, fan weight, and ease of main-

PERFORMANCE CURVES	PERFORMANCE CHARACTERISTICS*	APPLICATIONS
	Highest efficiencies occur 50 to 65% of wide open volume. This is also the area of good pressure characteristics; the horsepower curve reaches a maximum near the peak efficiency area and becomes lower toward free delivery, a self-limiting power characteristic as shown.	General heating, ventilating and air-conditioning systems. Used in large sizes for clean air industrial applications where power savings are significant.
	Operating characteristics of this fan are similar to the airfoil fan mentioned above. Peak efficiency for this fan is slightly lower than the airfoil fan. Normally unstable left of peak pressure.	Same heating, ventilating, and air-conditioning applications as the airfoil fan. Also used in some industrial applications where the airfoil blade is not acceptable because of corrosive and/or erosion environment.
	Higher pressure characteristics than the above mentioned fans. Power rises continually to free delivery.	Used primarily for material-handling applications in industrial plants. Wheel can be of rugged construction and is simple to repair in the field. Wheel is sometimes coated with special material. This design also used for high-pressure industrial requirements. Not commonly found in HVAC applications.
	Pressure curve is less steep than that of backward-curved bladed fans. There is a dip in the pressure curve left of the peak pressure point and highest efficiency occurs to the right of peak pressure, 40 to 50% of wide open volume. Fan should be rated to the right of peak pressure. Power curve rises continually toward free delivery and this must be taken into account when motor is selected.	Used primarily in low-pressure heating, ventilating, and air-conditioning applications such as domestic furnaces, central station units, and packaged air-conditioning equipment from room air-conditioning units to roof top units.

Types of fans: Performance characteristics and applications. (*These performance curves reflect the general characteristics of various fans as commonly employed. They are not intended to provide complete selection criteria for application purposes, since other parameters, such as diameter and speed, are not defined.)

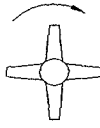
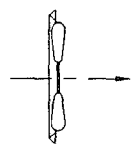
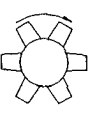
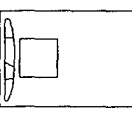
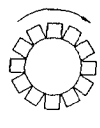
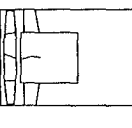
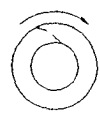
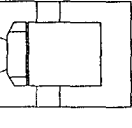

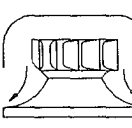
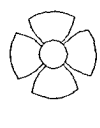
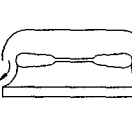
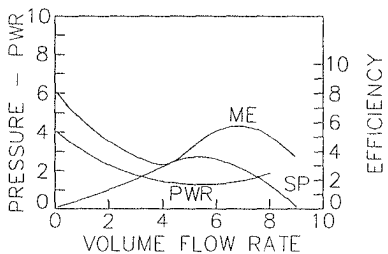
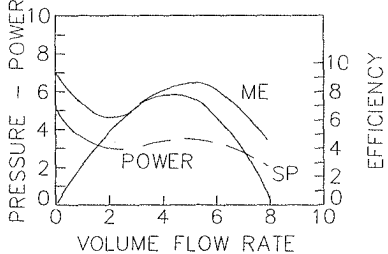
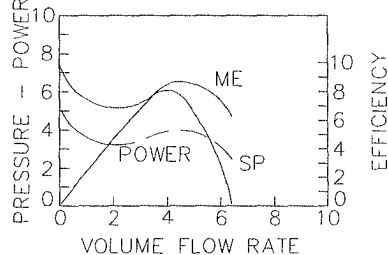
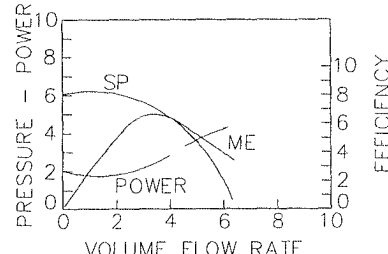
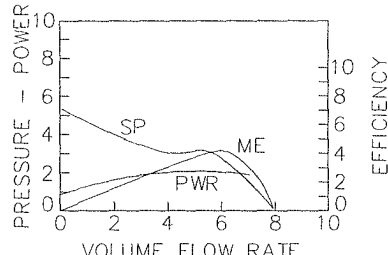
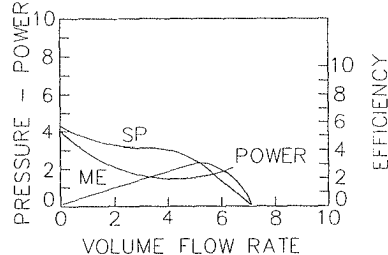
TYPE		IMPELLER DESIGN	HOUSING DESIGN
AXIAL FANS	PROPELLER	 <p>Efficiency is low. Impellers are usually of inexpensive construction and limited to low pressure applications. Impeller is of 2 or more blades, usually of single thickness attached to relatively small hub. Energy transfer is primarily in form of velocity pressure.</p>	 <p>Simple circular ring, orifice plate, or venturi design. Design can substantially influence performance and optimum design is reasonably close to the blade tips and forms a smooth inlet flow contour to the wheel.</p>
	TUBEAXIAL	 <p>Somewhat more efficient than propeller fan design and is capable of developing a more useful static pressure range. Number of blades usually from 4 to 8 and hub is usually less than 50% of fan tip diameter. Blades can be of airfoil or single thickness cross section.</p>	 <p>Cylindrical tube formed so that the running clearance between the wheel tip and tube is close. This results in significant improvement over propeller fans.</p>
	VANEAXIAL	 <p>Good design of blades permits medium-to high-pressure capability at good efficiency. The most efficient fans of this type have airfoil blades. Blades are fixed or adjustable pitch types and hub is usually greater than 50% of fan tip diameter.</p>	 <p>Cylindrical tube closely fitted to the outer diameter of blade tips and fitted with a set of guide vanes. Upstream or downstream from the impeller, guide vanes convert the rotary energy imparted to the air and increase pressure and efficiency of fan.</p>
SPECIAL DESIGNS	TUBULAR	 <p>This fan usually has a wheel similar to the airfoil backward-inclined or backward-curved blade as described above. (However, this fan wheel type is of lower efficiency when used in fan of type.) Mixed flow impellers are sometimes used.</p>	 <p>Cylindrical shell similar to a vaneaxial fan housing, except the outer diameter of the wheel does not run close to the housing. Air is discharged radially from the wheel and must change direction by 90 degrees to flow through the guide vane section.</p>
	CENTRIFUGAL	 <p>Many models use airfoil or backward-inclined impeller designs. These have been modified from those mentioned above to produce a low-pressure, high-volume flow rate characteristic. In addition, many special centrifugal impeller designs are used, including mixed-flow design.</p>	 <p>Does not utilize a housing in a normal sense since the air is simply discharged from the impeller in a 360 degree pattern and usually does not include a configuration to recover the velocity pressure component.</p>
	POWER ROOF VENTILATORS	 <p>A great variety of propeller designs are employed with the objective of high-volume flow rate at low pressure.</p>	 <p>Essentially a propeller fan mounted in a supporting structure with a cover for weather protection and safety considerations. The air is discharged through the annular space around the bottom of the weather hood.</p>

FIGURE 6-4 (continued). Types of fans: impeller and housing design

PERFORMANCE CURVES	PERFORMANCE CHARACTERISTICS*	APPLICATIONS
	High flow rate but very low-pressure capabilities and maximum efficiency is reached near free delivery. The discharge pattern of the air is circular in shape and the air stream swirls because of the action of the blades and the lack of straightening facilities.	For low-pressure, high-volume air moving applications such as air circulation within a space or ventilation through a wall without attached duct work. Used for replacement air applications.
	High flow-rate characteristics with medium-pressure capabilities. Performance curve includes a dip to the left of peak pressure which should be avoided. The discharge air pattern is circular and is rotating or whirling because of the propeller rotation and lack of guide vanes.	Low- and medium-pressure ducted heating, ventilating, and air-conditioning applications where air distribution on the downstream side is not critical. Also used in some industrial applications such as drying ovens, paint spray booths, and fume exhaust systems.
	High-pressure characteristics with medium volume flow rate capabilities. Performance curve includes a dip caused by aerodynamic stall to the left of peak pressure, which should be avoided. Guide vanes correct the circular motion imparted to the air by the wheel and improve pressure characteristics and efficiency of the fan.	General heating, ventilating, and air-conditioning systems in low-, medium-, and high-pressure applications is of advantage where straight-through flow and compact installation are required; air distribution on downstream side is good. Also used in industrial application similar to the tubeaxial fan. Relatively more compact than comparable centrifugal-type fans for same duty.
	Performance is similar to backward-curved fan, except lower capacity and pressure because of the 90 degree change in direction of the air flow in the housing. The efficiency will be lower than the backward-curved fan. Some designs may have a dip in the curve similar to the axial-flow fan.	Used primarily for low-pressure return air systems in heating, ventilating, and air-conditioning applications. Has straight-through flow configuration.
	Usually intended to operate without attached ductwork and therefore to operate against a very low-pressure head. It is usually intended to have a rather high-volume flow rate characteristic. Only static pressure and static efficiency are shown for this type of product.	For low-pressure exhaust systems such as general factory, kitchen, warehouse, and commercial installations where the low-pressure rise limitation can be tolerated. Unit is low in first cost and low in operating cost and provides positive exhaust ventilation in the space which is a decided advantage over gravity-type exhaust units. The centrifugal unit is somewhat quieter than the axial unit described below.
	Usually intended to operate without attached ductwork and therefore to operate against very low-pressure head. It is usually intended to have a high-volume flow rate characteristic. Only static pressure and static efficiency are shown for this type of product.	For low-pressure exhaust systems such as general factory, kitchen, warehouse, and some commercial installations where the low-pressure rise limitations can be tolerated. Unit is low in first cost and low in operating cost and provides positive exhaust ventilation in the space which is a decided advantage over gravity-type exhaust units.

tenance also must be considered. The most efficient fan size may not fit the physical space available.

DRIVE ARRANGEMENTS

All fans must have some type of power source — usually an electric motor. On packaged fans, the motor is furnished and mounted by the manufacturer. On larger units, the motor is mounted separately and coupled directly to the fan or indirectly by a belt drive. A number of standard drive arrangements are shown in Figures 6-5a, 6-5b, and 6-5c.

Direct Drive offers a more compact assembly and assures constant fan speed. Fan speeds are limited to available motor speeds (except in the case of variable frequency controllers). Capacity is set during construction by variations in impeller geometry and motor speed.

Belt Drive offers flexibility in that fan speed can be changed by altering the drive ratio. This may be important in some applications to provide for changes in system capacity or pressure requirements due to changes in process, hood design, equipment location or air cleaning equipment. V-belt drives must be maintained and have some power losses which can be estimated from the chart in Figure 6-6.

NOISE

Fan noise is generated by turbulence within the fan housing and will vary by fan type, flow rate, pressure, and fan efficiency. Because each design is different, noise ratings must be obtained from the fan manufacturer. Most fans produce a "white" noise which is a mixture of all frequencies. In addition to white noise, radial blade fans also produce a pure tone at a frequency equal to the blade passage frequency (BPF):

$$\text{BPF} = \text{RPM} \times \text{N} \times \text{CF} \quad [6.1]$$

where:

BPF = blade passage frequency, Hz

RPM = rotational rate, rpm

N = number of blades

CF = conversion coefficient, 1/60

This tone can be very noticeable in some installations and should be considered in the system design.

Because of its higher efficiency, the backward inclined impeller design is generally the quietest. However, for all fan types, non-uniform air flow at the fan inlet or outlet can increase the fan noise level. This is another problem related to "system effect" (see Section 6.4.1).

Most fan manufacturers publish sound ratings for their products. There are a variety of ways to present the ratings. One popular way is to list sound power levels for eight ANSI standard octave bands. The sound power levels are typically in units called "decibels" (dB). The sound **power** level is a characteristic of a fan that varies with the fan speed and point of operation.

For an installed fan, the surrounding environment affects the sound level that is measured or heard. Walls, floors, and other equipment reflect and absorb sound to varying degrees. The sound that reaches the listener will be different than the fan's rated sound power level. Typical sound measuring devices detect sound with a microphone and display sound pressure level in decibels. This sound **pressure** is an environment-dependent measurement that changes with listener location and/or environment changes.

While the decibel unit is used for sound power and sound pressure, the two measures are not interchangeable. For instance, 70 dB sound power is not 70 dB sound pressure. The decibel is not an absolute unit of measure. It is a ratio between a measured quantity and an agreed reference level. Both dB scales are logarithmic. The sound **power** is the log of the ratio of two power levels. The sound **pressure** is the log of the ratio of two pressure levels. The sound power scale uses a reference of 10^{-12} watts. The sound pressure scale uses a reference of 20×10^{-6} N/M².

For an installed fan, the sound pressure levels are usually measured in dB using the "A" weighting scale. The A-weighting is used to measure environmental noise as it most closely reflects the human auditory response to noise of various frequencies. A sound level meter set on the "A" scale automatically integrates the noise of all frequencies to give a single dBA noise measurement. Expanded detail can be obtained by taking noise measurements with a meter capable of measuring the sound pressure level in each octave band. Such detail can help indicate the predominant source of a noise.

The topic of sound is quite broad and there are many reference texts available to cover it. For a concise introduction, the *ASHRAE Fundamentals Handbook*^(6.4) is a good starting point.

SAFETY AND ACCESSORIES

Safety Guards are required. Consider all danger points such as inlet, outlet, shaft, drive and cleanout doors. Construction should comply with applicable governmental safety requirements, and attachment must be secure.

Accessories can help in the installation and in future maintenance requirements. Examples might include drains, cleanout doors, split housings, and shaft seals.

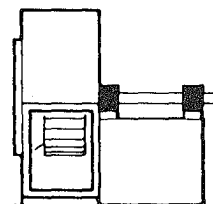
FLOW CONTROL

There are various accessories that can be used to change fan performance. Such changes may be required on systems that vary throughout the day or for reduction in flow rate in anticipation of some future requirement. Dampers, variable pitch blades, and speed control are three common accessories used with fans.

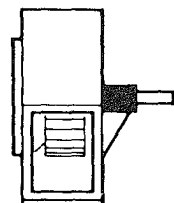
Dampers are installed directly on the fan inlet or outlet. Because they are in the air stream, dampers can build up with

SW - Single Width DW - Double Width
SI - Single Inlet DI - Double Inlet

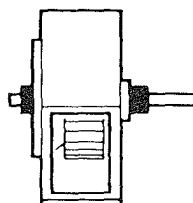
Arrangements 1, 3, 7 and 8 are also available with bearings mounted on pedestals or base set independent of the fan housing



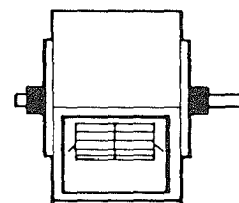
ARR. 1 SWSI For belt drive or direct connection. Impeller overhung. Two bearings on base



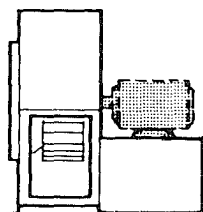
ARR. 2 SWSI For belt drive or direct connection. Impeller overhung. Bearings in bracket supported by fan housing.



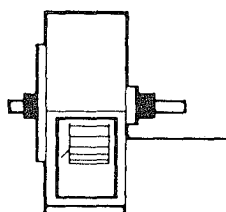
ARR. 3 SWSI For belt drive or direct connection. One bearing on each side and supported by fan housing.



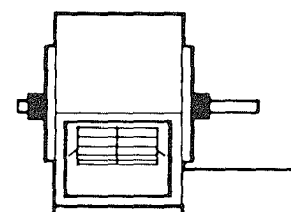
ARR. 3 DWDI For belt drive or direct connection. One bearing on each side and supported by fan housing.



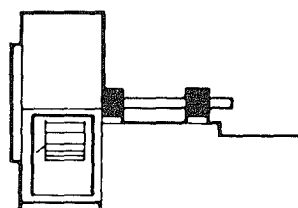
ARR. 4 SWSI For direct drive. Impeller overhung on prime mover shaft. No bearings on fan. Prime mover base mounted or integrally directly connected.



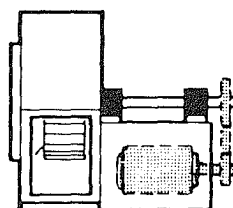
ARR. 7 SWSI For belt drive or direct connection. Arrangement 3 plus base for prime mover.



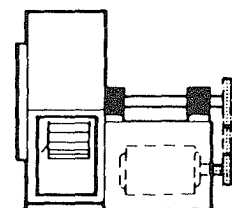
ARR. 7 DWDI For belt drive or direct connection. Arrangement 3 plus base for prime mover.



ARR. 8 SWSI For belt drive or direct connection. Arrangement 1 plus extended base for prime mover.



ARR. 9 SWSI For belt drive. Impeller overhung, two bearings, with prime mover outside base



ARR. 10 SWSI For belt drive. Impeller overhung, two bearings, with prime mover inside base

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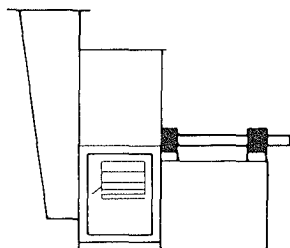
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DRIVE ARRANGEMENTS FOR CENTRIFUGAL FANS

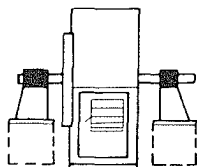
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FIGURE 6-5a

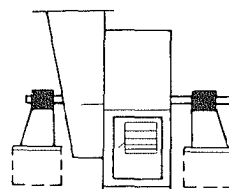
SW - Single Width DW - Double Width
SI - Single Inlet DI - Double Inlet



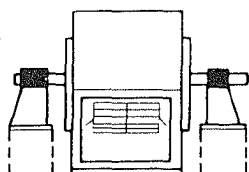
ARR. 1 SWSI WITH INLET BOX For belt drive or direct connection. Impeller overhung, two bearings on base. Inlet box may be self-supporting.



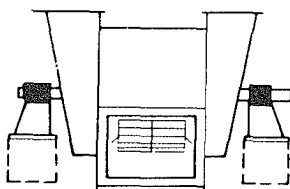
ARR. 3 SWSI WITH INDEPENDENT PEDESTAL For belt drive or direct connection fan. Housing is self-supporting. One bearing on each side supported by independent pedestals.



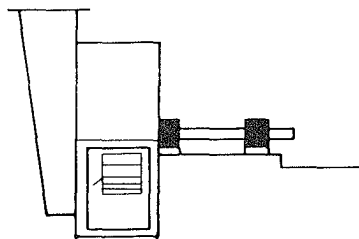
ARR. 3 SWSI WITH INLET BOX AND INDEPENDENT PEDESTALS For belt drive or direct connection fan. Housing is self-supporting. One bearing on each side supported by independent pedestals with shaft extending through inlet box.



ARR. 3 DWDI WITH INDEPENDENT PEDESTAL For belt drive or direct connection fan. Housing is self-supporting. One bearing on each side supported by independent pedestals.



ARR. 3 DWDI WITH INLET BOX AND INDEPENDENT PEDESTALS For belt drive or direct connection fan. Housing is self-supporting. One bearing on each side supported by independent pedestals with shaft extending through inlet box.



ARR. 8 SWSI WITH INLET BOX For belt drive or direct connection. Impeller overhung, two bearings on base plus extended base for prime mover. Inlet box may be self-supporting.

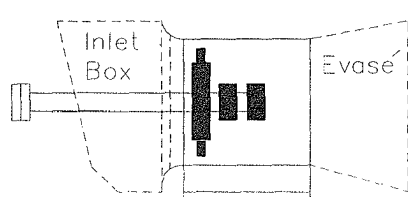
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DRIVE ARRANGEMENTS FOR CENTRIFUGAL FANS

DATE 5-92

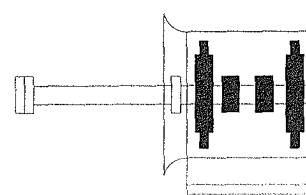
FIGURE 6-5b



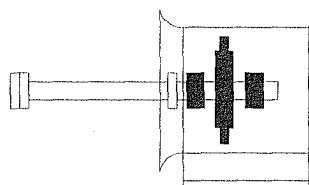
ARR. 1

For belt drive or direct connection. Impeller overhung. Two bearings located either upstream or downstream of impeller.

Inlet box and evase are optional on all arrangements.

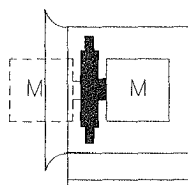


ARR. 1 TWO STAGE



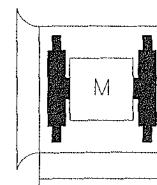
ARR. 3

For belt drive or direct connection. Impeller between bearings that are on internal supports. Drive through inlet.

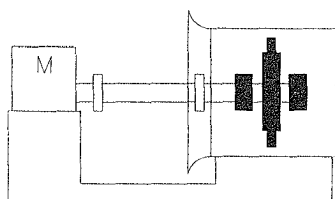


ARR. 4

For direct connection. Impeller overhung on motor shaft. No bearings on fan. Motor on internal supports.

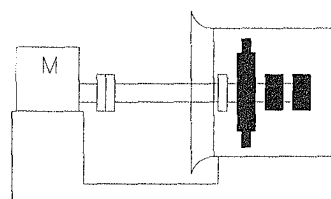


ARR. 4 TWO STAGE



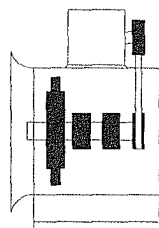
ARR. 7

For belt drive or direct connection. Arr. 3 plus common base for prime mover.



ARR. 8 (1 or 2 stage)

For belt drive or direct connection. Arr. 1 plus common base for prime mover.

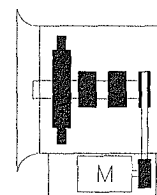


ARR. 9 Motor on Casing

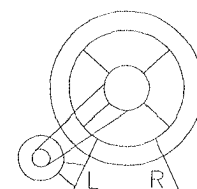
For belt drive. Impeller overhung. Two bearings on internal supports.

Motor on casing or on integral base. Drive through belt fairing.

NOTE: All fan orientations may be horizontal or vertical.



ARR. 9 Motor on Integral Base

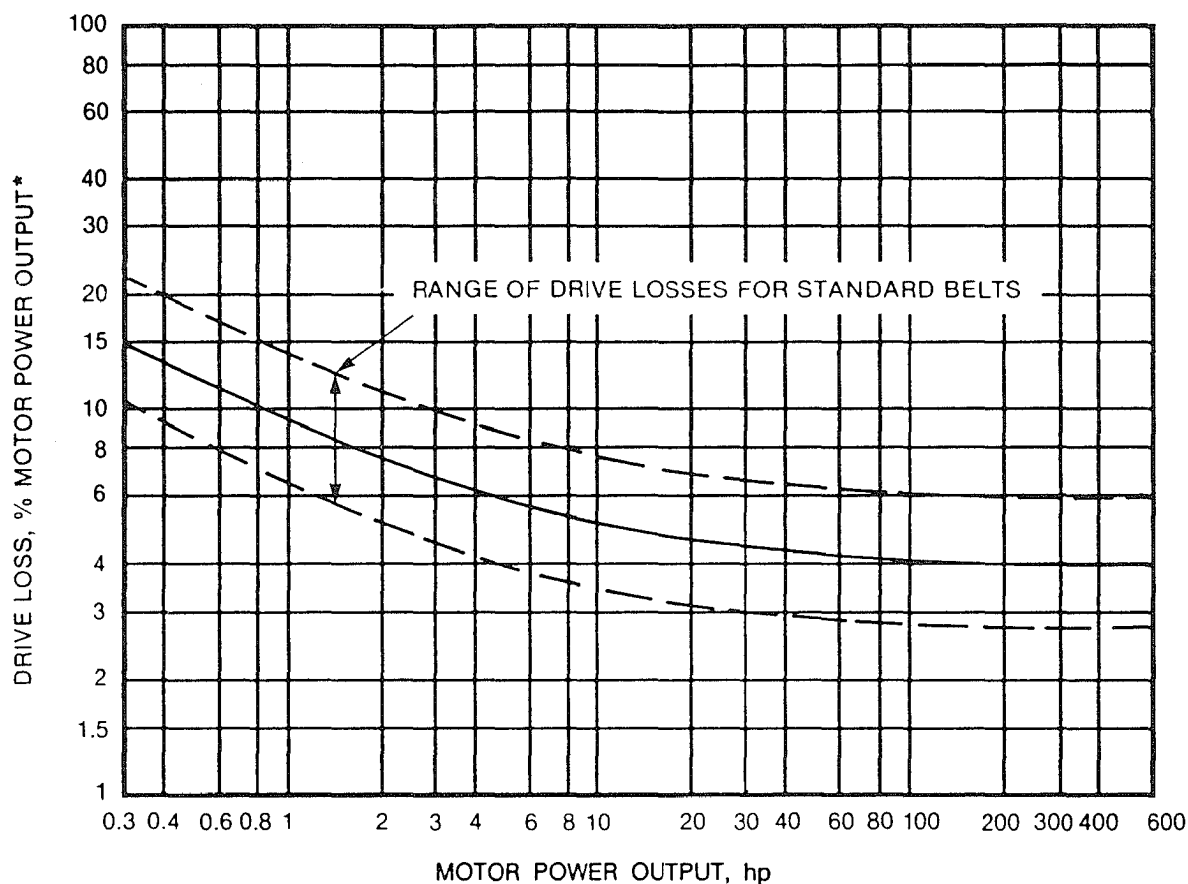


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DRIVE ARRANGEMENTS FOR
AXIAL FANS WITH OR WITHOUT
EVASE AND INLET BOX

DATE 11-96

FIGURE 6-5c



HIGHER BELT SPEEDS TEND TO HAVE HIGHER LOSSES
THAN LOWER BELT SPEEDS AT THE SAME HORSEPOWER

*Drive losses are based on the conventional V-belt which has been the "work horse" of the drive industry for several decades.

EXAMPLE

- Motor power output, H_{mo} , is determined to be 13.3 hp
- The belts are the standard type and just warm to the touch immediately after shutdown
- From chart, drive loss = 5.1%
- Drive loss, H_L = 0.051×13.3
= 0.7 hp
- Fan power input, H = $13.3 - 0.7$
= 12.6 hp.

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*ESTIMATED BELT
DRIVE LOSS*

DATE 5-92

FIGURE 6-6

material and may not be acceptable on material handling fans. Two types of dampers are available:

- *Outlet Dampers* mount on the fan outlet to add resistance to the system when partially closed. These are available with both parallel and opposed blades. Selection depends on the degree of control required (opposed blade dampers will control the flow more evenly throughout the entire range from wide open to closed).
- *Inlet Dampers* mount on the fan inlet to pre-spin air into the impeller. This reduces fan output and lowers operating horsepower. Because of the power savings, inlet dampers should be considered when the fan will operate for long periods at reduced capacities.

Variable pitch blades are available with some axial-type fans. The fan impellers are designed to allow manual or automatic changes to the blade pitch. "Adjustable" impellers have a blade pitch that can be manually changed when the fan is not running. "Variable" impellers include devices to allow the blade pitch to be changed pneumatically or hydraulically while the fan is operating.

A *Variable Frequency Drive* (VFD) may also be used to control flow. A VFD will control the fan speed, rather than varying the fan inlet flow conditions or the outlet area to change the fan's point of operation. This type of control varies both the flow rate and the fan static pressure.

The VFD control unit is connected in-line between the electric power source and the fan motor. It is used to vary the voltage and frequency of the power input to the motor. The motor speed will vary linearly with the line frequency. Most VFD applications use a direct drive arrangement; however, belt drives are occasionally used.

For a typical system with fixed physical characteristics, the attainable points of operation will fall on the system curve. For example, Figure 6-10 shows points A1 and A2 on a system curve. These two points of operation can be attained with a VFD by adjusting it for speeds of RPM₁ or RPM₂. This will result in fan curve PQ₁ or PQ₂, respectively.

VFDs do have disadvantages. They may have a low speed limitation. Most AC motors are designed to operate at their nameplate speeds. If a VFD is used to run a motor well below its nominal speed, the motor's efficiency will be reduced and losses will increase. This can increase motor heating and may cause damage.

The VFD can cause harmonic distortion in the electrical input lines from the power source. This may affect other electrical equipment on the same power system. Such distortion can be reduced with the addition of isolation transformers or line inductors.

To properly apply a VFD, the equipment supplier needs to know about its intended usage, about the building's power supply and about other electrical equipment in use. In general,

for applications where the minimum system air flow is 80% or more of the maximum system air flow, the VFD's losses and higher initial cost may make use of the inlet damper a better choice for flow control.

An advantage of the VFD or the Variable Pitch Blade over the dampers is often a dramatic power and noise reduction. However, these accessories usually require additional controlling equipment. An advantage of dampers is their relatively simple installation and use and their lower initial costs.

6.3.2 Rating Tables: Fan size and operating RPM and Power usually are obtained from a rating table based on required air flow and pressure. Tables are based on FTP or FSP:

$$\text{Fan TP} = (\text{SP}_{\text{outlet}} + \text{VP}_{\text{outlet}}) - (\text{SP}_{\text{inlet}} + \text{VP}_{\text{inlet}}) \quad [6.2]$$

$$\text{Fan SP} = \text{SP}_{\text{outlet}} - \text{SP}_{\text{inlet}} - \text{VP}_{\text{inlet}} \quad [6.3]$$

Fan Rating Tables are based on requirements for air at standard conditions (0.075 lbm/ft³). If other than standard conditions exist, the actual pressure must be converted to standard conditions. See Section 6.3.8, "Selection at Air Densities Other Than Standard."

The most common form of table is a "multi-rating table" (see Table 6-1) which shows a range of capacities for a particular fan size. For a given pressure, the highest mechanical efficiency usually will be in the middle third of the "CFM" column. Some manufacturers show the rating of maximum efficiency for each pressure by underscoring or similar indicator. In the absence of such a guide, the design engineer must calculate the efficiency from the efficiency equation

$$\eta = \frac{Q \times \text{FTP}}{\text{CF} \times \text{PWR}} = \frac{Q \times (\text{FSP} + \text{VP}_{\text{outlet}})}{\text{CF} \times \text{PWR}} \quad [6.4]$$

where:

η = Mechanical efficiency

Q = Volumetric flow rate, cfm

FTP = Fan total pressure, "wg

FSP = Fan Static Pressure, "wg

PWR = Power requirement, hp

CF = Conversion Coefficient, 6362

Even with a multi-rating table, it is usually necessary to interpolate in order to select fan RPM and BHP for the exact conditions desired. In many cases a double interpolation will be necessary. Straight line interpolations throughout the multi-rating table will introduce negligible errors.

Certain types of fans may be offered in various Air Movement and Control Association^(6.3) performance classes identified as I through IV. A fan designated as meeting the requirements of a particular class must be physically capable of operating at any point within the performance limits for that class. Performance limits for each class are established in terms of outlet velocity and static pressure. Multi-rating tables

TABLE 6-1. Example of Multi-Rating Table

CFM		OV	2"SP		4"SP		6"SP		8"SP		10"SP		12"SP		14"SP		16"SP		18"SP		20"SP		22"SP	
			RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
930	1000	843	0.57	1176	1.21	1434	1.93	1653	2.75	1846	3.64	2021	4.59	2184	5.62	2333	6.68	2475	7.81	2610	9.01	2738	10.2	
1116	1200	853	0.67	1183	1.35	1439	2.12	1656	2.98	1848	3.90	2022	4.89	2182	5.95	2333	7.07	2473	8.23	2606	9.45	2733	10.7	
1302	1400	866	0.77	1191	1.51	1445	2.33	1660	3.22	1852	4.20	2025	5.23	2183	6.31	2333	7.47	2474	8.68	2606	9.95	2731	11.2	
1488	1600	882	0.89	1201	1.69	1453	2.56	1668	3.50	1857	4.51	2030	5.59	2188	6.72	2337	7.92	2474	9.13	2606	10.4	2734	11.8	
1674	1800	899	1.01	1213	1.88	1463	2.81	1676	3.81	1863	4.86	2035	5.98	2194	7.16	2340	8.38	2479	9.67	2610	11.0	2735	12.4	
1860	2000	917	1.14	1227	2.09	1474	3.09	1685	4.13	1872	5.24	2040	6.39	2199	7.62	2344	8.89	2484	10.2	2613	11.6	2735	13.0	
2046	2200	937	1.29	1242	2.32	1484	3.37	1694	4.48	1879	5.63	2048	6.84	2206	8.13	2351	9.43	2487	10.8	2618	12.2	2741	13.6	
2232	2400	961	1.45	1257	2.56	1497	3.68	1704	4.85	1889	6.07	2056	7.33	2212	8.64	2357	10.0	2493	11.4	2622	12.8	2745	14.3	
2418	2600	984	1.62	1275	2.81	1513	4.02	1717	5.25	1900	6.53	2065	7.84	2222	9.22	2364	10.6	2501	12.1	2631	13.6	2750	15.1	
2790	3000	1038	2.02	1313	3.36	1543	4.73	1744	6.11	1924	7.52	2088	8.96	2241	10.4	2383	12.0	2517	13.5	2644	15.1	2766	16.7	
3162	3400	1099	2.50	1358	3.99	1580	5.52	1775	7.05	1952	8.60	2115	10.2	2265	11.8	2405	13.4	2538	15.1	2665	16.8	2783	18.5	
3534	3800	1164	3.07	1407	4.69	1620	6.37	1812	8.09	1984	9.79	2144	11.5	2290	13.3	2428	15.0	2562	16.8	2684	18.6	2803	20.5	
3906	4200	1232	3.75	1462	5.48	1665	7.31	1851	9.19	2018	11.0	2174	12.9	2320	14.8	2458	16.8	2587	18.7	2708	20.6	2825	22.5	
4278	4600	1306	4.56	1520	6.39	1717	8.38	1894	10.4	2058	12.4	2209	14.5	2355	16.5	2489	18.6	2614	20.6	2736	22.7	2852	24.8	
4650	5000	1380	5.49	1582	7.41	1770	9.53	1941	11.7	2100	13.9	2247	16.1	2390	18.3	2521	20.5	2645	22.7	2766	25.0	2883	27.3	
5022	5400	1457	6.56	1647	8.57	1827	10.8	1990	13.1	2146	15.5	2291	17.8	2428	20.2	2558	22.6	2681	25.0	2798	27.3			
5394	5800	1535	7.79	1719	9.93	1885	12.2	2045	14.7	2194	17.2	2334	19.7	2469	22.2	2594	24.7	2717	27.3	2830	29.8			

Performance shown is for fans with outlet ducts and with inlet ducts. BHP shown does not include belt drive losses.

usually will be shaded to indicate the selection zones for various classes or will state the maximum operating RPM. This can be useful in selecting equipment, but class definition is only based on performance and will not indicate quality of construction.

Capacity tables which attempt to show the ratings for a whole series of homologous fans on one sheet cannot be used accurately unless the desired rating happens to be listed on the chart. Interpolation is practically impossible since usually only one point of the fan curve for a given speed is defined in such a table.

Today, most fan manufacturers have "electronic catalogs" available. These catalogs are computer programs which can be used to calculate the correct fan speed and horsepower based on input data such as desired flow rate and fan static pressure or fan total pressure. Some electronic catalogs include estimates of the affects of various fan accessories such as dampers and inlet boxes.

6.3.3 Point of Operation: Fans are usually selected for operation at some fixed condition or single "Point of Operation." Both the fan and the system have variable performance characteristics which can be represented graphically as curves depicting an array of operating points. The actual "point of operation" will be the one single point at the intersection of the fan curve and the system curve.

Fan Performance Curves: Certain fan performance variables are usually related to volumetric flow rate in graphic form to represent a fan performance curve. Figure 6-7 is a typical representation where Pressure (P) and power requirement (PWR) are plotted against flow rate (Q). Other variables also may be included and more detailed curves representing various fan designs are provided in Figure 6-4. Pressure can be either FSP or FTP. This depends on the manufacturer's method of rating.

It should be noted that a fan performance curve is always specific to a fan of given size operating at a single rotation rate (RPM). Even with size and rotation rate fixed, it should be obvious that pressure and power requirements vary over a range of flow rates.

System Requirement Curves: The duct system pressure also varies with volumetric flow rate. Figure 6-8 illustrates the variation of pressure (P) with flow rate (Q) for three different situations. The turbulent flow condition is representative of

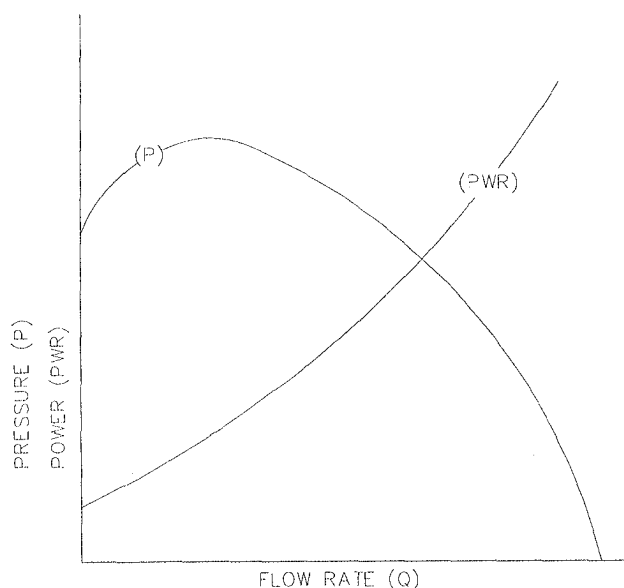


FIGURE 6-7. Typical fan performance curve

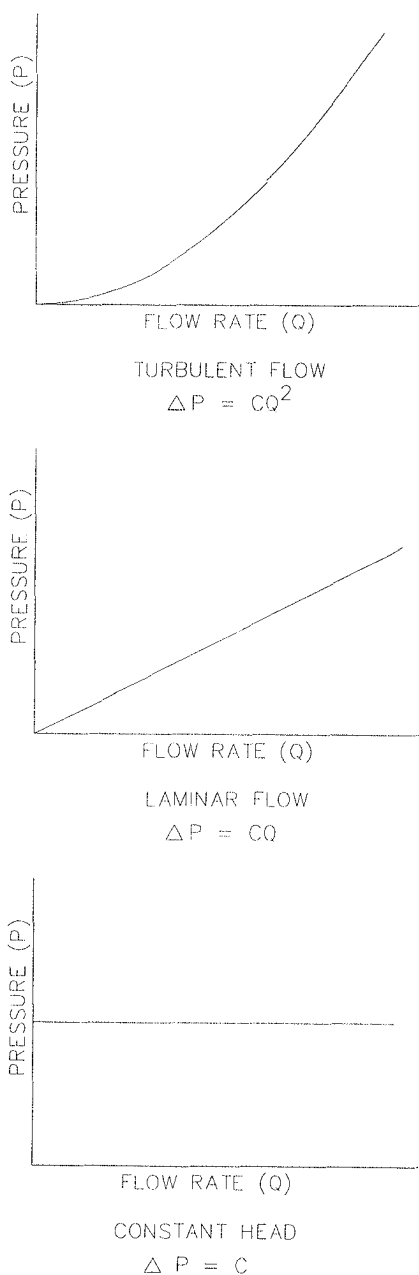


FIGURE 6-8. System requirement curves

duct losses and is most common. In this case, the pressure loss varies as the square of the flow rate. The laminar flow condition is representative of the flow through low velocity filter media. Some wet collector designs operate at or close to a constant loss situation.

The overall system curve results from the combined effects of the individual components.

6.3.4 Matching Fan Performance and System

Requirement: A desired point of operation results from the process of designing a duct system and selecting a fan. Con-

sidering the system requirement or fan performance curves individually, this desired point of operation has no special status relative to any other point of operation on the individual curve. Figure 6-9 depicts the four general conditions which can result from the system design fan selection process.

There are a number of reasons why the system design, fan selection, fabrication, and installation process can result in operation at some point other than design. When this occurs, it may become necessary to alter the system physically which will change the system requirement curve and/or cause a change in the fan performance curve. Because the fan performance curve is not only peculiar to a given fan but specific to a given rotation rate (RPM), a change of rotation rate can be relatively simple if a belt drive arrangement has been used. The "Fan Laws" are useful when changes of fan performance are required.

6.3.5 Fan Laws: Fan laws relate the performance variables for any homologous series of fans. A homologous series represents a range of sizes where all dimensional variables between sizes are proportional. The performance variables involved are fan size (SIZE), rotation rate (RPM), gas density (ρ), flow rate (Q), pressure (P), power requirement (PWR), and efficiency (η). Pressure (P) may be represented by total pressure (TP), static pressure (SP), velocity pressure (VP), fan static pressure (FSP), or fan total pressure (FTP).

At the same relative point of operation on any two performance curves in this homologous series, the efficiencies will be equal. The fan laws are mathematical expressions of these facts and establish the inter-relationship of the other variables. They predict the effect of changing size, speed, or gas density on capacity, pressure, and power requirement as follows:

$$Q_2 = Q_1 \left(\frac{\text{SIZE}_2}{\text{SIZE}_1} \right)^3 \left(\frac{\text{RPM}_2}{\text{RPM}_1} \right) \quad [6.5]$$

$$P_2 = P_1 \left(\frac{\text{SIZE}_2}{\text{SIZE}_1} \right)^2 \left(\frac{\text{RPM}_2}{\text{RPM}_1} \right)^2 \left(\frac{\rho_2}{\rho_1} \right) \quad [6.6]$$

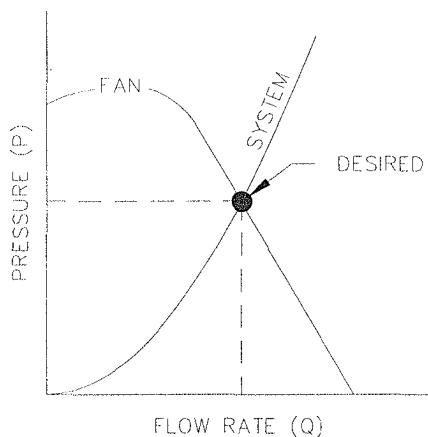
$$\text{PWR}_2 = \text{PWR}_1 \left(\frac{\text{SIZE}_2}{\text{SIZE}_1} \right)^5 \left(\frac{\text{RPM}_2}{\text{RPM}_1} \right)^3 \left(\frac{\rho_2}{\rho_1} \right) \quad [6.7]$$

As these expressions involve ratios of the variables, any convenient units may be employed so long as they are consistent. Size may be represented by any linear dimension since all must be proportional in homologous series. However, impeller diameter is the most commonly used dimension.

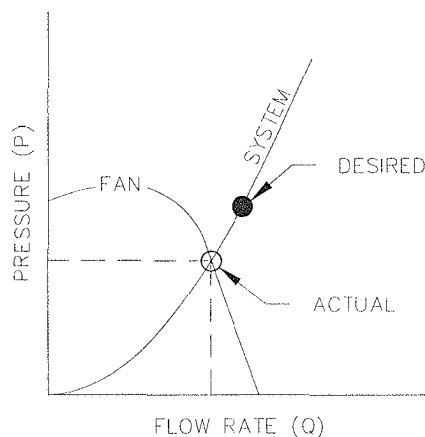
6.3.6 The Effect of Changing Rotation Rate or Gas

Density: In practice, these principles are normally applied to determine the effect of changing only one variable. Most often the fan laws are applied to a given fan size and may be expressed in the simplified versions which follow:

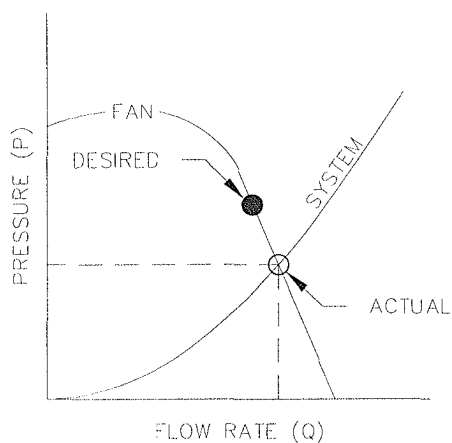
- For changes of rotation rate:



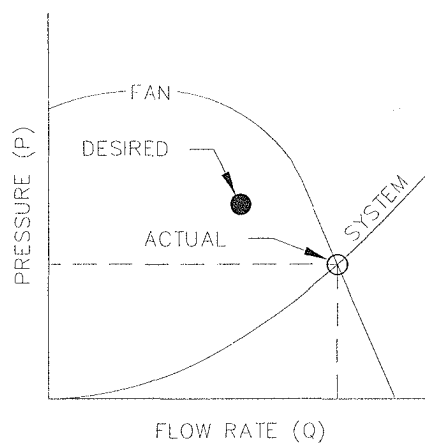
A. FAN AND SYSTEM MATCHED



B. WRONG FAN.



C. WRONG SYSTEM.



D. BOTH FAN AND SYSTEM WRONG

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*ACTUAL VERSUS DESIRED
POINT OF OPERATION*

DATE

1-88

FIGURE

6-9

Flow varies directly with rotation rate; pressure varies as the square of the rotation rate; and power varies as the cube of the rotation rate:

$$Q_2 = Q_1 \left(\frac{RPM_2}{RPM_1} \right) \quad [6.8]$$

$$P_2 = P_1 \left(\frac{RPM_2}{RPM_1} \right)^2 \quad [6.9]$$

$$PWR_2 = PWR_1 \left(\frac{RPM_2}{RPM_1} \right)^3 \quad [6.10]$$

- For changes of gas density:

Flow is not affected by a change in density; pressure and power vary directly with density:

$$Q_2 = Q_1 \quad [6.11]$$

$$P_2 = P_1 \left(\frac{\rho_2}{\rho_1} \right) \quad [6.12]$$

$$PWR_2 = PWR_1 \left(\frac{\rho_2}{\rho_1} \right) \quad [6.13]$$

6.3.7 Limitations on the Use of Fan Laws: These expressions are equations which rely on the fact that the performance curves are homologous and that the ratios are for the *same relative points of rating* on each curve. Care must be exercised to apply the laws between the same relative points of rating.

Figure 6-10 contains a typical representation of two homologous fan performance curves, PQ_1 and PQ_2 . These could

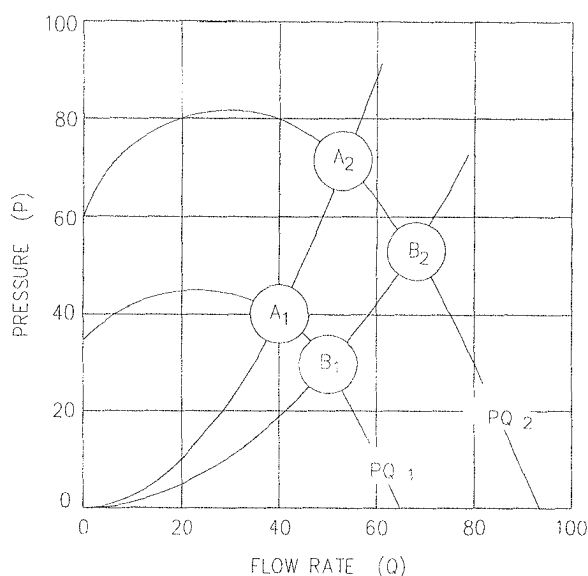


FIGURE 6-10. Homologous performance curves

be the performances resulting from two different rotation rates, RPM_1 and RPM_2 . Assuming a point of rating indicated as A_1 on PQ_1 there is only one location on PQ_2 with the same relative point of rating and that is at A_2 . The A_1 and A_2 points of rating are related by the expression

$$P_{A_2} = P_{A_1} \left(\frac{Q_{A_2}}{Q_{A_1}} \right)^2 \quad [6.14]$$

The equation can be used to identify every other point that would have the same relative point of rating as A_1 and A_2 . The line passing through " A_2, A_1 " and the origin locates all conditions with the same relative points of rating. These lines are more often called "system lines" or "system curves." As discussed in Section 6.3.3, there are a number of exceptions to the condition where system pressure varies as the square of flow rate. These lines representing the same relative points of rating are "system lines" or "system curves" for turbulent flow conditions only.

Where turbulent flow conditions apply, it must be understood that the system curves or lines of relative points of rating represent a system having fixed physical characteristics. For example the, " $B_2 - B_1$ " line defines another system which has lower resistance to flow than the " $A_2 - A_1$ " system.

Special care must be exercised when applying the fan laws in the following cases:

1. Where any component of the system does not follow the "pressure varies as the square of the flow rate" rule.
2. Where the system has been physically altered or for any other reason operates on a different system line.

6.3.8 Fan Selection at Air Density Other Than Standard:

Standard: As discussed in Section 6.3.6, fan performance is affected by changes in gas density. Variations in density due to normal fluctuations of ambient pressure, temperature, and humidity are small and need not be considered. Where temperature, humidity, elevation, pressure, gas composition or a combination of two or more cause density to vary by more than 5% from the standard 0.075 lbm/ft³, corrections should be employed.

Rating tables and performance curves as published by fan manufacturers are based on standard air. Performance variables are always related to conditions at the fan inlet. Fan characteristics are such that volumetric flow rate (Q) is unaffected but pressure (P) and power (PWR) vary directly with changes in gas density. Therefore, the selection process requires that rating tables are entered with *actual* volumetric flow rate but with a corrected or equivalent pressure.

The equivalent pressure is that pressure corresponding to standard density and is determined from Equation 6.12 as follows:

$$P_e = P_a \left(\frac{0.075}{\rho_a} \right)$$

where:

P_e = Equivalent Pressure

P_a = Actual Pressure

ρ_a = Actual density, lbm/ft³

The pressures (P_e and P_a) can be either Fan Static Pressure or Fan Total Pressure in order to conform with the manufacturer's rating method.

The fan selected in this manner is to be operated at the rotation rate indicated in the rating table and actual volumetric flow rate is that indicated by the table. However, the pressure developed is not that indicated in the table but is the actual value. Likewise, the power requirement is not that of the table as it also varies directly with density. The actual power requirement can be determined from Equation 6.13 as follows:

$$PWR_a = PWR_t \left(\frac{\rho_a}{0.075} \right)$$

where:

PWR_a = Actual Power Requirement

PWR_t = Power Requirement in Rating Table

ρ_a = Actual Density, lbm/ft³

Fan selection at non-standard density requires knowledge of the actual volumetric flow rate at the fan inlet, the actual pressure requirement (either FSP or FTP, depending on the rating table used) and the density of the gas at the fan inlet. The determination of these variables requires that the system design procedure consider the effect of density as discussed in Chapter 5.

EXAMPLE

Consider the system illustrated in Figure 6-11 where the heater causes a change in volumetric flow rate and density. For simplicity, assume the heater has no resistance to flow and that the sum of friction losses will equal FSP. Using the Multi-Rating Table, Table 6-1, select the rotation rate and determine power requirements for the optional fan locations ahead of or behind the heater.

Location 1: *Fan ahead of the heater* (side "A" to "B" in Figure 6-11).

Step 1. Determine actual FSP

$$\begin{aligned} \text{FSP} &= 1 \text{ "wg} + 3 \text{ "wg} \\ &= 4 \text{ "wg at } 0.075 \text{ lbm/ft}^3. \end{aligned}$$

Step 2a. Density at fan inlet is standard. Therefore, enter rating table with actual volumetric flow rate at fan inlet, 1000 acfm, and FSP of 4 "wg.

b. Interpolation from Table 6-1 results in:

$$\text{RPM} = 1182 \text{ rpm}$$

$$\text{PWR} = 1.32 \text{ bhp}$$

Step 3. The fan should be operated at 1182 rpm and actual power requirement will be 1.32 bhp.

Location 2: *Fan behind the heater* (side "B" to "C" in Figure 6-11).

Step 1. Determine actual FSP

$$\begin{aligned} \text{FSP} &= 1 \text{ "wg} + 3 \text{ "wg (as in explanation)} \\ &= 4 \text{ "wg at } 0.0375 \text{ lbm/ft}^3 \end{aligned}$$

Step 2a. Density at fan inlet is not standard and a pressure correction must be made (using Equation 6.12) to determine equivalent FSP.

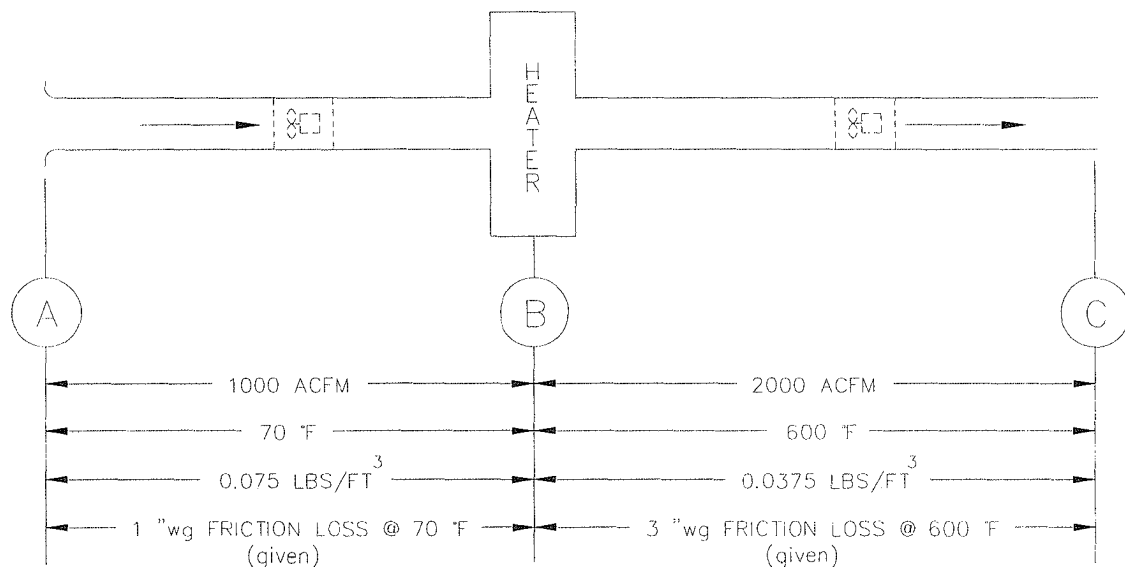


FIGURE 6-11. In-duct heater

$$FSP_e = FSP_a \left(\frac{0.075}{\rho_a} \right) = 4" \text{wg} \left(\frac{0.075}{0.0375} \right) = 8" \text{wg}$$

Now, enter rating table with actual volumetric flow rate at fan inlet, 2000 acfm, and equivalent FSP, 8"wg.

- b. Interpolation from Table 6-1 results in:

$$RPM = 1692 \text{ rpm}$$

$$PWR = 4.39 \text{ bhp}$$

- Step 3a. The fan should be operated at 1692 rpm, but actual power requirements will be affected by the density and can be determined by using Equation 6.13.

$$PWR_a = PWR_t \left(\frac{\rho_a}{0.075} \right) = 4.39 \left(\frac{0.0375}{0.075} \right)$$

$$= 2.2 \text{ bhp}$$

- b. It should also be noted that a measurement of FSP will result in the value of 4"wg (actual) and not the equivalent value of 8"wg.

It will be noted that, regardless of location, the fan will handle the same mass flow rate. Also, the actual resistance to flow is not affected by fan location. It may appear then that there is an error responsible for the differing power requirements of 1.32 bhp versus 2.2 bhp. In fact, the fan must work harder at the lower density to move the same mass flow rate. This additional work results in a higher temperature rise in the air from fan inlet to outlet. A fan located ahead of the heater will require less power and may be quieter due to the lower rotational speed.

6.3.9 Explosive or Flammable Materials: When conveying explosive or flammable materials, it is important to recognize the potential for ignition of the gas stream. This may be from airborne material striking the impeller or by the physical movement of the impeller into the fan casing. AMCA^(6.1) and other associations offer guidelines for both the manufacturer and the user on ways to minimize this danger. These involve more permanent attachment of the impeller to the shaft and bearings and the use of buffer plates or spark-resistant alloy construction. Because no single type of construction fits all applications, it is imperative that both the manufacturer and the user are aware of the dangers involved and agree on the type of construction and degree of protection that is being proposed.

NOTE: or many years, aluminum alloy impellers have been specified to minimize sparking if the impeller were to contact other steel parts. This is still accepted, but tests by the U. S. Bureau of Mines^(6.2) and others have demonstrated that impact of aluminum with rusty steel

creates a "Thermite" reaction and thus possible ignition hazards. Special care must be taken when aluminum alloys are used in the presence of steel.

6.4 FAN INSTALLATION AND MAINTENANCE

Fan rating tests for flow rate, static pressure, and power requirements are conducted under ideal conditions which include uniform straight air flow at the fan inlet and outlet. However, if in practice duct connections to the fan cause non-uniform air flow, fan performance and operating efficiency will be affected. Location and installation of the fan must consider the location of these duct components to minimize losses. If adverse connections must be used, appropriate compensation must be made in the system calculations. Once the system is installed and operating, routine inspection and maintenance will be required if the system is to continue to operate at original design levels.

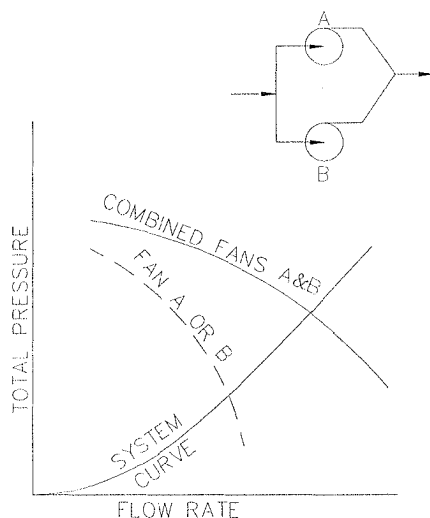
6.4.1 System Effect: System effect is defined as the estimated loss in fan performance from this non-uniform air flow. Figure 6-14 illustrates deficient fan system performance. The system pressure losses have been determined accurately and a suitable fan selected for operation at Point 1. However, no allowance has been made for the effect of the system connections on fan performance. The point of intersection between the resulting fan performance curve and the actual system curve is Point 3. The resulting flow rate will, therefore, be deficient by the difference from 1 to 3. To compensate for this system effect, it will be necessary to add a "system effect coefficient" to the calculated system pressure. This will be equal to the pressure difference between Points 1 and 2 and will have to be added to the calculated system pressure losses. The fan then will be selected for this higher pressure (Point 2) but will operate at Point 1 due to loss in performance from system effects.

One commonly neglected system effect is a duct elbow at the fan inlet. For example, consider the fan shown in Figure 6-16.

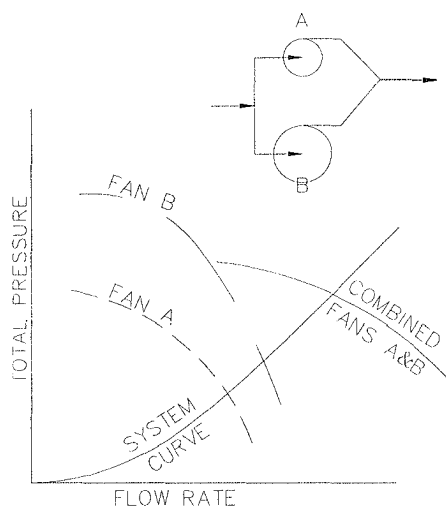
This fan has a four-piece 90° round duct elbow immediately in front of the inlet. There are no turning vanes inside the duct. The required flow rate is 5000 cfm and the system pressure losses are 8"wg at standard conditions (0.075 lb/ft³). Selecting a fan without the system effect, using Table 6-1, would result in a fan speed of 1987 rpm and power consumption of 13.02 hp.

With the elbow at the inlet, the air flow into the fan inlet will be degraded. Such a change in the air flow requires use of a system effect coefficient to select a fan that overcomes the degradation in performance. The system effect coefficient is used to determine a correction value, in inches water gauge, to be added to the system pressure losses.

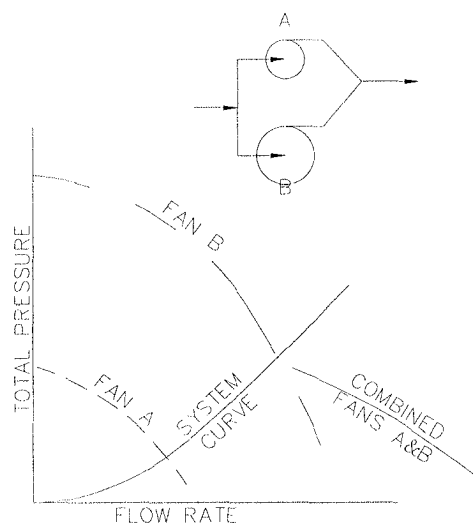
In this example, the duct diameter is 24" with a turning radius of 48". This is a radius-to-diameter (r/d) ratio of 2.0. In Figure 6-21, Item C, we find the system effect curve to use is



TWO IDENTICAL FANS
RECOMMENDED



TWO DIFFERENT FANS
SATISFACTORY



TWO DIFFERENT FANS
UNSATISFACTORY

WHEN SYSTEM CURVE DOES NOT CROSS COMBINED FAN CURVE, OR CROSSES PROJECTED COMBINED CURVE BEFORE FAN B, FAN B WILL HANDLE MORE AIR THAN FANS A AND B IN PARALLEL.

NOTES:

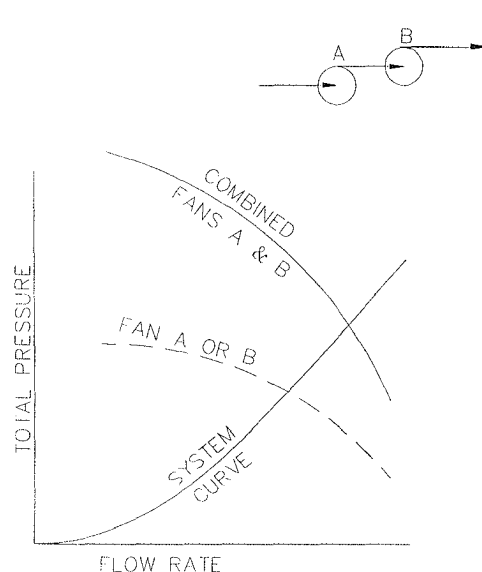
1. TO ESTABLISH COMBINED FAN CURVE, THE COMBINED AIR FLOW RATE, Q , IS THE SUM OF INDIVIDUAL FAN AIR FLOW RATES AT POINTS OF EQUAL PRESSURE
2. TO ESTABLISH SYSTEM CURVE, INCLUDE LOSSES IN INDIVIDUAL FAN CONNECTIONS.
3. SYSTEM CURVE MUST INTERSECT COMBINED FAN CURVE OR HIGHER PRESSURE FAN MAY HANDLE MORE AIR ALONE.

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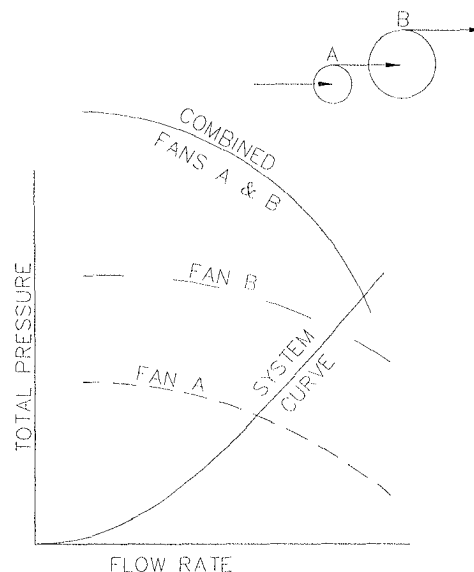
FANS
PARALLEL OPERATION

DATE 10-96

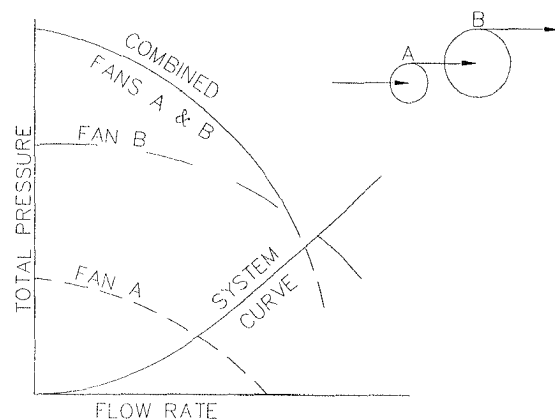
FIGURE 6-12



TWO IDENTICAL FANS
RECOMMENDED FOR BEST EFFICIENCY



TWO DIFFERENT FANS
SATISFACTORY



TWO DIFFERENT FANS
UNSATISFACTORY

NOTES:

1. TO ESTABLISH COMBINED FAN CURVE, THE COMBINED TOTAL PRESSURE IS THE SUM OF INDIVIDUAL FAN PRESSURES AT EQUAL AIR FLOW RATES, LESS THE PRESSURE LOSS IN THE FAN CONNECTIONS.
2. AIR FLOW RATE THROUGH EACH FAN WILL BE THE SAME, SINCE AIR IS CONSIDERED INCOMPRESSIBLE.
3. SYSTEM CURVE MUST INTERSECT COMBINED FAN CURVE OR LARGE FLOW RATE FAN MAY HANDLE MORE AIR ALONE.

WHEN SYSTEM CURVE DOES NOT INTERSECT COMBINED FAN CURVE, OR CROSSES PROJECTED COMBINED CURVE BEFORE FAN B CURVE, FAN B WILL MOVE MORE AIR THAN FAN A AND B IN SERIES.

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FIGURE

6-13

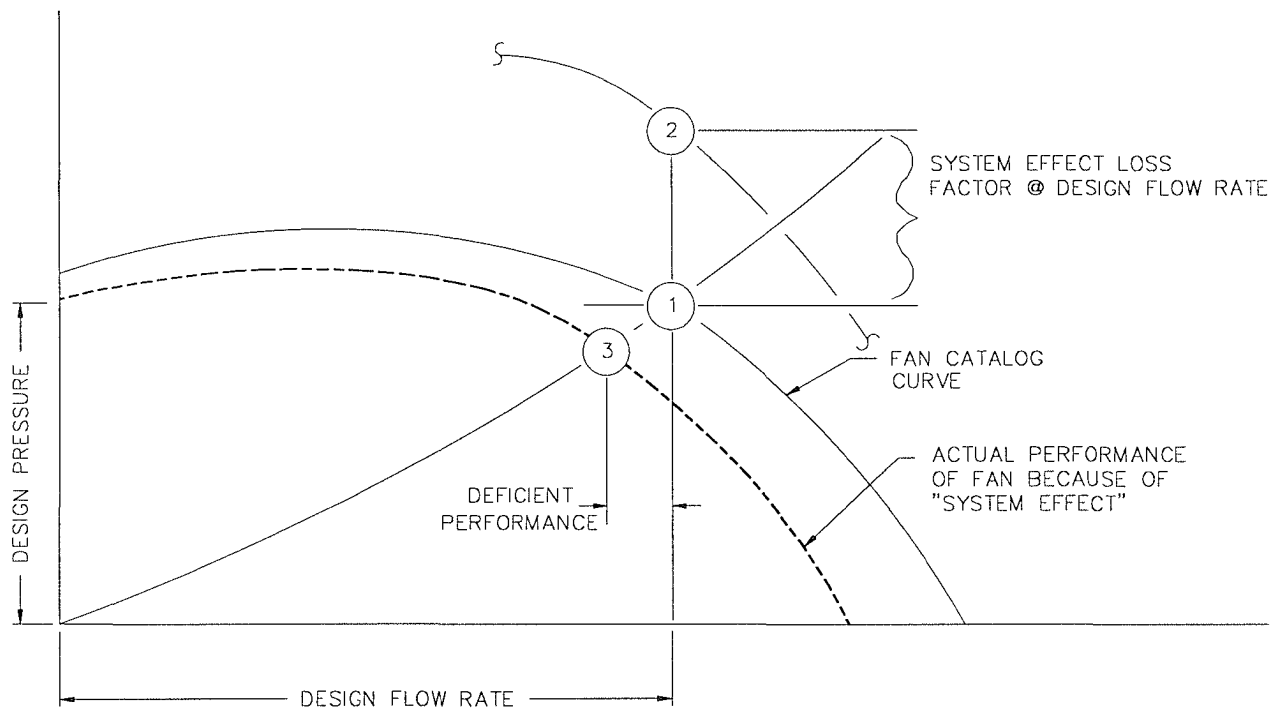


FIGURE 6-14. System effect factor

"R." To find the system effect correction value in inches water gauge, we use the fan inlet velocity with Figure 6-27. Since the duct area is 3.142 ft^2 , the velocity is 1592 fpm ($5000 \text{ cfm} \div 3.142 \text{ ft}^2 = 1592 \text{ fpm}$). From Figure 6-27 we get a correction value of 0.19 "wg. This 0.19" value is added to the fan static pressure when selecting the fan from the multi-rating table. Select the fan for a static pressure of 8.19 "wg. Interpolating in Table 6-1, we find a selection for 5000 cfm and 8.19 "wg at 2002 rpm and 13.25 hp. This selection for a fan with an elbow at the inlet will result in operation at 5000 cfm and 8 "wg drawing 13.25 hp.

Note: The system effect coefficient compensates for the affect on the fan of an irregular air stream. This system effect coefficient is taken in addition to the friction loss used to calculate the system loss (Figure 5-13.)

Figure 6-15 illustrates typical discharge conditions and the losses which may be anticipated. The magnitude of the change in system performance caused by elbows and other obstructions placed too close to a fan inlet or outlet can be estimated for the conditions shown on Figures 6-18 through 6-25.

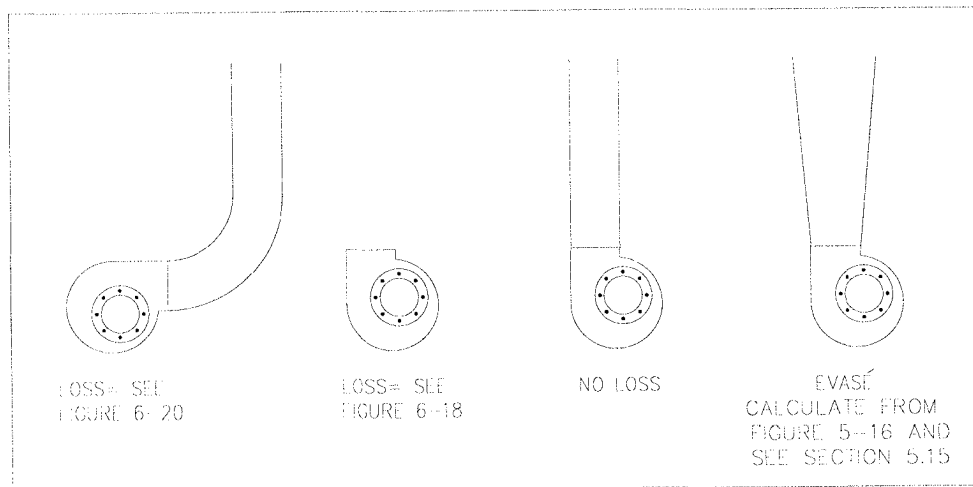


FIGURE 6-15. Fan discharge conditions

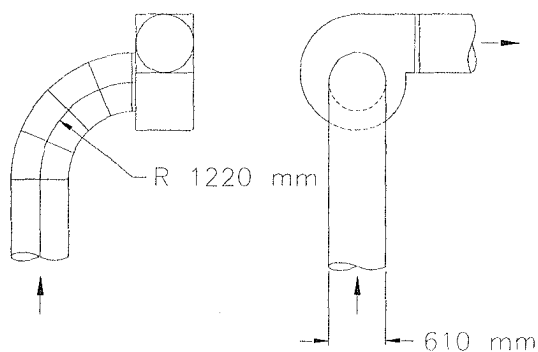


FIGURE 6-16. Inlet elbow

Addition to system static pressure is given by reference to lettered curves in all but Figure 6-23. The additional static pressure, in "wg, is determined by obtaining the appropriate system effect coefficient from Figure 6-26 or 6-27 and multiplying it by the fan inlet or discharge velocity pressure.

A vortex or spin of the air stream entering the fan inlet may be created by non-uniform flow conditions as illustrated in Figure 6-24. These conditions may be caused by a poor inlet box, multiple elbows or entries near the inlet, or by other spin-producing conditions. Since the variations resulting in inlet spin are many, no System Effect Coefficients are tabulated. Where a vortex or inlet spin cannot be avoided or is discovered at an existing fan inlet, the use of turning vanes, splitter sheets, or egg crate straighteners will reduce the effect.

6.4.2 Inspection and Maintenance: Material accumulation or abrasive wear on an impeller can cause a fan to "go out of balance." This unbalance will cause fan vibration. This may result in damage to or failure of the fan impeller, housing, bearings, or pedestal. Periodic cleaning and rebalancing of fans operating in air streams handling high material concentrations is recommended.

Regular observation of fan vibration levels can detect problems before they develop to a damaging amplitude (see Figure 6-17). Modern maintenance equipment permits the inspector to record vibration spectra. Review of changes in these spectra taken over time can indicate specific areas of developing problems with bearings, balance, belts or motors. Electronic or computerized vibration monitors are available to mount on fans used in critical operations. These devices can be set up with automatic alarm functions and/or to provide continuous information about a unit's vibration level.

It is not uncommon, during fan installation or motor/starter maintenance, for the fan impeller rotation direction to be

inadvertently reversed. Since fans do move a fraction of their rated capacity when running backward, incorrect rotation often goes unnoticed in spite of less effective performance of the exhaust system.

Scheduled inspection of fans is recommended. Items checked should include:

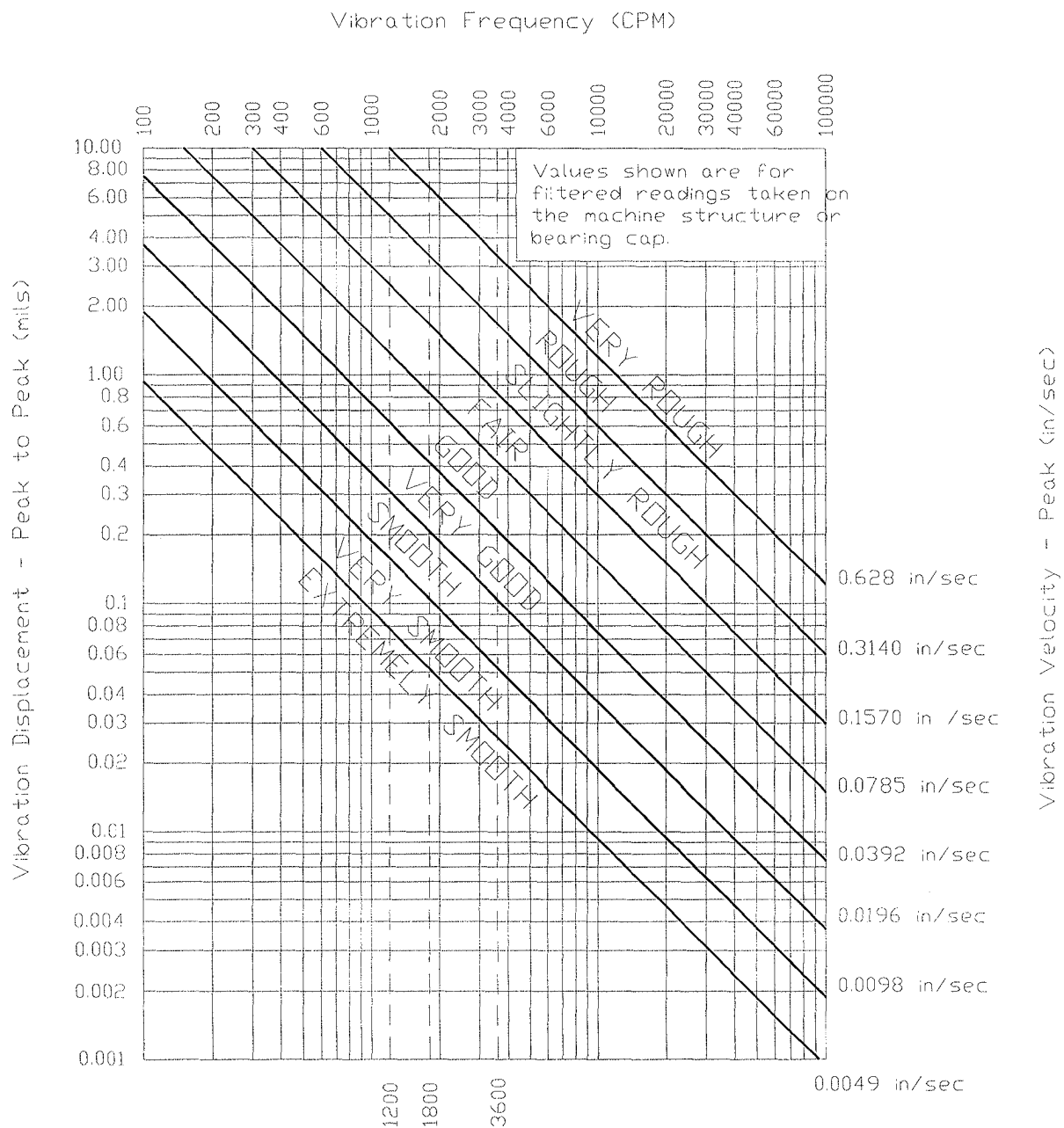
1. Bearings for proper operating temperature (lubricate them on the manufacturer's recommended schedule).
2. Excessive vibration of bearings or housing.
3. Belt drives for proper tension and minimum wear.
4. Correct coupling or belt alignment.
5. Fan impeller for proper alignment and rotation.
6. Impeller free from excess wear or material accumulation.
7. Tight fan hold-down bolts.
8. Tight fan impeller set screws or bushings.
9. Proper installation of safety guards.

Standard lockout/tagout procedures should be observed when servicing fan equipment or its associated duct. The electrical supply must be shut off and locked out at a disconnect near the fan. When opening access doors or reaching into the fan inlet or outlet, the fan must be mechanically locked out by blocking the impeller from rotating. A warning tag should be used when blocking a fan. Do not open an access door while the fan is operating or coasting down.

BESURE to remove any inserted obstructions used to block impeller rotation when servicing is complete.

REFERENCES

- 6.1. Air Movement and Control Association, Inc.: AMCA Publication 201-90, Fans and Systems. AMCA, Arlington Heights, IL (1990).
- 6.2. Gibson, N.; Lloyd, F.C.; Perry, G.R.: Fire Hazards in Chemical Plants from Friction Sparks Involving the Thermite Reaction. Symposium Series No. 25. Instn. Chem. Engrs., London (1968).
- 6.3. Air Movement and Control Association, Inc.: AMCA Publication 99-86, Standards Handbook. AMCA, Arlington Heights, IL (1986).
- 6.4. American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc.: 1993 ASHRAE Handbook, Fundamentals Volume. ASHRAE, Atlanta, GA (1993).



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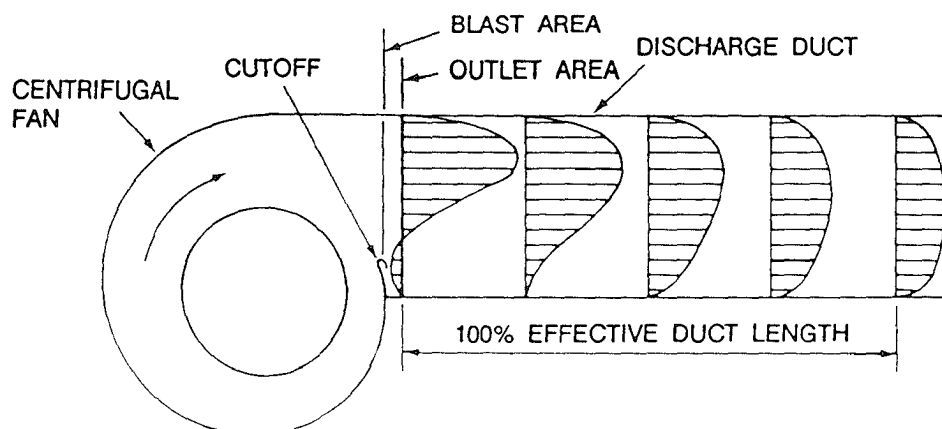
MACHINERY VIBRATION SEVERITY CHART

DATE

10-96

FIGURE

6-17



TO CALCULATE 100% EFFECTIVE DUCT LENGTH, ASSUME A MINIMUM OF 2-1/2 DUCT DIAMETERS FOR 2500 FPM OR LESS. ADD 1 DUCT DIAMETER FOR EACH ADDITIONAL 1000 FPM.

EXAMPLE: 5000 FPM = 5 EQUIVALENT DUCT DIAMETERS. IF THE DUCT IS RECTANGULAR WITH SIDE DIMENSIONS a AND b , THE EQUIVALENT DUCT DIAMETER IS EQUAL TO $(4ab/\pi)^{0.5}$

	No Duct	12% Effective Duct	25% Effective Duct	50% Effective Duct	100% Effective Duct
Pressure Recovery	0%	50%	80%	90%	100%
<u>Blast Area</u> <u>Outlet Area</u>	System Effect Curve				
0.4	P	R-S	U	W	—
0.5	P	R-S	U	W	—
0.6	R-S	S-T	U-V	W-X	—
0.7	S	U	W-X	—	—
0.8	T-U	V-W	X	—	—
0.9	V-W	W-X	—	—	—
1.0	—	—	—	—	—

DETERMINE SEF BY USING FIGURE 6-26 OR 6-27

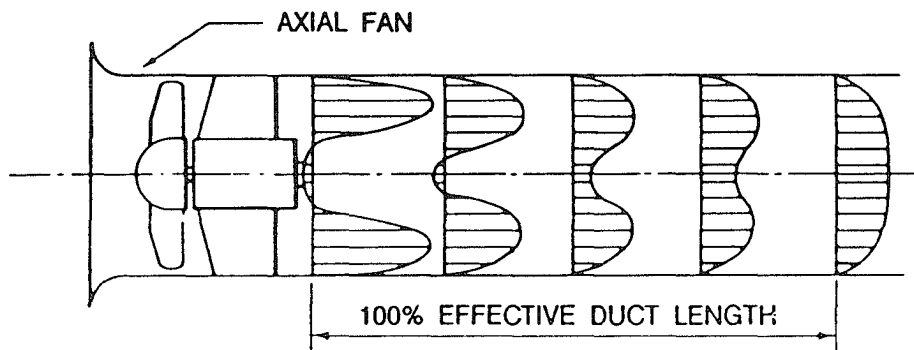
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SYSTEM EFFECT CURVES
FOR OUTLET DUCTS—
CENTRIFUGAL FANS

DATE 5-92

FIGURE 6-18



TO CALCULATE 100% EFFECTIVE DUCT LENGTH, ASSUME A MINIMUM OF 2-1/2 DUCT DIAMETERS FOR 2500 FPM OR LESS. ADD 1 DUCT DIAMETER FOR EACH ADDITIONAL 1000 FPM.

EXAMPLE: 5000 FPM = 5 EQUIVALENT DUCT DIAMETERS

	No Duct	12% Effective Duct	25% Effective Duct	50% Effective Duct	100% Effective Duct
Tubeaxial Fan	---	---	---	---	---
Vaneaxial Fan	U	V	W	---	---

DETERMINE SEF BY USING FIGURE 6-26 OR 6-27

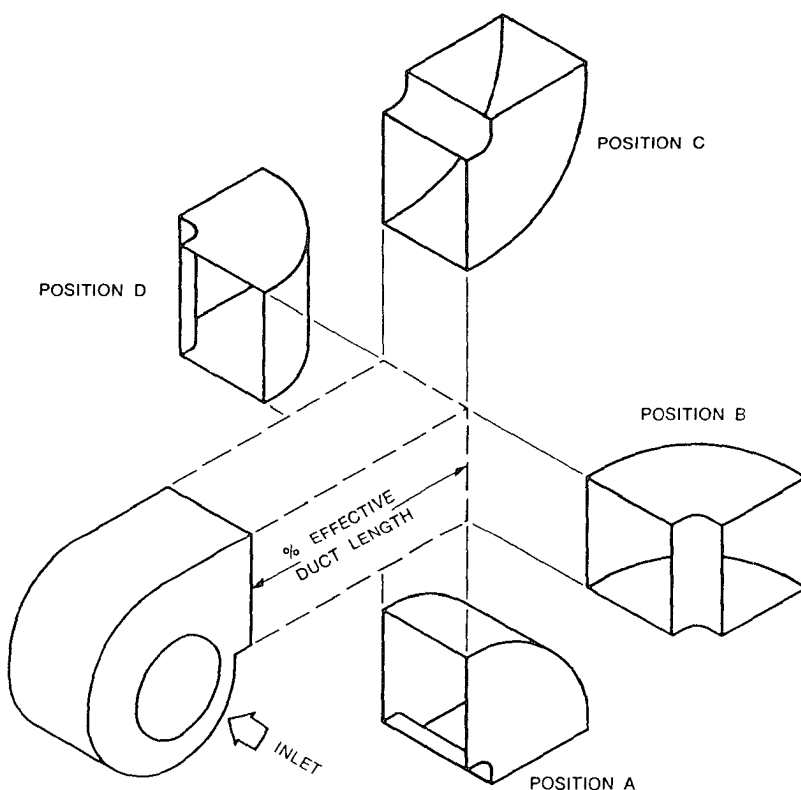
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*SYSTEM EFFECT CURVES
FOR OUTLET DUCTS—
AXIAL FANS*

DATE 5-92

FIGURE 6-19



DETERMINE SEF BY USING FIGURES 6-26 AND 6-18

For **DWDI** fans determine SEF using the curve for SWSI fans. Then apply the appropriate multiplier from the tabulation below

MULTIPLIERS FOR DWDI FANS

- ELBOW POSITION A = $\Delta P \times 1.00$
- ELBOW POSITION B = $\Delta P \times 1.25$
- ELBOW POSITION C = $\Delta P \times 1.00$
- ELBOW POSITION D = $\Delta P \times 0.85$

Blast Area Outlet Area	Outlet Elbow Position	No Outlet Duct	12% Effective Duct	25% Effective Duct	50% Effective Duct	100% Effective Duct
0.4	A B C D	N M-N L-M L-M	O N M M	P-Q O-P N N	S R-S Q Q	NO SYSTEM EFFECT FACTOR
0.5	A B C D	O-P N-O M-N M-N	P-Q O-P N N	R Q O-P O-P	T S-T R-S R-S	
0.6	A B C D	Q P N-O N-O	Q-R Q O O	S R Q Q	U T S S	
0.7	A B C D	R-S Q-R P P	S R-S Q Q	T S-T R-S R-S	V U-V T T	
0.8	A B C D	S R-S Q-R Q-R	S-T S R R	T-U T S S	W V U-V U-V	
0.9	A B C D	T S R R	T-U S-T S S	U-V T-U S-T S-T	W V V V	
1.0	A B C D	T S-T R-S R-S	T-U T S S	U-V U T T	W V V V	

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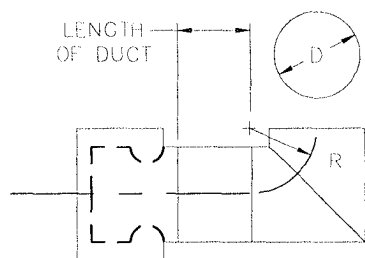
*SYSTEM EFFECT CURVES
FOR OUTLET ELBOWS
ON CENTRIFUGAL FANS*

DATE

5-92

FIGURE

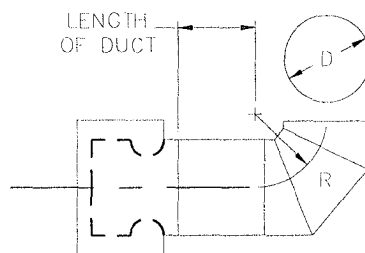
6-20



SYSTEM EFFECT FACTORS

R/D	NO DUCT	2D DUCT	5D DUCT
—	N	P	R-S

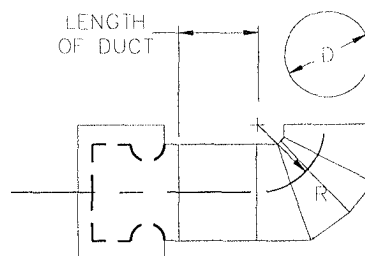
A. TWO-PIECE MITERED 90° ROUND SECTION ELBOW --- NOT VANED



SYSTEM EFFECT FACTORS

R/D	NO DUCT	2D DUCT	5D DUCT
0.5	O	Q	S
0.75	Q	R-S	T-U
1.0	R	S-T	U-V
2.0	R-S	T	U-V
3.0	S	T-U	V

B. THREE-PIECE MITERED 90° ROUND SECTION ELBOW -- NOT VANED



SYSTEM EFFECT FACTORS

R/D	NO DUCT	2D DUCT	5D DUCT
0.5	P-Q	R-S	T
0.75	Q-R	S	U
1.0	R	S-T	U-V
2.0	R-S	T	U-V
3.0	S-T	U	V-W

C. FOUR OR MORE PIECE MITERED 90° ROUND SECTION ELBOW -- NOT VANED

D=Diameter of the inlet collar.

The inside area of the square duct (H X H) should be equal to the inside area of the fan inlet collar.

+The maximum permissible angle of any converging element of the transition is 15°, and for a diverging element 7°.

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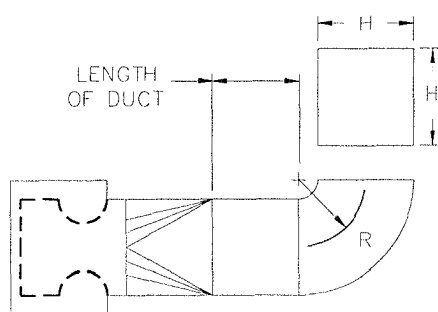
SYSTEM EFFECT CURVES FOR
VARIOUS MITERED ELBOWS
WITHOUT TURNING VANES

DATE

1-98

FIGURE

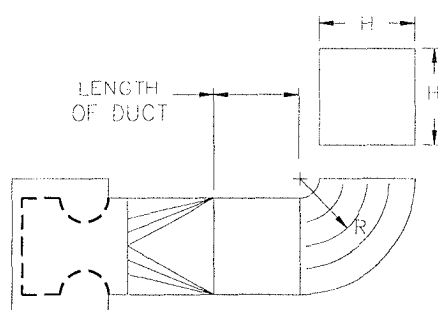
6-21



SYSTEM EFFECT FACTORS

R/D	NO DUCT	2D DUCT	5D DUCT
0.5	O	Q	S
0.75	P	R	S-T
1.0	R	S-T	U-V
2.0	S	T-U	V

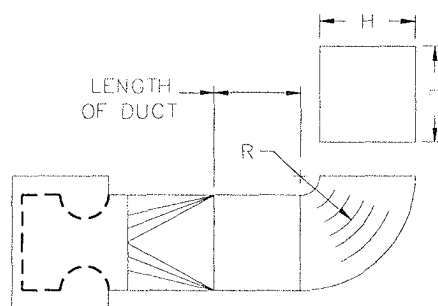
A. SQUARE ELBOW WITH INLET TRANSITION -- NO TURNING VANES.



SYSTEM EFFECT FACTORS

R/D	NO DUCT	2D DUCT	5D DUCT
0.5	S	T-U	V
1.0	T	U-V	W
2.0	V	V-W	W-X

B. SQUARE ELBOW WITH INLET TRANSITION -- 3 LONG TURNING VANES.



SYSTEM EFFECT FACTORS

R/D	NO DUCT	2D DUCT	5D DUCT
0.5	S	T-U	V
1.0	T	U-V	W
2.0	V	V-W	W-X

C. SQUARE ELBOW WITH INLET TRANSITION -- SHORT TURNING VANES.

$$D = \frac{2H}{\sqrt{\pi}}$$

THE INSIDE AREA OF THE SQUARE DUCT ($H \times H$) IS EQUAL TO THE INSIDE AREA CIRCUMSCRIBED BY THE FAN INLET COLLAR. THE MAXIMUM PERMISSIBLE ANGLE OF ANY CONVERGING ELEMENT OF THE TRANSITION IS 15° , AND FOR A DIVERGING ELEMENT 7.5° .

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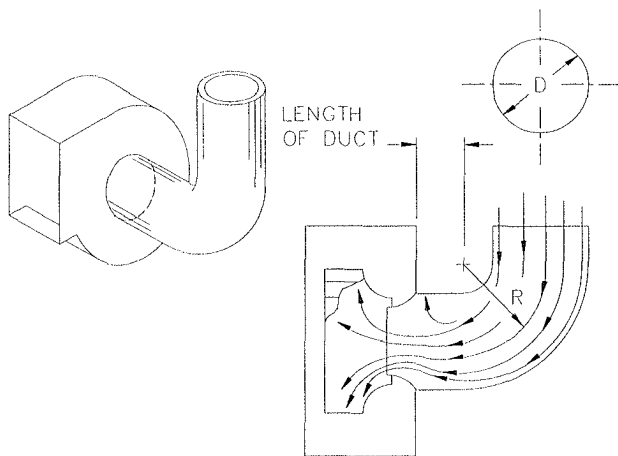
SYSTEM EFFECT CURVES
FOR OUTLET DUCTS--
AXIAL FANS

DATE

5-92

FIGURE

6-22

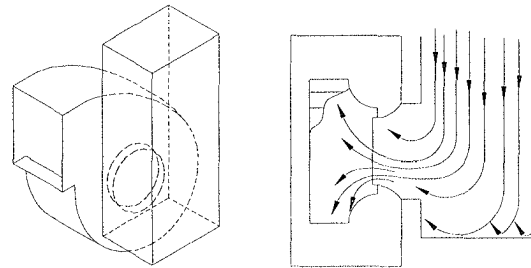


SYSTEM EFFECT FACTORS*

R/D	NO DUCT	2D DUCT	5D DUCT
0.75	1.4	0.8	0.40
1.0	1.2	0.66	0.33
2.0	1.0	0.53	0.33
3.0	0.66	0.40	0.22

A. NON-UNIFORM FLOW INTO A FAN INLET
BY A 90° ROUND SECTION ELBOW - NO
TURNING VANES.

*Values shown are in modification of the
original chart.



THE REDUCTION IN FLOW RATE AND PRESSURE FOR THIS TYPE OF INLET CONDITION IS IMPOSSIBLE TO TABULATE. THE MANY POSSIBLE VARIATIONS IN WIDTH AND DEPTH OF THE DUCT INFLUENCE THE REDUCTION IN PERFORMANCE TO VARYING DEGREES AND THEREFORE THIS INLET SHOULD BE AVOIDED. FLOW RATE LOSSES AS HIGH AS 45% HAVE BEEN OBSERVED. EXISTING INSTALLATIONS CAN BE IMPROVED WITH GUIDE VANES OR THE CONVERSION TO SQUARE OR MITERED ELBOWS WITH GUIDE VANES.

B. NON-UNIFORM FLOW INDUCED INTO FAN
INLET BY A RECTANGULAR INLET DUCT.

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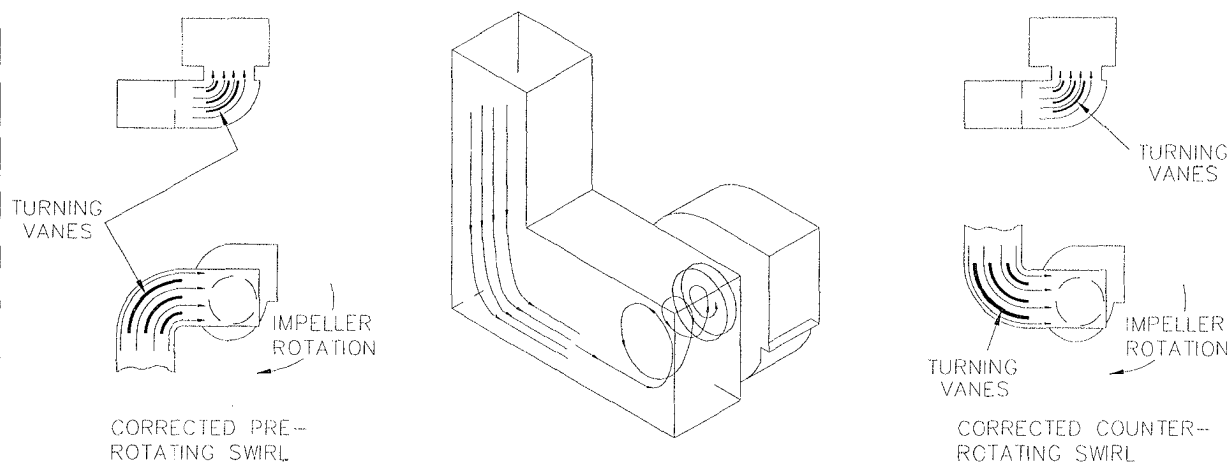
NON-UNIFORM INLET
FLOWS

DATE

5-92

FIGURE

6-23



C. NON-UNIFORM FLOW INTO A FAN INLET BY AN INDUCED VORTEX, SPIN OR SWIRL.

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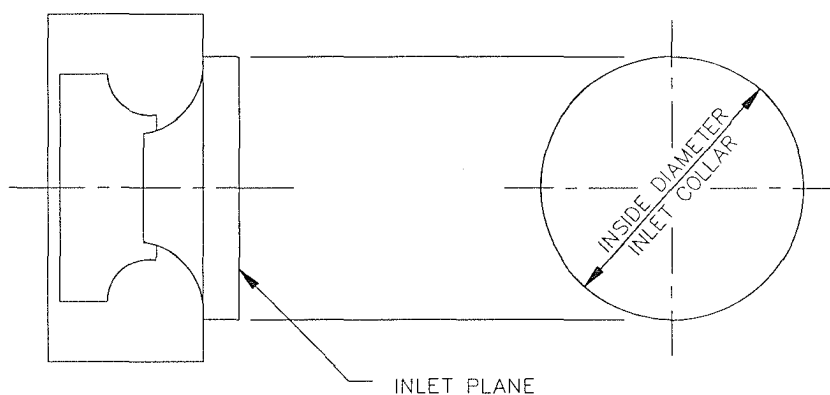
NON-UNIFORM INLET
CORRECTIONS

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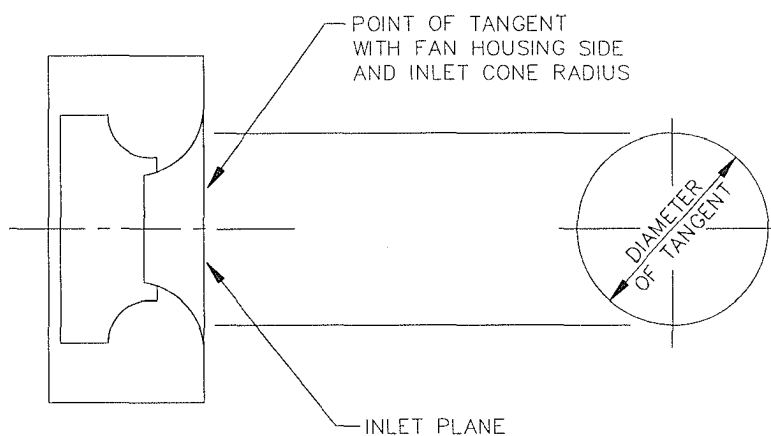
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FIGURE

6-24



A. FREE INLET AREA PLANE -- FAN WITH INLET COLLAR.



B. FREE INLET AREA PLANE -- FAN WITHOUT INLET COLLAR.

PERCENTAGE OF UNOBSTRUCTED INLET AREA	SYSTEM EFFECT FACTORS
100	NO LOSS
95	0.26
90	0.40
85	0.53
75	0.8
50	1.6
25	2.0

DETERMINE SEF BY CALCULATING INLET VELOCITY AND USING FIGURE 6-26

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*SYSTEM EFFECT CURVES
FOR INLET OBSTRUCTIONS*

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FIGURE

6-25

**Loss Factor Equivalents
for System Effect Curves***

Curve	F_{sys}	Curve	F_{sys}
F	16.0	P	1.98
G	14.3	Q	1.60
H	12.8	R	1.20
I	11.3	S	0.80
J	9.62	T	0.53
K	8.02	U	0.40
L	6.42	V	0.26
M	4.63	W	0.18
N	3.20	X	0.10
O	2.51		

To use this table:

- 1) Obtain the curve letter from Figures 6-18 through 6-22 or Figure 6-25.
- 2) For inlet system effects, multiply the equivalent loss coefficient from the above table by the fan inlet velocity pressure.
- 3) For outlet system effects, multiply the equivalent loss coefficient from the above table by the fan outlet velocity pressure.

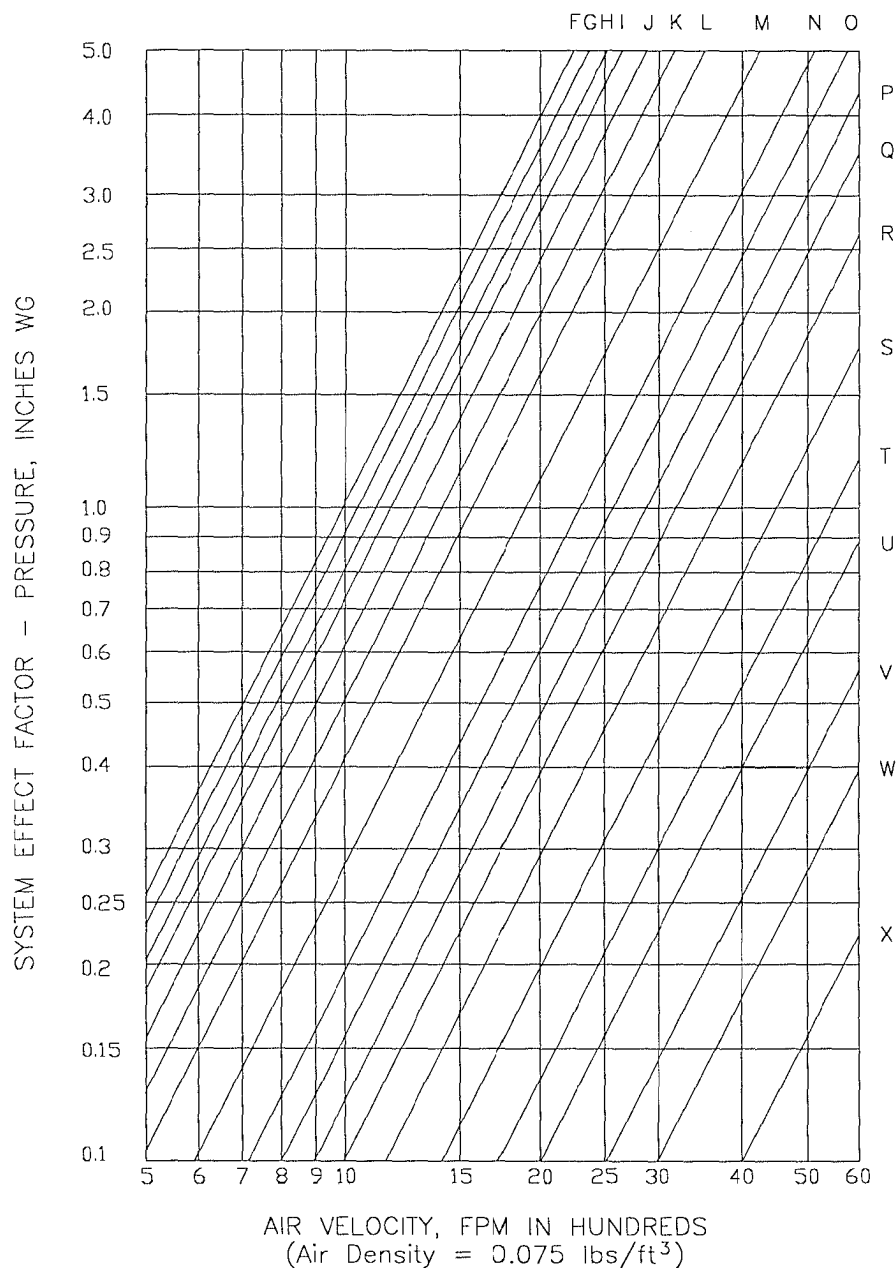
* F_{sys} values are in number of velocity pressures. For loss directly in "Wg, refer to Figure 6-27.

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SYSTEM EFFECT CURVES

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FIGURE 6-26



*Enter the chart at the appropriate air velocity (on the abscissa) read up to the applicable curve, then across from the curve (to the ordinate) to find the SEF at standard air density.

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SYSTEM EFFECT CURVES

DATE

6-92

FIGURE

6-27

Chapter 7

REPLACEMENT AND RECIRCULATED AIR

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7.1 INTRODUCTION

Chapters 1 through 6 describe the purpose, function, and design of industrial exhaust systems. As mentioned in Chapter 1, Section 1.2, supply systems are used for two basic purposes: to create a comfortable environment and to replace air exhausted from the building. It is important to note that while properly designed exhaust systems will remove toxic contaminants, they should not be relied upon to draw outdoor air into the building. If the amount of replacement air supplied to the building is lower than the amount of air exhausted, the pressure in the building will be lower than atmospheric. This condition is called "negative pressure" and results in air entering the building in an uncontrolled manner through window sashes, doorways, and walls. In turn, this may lead to many undesirable results such as high velocity drafts, backdrafting, difficulty in opening doors, etc.

To minimize these effects, design the mechanical supply systems to introduce sufficient outside air to avoid excessive negative or positive pressure conditions. A properly designed and installed air supply system can provide both replacement air and effective environmental control. Provided that important health and safety measures are taken, recirculation of the exhaust air may be an effective method that can substantially reduce heating and/or cooling costs.

7.2 REPLACEMENT AIR

Air will enter a building in an amount to equal the flow rate of exhaust air whether or not provision is made for this replacement. However, the actual exhaust flow rate will be less than the design value if the plant is under negative pressure. If the building perimeter is tightly sealed, thus blocking effective infiltration of outdoor air, a severe decrease of the exhaust flow rate will result. If, on the other hand, the building is relatively old with large sash areas, air infiltration may be quite pronounced and the exhaust system performance will decrease only slightly and other problems may occur.

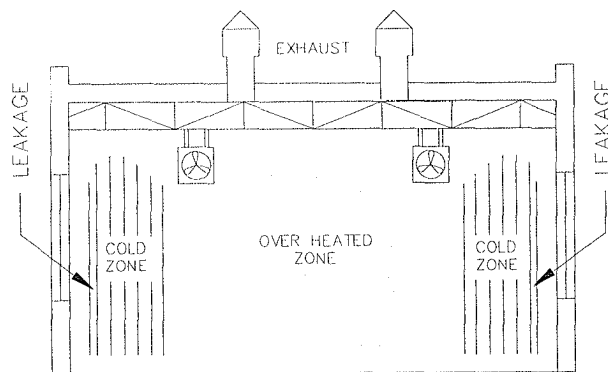


Figure 7-1. Under negative pressure conditions, workers in the cold zones turned up thermostats in an attempt to get heat. Because this did nothing to stop leakage of cold air, they remained cold while the center of plant was overheated.

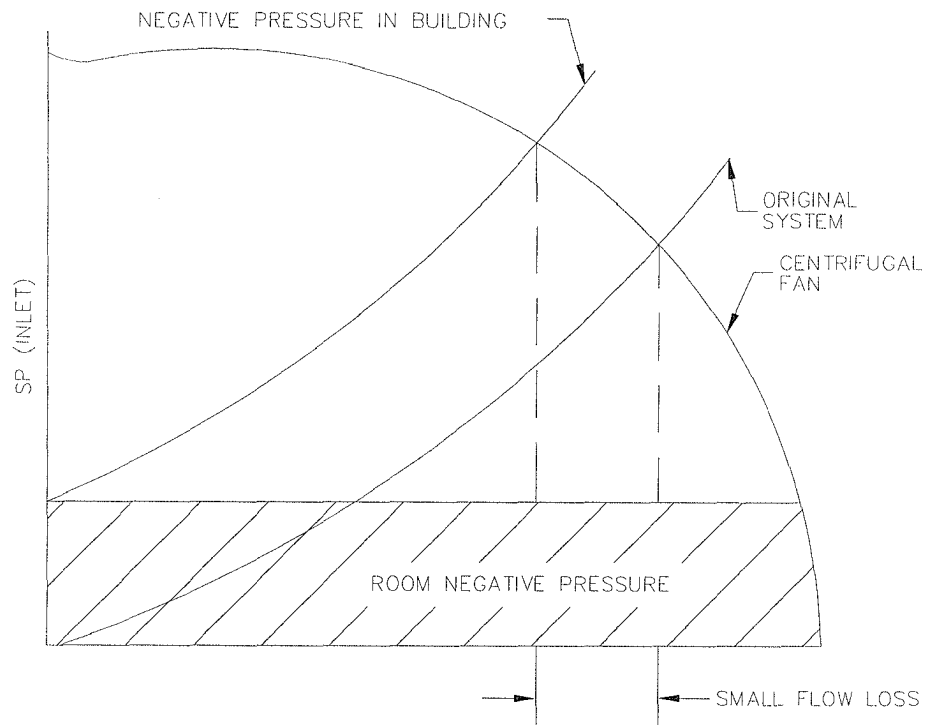
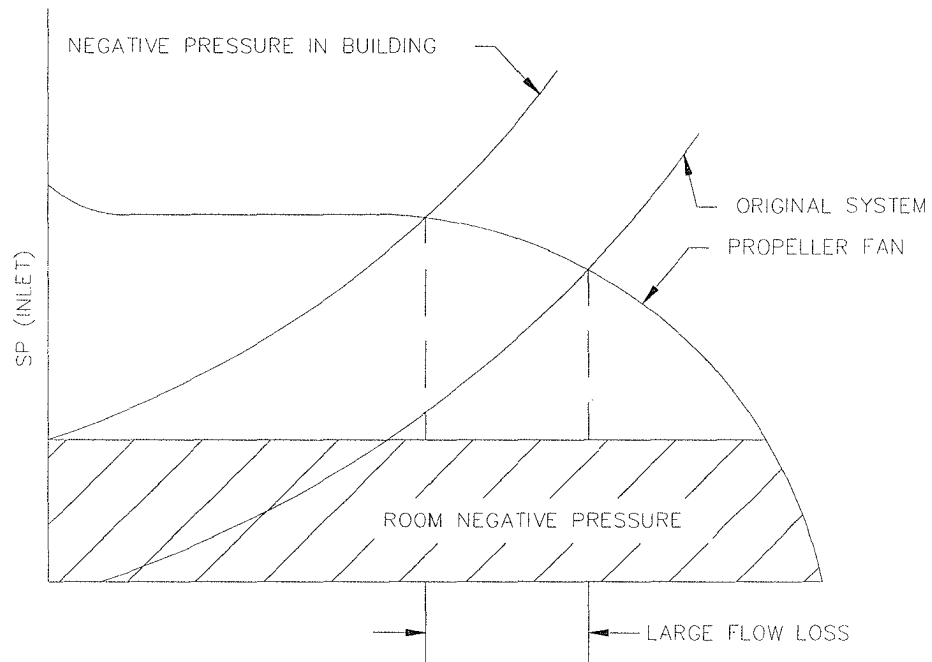
TABLE 7-1. Negative Pressures and Corresponding Velocities Through Crack Openings (Calculated with air at room temperature, standard atmospheric pressure, $C_e = 0.6$)

Negative Pressure, "wg	Velocity, fpm
0.004	150
0.008	215
0.010	240
0.014	285
0.016	300
0.018	320
0.020	340
0.025	380
0.030	415
0.040	480
0.050	540
0.060	590
0.080	680
0.100	760
0.150	930
0.200	1080
0.250	1200
0.300	1310
0.400	1520
0.500	1700
0.600	1860

When the building is relatively open, the resultant in-plant environmental condition is often undesirable since the influx of cold outdoor air in the northern climates chills the perimeter of the building. Exposed workers are subjected to drafts, space temperatures are not uniform, and the building heating system is usually overtaxed (see Figure 7-1). Although the air may eventually be tempered to acceptable conditions by mixing as it moves to the building interior, this is an ineffective way of transferring heat to the air and usually results in fuel waste.

Experience has shown that replacement air is necessary for the following reasons:

1. *To insure that exhaust hoods operate properly.* A lack of replacement air and the attendant negative pressure condition results in an increase in the static pressure the exhaust fans must overcome. This can cause a reduction in exhaust flow rate from all fans and is particularly serious with low-pressure fans such as wall fans and roof exhausters (see Figure 7-2).
2. *To eliminate high-velocity cross-drafts through windows and doors.* Depending on the negative pressure created, cross drafts may be substantial (see Table 7-1). Cross-drafts not only interfere with the proper opera-



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*HOW FAN PERFORMANCE FALLS
OFF UNDER NEGATIVE PRESSURE*

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FIGURE 7-2

TABLE 7-2. Negative Pressures Which May Cause Unsatisfactory Conditions Within Buildings

Negative Pressure "wg	Adverse Conditions
0.01 to 0.02	Worker Draft Complaints – High velocity drafts through doors and windows.
0.01 to 0.05	Natural Draft Stacks Ineffective – Ventilation through roof exhaust ventilators, flow through stacks with natural draft greatly reduced.
0.02 to 0.05	Carbon Monoxide Hazard – Back drafting will take place in hot water heaters, unit heaters, furnaces, and other combustion equipment not provided with induced draft.
0.03 to 0.10	General Mechanical Ventilation Reduced – Air flows reduced in propeller fans and low pressure supply and exhaust systems.
0.05 to 0.10	Doors Difficult to Open – Serious injury may result from non-checked, slamming doors.
0.10 to 0.25	Local Exhaust Ventilation Impaired – Centrifugal fan exhaust flow reduced.

tion of exhaust hoods, but also may disperse contaminated air from one section of the building to another and can interfere with the proper operation of process equipment such as open top solvent degreasers. In the case of dusty operations, settled material may be dislodged from surfaces and result in recontamination of the workroom.

3. *To insure operation of natural draft stacks such as combustion flues.* Moderate negative pressures can result in backdrafting of flues which may cause a dangerous health hazard from the release of combustion products, principally carbon monoxide, into the workroom. Back drafting may occur in natural draft stacks at negative pressures as low as 0.02 "wg (see Table 7-2). Secondary problems include difficulty in maintaining pilot lights in burners, poor operation of temperature controls, corrosion damage in stacks and heat exchangers due to condensation of water vapor in the flue gases.
4. *To eliminate cold drafts on workers.* Drafts not only cause discomfort and reduce working efficiency but also may result in lower overall ambient temperatures.
5. *To eliminate differential pressure on doors.* High differential pressures make doors difficult to open or shut and, in some instances, can cause personnel safety hazards when the doors move in an uncontrolled fashion (see Figure 7-3 and Table 7-2).
6. *To conserve fuel.* Without adequate replacement air, uncomfortable cold conditions near the building perimeter frequently lead to the installation of more heating equipment in those areas in an attempt to correct the problem. These heaters take an excessive amount of time to warm the air and the over-heated air moving toward the building interior makes those areas uncomfortably warm (see Figure 7-1). This in turn leads to the installation of more exhaust fans to remove the excess heat, further aggravating the problem. Heat is wasted without curing the problem. The fuel consumption with a replacement air heating system usu-

ally is lower than when attempts are made to achieve comfort without replacement air (see Section 7.10).

7.3 REPLACEMENT AIR DISTRIBUTION

Replacement air distribution is as critical as air volume (quantity) in industrial ventilation system design. Poor air distribution can destroy the control provided by a well-designed exhaust system. Non-turbulent air flow is particularly critical in indoor firing ranges, pharmaceutical plants, electronic components plants, some paint shops, and similar facilities.

Designers often use the general heating, ventilating, and air conditioning system or plant air system to provide the supply (or replacement) air to replace air exhausted by local exhaust hoods. Unfortunately, plant air systems use high throw diffusers to mix the warmer/cooler air with the air already in the plant. Many times the throw distance is over 40 feet. This mixing effect causes turbulence near local exhaust systems. The local exhaust hoods must then be redesigned to draw in more air to control the contaminants and overcome the turbulence. This increases energy costs due to the need for larger fans and motors. Sometimes hoods cannot capture contaminants and overcome turbulence. Hence, workers could still be overexposed even with a local exhaust system in place. There-

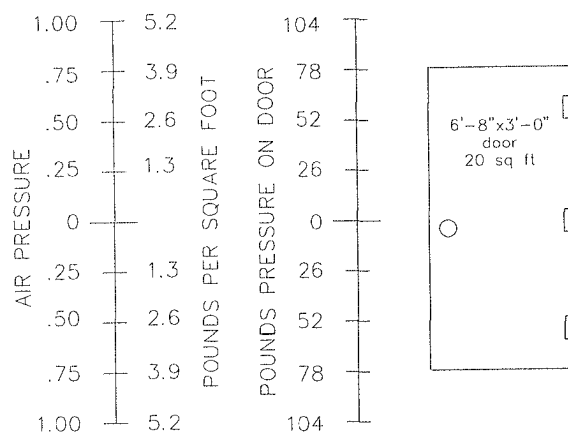


Figure 7-3. Relationship between air pressure and amount of force needed to open or close an average-sized door

fore, locate local exhaust hoods away from the turbulent effects of the plant air distribution system.

If the supply air system does not sufficiently cool the employees, pedestal fans are often used. Pedestal fans also destroy contaminant control by causing turbulence near the local exhaust hood and should not be allowed.

One method of providing non-turbulent air to the facility is to pass air through a supply air plenum built as part of the ceiling and/or through perforated duct. Cover the plenum face with perforated sheet metal. The ceiling plenum or duct runs should cover as large an area as possible to diffuse the air flow. A plenum wall providing cross-flow ventilation can be used only if the workers position themselves *between* the supply air system and the contaminant source and should not be used if the design velocity is over 100 fpm.

Perforated drop-type ceilings work best in facilities with ceiling heights of less than 15 feet. Hoist tracks, lighting, and fire protection systems can be built into the ceiling. In some cases, fire protection will be required above and below the ceiling. Use perforated duct for ceilings over 15 feet. Perforated duct manufacturers have computer programs to assist designers in determining duct sizes, shapes, and types as well as the location of pressure adjusting devices such as orifice plates and reducers. Air flow delivery in large bays may require supplemental air delivered at work stations to provide comfortable conditions for the workers.

Feeding the air into the plenum is also critical. High velocity flow into the plenum will cause the same turbulence problems as the large diffusers commonly found in plants that attempt to throw the air to the floor. Consider feeding the plenum with perforated duct to diffuse the air inside the plenum. Another method of distributing air flow, from either a ceiling or wall-mounted plenum, is to design the plenum with two perforated plates, one fixed and one adjustable, located 2–6 inches apart. Air flowing through slightly offset holes will encounter more resistance; thus air quantities passing through the low-flow areas will increase. The adjustable plates must be small enough to fine tune the air flow from the plenum.

7.4 REPLACEMENT AIR FLOW RATE

In most cases, replacement air flow rate should approximate the total air flow rate of air removed from the building by exhaust ventilation systems, process systems and combustion processes. Determination of the actual flow rate of air removed usually requires an inventory of air exhaust locations and any necessary testing. When conducting the exhaust inventory, it is necessary not only to determine the quantity of air removed, but also the need for a particular piece of equipment. At the same time, reasonable projections should be made of the total plant exhaust requirements for the next one to two years, particularly if process changes or plant expansions are contemplated. In such cases it can be practical to purchase a replacement air unit slightly larger than immedi-

ately necessary with the knowledge that the increased capacity will be required within a short time. The additional cost of a larger unit is relatively small and in most cases the fan drive can be regulated to supply only the desired quantity of air.

Having established the minimum air supply quantity necessary for replacement air purposes, many plants have found that it is wise to provide additional supply air flow rate to overcome natural ventilation leakage and further minimize drafts at the perimeter of the building.

7.5 ROOM PRESSURE

While negative pressure can cause adverse conditions as described in Sections 7.1–7.3, there are situations where negative pressures are desired. An example is a room or area where a contaminant must be prevented from escaping into the surrounding area. It also may be desirable to maintain a room or area under positive pressure to maintain a clean environment. Either of these conditions can be achieved by setting and maintaining the proper exhaust/supply flow differential. Negative pressure can be achieved by setting the exhaust volumetric flow rate (Q) from the area to a level higher than the supply rate. A good performance standard for industrial processes is to set a negative pressure differential of 0.04 ± 0.02 "wg. Conversely, positive pressure is achieved by setting the supply air flow rate higher than the exhaust rate. The proper flow differential will depend on the physical conditions of the area, but a general guide is to set a 5% flow difference but no less than 50 cfm. If the volume flows vary during either negatively or positively pressurized processes, it is easier to maintain the desired room pressure by adjusting the supply air.

Some designers use transfer grilles and a pressure sensor in the room to maintain a desired room pressure. Air is allowed to seep from adjacent hallways, offices and other non-industrial areas. Do not use transfer grilles between areas where contaminant migration is possible.

7.6 ENVIRONMENTAL CONTROL

There are generally three types of industrial ventilation systems in most plants: 1) return air for the clean plant air; 2) a return air system where low level contaminants are diluted with fresh air (dilution ventilation); and 3) contaminant-laden air drawn through a local exhaust hood or ventilation system. In addition to toxic contaminants which are most effectively controlled by hoods, industrial processes may create an undesirable heat load in the work space. Modern automated machining, conveying, and transferring equipment require considerable horsepower. Precision manufacturing and assembling demand increasingly higher light levels in the plant with correspondingly greater heat release. The resulting in-plant heat burden raises indoor temperatures, often beyond the limits of efficient and healthful working conditions and, in some cases, beyond the tolerance limits for the product.

Environmental control of these factors can be accommodated through the careful use of the supply system. Industrial air conditioning may be required to maintain process specifications and employee health. Many times the designer can use a setpoint higher than the 50-55 F used in conventional HVAC designs. ASHRAE gives basic criteria for industrial air conditioning in HVAC applications.^(7,9) (It must be noted that radiant heat cannot be controlled by ventilation and methods such as shielding, described in Chapter 2, are required.) Sensible and latent heat released by people and the process can be controlled to desired limits by proper use of ventilation.

The HVAC industry uses automated building control and direct digital control (DDC) in many facilities. The technology can be applied to industrial ventilation with careful planning. DDC uses computers and microprocessors tied to sensors and actuators to form a feedback and control system. DDC can be useful in industrial ventilation systems to control temperature, humidity, and relative room pressures. DDC systems can also track the system performance at hoods, fans, heating and cooling, and air pollution control equipment. DDC is especially useful in preventive maintenance. However, DDC systems for industrial ventilation systems are complicated. Many are "one-of-a-kind" systems designed by a controls manufacturer and they require trained personnel to operate.

Many industrial processes release minor amounts of "nuisance" contaminants which, at low concentrations, have no known health effects but which are unpleasant or disagreeable to the workers or harmful to the product. The desire to provide a clean working environment for both the people and the product often dictates controlled air flow between rooms or entire departments. Evaluate the air streams returned into the facility to determine if the air pollution control devices (e.g., filters, cyclones) provide sufficient cleaning to prevent employee exposure to "nuisance" contaminants. In addition, systems with known contaminants require controls listed in Section 7.12 and 7.13. The facility must employ trained mechanics and support a preventive maintenance program to sufficiently protect the workers.

7.7 ENVIRONMENTAL CONTROL AIR FLOW RATE

The design supply air flow rate depends on several factors including the health and comfort requirements. *Sensible heat* can be removed through simple air dilution (see Chapter 2 under ventilation).

"Nuisance" or undesirable contaminants can also be reduced by dilution with outdoor air. The control of odors from people at various conditions of rest and work can be accomplished with the outdoor air flow rate described in Chapter 2. However, these data apply mainly to offices, schools and similar types of environment and do not correspond well with the usual industrial or commercial establishment. Experience shows that when the air supply is properly distributed to the

TABLE 7-3. Air Exchanges Vs. Room Sizes

Room Size	Room ft ³	Air changes/ minute	Air changes/ hour
40 x 40 x 12 high	19,200	11,650/19,200 = 0.61	36
40 x 40 x 20 high	32,000	11,650/32,000 = 0.364	22

working level (i.e., in the lower 8-10 ft of the space), outdoor air supply of 1-2 cfm/ft² of floor space will give good results. Specific quantities of outdoor air must be obtained from criteria developed by groups such as ASHRAE.

7.8 AIR CHANGES

"Number of air changes per minute or per hour" is the ratio of the ventilation rate (per minute or per hour) to the room volume. "Air changes per hour" or "air changes per minute" is a poor basis for ventilation criteria where environmental control of hazards, heat, and/or odors is required. The required ventilation depends on the problem, not on the size of the room in which it occurs. For example, let us assume a situation where 11,650 cfm would be required to control solvent vapors by dilution. The operation may be conducted in either of two rooms, but in either case, 11,650 cfm is the required ventilation. The "air changes," however, would be quite different for the two rooms. As can be seen in Table 7-3, for the same "air change" rate, a high ceiling space will require more ventilation than a low ceiling space of the same floor area. Thus, there is little relationship between "air changes" and the required contaminant control.

The "air change" basis for ventilation does have some applicability for relatively standard situations such as office buildings and school rooms where a standard ventilation rate is reasonable. It is easily understood and reduces the engineering effort required to establish a design criteria for ventilation. It is this ease of application, in fact, which often leads to lack of investigation of the real engineering parameters involved and correspondingly poor results.

7.9 AIR SUPPLY TEMPERATURES

Supply air temperature is controlled by the demand for heating and cooling. Factors to consider in maintaining a comfortable work environment for occupants are: setpoint temperature, humidity control, air distribution, and air flow rate. Where high internal heat loads are to be controlled, however, the temperature of the air supply can be appreciably below that of the space by reducing the amount of heat supplied to the air during the winter months and by deliberately cooling the air in the summer. When a large air flow rate is delivered at approximately space temperatures or somewhat below, the distribution of the air becomes vitally important in order to maintain satisfactory environmental conditions for the persons in the space.

Maximum utilization of the supply air is achieved when the

air is distributed in the "living zone" of the space, below the 8–10 foot level (see Figure 7-4). When delivered in this manner — where the majority of the people and processes are located — maximum ventilation results with minimum air handling. During the warm months of the year, large air flow in the working space at relatively high velocities is welcomed by the workers. During the winter months, however, care must be taken to insure that air velocities over the person, except when extremely high heat loads are involved, are kept within acceptable values (see Chapter 2, Table 2-5). To accomplish this, the air can be distributed uniformly in the space or where required for worker comfort. Heavy-duty, adjustable, directional grilles and louvers have proven to be very successful in allowing individual workers to direct the air as needed.^(7.1) Light gauge, stamped grilles intended for commercial use are not satisfactory. Suitable control must be provided to accommodate seasonal and even daily requirements with a minimum of supervision or maintenance attention.

Chapter 2 describes the relative comfort that can be derived through adequate air flow control. Published tables of data by register and diffuser manufacturers indicate the amount of throw (projection) and spread that can be achieved with different designs at different flow rates (see Figure 7-4). Terminal velocities at the throw distance can also be determined.

Multiple point distribution is usually best since it provides uniformity of air delivery and minimizes the re-entrainment of contaminated air that occurs when large volumes are "dumped" at relatively high velocities. Depending on the size and shape of the space and the amount of air to be delivered, various distributional layouts are employed. Single point distribution can be used; however, it is usually necessary to redirect the large volume of air with a baffle or series of baffles in order to reduce the velocity close to the outlet and minimize re-entrainment. In determining the number and types of registers or outlet points, it also is necessary to consider the effect of terminal air supply velocity on the performance of local exhaust hoods.

When large amounts of sensible heat are to be removed from the space during the winter months, it is most practical to plan for rapid mixing of the cooler air supply with the warmer air in the space. During the summer months, the best distribution usually involves minimum mixing so that the air supply will reach the worker at higher velocities and with a minimum of heat pickup. These results can be obtained by providing horizontal distribution of winter air over the worker's head, mixing before it reaches the work area and directing the air toward the worker through register adjustment for the summer months (see Figure 7-5).

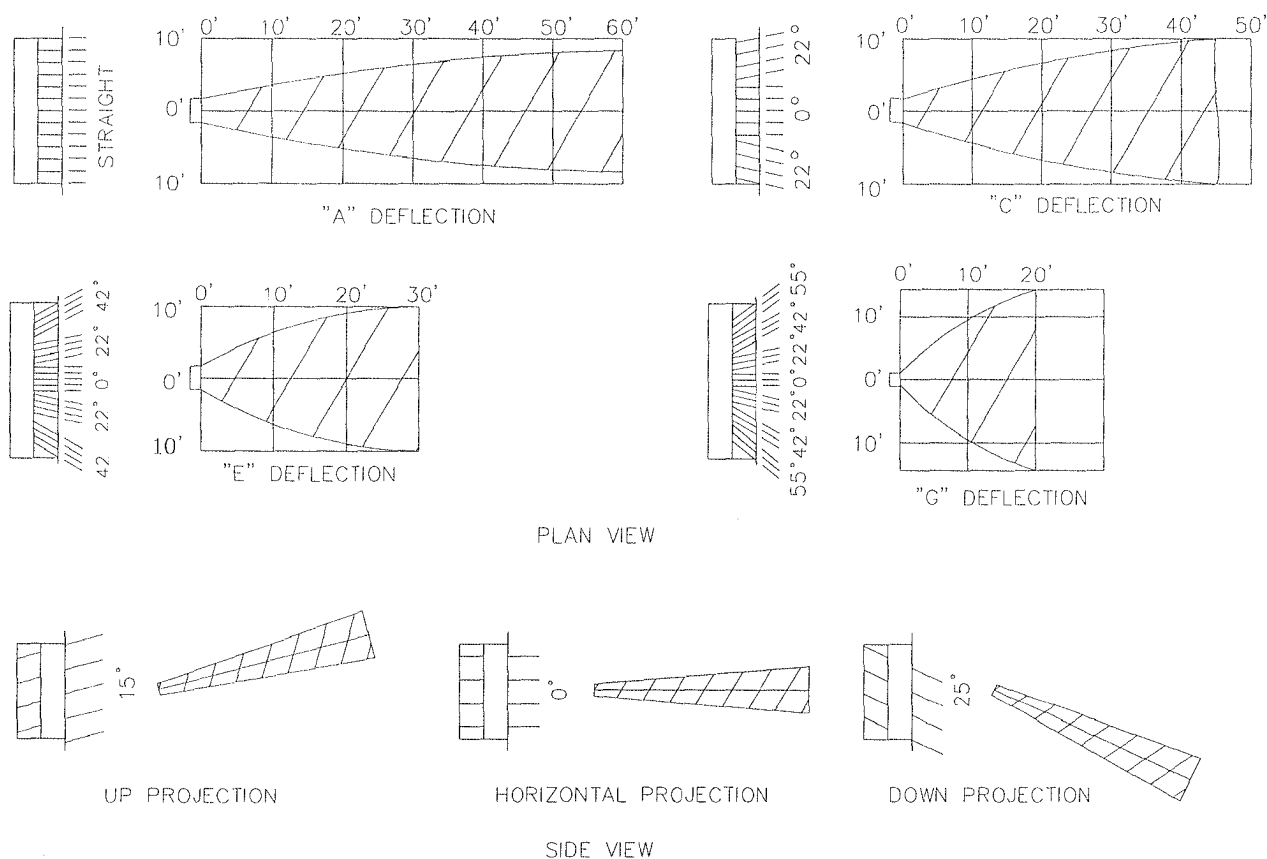


FIGURE 7-4. Throw patterns and distance from different register adjustments (REF. 7-2)

Delivered air temperatures during the winter usually range from 65 F–68 F for work areas without much process heat or vigorous work requirement downward to 60 F or even 55 F where hard work or significant heat sources are involved. For summer operation, the temperature rise in indoor air can be estimated as described in Chapter 2. Evaporative cooling should be considered for summer operation. Although not as effective as mechanical refrigeration under all conditions, evaporative cooling significantly lowers the temperature of

the outdoor air even in humid climates, improves the ability of the ventilation air to reduce heat stress, and costs much less to install and operate.

7.10 AIR SUPPLY VS. PLANT HEATING COSTS

Even if the supply air were drawn into the building simply by the action of the exhaust fans, during the winter months there will be an added burden on the plant heating system and fuel costs will rise. Experience has shown, however, that when

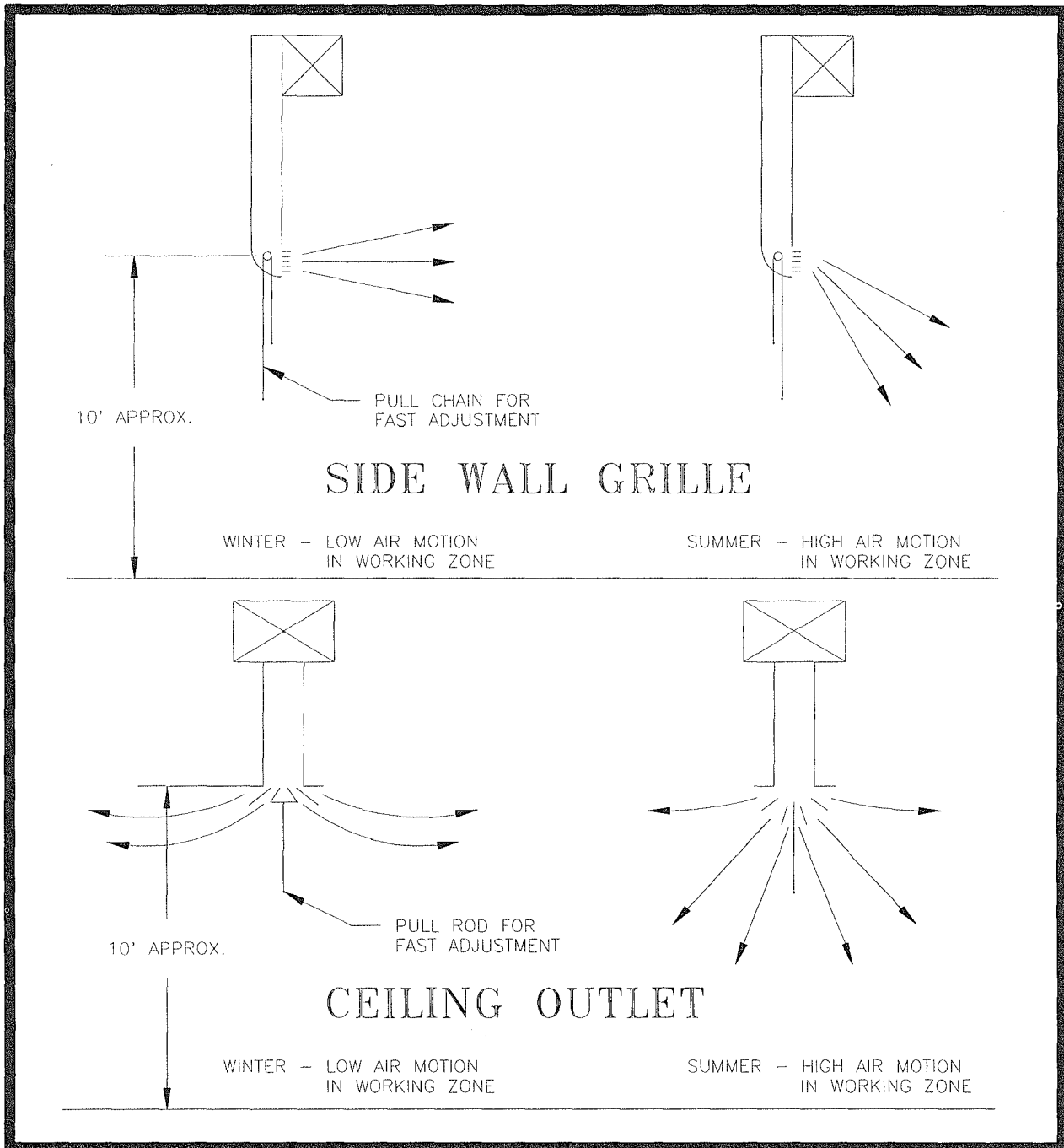


FIGURE 7-5. Seasonal air ventilation

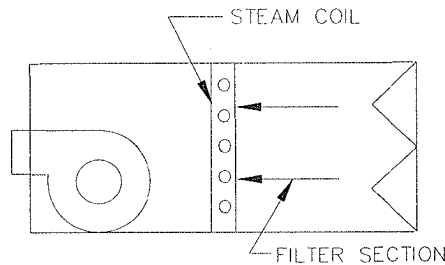


FIGURE 7-6. Single coil steam unit

the same flow rate of outdoor air is introduced through properly designed replacement air heaters, the overall fuel cost does not exceed previous levels and often is decreased. A partial explanation of this savings is more efficient heat transfer. The most important factor, however, is that a well-designed air supply system is not dependent on the plant space heating system; rather, the two systems operate in an independent fashion. The air supply system and the plant heating system can be understood best by considering the building as a whole. In order for an equilibrium to be established, the heat outflow from the building must balance the heat inflow. To obtain additional energy saving during downtime, design the supply system to provide sufficient heating to counter air entering the building through infiltration and to prevent freezing.

7.11 REPLACEMENT AIR HEATING EQUIPMENT

Replacement air heaters are usually designed to supply 100% outdoor air. The basic requirements for an air heater are that it be capable of continuous operation, constant delivered air flow rate, and constant preselected discharge temperature. The heater must meet these requirements under varying conditions of service and accommodate outdoor air temperatures which vary as much as 40 F daily. Standard design heating and ventilating units are usually selected for mixed air applications, i.e., partial outdoor air and partial recirculated air; it is rare that their construction and operating capabilities will meet the requirements of industry. Such units are applicable in commercial buildings and institutional facilities where the requirements are less severe and where mixed air service is more common.

Air heaters are usually categorized according to the source of heat: steam and hot water units, indirect-fired gas and oil units, and direct-fired natural gas and Liquefied Petroleum Gas (LPG) units. Each basic type is capable of meeting the first two requirements — constant operation and constant delivered air flow rate. Variations occur within each type in relation to the third requirement, that of constant preselected discharge temperature. One exception to this rule is the direct-fired air heater where the inherent design provides a wide range of temperature control. Each type of air heater has specific advantages and limitations which must be understood by the designer in making a selection.

Steam coil units were probably the earliest air heaters

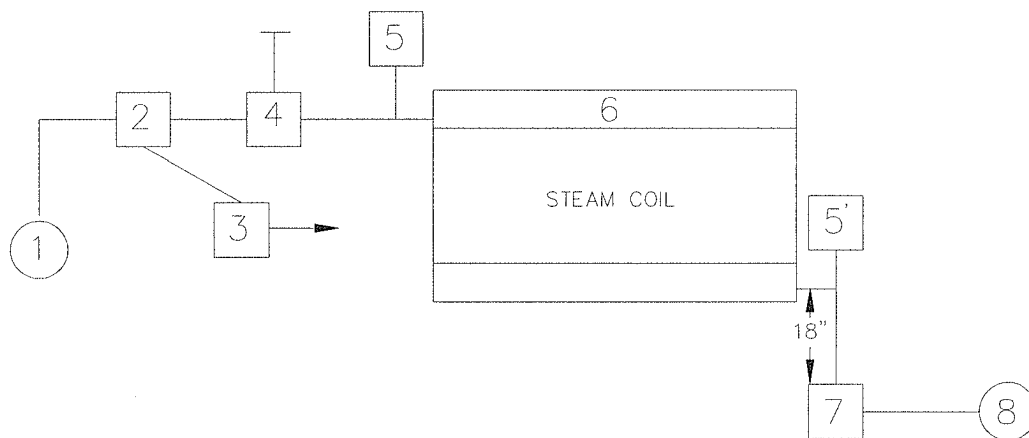
applied to general industry as well as commercial and institutional buildings (see Figure 7-6). When properly designed, selected, and installed, they are reliable and safe. They require a reliable source of clean steam at dependable pressure. For this reason they are applied most widely in large installations; smaller industrial plants often do not provide a boiler or steam capacity for operating a steam air heater. Principal disadvantages of steam units are potential damage from freezing or water hammer in the coils, the complexity of controls when close temperature limits must be maintained, high cost, and excessive piping.

Freezing and water hammer are the result of poor selection and installation and can be minimized through careful application. The coil must be sized to provide desired heat output at the available steam pressure and flow. The coil preferably should be of the steam distributing type with vertical tubes. The traps and return piping must be sized for the maximum condensate flow at minimum steam pressure plus a safety factor. Atmospheric vents must be provided to minimize the danger of a vacuum in the coil which would hold up the condensate. Finally, *the condensate must never be lifted by steam pressure*. The majority of freeze-up and water hammer problems relate to the steam modulating type of unit which relies on throttling of the steam supply to achieve temperature control. When throttling occurs, a vacuum can be created in the coil and unless adequate venting is provided, condensate will not drain and can freeze rapidly under the influence of cold outdoor air. Most freeze-ups occur when outdoor air is in the range of 20–30 F and the steam control valve is partially closed, rather than when the outdoor air is a minimum temperature and full steam supply is on (see Figure 7-7).

"Safety" controls are often used to detect imminent danger from freeze-up. A thermostat in the condensate line or an extended bulb thermostat on the downstream side of the coil can be connected into the control circuit to shut the unit down when the temperature falls below a safe point. As an alternate, the thermostat can call for full steam flow to the coil with shutdown if a safe temperature is not maintained. An obvious disadvantage is that the plant air supply is reduced; if the building should be subjected to an appreciable negative pressure, unit freeze-up still may occur due to cold air leakage through the fresh air dampers.

The throttling range of a single coil unit can be extended by using two valves: one valve is usually sized for about two-thirds the capacity and the other valve one-third. Through suitable control arrangements both valves will provide 100% steam flow when fully opened and various combinations will provide a wide range of temperature control. Controls are complex in this type of unit and care must be taken to insure that pressure drop through the two valve circuits is essentially equal so as to provide expected steam flow.

Multiple coil steam units (Figure 7-8) and bypass designs (Figure 7-9) are available to extend the temperature control



1. STEAM SUPPLY
 PROVIDE STEAM FROM A CLEAN SOURCE
 MAINTAIN CONSTANT PRESSURE WITH REDUCING VALVES IF REQUIRED
 PROVIDE TRAPPED DRIPS FOR SUPPLY LINES
 SIZE SUPPLY PIPING FOR FULL LOAD AT AVAILABLE PRESSURE
2. STRAINER
 1/32" DIAMETER MINIMUM PERFORATIONS
3. DRIP TRAP
 INVERTED BUCKET TRAP PREFERRED
4. CONTROL VALVE
 SIZE FOR MAXIMUM STEAM FLOW
 MAXIMUM PRESSURE DROP EQUAL TO 50% INLET STEAM PRESSURE
5. VACUUM BREAKER
 1/2" CHECK VALVE TO ATMOSPHERE
- 5'. ALTERNATE VACUUM BREAKER
6. STEAM COIL
 A. SIZE FOR DESIGN CAPACITY AT INLET STEAM PRESSURE (SUPPLY-VALVE DROP)
 B. VERTICAL COILS PREFERRED
 C. HORIZONTAL COILS MUST BE PITCHED 1/4" PER FOOT TOWARD DRAIN.
 6' MAXIMUM LENGTH RECOMMENDED
7. CONDENSATE TRAP
 A. INVERTED BUCKET PREFERRED
 B. SIZE TRAP FOR THREE TIMES MAXIMUM CONDENSATE LOAD AT PRESSURE
 DROP EQUAL TO 50% INLET PRESSURE
 C. INDIVIDUAL TRAP FOR EACH COIL
8. CONDENSATE RETURN
 ATMOSPHERIC DRAIN ONLY

AMERICAN CONFERENCE
 OF GOVERNMENTAL
 INDUSTRIAL HYGIENISTS

STEAM COIL PIPING

DATE 1-74

FIGURE 7-7

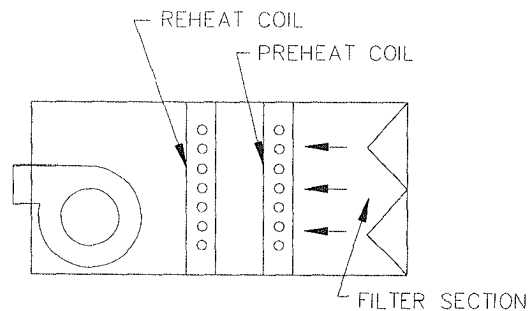


FIGURE 7-8. Multiple coil steam unit

range and help minimize freeze-up. With multiple coil units, the first coil (preheat) is usually sized to raise the air temperature from the design outdoor temperature to at least 40 F. The coil is controlled with an on-off valve which will be fully open whenever the outdoor temperature is below 40 F. The second (reheat) coil is designed to raise the air temperature from 40 F to the desired discharge condition. Temperature control will be satisfactory for most outdoor conditions, but overheating can occur when the outdoor air temperature approaches 40 F (39 F + the rise through the preheat coil can give temperatures of 79–89 F *entering* the reheat). Refined temperature control can be accomplished by using a second preheat coil to split the preheat load.

Bypass units incorporate dampers to direct the air flow. When maximum temperature rise is required, all air is directed through the coil. As the outdoor temperature rises, more and more air is diverted through the bypass section until finally all air is bypassed. Controls are relatively simple. The principal disadvantage is that the bypass is not always sized for full air flow at the same pressure drop as through the coil, thus (depending on the damper position) the unit may deliver

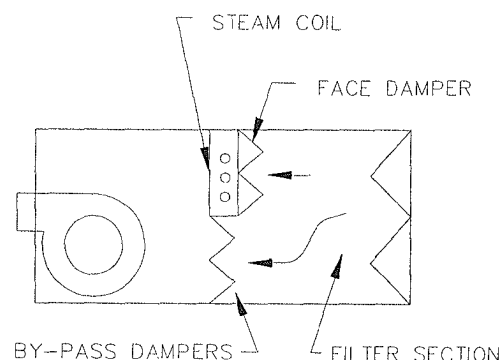
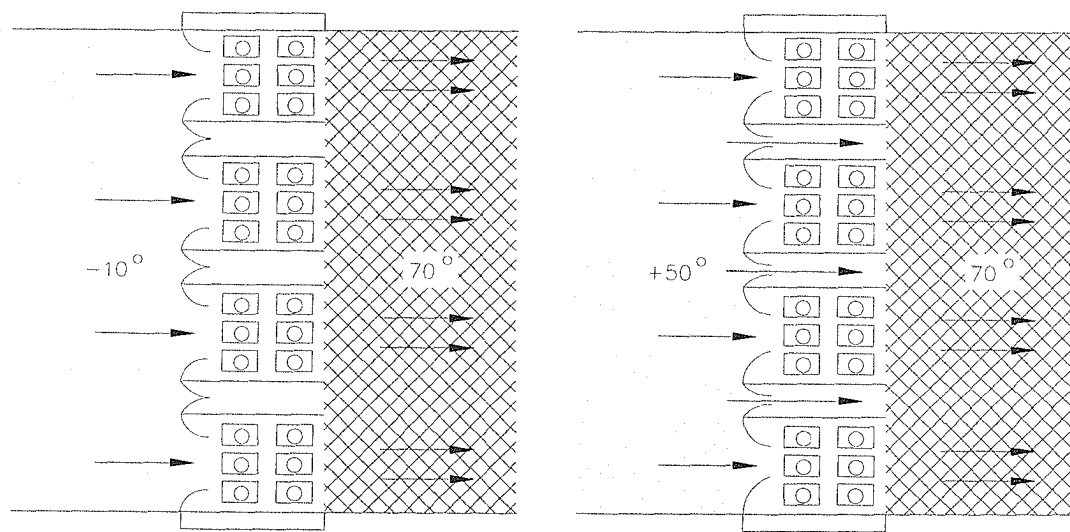


FIGURE 7-9. By-pass steam system

differing air flow rates. Damper air flow characteristics are also a factor. An additional concern is that in some units the air coming through the bypass and entering the fan compartment may have a nonuniform flow and/or temperature characteristic which will affect the fan's ability to deliver air.

Another type of bypass design, called integral face and bypass (Figure 7-10), features alternating sections of coil and bypass. This design promotes more uniform mixing of the air stream, minimizes any nonuniform flow effect and, through carefully engineered damper design, permits minimum temperature pickup even at full steam flow and full bypass.

Hot water is an acceptable heating medium for air heaters. As with steam, there must be a dependable source of water at predetermined temperatures for accurate sizing of the coil. Hot water units are less susceptible to freezing than steam because of the forced convection which insures that the cooler water can be positively removed from the coil. Practical difficulties and pumping requirements thus far have limited

FIGURE 7-10. Integral face and by-pass coil^(7.4)

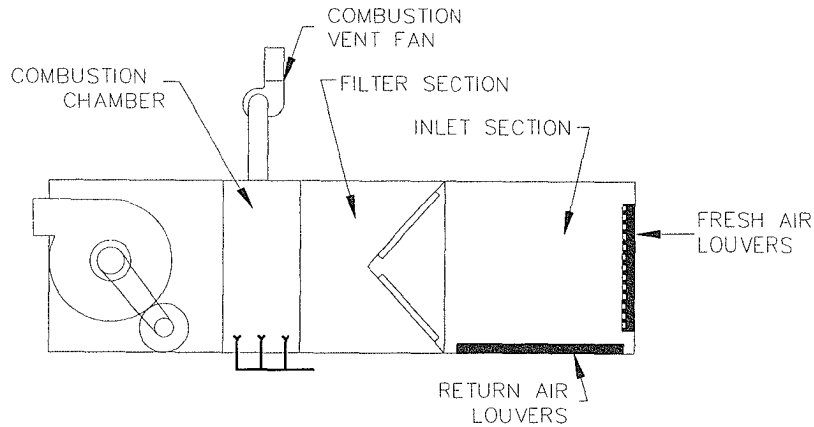


FIGURE 7-11. Indirect-fired unit

the application of hot water to relatively small systems: for a 100 F air temperature rise and an allowable 100 F water temperature drop, 1 GPM of water will provide heat for only 450 cfm of air. This range can be extended with high temperature hot water systems.

Hybrid systems using an intermediate heat exchange fluid, such as ethylene glycol, have also been installed by industries with critical air supply problems and a desire to eliminate all freeze-up dangers. A primary steam system provides the necessary heat to a converter which supplies a secondary closed loop of the selected heat exchange fluid. The added equipment cost is at least partially offset by the less complex control system.

Indirect-fired gas and oil units (Figure 7-11) are widely applied in small industrial and commercial applications. Economics appear to favor their use up to approximately 10,000 cfm; above this size the capital cost of direct-fired air heaters is lower. Indirect-fired heaters incorporate a heat exchanger, commonly stainless steel, which effectively separates the incoming air stream from the products of combustion of the fuel being burned. Positive venting of combustion products is usually accomplished with induced draft fans. These precautions are taken to minimize interior corrosion damage from condensation in the heat exchanger due to the chilling effect of the incoming cold air stream. The indirect-fired air heater permits the use of oil as a heat source and room air recirculation is permitted with this type of unit since the air stream is separated from the products of combustion. A third major advantage is that this type of unit is economical in the smaller sizes and is widely applied as a "package" unit in small installations such as commercial kitchens and laundries.

Temperature control, "turn-down ratio," is limited to about 3:1 or 5:1 due to burner design limitations and the necessity to maintain minimum temperatures in the heat exchanger and flues. Temperature control can be extended through the use of a bypass system similar to that described for single coil steam air heaters. Bypass units of this design offer the same

advantages and disadvantages as the steam bypass units.

Another type of indirect-fired unit incorporates a rotating heat exchanger. Temperature control can be as high as 20:1.

Direct-fired heaters wherein the fuel, natural or LPG gas, is burned directly in the air stream and the products of combustion are released in the air supply have been commercially available for some years (Figure 7-12). These units are economical to operate since all of the net heating value of the fuel is available to raise the temperature of the air resulting in a net heating efficiency approaching 90%. Commercially available burner designs provide turndown ratios from approximately 25:1 to as high as 45:1 and permit excellent temperature control. In sizes above 10,000 cfm, the units are relatively inexpensive on a cost per cfm basis; below this capacity, the costs of the additional combustion and safety controls weigh heavily against this design. A further disadvantage is that governmental codes often prohibit the recirculation of room air across the burner. Controls on these units are designed to provide a positive proof of air flow before the burner can ignite, a timed pre-ignition purge to insure that any leakage gases will be removed from the housing, and constantly supervised flame operation which includes both flame controls and high temperature limits.

Concerns are often expressed with respect to potentially toxic concentrations of carbon monoxide, oxides of nitrogen, aldehydes, and other contaminants produced by combustion and released into the supply air stream. Practical field evalu-

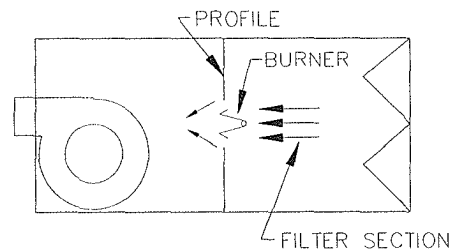


FIGURE 7-12. Direct-fired unit

ations and detailed studies show that with a properly operated, adequately maintained unit, carbon monoxide concentrations will not be expected to exceed 5 ppm and that oxides of nitrogen and aldehydes are well within acceptable limits.^(7.3)

A variation of this unit, known as a bypass design, has gained acceptance in larger plants where there is a desire to circulate large air flows at all times (see Figure 7-13). In this design, controls are arranged to reduce the flow of outdoor air across the burner and permit the entry of room air into the fan compartment. In this way, the fan air flow rate remains constant and circulation in the space is maintained. It is important to note that the bypass air does not cross the burner — 100% outdoor air only is allowed to pass through the combustion zone. Controls are arranged to regulate outdoor air flow and also to insure that burner profile velocity remains within the limits specified by the burner manufacturer, usually in the range of 2,000 to 3,000 fpm. This is accomplished by providing a variable profile which changes area as the damper positions change.

Inasmuch as there are advantages and disadvantages to both direct-fired and indirect-fired replacement air heaters, a careful consideration of characteristics of each heater should be made. A comparison of the heaters is given in Table 7-4.

7.12 COST OF HEATING REPLACEMENT AIR

As noted above, the cost of heating replacement air is

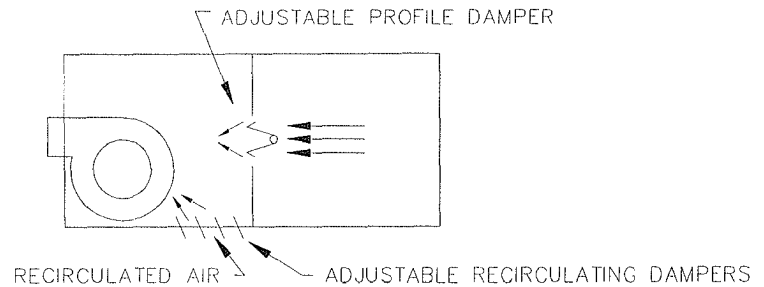


FIGURE 7-13. Direct fired by-pass unit

probably the most significant annual cost of a ventilation system. Newer processes requiring cooling during the process must also be evaluated. Occupant comfort is more important than saving a few dollars in energy costs. Recent indoor air quality studies quantify diminished productivity when workers are uncomfortable. In addition to the equipment first cost, local building codes, and environmental regulations, designer experience in utility incentives and operating costs are involved in purchasing decisions.

The American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE), the U. S. Department of Energy, and others develop formulae and computer programs to determine the life-cycle costs of various equipment. The formulae and programs should not be used to determine the annual utility bill. Instead,

TABLE 7-4. Comparison of Heater Advantages and Disadvantages

Advantages	Disadvantages
Direct-fired Unvented: <ol style="list-style-type: none"> 1. Good turndown ration —8:1 in small sizes; 25:1 in large sizes. Better control; lower operating costs. 2. No vent stack, flue, or chimney necessary. Can be located inside walls of building. 3. Higher efficiency (90%). Lower operating costs. (Efficiency based on available sensible heat.) 4. Can heat air over a wide temperature range. 5. First cost lower in large size units. 	<ol style="list-style-type: none"> 1. Products of combustion in heater air stream (some CO₂, CO, oxides of nitrogen, and water vapor present). 2. First cost higher in small size units. 3. May be limited in application by governmental regulations. Consult local ordinances. 4. Extreme care must be exercised to prevent minute quantities of chlorinated or other hydrocarbons from entering air intake, or toxic products may be produced in heated air. 5. Can be used only with natural gas or LPG. 6. Burner must be tested to assure low CO and oxides of nitrogen content in air stream.
Indirect Exchanger: <ol style="list-style-type: none"> 1. No products of combustion; outdoor air only is discharged into building. 2. Allowable in all types of applications and buildings if provided with proper safety controls. 3. Small quantities of chlorinated hydrocarbons will not normally break down on exchanger to form toxic products in heated air. 4. Can be used with oil, LPG, and natural gas as fuel. 5. First cost lower in small size units. 6. Can be used for recirculation as well as replacement. 	<ol style="list-style-type: none"> 1. First cost higher in large size units. 2. Turn down ratio is limited — 3:1 usual, maximum 5:1. 3. Flue or chimney required. Can be located only where flue or chimney is available. 4. Low efficiency (80%). Higher operating cost. 5. Can heat air over a limited range of temperatures. 6. Heat exchanger subject to severe corrosion condition. Needs to be checked periodically for leaks after a period of use. 7. Difficult to adapt to all combustion air from outdoors unless roof or outdoor mounted.

they are useful tools to compare the costs of various options in providing heating and cooling to an industrial ventilation system.

The following two equations may be used to estimate replacement air heating costs on an hourly and yearly basis. They are based on average usage schedules and typical weather conditions rather than worst case conditions and maximum usage.

Since there is an allowance for the efficiency of the replacement air unit, these equations will tend to give a low result if air is allowed to enter by infiltration only. They are also based on normal temperatures and moisture ratios and standard atmospheric pressure of 14.7 (101.4 kPa). Due to the heavy nature of the work in many industrial facilities, supply air may be cooler than for an office setting. Table 7-5 gives equation values (N) for supply air delivered at 70 F and 65 F. The humidity ratio (W) is assumed to be 0.01 pounds of water per pound of dry air.

$$C_1 = \text{Hourly cost} = 0.001 \frac{QN}{q} c \quad [7.1]$$

$$C_2 = \text{Yearly cost} = \frac{0.154 (Q)(dg)(T)(c)}{q} \quad [7.2]$$

where:

- Q = air flow rate, cfm
- N = required heat, BTU/hr/1000 cfm (Table 7-5 and Table 7-7)
- T = operating time, hours/week
- q = available heat per unit of fuel (Table 7-6)
- dg = annual degree days (Table 7-7)
- c = cost of fuel, \$/unit

EXAMPLE PROBLEM 1

Find the hourly and yearly cost of tempering 10,000 cfm of replacement air to 70 F in St. Louis, Missouri, using oil at \$1.35/gallon.

Average winter temperature = 31 F

$$\text{Hourly cost} = \frac{0.001 QN}{q} c$$

$$= 0.001 \times 10^4 \times \frac{42,000}{106,500} \times \$1.35 = \$5.32$$

$$\text{Yearly cost} = \frac{(0.154)(10^4)(6023)(40)}{106,500} \times \$1.35$$

$$= \$4,700 \text{ (assuming 40 hr/week of operation)}$$

The yearly cost is more representative because both the length and severity of the heating season are taken into account.

TABLE 7-5. Required Heat for Outside Air Temperatures

Avg. Outside Air Temperature, F	N, Required Heat, BTU/hr/1,000 cfm @ 70 F	N, Required Heat, BTU/hr/1,000 cfm @ 65 F
0	77,000	71,500
5	71,500	66,000
10	66,000	60,500
15	60,500	55,000
20	55,000	49,500
25	49,500	44,000
30	44,000	38,500
35	38,500	33,000
40	33,000	27,500
45	27,500	22,000
50	22,000	16,500
55	16,500	11,000
60	11,000	5,500
65	5,500	—

NOTE: Sensible Heat Equation used: $q = 1.1 \text{ (cfm) } \Delta t$. Humidity ratio is assumed to be 0.01 pounds of moisture per pound of dry air.

7.13 AIR CONSERVATION

The supply and exhaust of air represent both a capital cost for equipment and an operating cost which is often sizable in northern climates. Concerned designers, recognizing these cost and energy conservation needs, are unanimous in their desire for reduced ventilation rates.

There are four methods by which the cost of heating and cooling a large flow of outdoor air can be reduced: 1) reduction in the total flow of air handled, 2) delivery of untempered outdoor air to the space, 3) recovery of energy from the exhaust air, and 4) recovery of warm, uncontaminated air from processes. The successful application of these engineering methods without reduction in health hazard control and without impairing the implant environment requires careful consideration.

7.13.1 Reduced Flow Rate: A reduction of total air flow

TABLE 7-6. Available Heat per Unit of Fuel

Fuel	Btu Per Unit	Efficiency %	Available Btu Per Unit
Coal	12,000 Btu/lb 142,000 Btu/gal	50	6,000
Oil		75	106,500
Gas			
Heat Exchanger	1,000 Btu/ft ³	80	800
Direct Fired		90	900

TABLE 7-7. Heating Degree Day Normals and Average Winter Temperatures

City	Albany	Boston	Chicago	Cleveland	Detroit	Minneapolis	NY	Phila- delphia	Pitts- burgh	St. Louis	Wash., DC
Avg Temp (F) Dec-Feb	24	22.4	25	28	25.9	16	33.2	33.3	29	32.2	33.4
Discharge Air Temp (F)	Heating Degree Days										
80	11782	10409	10613	11343	10959	13176	9284	9652	10797	8943	8422
79	11425	10049	10277	10982	10605	12826	8937	9300	10436	8624	8089
78	11062	9690	9940	10621	10256	12478	8596	8954	10076	8310	7764
77	10709	9242	9610	10265	9914	12135	8265	8619	9726	8003	7446
76	10356	8994	9283	9915	9581	11797	7938	8285	9379	7702	7139
75	10009	8652	8972	9570	9247	11475	7620	7959	9036	7413	6835
74	9669	8317	8656	9229	8920	11142	7308	7641	8702	7121	6538
73	9333	7790	8349	8898	8599	10816	7004	7328	8372	6839	6250
72	9007	7668	8046	8567	8291	10496	6706	7028	8050	6560	5974
71	8682	7354	7750	8248	7981	10180	6421	6728	7740	6289	5703
70	8364	7046	7468	7928	7678	9870	6146	6438	7429	6023	5438
69	8256	6749	7183	7617	7383	9567	5871	6158	7127	5767	5179
68	7750	6458	6905	7313	7100	9269	5606	5886	6833	5523	4929
67	7452	6175	6635	7016	6816	8975	5349	5618	6546	5277	4690
66	7162	5903	6373	6722	6543	8687	5101	5360	6272	5053	4455
65	6881	5633	6122	6445	6278	8410	4858	5109	5997	4822	4229
64	6607	5370	5875	6165	6020	8131	4621	4864	5734	4595	4014
63	6340	5118	5638	5897	5772	7858	4394	4628	5483	4379	3798
62	3081	4873	5399	5636	5533	7590	4176	4397	5234	4168	3588
61	5829	4643	5164	5381	5290	7339	3957	4172	5006	3963	3383
60	5586	4399	4936	5140	5054	7086	3747	3952	4769	3761	3182

rate handled can be accomplished by conducting a careful inventory of all exhaust and supply systems in the plant. Determine which are necessary, which can be replaced with more efficient systems or hood designs, and which systems may have been rendered obsolete by changes.

Numerous hood designs presented in Chapter 10 are intended specifically to provide for adequate contaminant capture at reduced air flow rates. For instance, the use of horizontal sliding sash in the laboratory hood can provide a 30% saving in exhaust air flow rate without impairing capture velocity. The use of a tailored hood design, such as the evaporation hood shown in VS-35-40, provides good contaminant capture with far lower exhaust flow rates than would be required for a typical laboratory bench hood. Low Volume-High Velocity hoods and systems such as those illustrated in VS-

40-01 through VS-40-20 are used for many portable hand tool and fixed machining operations and can provide contaminant capture at far lower air handling requirements.

Throughout industry there are many applications of window exhaust fans and power roof exhausters to remove heat or nuisance contaminants which would be captured more readily at the source with lower air flow rates. Many roof exhausters, as noted earlier, have been installed initially to combat problems which were really caused by a lack of replacement air. When air supply and balanced ventilation conditions are established, their use no longer may be necessary.

Good design often can apply proven principles of local exhaust capture and control to reduce air flow rates with improved contaminant control.

7.13.2 Untempered Air Supply: In many industries utilizing hot processes, cold outdoor air is supplied untempered or moderately tempered to dissipate sensible heat loads on the workers and to provide effective temperature relief for workers exposed to radiant heat loads. The air required for large compressors, as well as for cooling tunnels in foundries, also can come directly from outside the plant and thus eliminate a load that is otherwise replaced with tempered air.

7.13.3 Energy Recovery: Energy recovery from exhaust air can be considered in two aspects: 1) the use of heat exchange equipment to extract heat from the air stream before it is exhausted to the outside and 2) the return (recirculation) of cleaned air from industrial exhaust systems. Heat exchanger application to industrial exhaust systems has been limited primarily by the ratio of installed cost to annual return.

Heat Exchangers — Air-to-air *heat exchangers* have been used to reduce energy consumption. This is achieved by transferring waste energy from the exhaust to replacement air streams of a building or process. The methods and equipment used will depend on the characteristics of the air streams. Major categories of equipment include heat wheels, fixed plate exchangers, heat pipes, and run-around coils.

A *heat wheel* is a revolving cylinder filled with an air permeable media. As the exhaust air passes through the media, heat is transferred to the media. Since the media rotates, the warm media transfers heat to the cooler replacement air. Special care is required to ensure that this transfer does not cause a transfer of contaminants.

A *fixed plate* exchanger consists of intertwined tunnels of exhaust and replacement air separated by plates (or sometimes a combination of plates and fins). The warm exhaust air heats the plates which in turn heat the cool replacement air on the other side of the plate. This exchanger uses no transfer media other than the plate forming wall of the unit.

A *heat pipe*, or thermo siphon, uses a pipe manifold with one end in the warm exhaust air stream and the other in the cool replacement air stream. The pipe contains a fluid which boils in the warm exhaust air stream extracting heat and condenses in the cool replacement air stream releasing heat. Thus the heat pipe operates in a closed loop evaporation/condensation cycle.

A *run-around coil* exchanger uses a pair of finned-tube coils. A fluid circulates through the coils extracting heat from the warm exhaust air releasing heat to the cool replacement air. An advantage of the run-around coil is that the exhaust and supply duct systems can be separated by a significant distance which results in a reduced potential for re-entry; usually less duct in the systems and usually less roof area occupied by the units.

Several factors are important in the selection of the appropriate heat exchanger. A partial list is as follows:

1. The nature of the exhaust stream. A corrosive or dust

laden stream may need to be precleaned.

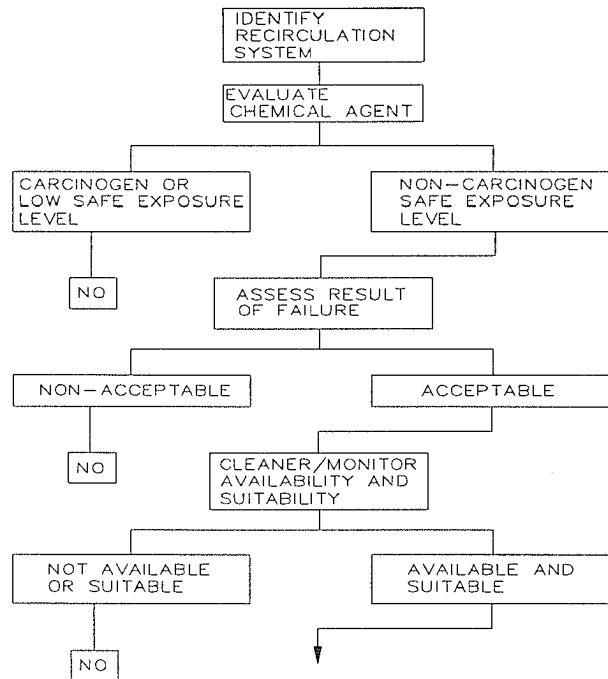
2. The need to isolate the contaminated exhaust stream from the clean replacement air stream.
3. The temperature of the exhaust stream. Unless the hot air stream is well above the desired delivery temperature of the replacement air stream and the exhaust air stream is at elevated temperatures whenever heat is demanded by the replacement air stream, additional heating capacity will be required.
4. Space requirements. Space requirements for some heat exchangers can be very extensive, especially when the additional duct runs are considered.
5. The nature of the air stream. Many exhaust air streams are corrosive or dirty and special construction materials may be required.
6. The need for a by-pass. During failure mode or summer conditions, a by-pass will be required.

Recirculation of Air from Industrial Exhaust Systems:

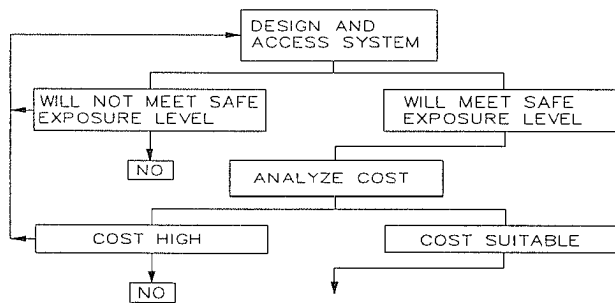
Where large amounts of air are exhausted from a room or building in order to remove particulates, gases, fumes, or vapors, an equivalent amount of fresh tempered replacement air must be supplied to the room. If the amount of replacement air is large, the cost of energy to condition the air can be very high. Recirculation of the exhaust air after *thorough* cleaning is one method that can reduce the amount of energy consumed. Acceptance of such recirculating systems will depend on the degree of health hazard associated with the particular contaminant being exhausted as well as other safety, technical and economic factors. A logic diagram listing the factors that must be evaluated is provided in Figure 7-14.^(7,6)

Essentially this diagram states that recirculation may be permitted if the following conditions are met:

1. The chemical, physical, and toxicological characteristics of the chemical agents in the air stream to be recirculated must be identified and evaluated. Exhaust air containing chemical agents whose toxicity is unknown or for which there is no established safe exposure level should not be recirculated.
2. All governmental regulations regarding recirculation must be reviewed to determine whether it is restricted or prohibited for the recirculation system under review.
3. The effect of a recirculation system malfunction must be considered. Recirculation should not be attempted if a malfunction could result in exposure levels that would cause worker health problems. Substances which can cause permanent damage or significant physiological harm from a short overexposure shall not be recirculated.
4. The availability of a suitable air cleaner must be determined. An air cleaning device capable of providing an



II. DESIGN AND ASSESSMENT



III. SYSTEM EVALUATION

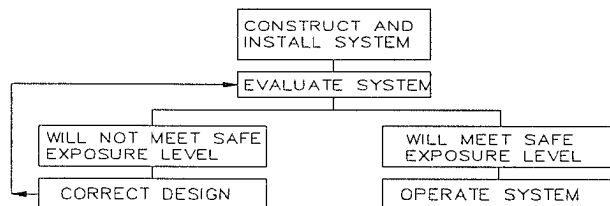


FIGURE 7-14. Recirculation decision logic

effluent air stream contaminant concentration sufficiently low to achieve acceptable workplace concentrations must be available.

5. The effects of minor contaminants should be reviewed. For example, welding fumes can be effectively removed from an air stream with a fabric filter; however if the welding process produces oxides of nitrogen, recirculation could cause a concentration of these gases to reach an unacceptable level.
6. Recirculation systems must incorporate a monitoring

system that provides an accurate warning or signal capable of initiating corrective action or process shut-down before harmful concentrations of the recirculated chemical agents build up in the workplace. Monitoring may be accomplished by a number of methods and must be determined by the type and hazard of the substance. Examples include area monitoring for nuisance type substances and secondary high efficiency filter pressure drop and on-line monitors for more hazardous materials.

While all system components are important, special consideration should be given to the monitor. The prime requisites are that the monitor be capable of sensing a system malfunction or failure and of providing a signal which will initiate an appropriate sequence of actions to assure that overexposure does not occur. The sophistication of the monitoring system can vary widely. The type of monitor selected will depend on various parameters (i.e., location, nature of contaminant — including shape and size — and degree of automation).

7.13.4 Selection of Monitors: The safe operation of a recirculating system depends on the selection of the best monitor for a given system. Reference 7.7 describes four basic components of a complete monitoring system which includes signal transfer, detector/transducer, signal conditioner, and information processor. Figure 7-15 shows a schematic diagram of the system incorporating these four components. It is quite likely that commercially available monitors may not contain all of the above four components and may have to be custom engineered to the need.

In addition, the contaminant must be collected from the air stream either as an extracted sample or in toto. If a sample is taken, it must be representative of the average conditions of the air stream. At normal duct velocities, turbulence assures perfect mixing so gas and vapor samples should be representative. For aerosols, however, the particle size discrimination produced by the probe may bias the estimated concentration unless isokinetic conditions are achieved.

The choice of detection methods depends on the measurable chemical and physical properties of the contaminants in the air stream. Quantifying the collected contaminants is generally much easier for particulate aerosols than for gases, vapors or liquid aerosols.

Particulates: Where the hazardous contaminant constitutes a large fraction of the total dust weights, filter samples may allow adequate estimation of concentration. Better, if the primary collector (e.g., bag filters, cartridge filters) allows very low penetration rates, it may be economical to use high efficiency filters as secondary filters. If the primary filter fails, the secondary filter not only will experience an easily measured increase in pressure drop, but will filter the penetrating dust as well — earning this design the sobriquet, "safety

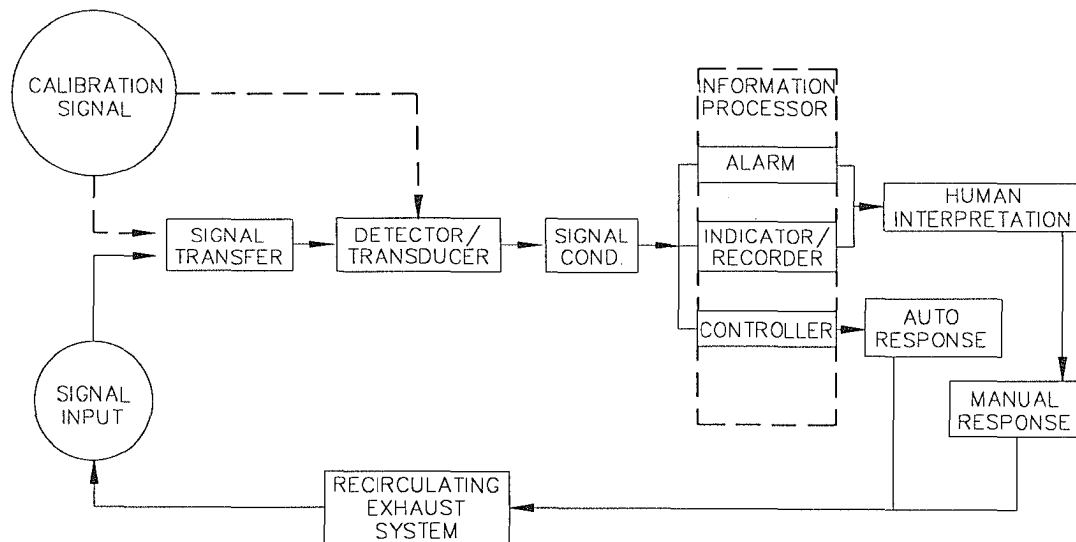


FIGURE 7-15. Schematic diagram of recirculation monitoring system

monitor" systems (see Figure 7-16).

Non-particulates: Continuously detecting and quantifying vapor and gas samples reliably and accurately is a complex subject beyond the scope of this manual.

Air Sampling Instruments for Evaluation of Atmospheric Contaminants, published by ACGIH,^(7,8) describes and evaluates different air monitoring devices. The monitor in a recirculating system must be capable of reliably monitoring continuously and unattended for an extended period of time. It must also be able to quickly and accurately sense a change in system performance and provide an appropriate warning if a preselected safety level is reached. In order to function properly, monitors must be extremely reliable and properly maintained. Monitors should be designed so that potential malfunctions are limited in number and can be detected easily by following recommended procedures. Required maintenance should be simple, infrequent, and of short duration.

7.14 EVALUATION OF EMPLOYEE EXPOSURE LEVELS

Under equilibrium conditions, the following equations may be used to determine the concentration of a contaminant permitted in the recirculation return air stream:

$$C_R = \frac{(1-\eta)(C_E - K_R C_M)}{1 - [(K_R)(1-\eta)]} \quad [7.3]$$

where:

C_R = air cleaner discharge concentration after recirculation, mg/m^3

η = fractional air cleaner efficiency

C_E = local exhaust duct concentration before recirculation, mg/m^3

K_R = coefficient which represents a fraction of the recirculated exhaust stream that is composed

of the recirculation return air (range 0 to 1.0)

C_M = replacement air concentration, mg/m^3

$$C_B = \frac{Q_B}{Q_A} (C_G - C_M) (1-f) + (C_O - C_M) f + K_B C_R + (1-K_B)(C_M) \quad [7.4]$$

where:

C_B = 8-hr TWA worker breathing zone concentration after recirculation, mg/m^3

Q_B = total ventilation air flow before recirculation

Q_A = total ventilation air flow after recirculation

C_G = general room concentration before recirculation, mg/m^3

f = coefficient which represents the fraction of time the worker spends at the work station

C_O = 8-hr TWA breathing zone concentration at work station before recirculation

K_B = fraction of worker's breathing zone air that is composed of recirculation return air (range 0 to 1.0)

The coefficients K_R , K_B , and f are dependent on the work station and the worker's position in relation to the source of the recirculation return air and the worker's position in relation to the exhaust hood. The value of K_R can range from 0 to 1.0 where 0 indicates no recirculation return air entering the hood and 1.0 indicates 100% recirculation air entering the hood. Similarly, the value of K_B can range from 0 to 1.0 where 0 indicates there is no recirculation return air in the breathing zone and 1.0 indicates that the breathing zone air is 100% recirculated return air. The coefficient "f" varies from 0 where the worker does not spend any time at the work station where the air is being recirculated to 1.0 where the worker spends

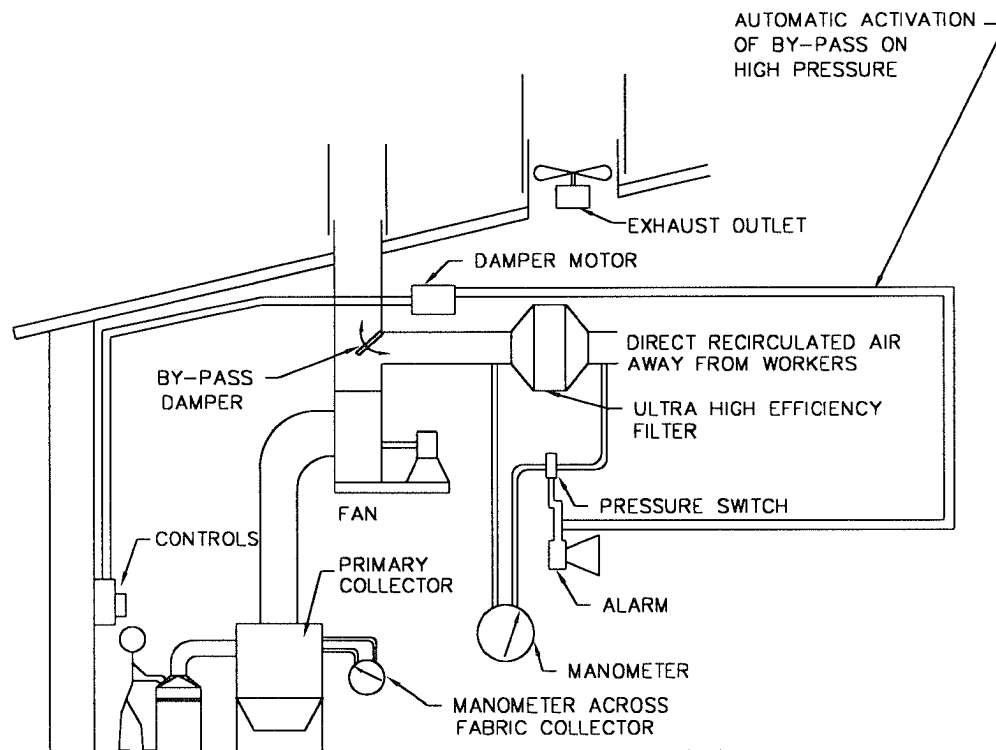


FIGURE 7-16. Schematic of recirculation from air cleaning devices (particulates)

100% time at the work station.

In many cases it will be difficult to attempt quantification of the values required for solution of these equations for an operation not yet in existence. Estimates based on various published and other available data for the same or similar operations may be useful. The final system must be tested to demonstrate that it meets design specifications.

An example of use of Equations 7.3 and 7.4 and the effect of the various parameters is as follows:

Consider a system with 10,000 cfm total ventilation before recirculation (Q_B) consisting of 5,000 cfm of general exhaust and 5,000 cfm of local exhaust. The local exhaust is recirculated resulting in 10,000 cfm after recirculation air flow consisting of 5,000 cfm recirculated and 5,000 cfm fresh air flow.

Assume poor *placement* of the recirculation return (K_R and $K_B = 1$) and that the worker spends all his time at the work station ($f = 1$); the air cleaner efficiency (η) = 0.95; exhaust duct concentration (C_E) = 500 ppm; general room concentration (C_G) = 20 ppm; replacement air concentration (C_M) = 5 ppm; work station (breathing zone) concentration before recirculation (C_O) = 35 ppm; and a contaminant TLV of 50 ppm.

Equation 7.3 gives recirculation air return concentration:

$$C_R = \frac{(1 - 0.95)(500 - 1 \times 5)}{1 - [(1)(1 - 0.95)]} = 26.1 \text{ ppm}$$

Equation 7.4 gives the worker breathing zone concentra-

tion:

$$\begin{aligned} C_B &= \frac{Q_B}{Q_A} (C_G - C_M) (1 - f) + (C_O - C_M) f + K_B C_R \\ &+ (1 - K_B)(C_M) \\ &= \left(\frac{10^4}{10^4} \right) (20 - 5) (1 - 1) + (35 - 5) (1) + (26.1) \\ &+ (1 - 1) (5) \\ &= 56.1 \text{ ppm} \end{aligned}$$

Obviously, 56.1 ppm exceeds the TLV of 50 ppm and therefore is unacceptable.

In order to achieve lower concentrations (C_B), the system configuration must be redesigned so that only 50% of the recirculation return air reaches the work station. Thus, K_R and K_B are reduced to 0.5. Substituting these new data in Equation 7.4, the breathing zone concentration calculates as 45.3 ppm. This is lower than the TLV of 50 ppm and therefore acceptable.

Several potential problems may exist in the design of recirculated air systems. Factors to be considered are:

1. Recirculating systems should, whenever practicable, be designed to bypass to the outdoors, rather than to

recirculate, when weather conditions permit. If a system is intended to conserve heat in winter months and if adequate window and door openings permit sufficient replacement air when open, the system can discharge outdoors in warm weather. In other situations where the work space is conditioned or where mechanically supplied replacement air is required at all times, such continuous bypass operation would not be attractive.

2. Wet collectors also act as humidifiers. Recirculation of humid air from such equipment can cause uncomfortably high humidity and require auxiliary ventilation or some means must be used to prevent excess humidity.
3. The exit concentration of typical collectors can vary with time. Design data and testing programs should consider all operational time periods.
4. The layout and design of the recirculation duct should provide adequate mixing with other supply air and avoid uncomfortable drafts on workers or air currents which would upset the capture velocity of local exhaust hoods.
5. A secondary air cleaning system, as described in the example on particulate recirculation, is preferable to a monitoring device because it is usually more reliable and requires a less sophisticated degree of maintenance.
6. Odors or nuisance value of contaminants should be considered as well as the official TLV values. In some areas, adequately cleaned recirculated air, provided by a system with safeguards, may be of better quality than the ambient outdoor air available for replacement air supply.
7. Routine testing, maintenance procedures, and records should be developed for recirculating systems.
8. Periodic testing of the workroom air should be provided.
9. An appropriate sign shall be displayed in a prominent place reading as follows:

CAUTION

AIR CONTAINING HAZARDOUS SUBSTANCES IS BEING CLEANED TO A SAFE LEVEL IN THIS EQUIPMENT AND RETURNED TO THE BUILDING. SIGNALS OR ALARMS INDICATE MALFUNCTIONS AND MUST RECEIVE IMMEDIATE ATTENTION: STOP RECIRCULATION, DISCHARGE THE AIR OUTSIDE, OR STOP THE PROCESS IMMEDIATELY.

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Chapter 8

VENTILATION ASPECTS OF INDOOR AIR QUALITY

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8.1 INTRODUCTION

There are two ventilation aspects which are major causes of the complaints noted in the vast majority of reported problems from all parts of this and other countries. They are complaints of unsatisfactory indoor air quality (which may be due to the lack of sufficient outdoor air for dilution of "normal" indoor airborne contaminants) and the failure to deliver supply air properly to the occupied zones.

Indoor air quality is defined as the overall quality of the indoor air and includes biological, chemical, and comfort factors. This chapter is designed to familiarize the reader with heating, ventilation, and air conditioning (HVAC) systems used in office and similar spaces. The individual components of a typical HVAC system are defined, and the operation of the more common types of HVAC systems found are discussed.

8.2 DILUTION VENTILATION FOR INDOOR AIR QUALITY

The oil shortage and the resulting energy crisis of the late 1960s and early 1970s is considered by some as the most significant cause of the current indoor air quality concern. In the past, when energy costs were relatively low, the design of heating, ventilation and air conditioning (HVAC) systems for buildings included the infiltration of outside air through doors, windows, and other sources. Also, up to 25%^(8.1) outdoor air was supplied by the system, in addition to the infiltration, for general ventilation purposes. The outdoor air had the effect of diluting the "normal" indoor contaminants to a very low level of concentration, which had little effect on the occupants.

Since the energy crisis resulted in major increases in energy costs, an extensive effort was made to reduce the infiltration of outdoor air by constructing the building as airtight as possible. Outdoor air supplied by the HVAC system was reduced to a minimum and in some instances eliminated entirely. Airborne contaminants found in indoor environments were present in extremely small quantities and had not been a health problem in the past due to the dilution effect of the outdoor air. New concepts of office design that utilize fabric partitions, particle board furniture, increased use of carpets, office copy machines, etc., have increased the potential for indoor contaminants. As buildings became more energy efficient, there was an increase in complaints of stuffiness, drowsiness, tiredness, eye irritation, throat irritation, and stale air.

Existing health standards are not usually violated by the low-level concentrations, and the only current legal requirement for outdoor air is found in the building codes. The Uniform Building Code^(8.2) is the most widely accepted standard for providing outdoor air. Section 605 states that 5 cfm of outdoor air per occupant shall be mechanically supplied to all parts of the building during occupancy. Carbon dioxide concentrations from occupant respiration within a space are often

used as an indicator of the quantity of outdoor air being supplied to that space. When the indoor air concentration reaches approximately 800–1000 ppm (excluding external combustion sources), complaints may escalate. As the carbon dioxide levels increase, the number of complaints will increase more rapidly.

In 1989, the American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE) developed and adopted ASHRAE Standard 62-1989, "Ventilation for Acceptable Indoor Air Quality."^(8.3) The standard recognized the health problems resulting from the changes in construction and HVAC methods. It is based on occupancy of spaces and provides the outdoor air requirement for that space. Requirements for outdoor air for offices based on an occupancy of seven people per 1000 square feet is currently 20 cfm per person. This is based on a total occupancy, including transients, and is in addition to the usual HVAC requirements. The standard is expected to satisfy the requirements for 80% or more of the occupants.

Provision for delivery of the outdoor air for dilution of the normal indoor airborne contaminants in the occupied space is a major factor of indoor air quality considerations. It is obvious that if the outdoor air included as part of the total supply air is not delivered to the occupied zone, the potential for unsatisfactory indoor air quality increases. Another important factor in the delivery of the air to the occupied zone is the location of the supply and return air grilles to avoid short-circuiting. Ideally, the supply air diffusers and the air grilles are so located that a uniform flow of air through the space occurs to avoid both stagnant air and drafts.

Temperature and humidity can play a role in how people perceive indoor environment. ASHRAE Standard 55-1992^(8.4) provides guidance in design and maintenance of indoor thermal environments. ASHRAE recommends temperature ranges of 67 to 76 F in winter (heating season) and 72 to 81 F in summer (cooling season). However, complaints may increase when temperatures rise above 74 F. Similarly, it is preferable to keep relative humidities above 20–30% during the heating season and below 60% during the cooling season. ASHRAE also suggests limits on air movement. The average air movement in an occupied space should not exceed 30 fpm in winter or 50 fpm in summer.

8.3 HVAC COMPONENTS AND SYSTEM TYPES

When considering the ventilation aspects of HVAC systems, the type of system and its components should be reviewed for potential sources or causes of complaints regarding indoor air quality. Detailed descriptions of the systems and components can be found in the *Systems and Equipment* volume of the ASHRAE Handbook.^(8.5)

8.3.1 Components: The components that make up HVAC systems generally include the following:

- A. **HVAC System:** HVAC system refers to the equipment and distribution system used for heating, ventilating, cooling, humidifying, dehumidifying, and cleansing air for a building or building zone for the purpose of comfort, safety, and health of the occupants.
 - B. **Dampers:** Dampers are devices of various types used to vary the volume of air passing through an outlet, inlet, or duct.
 - C. **Outdoor Air (Fresh Air; Replacement Air; Compensating Air):** Outdoor air used to replace all or part of the air in a building or building space.
 - D. **Return Air:** Return air is air that has been in the building for a period of time and is returned to the HVAC system. Varying percentages of return air are exhausted outdoors with the remaining air (recirculated air) mixed with outdoor air for conditioning and distribution.
 - E. **Mixing Plenum:** A mixing plenum is a chamber within an HVAC system where outdoor air is mixed with returned air. The mixed air, after cleaning and conditioning, comprises the supply air for the building.
 - F. **Air Cleaners:** Air cleaners are devices designed to remove atmospheric airborne impurities such as dusts, gases, vapors, fumes, and smoke. (Air cleaners include air washers, air filters, electrostatic precipitators and charcoal filters.)
 - G. **Heating Coils:** Heating coils are heat transfer devices which utilize hot water, steam, or electricity to heat the supply air.
 - H. **Cooling Coils:** Cooling coils are heat transfer devices which utilize chilled water or a refrigerant to cool the supply air.
 - I. **Condensate Pan (Drip Tray; Defrost Pan):** A vessel or tray under the cooling coil to receive water extracted from the supply air by condensation from the cooling coil.
 - J. **Humidifier/Dehumidifier:** Humidifier/dehumidifiers are devices to add/remove moisture to/from the supply air.
 - K. **Fans (Supply and Return):** Fans are devices for moving ventilation air through the HVAC system.
 - L. **Supply Air:** Supply air is conditioned ventilation air delivered to zones within a building.
 - M. **Control Zone:** Control zone is a space or group of spaces within a building served by an HVAC system. Depending on the space requirements, the control zone may be designated as core or interior zone and/or perimeter zone.
 - N. **Occupied Zone:** The occupied zone is the region within an occupied space between 3 and 72 inches above the floor.
 - O. **Control Box (Variable Air Volume, Bypass, Dual Duct):** Control boxes are devices to which the supply air may be delivered by the HVAC system prior to delivery to the supply diffuser. These boxes may include means of controlling supply air temperature and volume to the diffuser or multiple diffusers within a HVAC zone.
 - P. **Supply Air Diffusers:** Supply air diffusers are devices whose function is to deliver the supply air to the occupied zone and to provide a desired distribution pattern. The diffusers may be circular, square, rectangular, linear slots, louvered, fixed, adjustable, or a combination.
 - Q. **Return Air Grilles:** Return air grilles may be louvered or perforated coverings for openings located in the sidewall, ceiling or floor of a zone through which the return air enters. The return air grilles may be directly connected to an open return air plenum or to a ducted return air system.
 - R. **Return Air Plenum:** A return air plenum is the space usually located above the ceiling where the return air is collected from a zone prior to entering the return air system.
 - S. **Economizer:** An economizer is a control system which reduces the heating and cooling load through the use of outdoor air for free cooling when the total heat of the return air exceeds the total heat of the outdoor air.
- 8.3.2 Types of Systems:** There are different types of HVAC systems: single-duct systems, dual-duct systems, multi-zone systems, and special systems. These systems may be considered basic and subject to variations that are necessary to meet specific requirements. The following descriptions of the basic systems are intended as a guide and the referenced ASHRAE Handbook should be reviewed for system details and variations.
- Single-Duct Systems** may be either a constant or a variable air volume system. The constant volume system maintains constant air flow with the temperature of the supply air controlled in response to the space load. See Figures 8-1 and 8-2. The system may be a single zone, a zoned reheat, multiple-zone modification or a by-pass variation using a by-pass box in lieu of reheat constant volume primary system with a variable air volume secondary system. A variable air volume (VAV) system controls the temperature within a zone by varying the supply air volume. See Figures 8-3, 8-4, and 8-5. This type of system may include reheat at the terminals, induction unit, fan-powered distribution box, dual conduit, and variable diffusers.

Dual-Duct Systems condition all the air in a central apparatus and distribute it to the conditioned zones through two parallel mains, one carrying cold air and the other warm air. The system may be a constant volume type single fan and with or without reheat capability. See Figure 8-6. Also, the system may be VAV which mixes the cold and warm air in various volume combinations depending on the zone load. In both system types, the cold and warm air is delivered to a dual duct box which mixes the air prior to delivery to the supply air diffuser.

Multizone Systems supply several zones from a centrally located HVAC unit. Supply air for the different zones consists of mixed cold and warm air through zone dampers in the central HVAC unit in response to zone thermostat control. From there, the supply air is distributed through the building by single zone ducts which, in turn, supply the air to the zone diffusers. See Figure 8-7.

Fan Coil Units are usually located along the outdoor wall of a building for heating and cooling the perimeter up to 15 feet from the outdoor wall. These units may have a through-wall duct for outdoor air and can be totally self contained or have the heating and cooling media supplied from a central mechanical room. See Figure 8-8. Controls for temperature and operation will vary although control of the outdoor air is usually at the unit and beside the nearest occupant.

Zone Heat Pumps are packaged HVAC units that may provide the heating and cooling for individual zones within a building. These units vary in how the heating and cooling media is provided, but the function is generally constant (see Figure 8-9). Also, these units may be located within the individual zone above the ceiling in the return air space or remotely such as on the building roof. The supply air is delivered to the entire zone through a duct distribution and diffuser system. Return air for a pump located in the building enters the return air plenum above the ceiling due to zone pressure and migrates to the unit for reconditioning. For the remote unit, the return air is ducted from the ceiling plenum or from return air grilles to the unit. Outdoor air for interior units may be provided by a separate system and delivered to the return air plenum above the ceiling. Some building codes require that the outdoor air be directly supplied to the interior units. For the remote unit located on the roof, the outdoor air may be provided by the unit on the return air side through a damper that usually is set manually.

8.4 HVAC COMPONENTS, FUNCTIONS, AND MALFUNCTIONS

8.4.1 Outdoor Air: The outdoor air requirement for a space or an entire building must satisfy the need for acceptable indoor air quality and the need to replace air removed from the space or building by process or other exhaust. For indoor environment, ASHRAE Standard 62-1989, "Ventilation for Acceptable Indoor Air Quality,"^(8.3) is the accepted design

criteria. Replacement air, however, will depend on factors such as total exhaust volume and pressure differential requirements of the space or building plus the evaluation of potential airborne contaminants that may be generated inside or outside the building. For example, in the "open concept" type of office layout where partitions approximately five feet high enclose office spaces, the supply air has a tendency to ventilate only the space between the partitions and the ceiling. Very little, if any, of the supply air enters the actual occupied space directly to provide the necessary dilution. This allows contaminants in the occupied space to increase in concentration resulting in the potential for unsatisfactory air quality complaints.

ASHRAE Standard 62-1989 recommends the measurement and documentation of the outdoor air intake volumetric flow rate on all configurations of HVAC systems. The primary purpose of this requirement is to control the level of carbon dioxide, human odors, and the normal airborne contaminants generated within the space. The published ventilation rates are based on occupancy or space usage and on an assumed occupant density. If the occupant density increases or the space usage increases, a degradation of the indoor air quality will occur which, in turn, will require an increase in outdoor air.

It is possible to estimate the percentage of outdoor air by equation using the return, outdoor, and mixed air temperatures. The percentage of outdoor air also can be determined by equation using the carbon dioxide concentrations in the same air flow areas. Using these results, the estimated volumetric flow rate of the outdoor air can be determined. The equations are as follows:

Temperature Method:

$$\% \text{ outdoor air} = \frac{t_{RA} - t_{MA}}{t_{RA} - t_{OA}} \times 100$$

where:

t_{RA} = temperature, return air

t_{MA} = temperature, mixed air

t_{OA} = temperature, outdoor air

Carbon Dioxide Method:

$$\% \text{ outdoor air} = \frac{\text{ppm/RA} - \text{ppm/MA}}{\text{ppm/RA} - \text{ppm/OA}} \times 100$$

where:

ppm/RA = CO₂ concentration, return air

ppm/MA = CO₂ concentration, mixed air

ppm/OA = CO₂ concentration, outdoor air

Given the legal implications, the direct measurement and documentation of the outdoor air volumetric flow rate is recommended.

TABLE 8-1^(8.6) Relationships among extent of complaints regarding indoor air quality, CO₂ levels, and outdoor air ventilation rates

COMMENTS	CO ₂ (ppm)	Outdoor Air Ventilation Rate/Person	
		CFM	L/s
Occasional complaints, particularly if the air temperature rises	600	35	16.5
Complaints are more prevalent	800	21	10
Insufficient replacement air, complaints more general	1000	15	7

The concentration of carbon dioxide within a space may provide a good indication of the outdoor air being delivered to the space. A study conducted by the Ontario Inter-Ministerial Committee on Indoor Air Quality reported on the relationship between levels of complaints, carbon dioxide concentrations, and the outdoor air ventilation rates.^(8.6) The results are indicated in Table 8-1.

Location of outdoor air intake may be found on the building roof, sidewall, at ground level, or possibly at all three locations for very large building complexes. Figure 5-28, "Air Flow Around Buildings," clearly illustrates the potential for airborne contaminants to enter the building through any of the outdoor air intakes. Sources of potential airborne contaminants from the building and from sources remote or adjacent to the building should be thoroughly investigated. Assistance with this investigation should be requested of industrial hygiene and environmental organizations who have responsibilities for the building. The location of the outdoor air intakes will be affected by the atmospheric air flow over the building as will the location and height of any exhaust stacks. Criteria for the atmospheric air flow characteristics and stack heights may be found in Figures 5-28, 5-29, and 5-30 of this manual or in the *Fundamentals* volume of the 1993 ASHRAE Handbook, Chapter 14, "Airflow Around Buildings."^(8.7)

Roof intakes generally are located within a few feet of the roof surface. Standing water on the roof from weather conditions, HVAC equipment drains, or other sources present the potential for biological growth and entry into the intake. Unless discharged vertically above the recirculation region (see Figure 5-28), building exhaust systems from restrooms, processes within the building, and restaurant kitchens have significant potential for re-entry.

Building sidewall intakes have the potential for entry of airborne contaminants from street level automotive traffic, shipping and receiving docks, and adjacent buildings. In the building wake region, see Figure 5-28, the potential for re-entry increases for the outdoor air intakes and open windows or doors since the pressure in the recirculation region is lower

than the surrounding area. Airborne debris such as leaves, paper, and atmospheric dirt may tend to collect on the intake bird screens which can reduce the intake area and may reduce the flow rate into the intakes.

Ground level outdoor air intakes are possibly the least desirable location of the three described. This location offers the potential for air quality problems caused by standing water, automotive emissions, and as a collection point for dirt and debris. The security of a building can be compromised through the ground level intakes by the deliberate addition of foreign materials.

8.4.2 Dampers: A typical office building HVAC system will include outdoor and return air dampers. Air flow through these dampers will vary over a wide range depending on the damper opening settings and the space or building requirements. Indoor air quality problems often result if the outdoor air damper is not designed or adjusted to allow introduction of sufficient outdoor air for the current use of the building. Outdoor air requirements for acceptable indoor air quality indicate that the actual volumetric flow rate through the damper sections be monitored. When the outdoor air and return air dampers are combined in an HVAC system, there may be an imbalance in volumetric flow rates.

It is customary to report the volumetric air flow rate through dampers in terms of damper opening. However, damper opening is not linearly proportionate to volumetric flow rate. Another misconception regarding dampers is that a "closed" damper will leak approximately 10%. Closed dampers may not leak at all.

Dampers are mechanical devices (either parallel or opposed blade, see Figure 8-11) that require routine maintenance and periodic settings checks to assure proper air flow passage. Actuators, connecting arms and damper bearings are components that can affect the air flow if not properly connected or adjusted. This is an area where potential problems affecting the indoor air quality are not uncommon. In older buildings, the practice of disconnecting the outdoor air dampers to conserve energy is fairly common. This practice has been found by current surveys in some older buildings and also in newer buildings which have been occupied for an extended period. The result, of course, is a significant potential source of unsatisfactory indoor air quality complaints.

8.4.3 Air Cleaning: The requirements for air cleaning vary according to the space or building requirements. There are, however, some basic factors that should be included in the design of the HVAC filter section. It is considered essential that all ventilation supply air, including outdoor and recirculated air, pass through a prefilter and a high efficiency final filter. Depending on requirements, filters such as charcoal, potassium permanganate, HEPA, and others may be specified. See Figure 8-12 for various types.

For example, paper dust is one of the contributors to unsatisfactory indoor air quality. The paper dust in itself is an irritant to the eyes and respiratory system. Also, many papers are chemically treated which tends to compound the irritant effect. The dust generated enters the return air section of the HVAC system, and may be reintroduced to the space being served by the system. Studies^(8.8-8.10) have indicated that a significant percentage of the paper dust will be removed by high efficiency type filters which, as stated, should be included in the HVAC filter section. Figure 8-10(8.5) shows approximate efficiency versus particle size for typical air filters.

In some older HVAC equipment, also in perimeter fan coil units and self-contained heat pumps, low efficiency filters are noted. It may be possible to replace these filters with medium efficiency filters of the same dimensions. The medium efficiency pleated filter has more than twice the filtering area, thereby increasing the interception of the airborne particulates without a significant increase in the static pressure requirements.

The air cleaning or filter section of the HVAC system requires routine maintenance for replacing dirty filters, or in some instances, cleaning a reusable type of air cleaner. Routine maintenance would also bring attention to damaged filters or filter frames and uneven air flow (by the dirt pattern on the face of the filters.) Even though a regular maintenance program may be in force, the filter sections, both pre- and final filters, present a potential source for indoor air quality complaints.

Some HVAC systems utilize self-contained heat pumps to control conditions in specific zones. These heat pumps can be located above the ceiling in the return air plenum near the zone being served. This location results in a difficult situation in terms of providing service for the unit. The zonal heat pump usually has a low efficiency filter which can be completely blocked or missing due to the difficulty of servicing.

8.4.4 Heating/Cooling Coils: Heating and cooling coils must be free of damage, especially the heat transfer fins. Irregularities in the fins will result in unequal heat transfer and will provide an area for dirt and other materials to accumulate. Air cleaning sections are not 100% perfect in removing the airborne contaminants regardless of efficiency ratings. Convenient access to the coil section for inspection, cleaning and maintenance is essential to the proper functioning of the coil. See Figure 8-13.

Cooling coils require some additional considerations. The supply air will pass through at a relatively low velocity, and the heat transfer will condense moisture on the coil fins. This moisture will drain to the condensate pan below the coil. Provision must be made to properly discharge the condensate. Since the air cleaning section is not perfect, some airborne contaminants will reach the cooling coils. The moisture accumulating on the coil fins will collect a significant percentage of these contaminants, which may adhere to the fins or drain

to the pan below with the condensate. Accumulations of these contaminants create a source of molds, spores, bacteria, etc., that may enter the supply air stream. The condensate pan drain may allow condensate to accumulate at or near the outdoor air intake and can re-enter the HVAC system. It is essential that the cooling coil condensate pan be properly drained. The pan drain must be directed away from any outdoor air intake. The coil and pan must be inspected and cleaned on a regular basis. Microorganisms may proliferate if this is not done.

8.4.5 Fans: HVAC systems vary in size and complexity over a wide range as do the fans as the system prime mover of volumetric flow for both supply, outdoor and return air. The fans may be the axial or centrifugal type with inlet vanes, outlet dampers, variable speed, direct or belt drive (see Figure 8-14). Also, the fan or fans may be inside the housing of a self-contained HVAC unit or a separate component in a mechanical room or penthouse. See Figure 8-16 for a typical layout of the self-contained and mechanical room system.

Since the fan is the prime mover of the HVAC system, a preventive maintenance program usually will reveal any potential malfunctions before they occur. Failures of the fan are usually noted immediately and corrected by maintenance. There is, however, a maintenance procedure (lubrication of moving components) that may be a source of odor complaints by the building occupants. Over lubrication, which is not an uncommon practice, may place a small quantity of the lubricant in the air flow into the fan. This may cause the blades to become coated and the lubricant odor to be carried into occupied areas.

8.4.6 Humidifiers/Dehumidifiers: The incorporation of humidifiers/dehumidifiers is dependent on the space requirements of the building. Humidifiers add moisture to the supply air by direct water or by steam spray. Dehumidifiers remove moisture from the supply air by a desiccant-type filter or by cooling coils. Of the two processes, humidification is more widely used in HVAC systems (see Figure 8-15). The equipment used for humidification has a reputation for requiring a high level of maintenance for proper operation. For this reason, it is fairly common to find that the humidifier has been shut off — especially in office buildings.

Both humidifying and dehumidifying are associated with water and dampness. This association presents the potential for the growth of molds, spores, bacteria, etc., that may enter the supply air flow. Proper drainage of any water or moisture generated by either process is essential.

8.4.7 Supply Air Distribution: The air supply distribution system should be through sheet metal, steel and aluminum, or some type of non-fibrous duct material. There is increasing concern that fibrous materials such as fiberglass board ducts may produce fibers that may be potentially harmful. The use of interior duct insulation should also be avoided to eliminate the possibility of fibers entering the supply air stream. Supply

air duct systems should be designed in accordance with accepted standards as detailed in current publications such as the ASHRAE Handbook series, standards of the Sheet Metal and Air Conditioning Contractors National Association (SMACNA), National Fire Protection Association, and other applicable criteria sources.

The physical condition of the air supply duct system is important in the overall evaluation. Duct systems are usually located above the ceiling in the return air space together with utility lines, sprinkler lines, computer cables, etc. This space is relatively small, and when repairs, rearrangements, installations, etc., occur, damage to the duct system may not be noticed but may affect the air distribution.

It is common practice to connect the supply air duct to mixing boxes and/or diffusers with flexible duct. The frictional resistance can be up to five times that of sheet metal, and the manufacturer's data should be reviewed. Bends and turns using flexible duct will compound the losses and have a tendency to reduce the cross-section, which may in turn reduce the air volume. Improper hangers and supports also have the same tendency and results. The use of flexible ducts should be limited to minimum lengths, properly supported and securely fastened at each end.

8.4.8 Supply Air Diffuser: The function of the supply air diffuser is to deliver and distribute the supply air throughout the occupied zone. Diffusers are available in a wide variety of types, shapes, and sizes — all of which will provide the proper air volume according to the supplier. See Figure 8-17 for illustrations of various types of diffusers. The suppliers or manufacturers usually rate the diffusers in terms of supply air volume, static pressure drop, and the "throw" or pattern of the air delivery. Also, the published data will include illustrations of the air flow pattern created by the diffuser in a totally empty space and rely on the "coanda effect" for mixing the air in the space. However, when the space is occupied by people, equipment, file cabinets, cubicle partitions, library shelves, etc., the supply air pattern changes dramatically. This results in less supply air to the occupied zone. This ventilation aspect can be easily recognized through the use of a simple hand-held smoke test.

Some of the more common problems associated with diffusers are as follows:

1. Variable air volume HVAC systems with fixed-supply air diffusers vary the air flow rate depending on temperature demand. Even if the minimum outdoor air is provided at all times, the reduced flow rate through the diffusers will reduce the throw and flow pattern. This may result in some areas within the occupied zone receiving little or no supply air. There are diffusers that automatically adjust for reduced air flow rates to maintain a constant throw and flow pattern utilizing the "coanda effect." Reports from the field vary over a wide

range. Some reports state that technical maintenance is relatively high to assure proper functioning. Others report that the pressure required increases as the slot area decreases which decreases the air flow rate.

2. Variable air volume systems may include constant air flow to the diffusers through terminal boxes serving specific zones. These boxes contain an air supply fan with sensors and controls that draw air from the return air system or the specific zone based on the flow rate from the main system. Even though this will assure a constant air flow rate to the diffuser, it also presents some potential problems by localized recirculation within the specific zone.
3. Supply air diffusers with fixed blades, diffusers covered by a perforated plate, linear fixed diffusers, and fluorescent light troffers direct the supply air across the ceiling depending on the coanda effect for delivery to the occupied zone. In the "open concept" office layout with 5-ft high partitions enclosing office spaces, the supply air from the diffusers described will have a tendency to provide continuous supply only to the space between the partitions and the ceiling. See Figure 8-18. Very little if any of the supply air enters the occupied zone directly which may result in complaints. This particular ventilation aspect may occur even though the system is providing 100% outdoor air and is often referred to as "short circuiting." The flow pattern above the partitions can usually be observed by using the simple smoke test. Another test that is done which may give a more qualitative result is the measurement of the carbon dioxide concentrations in the occupied zone and in the space between the partitions and ceiling. A concentration in the occupied zone that is significantly higher than the concentration above the partitions indicates that possibly up to 75% of the supply is above the partitions.
4. Supply air diffusers with adjustable blades are available in the multi-directional ceiling type, linear diffusers with adjustable T-bars, sidewall supply grilles, and other types. The adjustable feature does offer a means of better directing the supply air to the occupied zone. However, locations of the diffusers and the adjustment of the blades is critical to the distribution of the supply air. Improper adjustment may result in complaints by the occupants of excessive drafts.

Location and type of supply air diffusers should be such that a continuous flow of air through the occupied space will occur at all times. Avoid situations that result in localized recirculation or short circuiting to the return air system. In general, the air flow pattern through a space by the supply air should receive critical attention and can be characterized in terms of ventilation efficiency. Two efficiencies should be considered: system efficiency and ventilation efficiency. Sys-

tem efficiency is defined as the ratio of the actual volumetric flow rate to a specific space to the design volumetric flow rate for that specific space. Ventilation efficiency is defined as the ratio of the actual volumetric flow rate to a specific occupied zone to the design volumetric flow rate for that specific occupied space. Location and type of supply air diffusers are critical in the development of good ventilation efficiency. Design criteria in ASHRAE Standard 62-1989 will assist the design engineer in this effort.

8.4.9 Return Air Grilles: The return grilles have the function of receiving or exhausting air from a space through the return air system. Also, it is the function of the return air grilles to enhance the flow of the supply air through the space. The size and number of return grilles must be such that 100% of the supply air can be returned to the return air system. Location of the return air grilles influences the air flow pattern through the space and proper location will minimize localized recirculation zones.

There is little design data available on the placement of the return air grilles but the location should be considered as important as the location of the supply air diffusers. The short-circuiting of the supply air directly to the ceiling return grilles may result in less than 50% of the supply air reaching the occupied zone. Development of an air flow pattern through an occupied zone from the supply diffusers to the return air grilles is a primary consideration.

8.4.10 Return Air: The return air system may be either an open plenum type or a ducted system, both of which are typically located above the ceiling. In the return system, a static pressure balance between return air points must be part of the system design. It is obvious that the open ceiling plenum cannot be balanced by design which accounts for difficulty in providing a balanced supply air volume. For ducted return, the approach is similar to an industrial exhaust system. The static pressure in each run must be balanced by design at their junction which also accounts for difficulty in providing a balanced supply air volume.

Pressure differentials at any junction are limited to 20% which is the maximum correction possible by damper. For differentials over this limit, redesign is necessary. Flexible duct is used at times to connect return air grilles to the ducted return. Since the negative pressure will tend to collapse the flexible duct, this practice should be avoided.

8.4.11 Fan Coil Unit: The fan coil units used for HVAC are commonly located around the perimeter of a building and serve up to 15 feet from the outdoor wall. See Figure 8-8 for an illustration of a typical fan coil unit. These units may be totally self-contained with automatic controls; may include a

through-wall duct for outdoor air; may have remote heating and cooling media or may be controlled manually at the unit. Since the fan coil units are rather compact, the filters are relatively small and in the low efficiency range. This will tend to increase the maintenance requirements since the return air is at the floor level — a potential significant source for dirt and possibly other contaminants. A provision for outdoor air may be a feature of the fan coil unit especially for units used to provide the HVAC for the building perimeter. The outdoor air intake is normally screened and may, over time, become blocked by dirt and debris from the outside atmosphere. Also, the intake may be located on an outside ledge of the building, depending on the building design, which may be a roosting area for birds. The outdoor air intake presents a significant source for contaminants and a difficult location to maintain.

8.5 HVAC COMPONENT SURVEY OUTLINE

The responsibility for monitoring the indoor air quality within a building may be assigned to an office individual, the building maintenance department, an outside environmental firm, or an HVAC maintenance contractor. In order to meet this responsibility, the assignee should conduct periodic walk-through surveys of the HVAC system and its components. The assignee should have a procedure or outline of the system components in order to conduct the survey. Basic information required to develop the procedure would include the following:

1. The mechanical plans and specifications for the HVAC system to be surveyed including modifications or rearrangements, which are essential to conducting the survey.
2. A detailed description of the type of HVAC system, its features and functions, especially for those who are not thoroughly acquainted with the system.
3. The current test and balance reports which can provide information on air distribution and design vs. performance data. These reports may also indicate a specific component problem such as outdoor air requirements.
4. Reports of complaints regarding the indoor air quality (which should include the nature and location). These reports are essential to conducting this survey. They may indicate a component problem such as a disconnected diffuser and the lack of air movement in an occupied zone.

In addition to the basic information, the walk-through survey includes observation or inspection of each of the HVAC system components for potential malfunction. The procedure or survey outline of the components together with specific notes follows.

WALK-THROUGH SURVEY OUTLINE**1. Outdoor Air** (see Figure 5-28):

- A. Intake location and physical condition
- B. Building exhaust stacks and vent pipes adjacent to intake
- C. Cooling tower; type and location
- D. Building entryways, doors, and windows as potential entries for airborne contaminants
- E. Areas adjacent to the building as potential sources: shipping/receiving docks, parking lots, high traffic roads, adjacent buildings and operations, etc.

2. Dampers (see Figure 8-11):

- A. Outdoor air; type and physical condition
- B. Return air; type and physical condition
- C. Face and bypass; type and physical condition
- D. Exhaust/pressure relief; type and physical condition

3. Air Cleaning (see Figure 8-12):

- A. Type and general condition
- B. Prefilter; type, efficiency, and condition
- C. Final filter; type, efficiency, and condition

4. Heating/Cooling Coils (see Figure 8-13):

- A. Pre-heat; type and condition
- B. Cooling; type and condition
- C. Condensate pan and drain
- D. Re-heat; type and condition

5. Fans/Blowers (see Figure 8-14):

- A. Supply air; type and condition
- B. Return air; type and condition
- C. Exhaust/pressure relief; type and condition

6. Humidifier/Dehumidifier (see Figure 8-15):

- A. Type and general condition
- B. Condensate pan and drain

7. Supply Air Distribution:

- A. Duct system; type and general condition
- B. Control box; type and condition
- C. Control box function
- D. Control box/diffuser connection; type and condition

8. Supply Air Diffusers (see Figure 8-17):

- A. Type and general condition
- B. Characteristics of area served
- C. Number of diffusers this area
- D. Occupied zone air flow pattern; smoke test results
- E. Obstructions to flow pattern

9. Return Air Grilles (see Figure 8-17):

- A. Type, location, and general condition
- B. Air flow pattern, supply to return; smoke test
- C. Obstructions to flow pattern

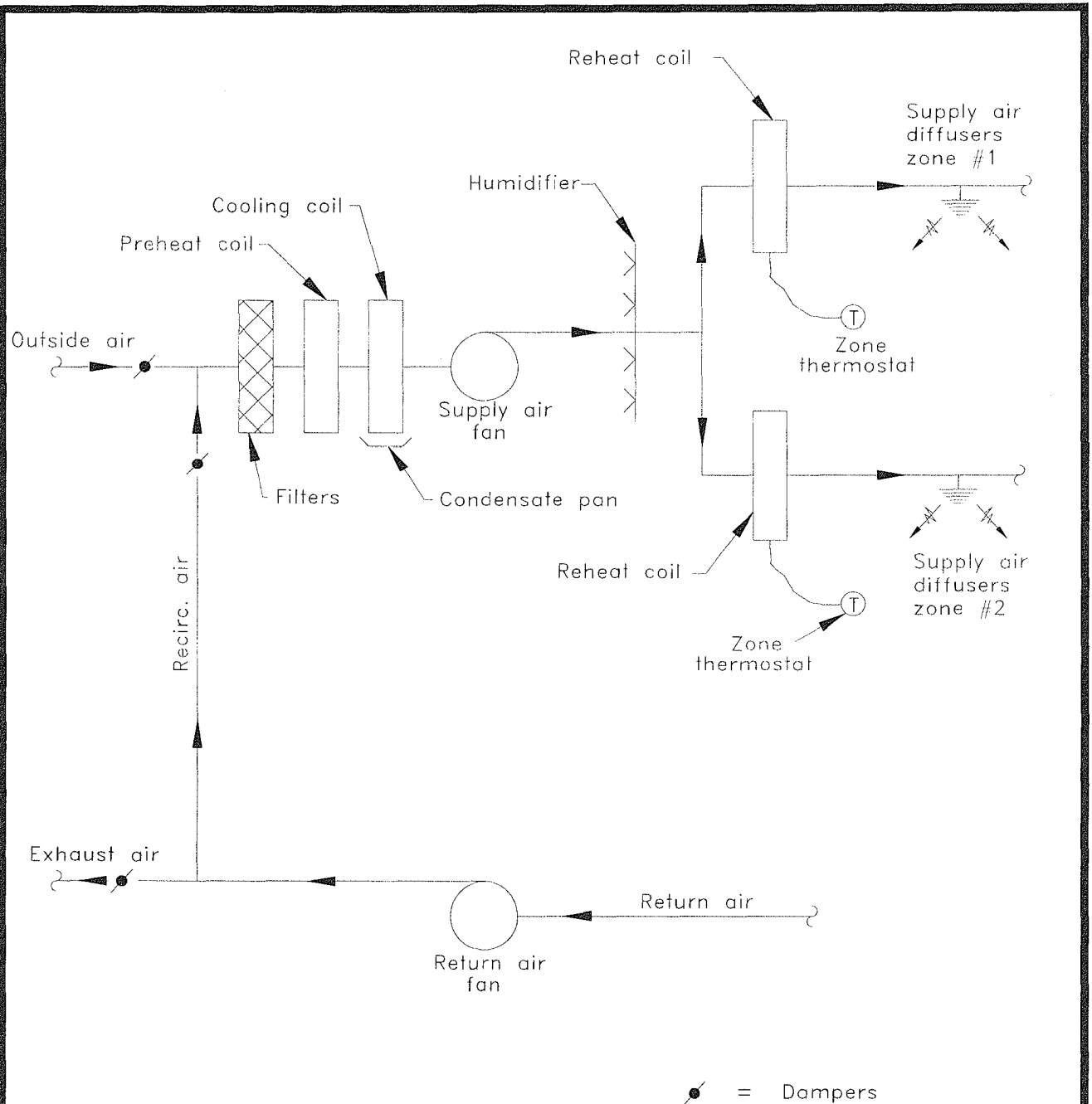
10. Return Air System:

- A. Open plenum; general condition, location of return air opening; return air fan/duct
- B. Ducted return; location and general condition
- C. Balancing dampers; type and condition

11. Miscellaneous Potential Contaminant Sources:**12. General Comments and Notes:**

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NOTE: See text regarding outside air requirements and distribution.

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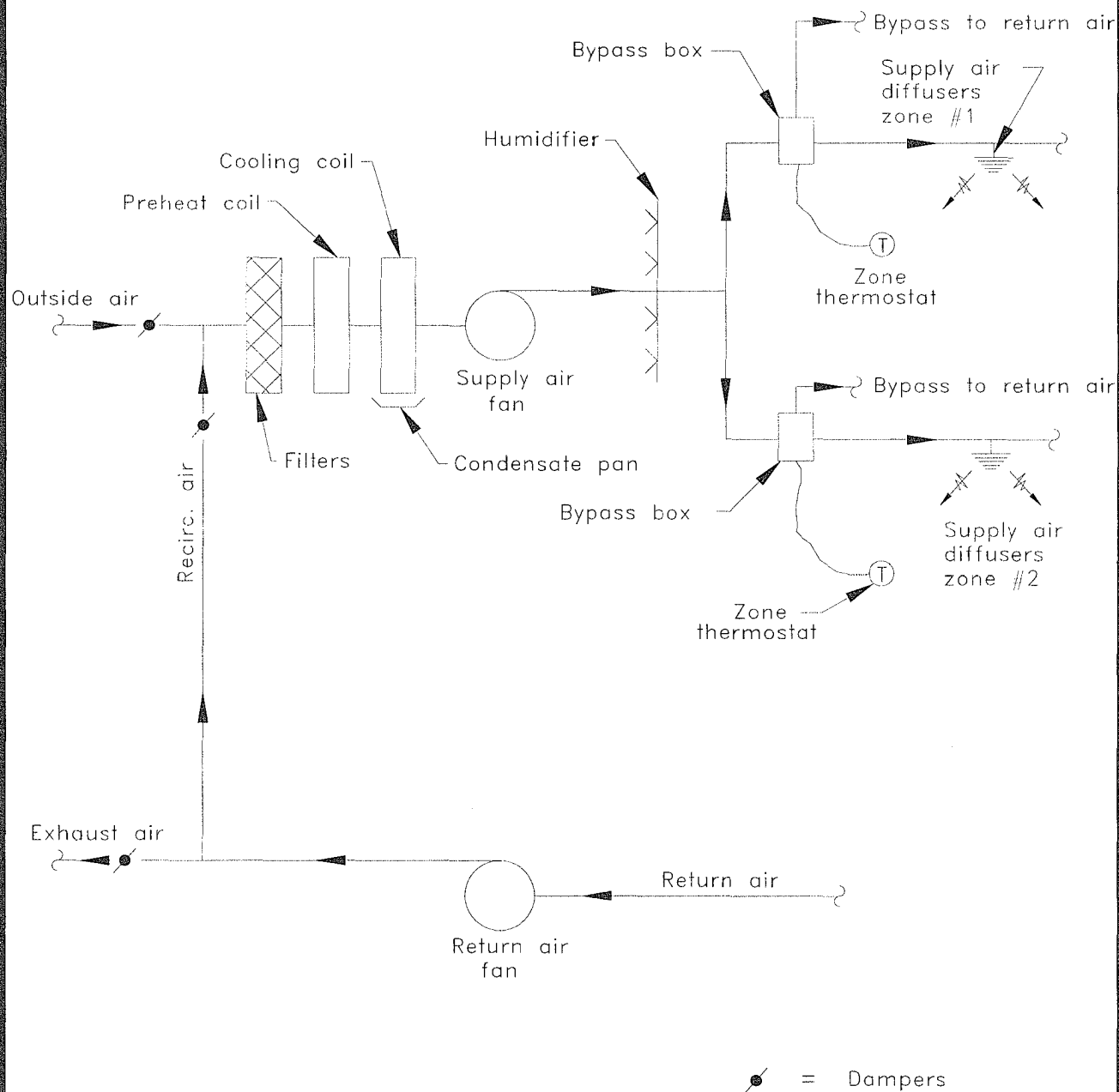
*SINGLE DUCT CONSTANT
VOLUME WITH REHEAT*

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4-94

FIGURE

8-1



NOTE: See text regarding outside air requirements and distribution.

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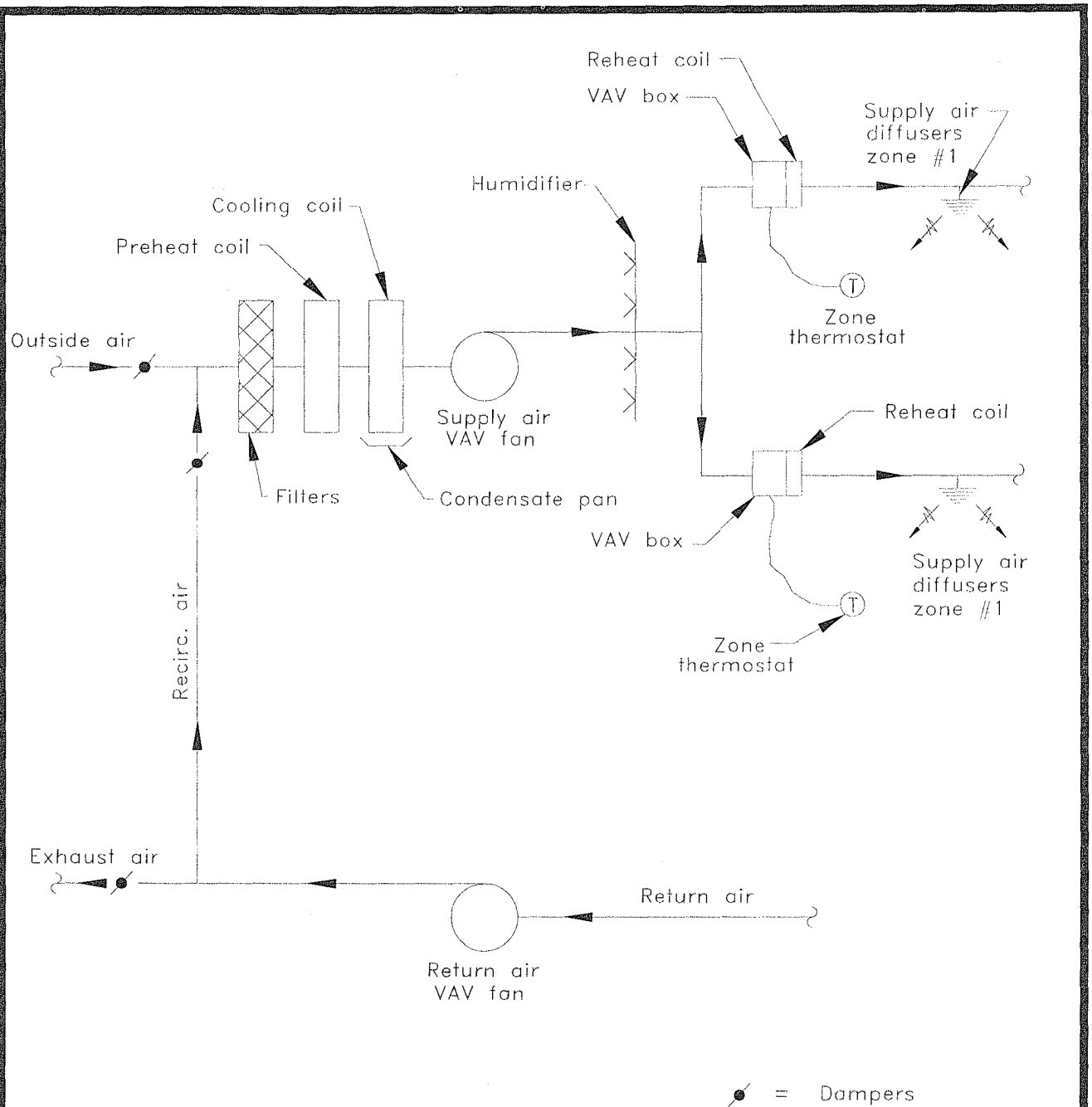
*SINGLE DUCT CONSTANT
VOLUME WITH BYPASS*

DATE

4-94

FIGURE

8-2



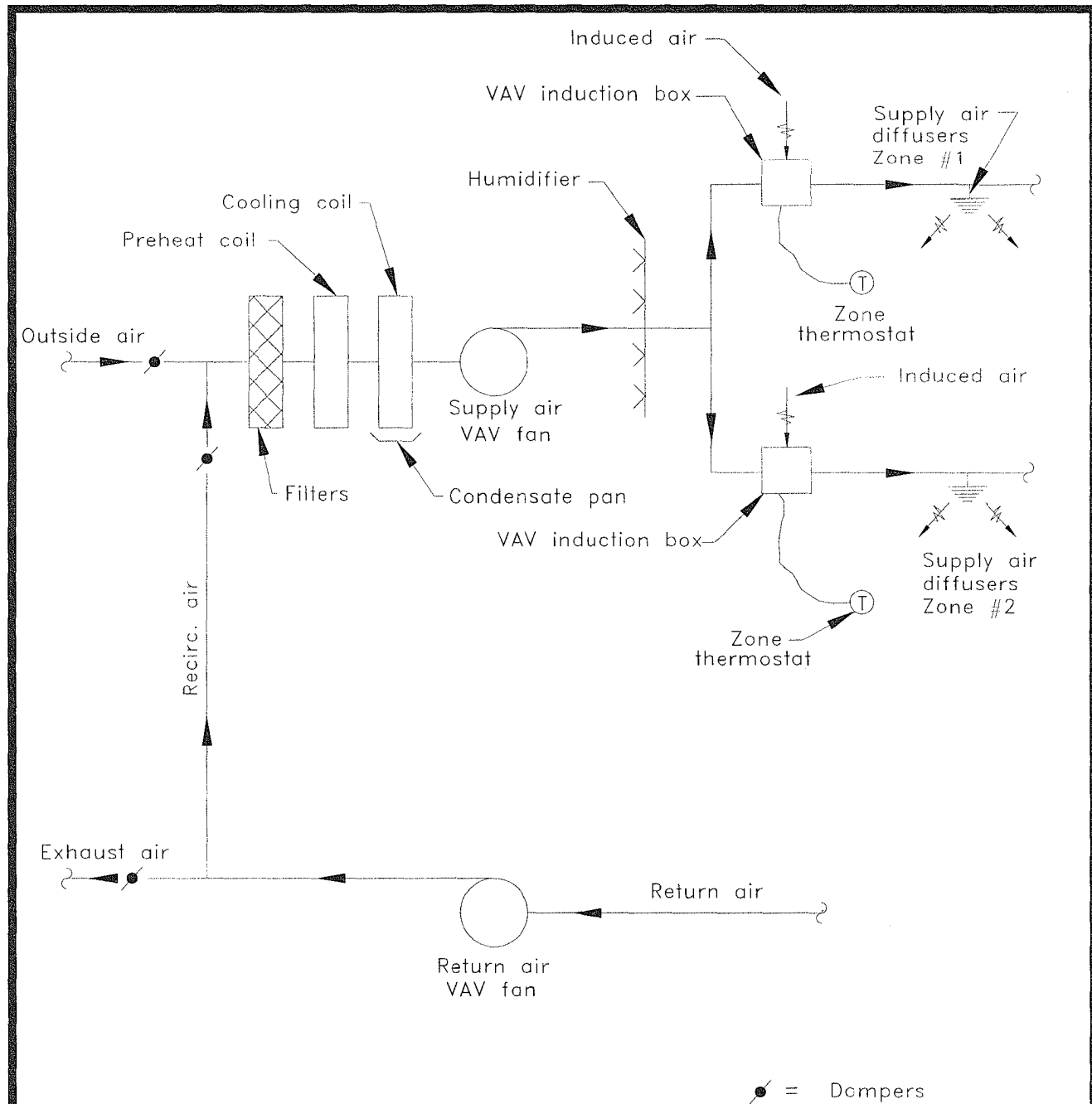
NOTE: See text regarding outside air requirements and distribution.

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*SINGLE DUCT VARIABLE
AIR VOLUME WITH REHEAT*

DATE 4-94

FIGURE 8-3



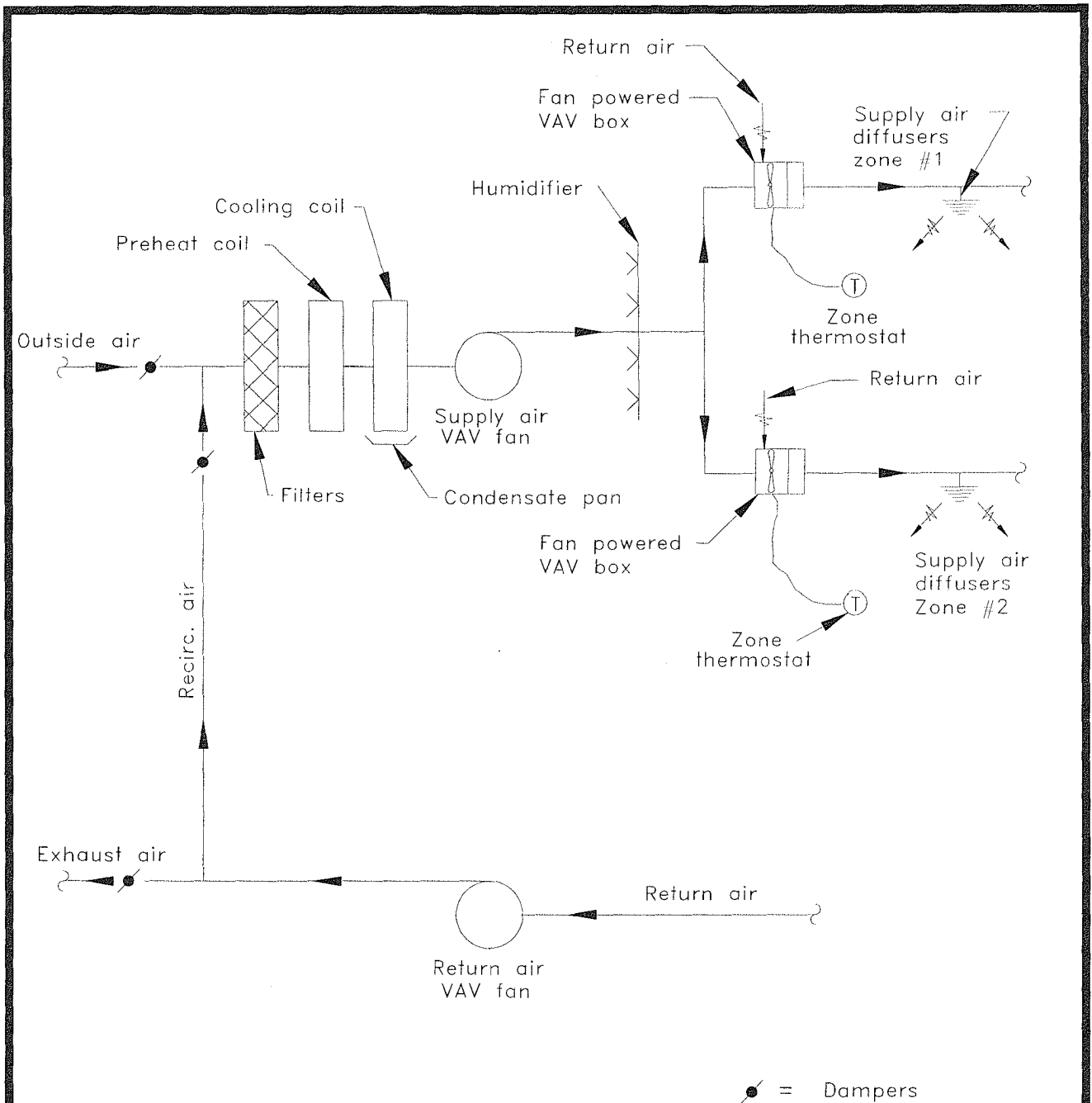
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*SINGLE DUCT VARIABLE
AIR VOLUME WITH INDUCTION*

DATE 4-94

FIGURE 8-4



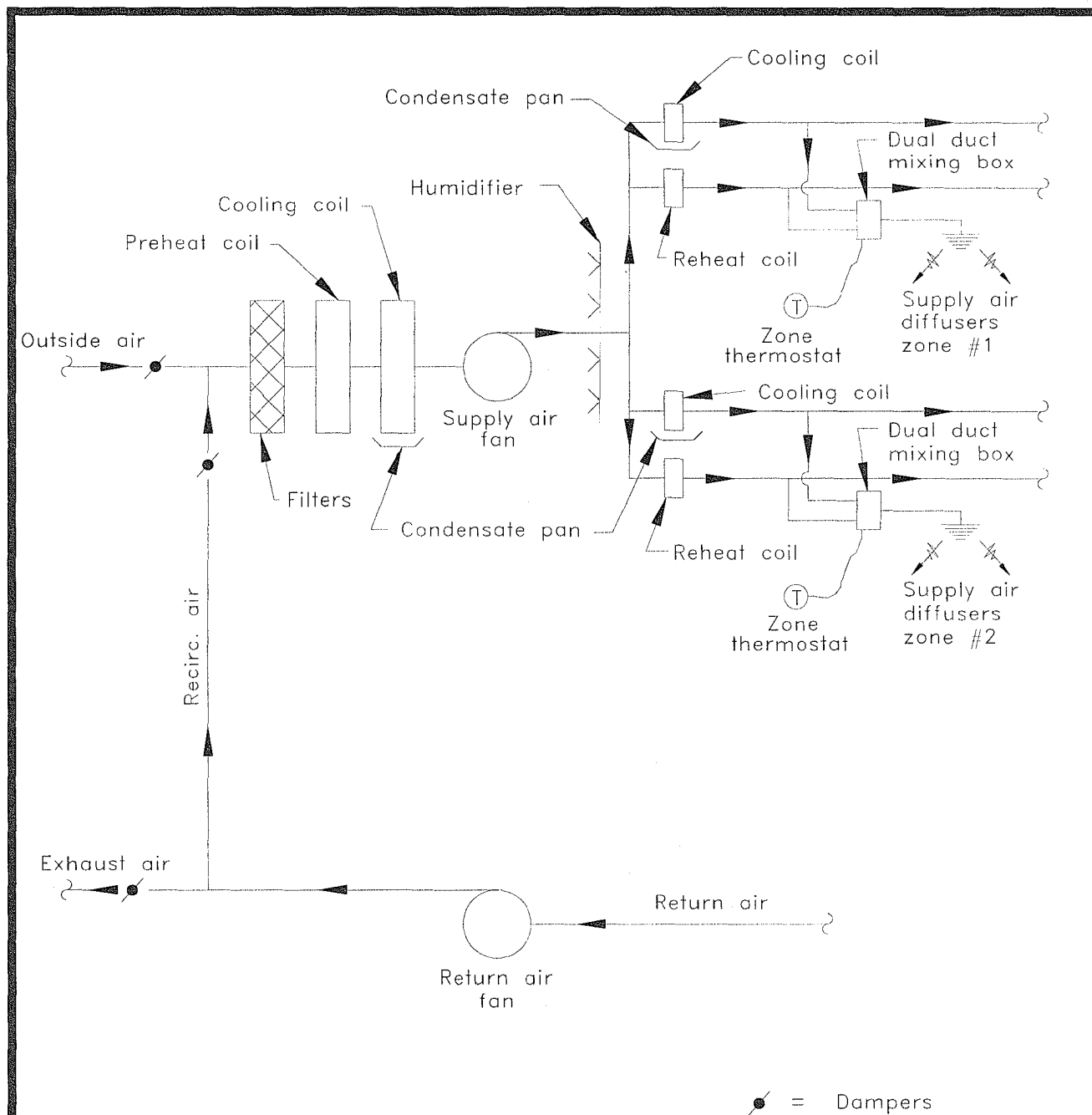
NOTE: See text regarding outside air requirements and distribution.

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*SINGLE DUCT VARIABLE AIR
VOLUME WITH FAN POWERED
DEVICES AND REHEAT*

DATE 4-94

FIGURE 8-5



NOTE: See text regarding outside air requirements and distribution.

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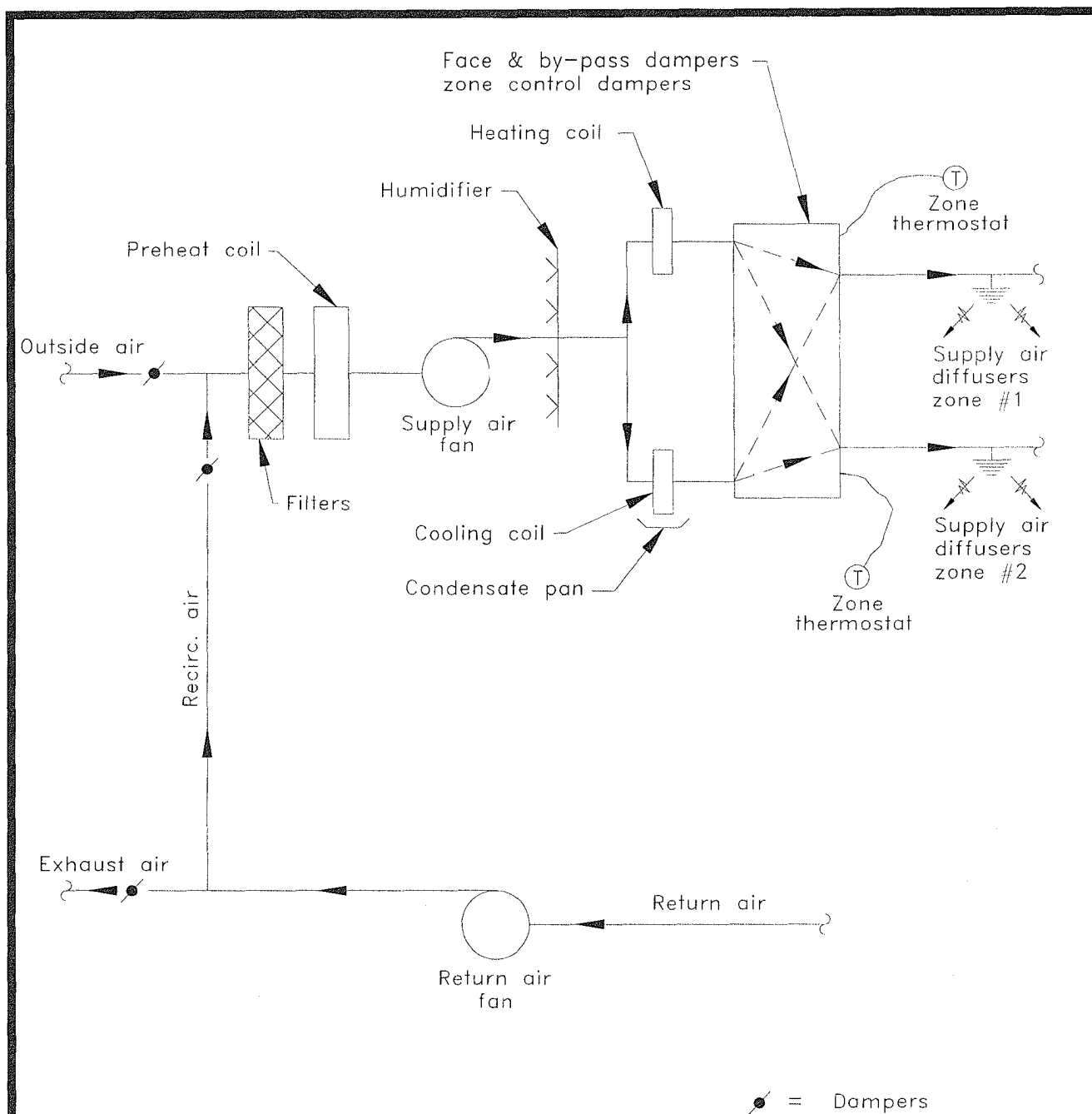
*DUAL DUCT CONSTANT
AIR VOLUME*

DATE

4-94

FIGURE

8-6



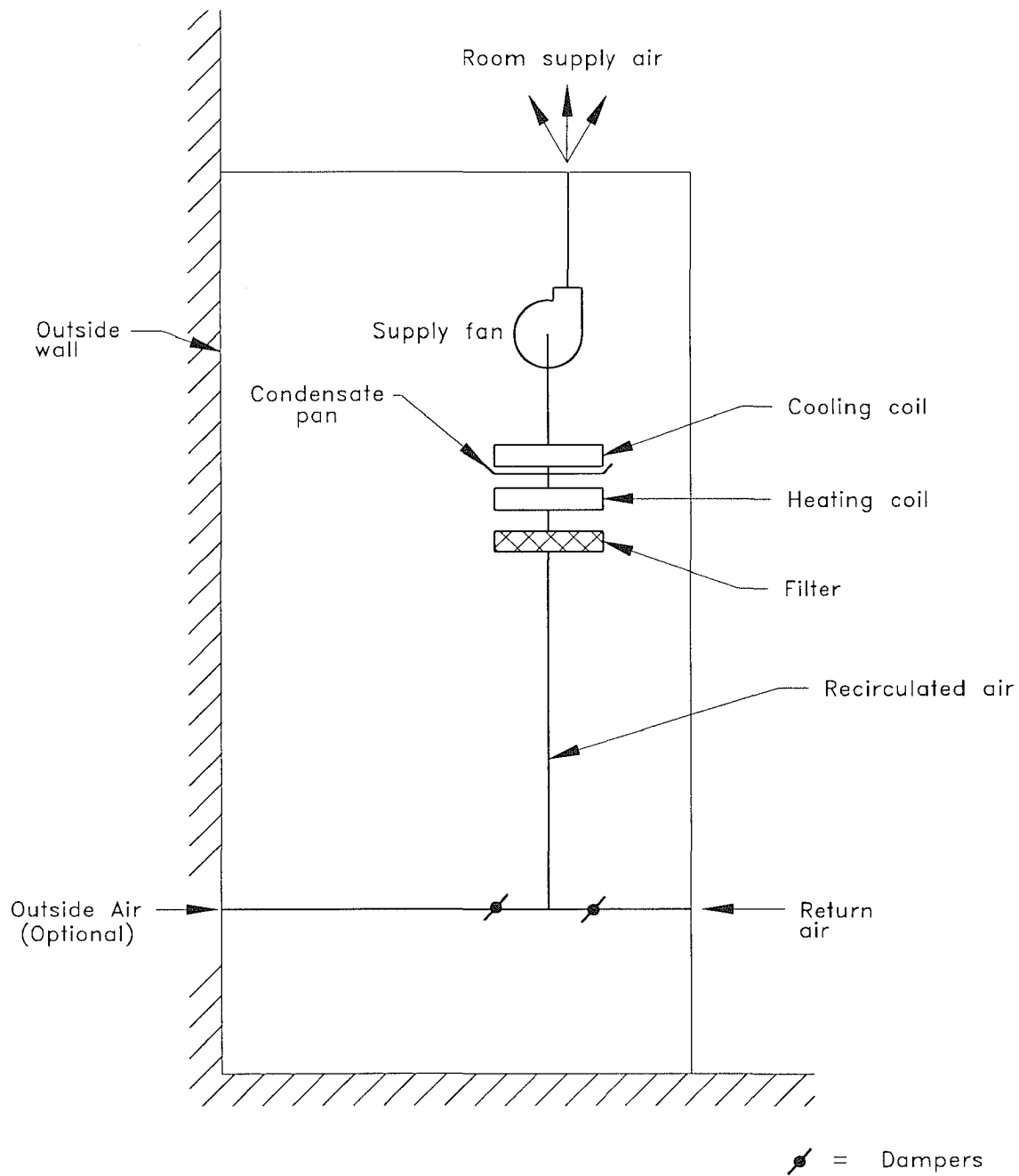
NOTE: See text regarding outside air requirements and distribution.

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*SINGLE DUCT CONSTANT
AIR VOLUME MULTI-ZONE*

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FIGURE 8-7

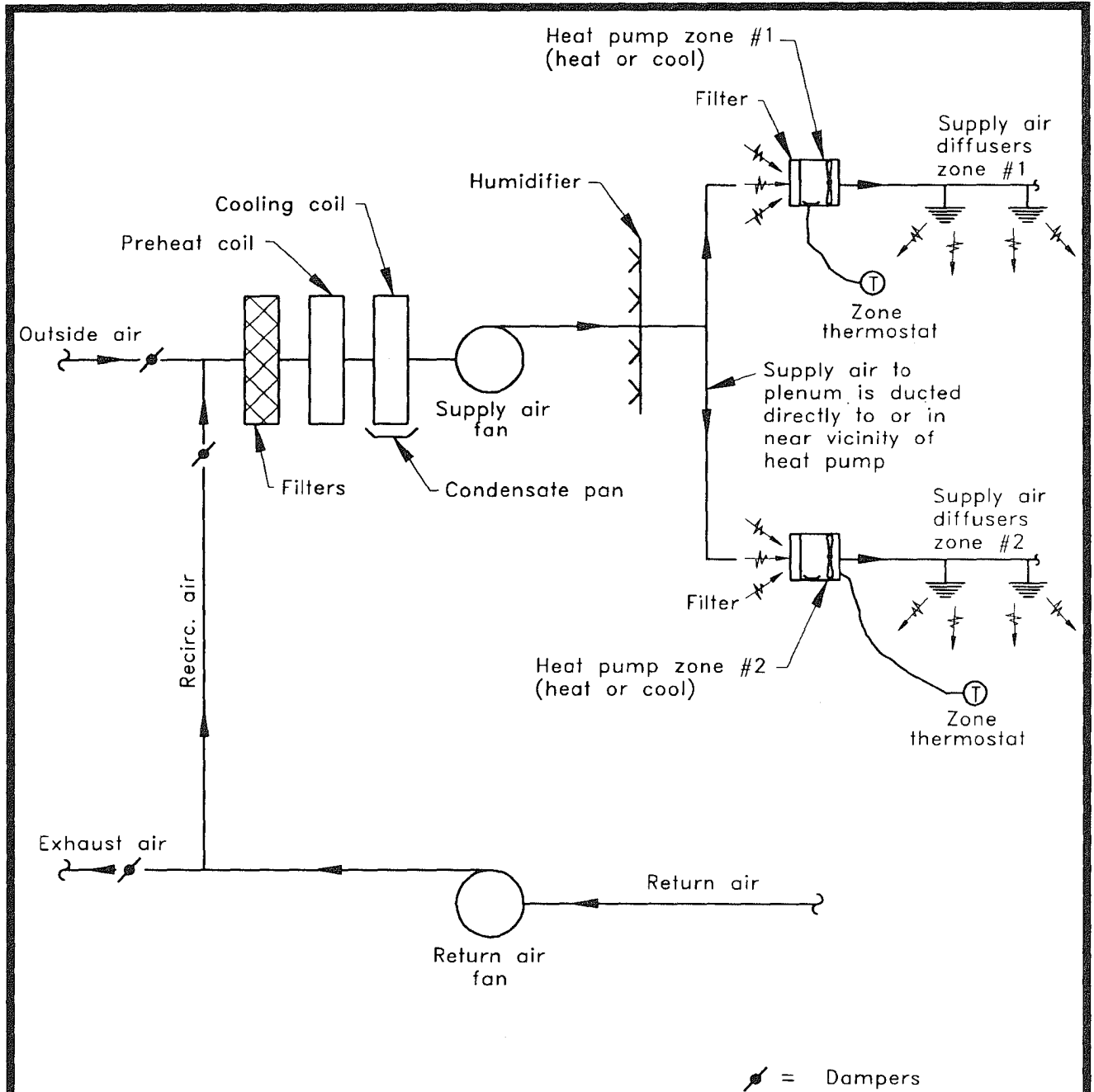


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AIR COIL UNIT

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FIGURE 8-8



NOTE: See text regarding outside air requirements and distribution.

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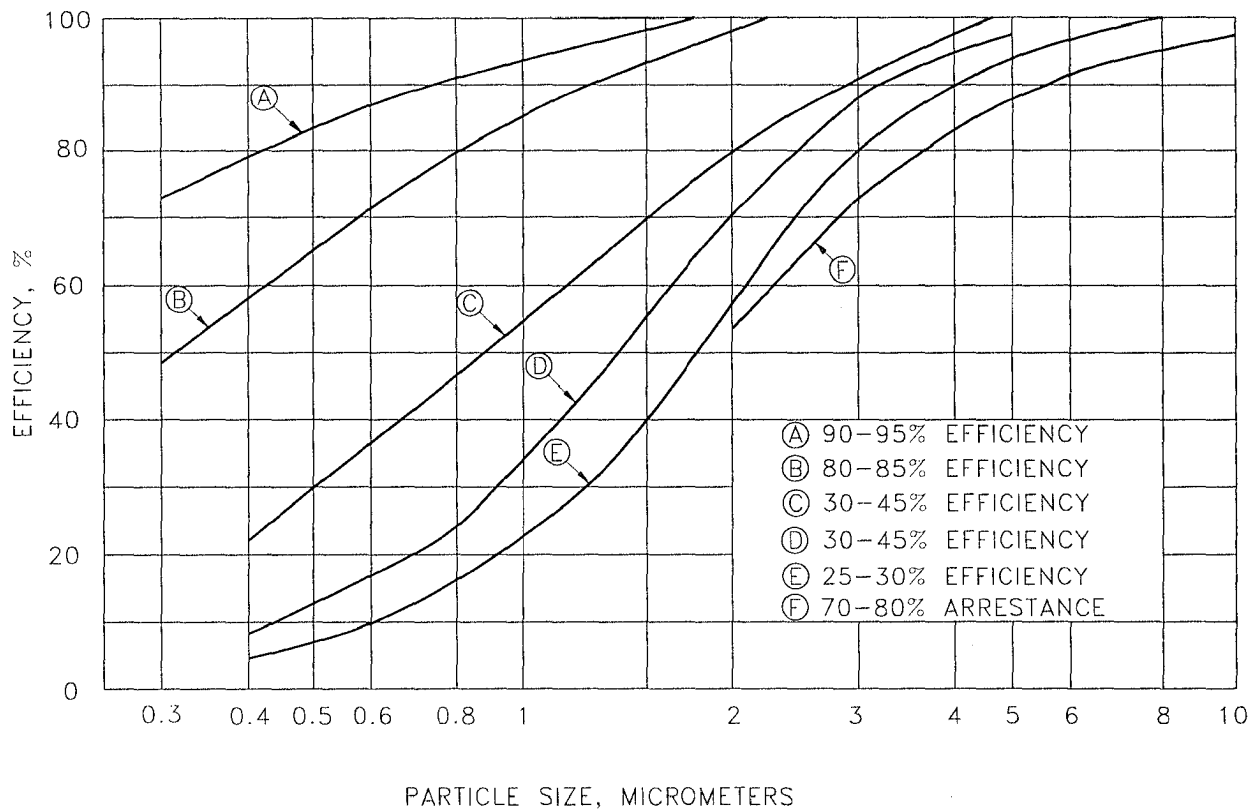
ZONE HEAT PUMP SYSTEM

DATE

4-94

FIGURE

8-9



Approximate Efficiency Versus Particle Size
for Typical Air Filters (See notes 1 & 2)

- NOTE: 1. Compiled and averaged from manufacturer data
Efficiency and arrestance per ASHRAE Standard
52 - 76 Test Methods.
2. Caution: Curves are approximations only for general guidance.
Values from them must not be used to specify air filters,
since a generally recognized test standard does not exist.

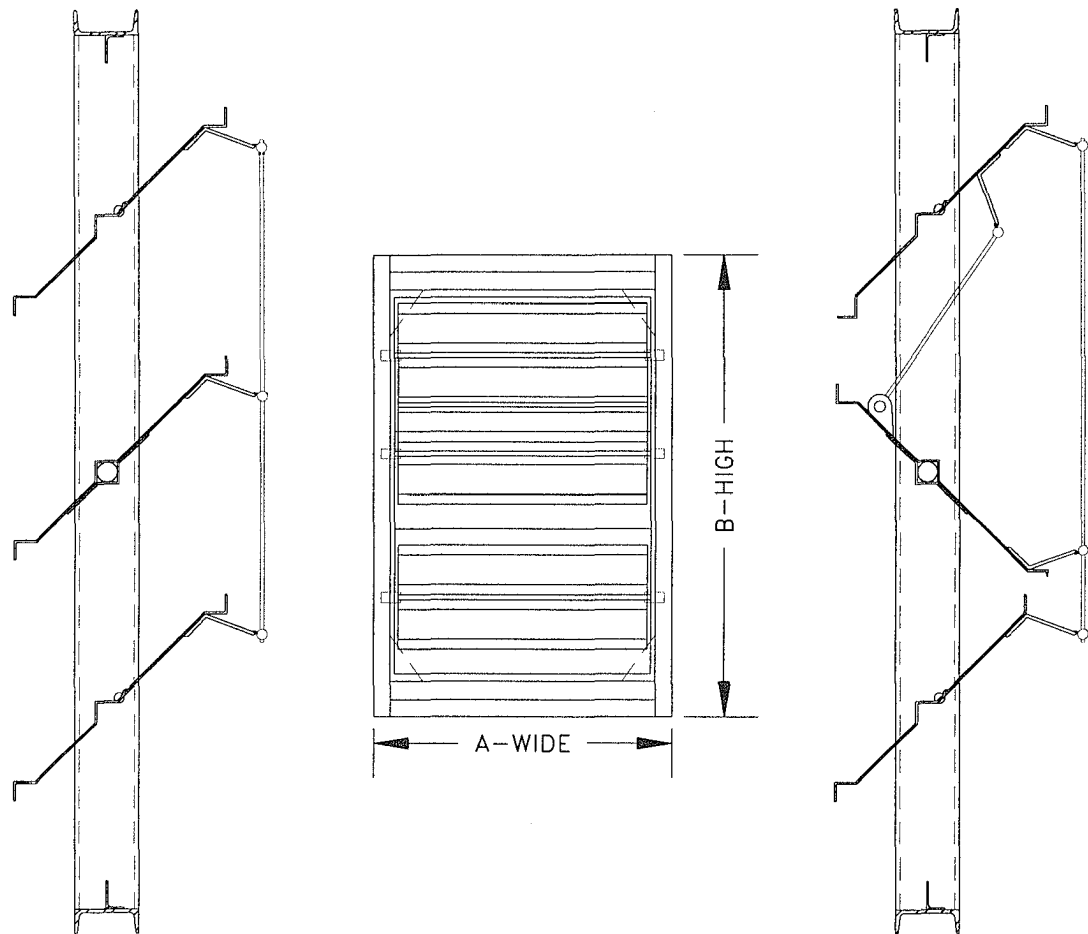
From: ASHRAE Equipment

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*FILTER EFFICIENCY
VS PARTICLE SIZE*

DATE 8-96

FIGURE 8-10

PARALLEL BLADEOPPOSED BLADE

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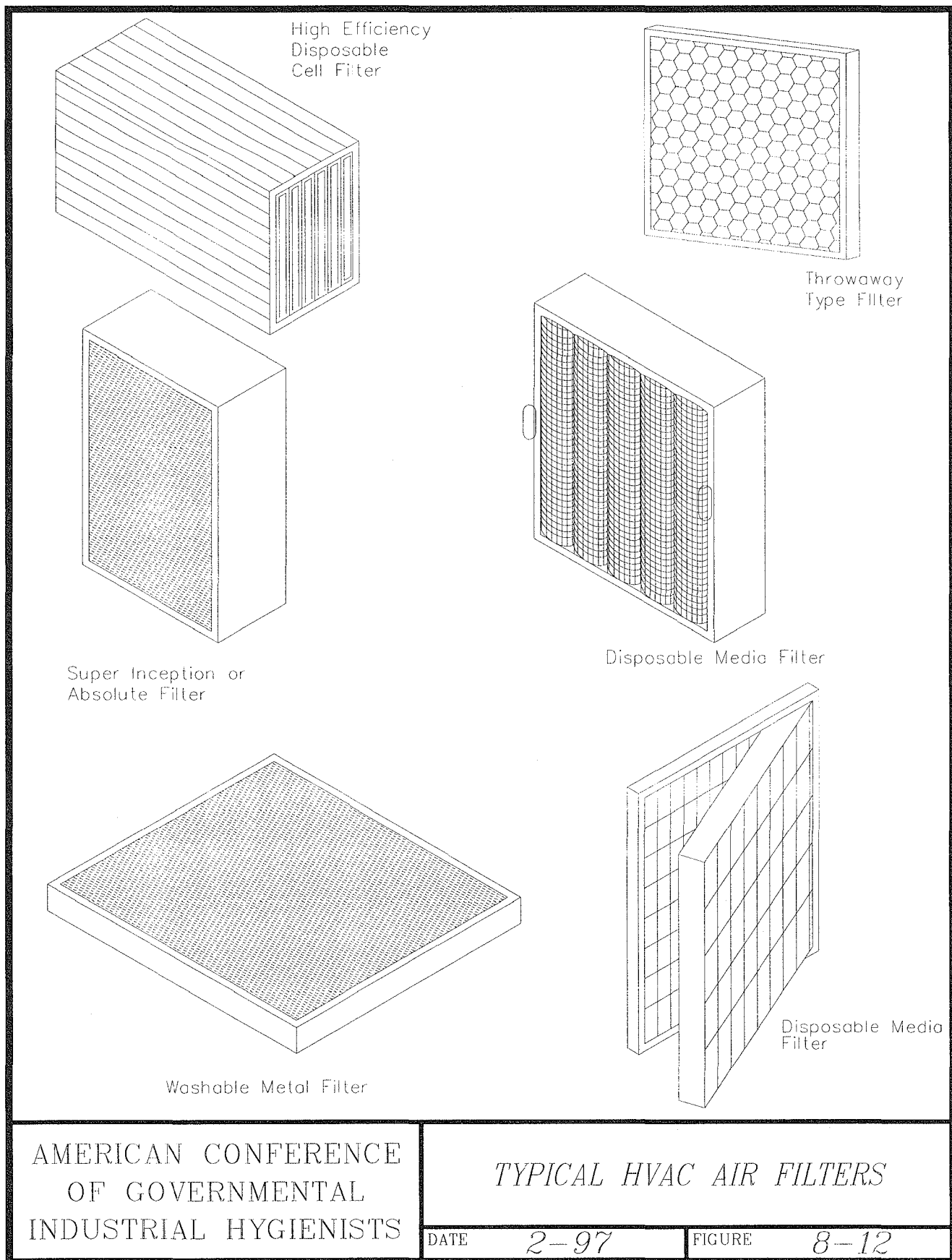
***DAMPERS:
PARALLEL AND OPPOSED BLADE***

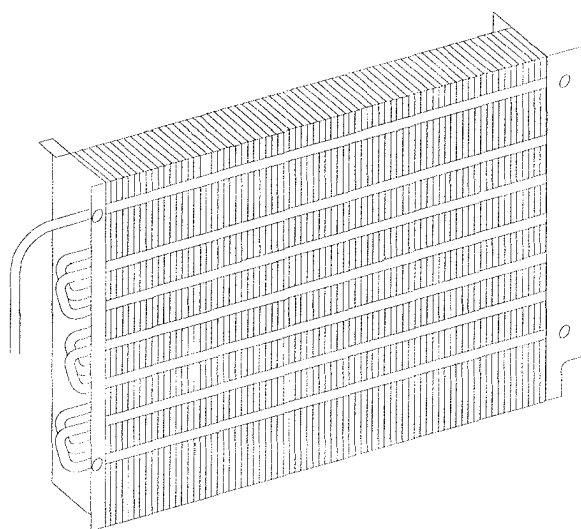
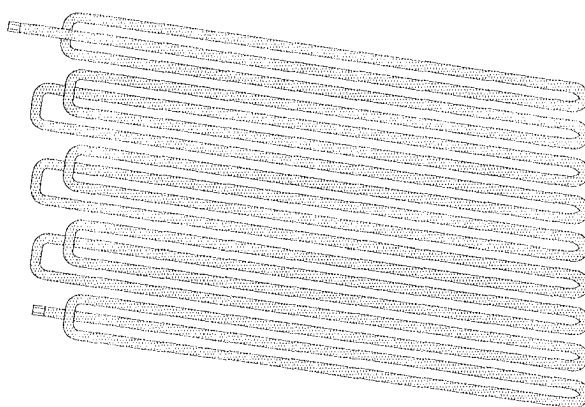
DATE

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FIGURE

8-11



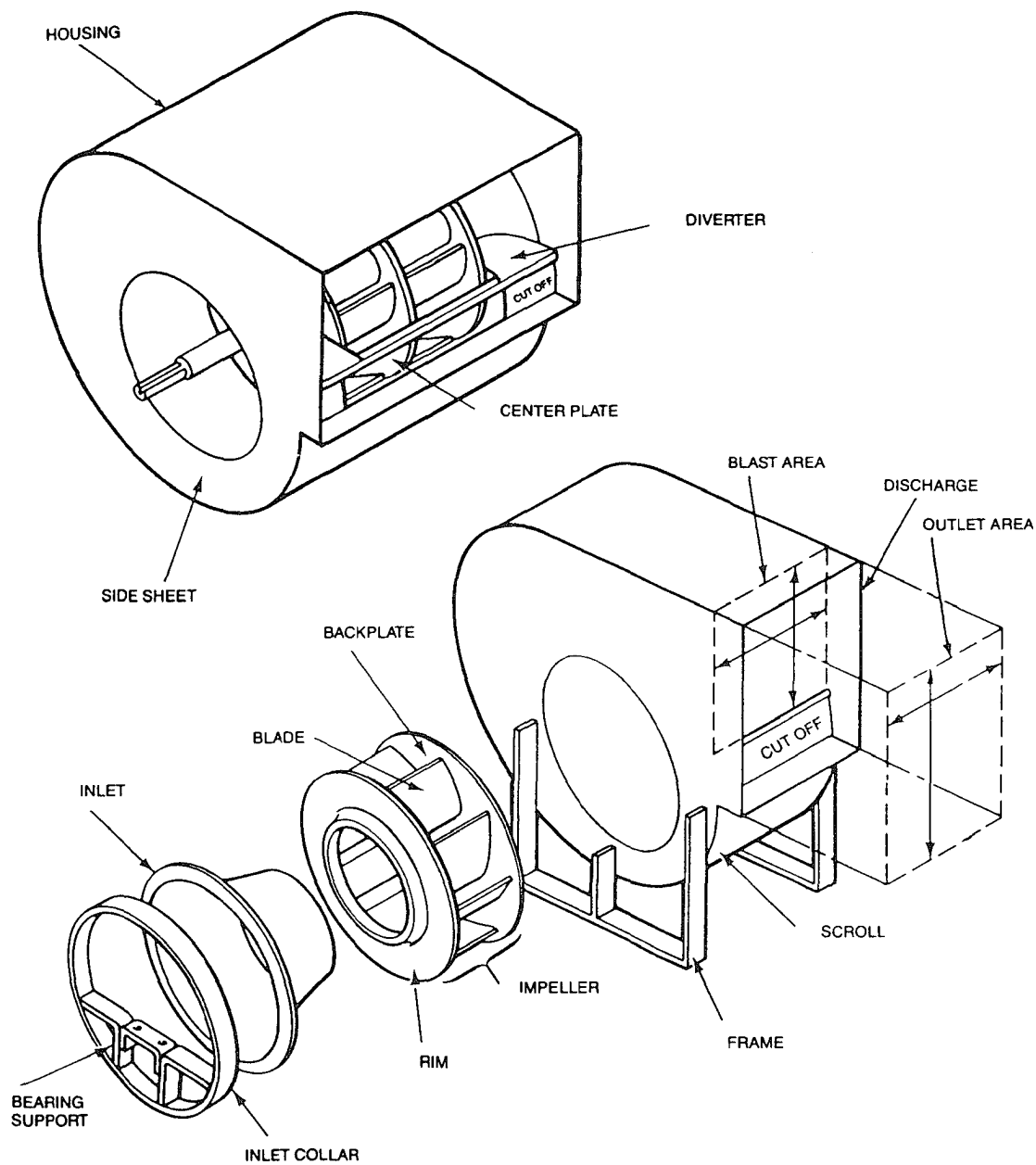


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HEATING/COOLING COILS

DATE 02-97

FIGURE 8-13



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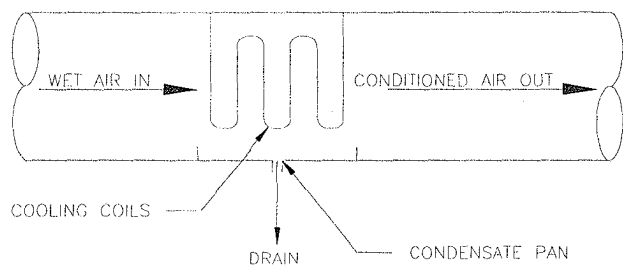
*TERMINOLOGY FOR
CENTRIFUGAL FAN
COMPONENTS*

DATE

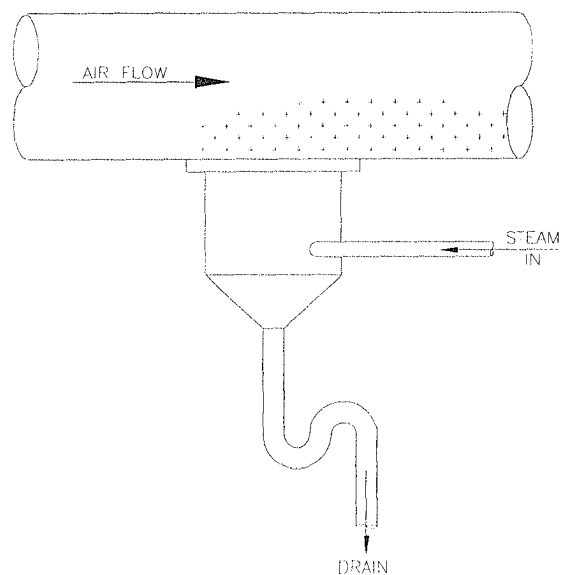
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FIGURE

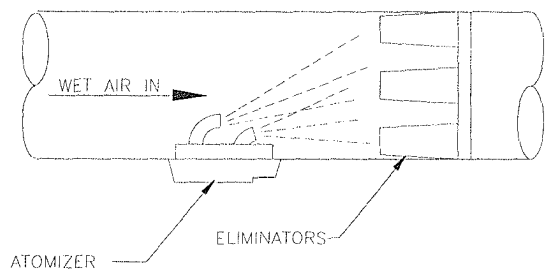
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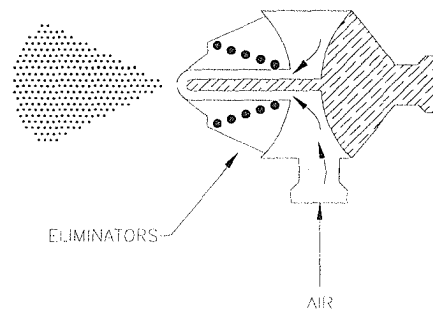
REFRIGERATION DEHUMIDIFICATION



CUP STEAM HUMIDIFIER



ATOMIZING HUMIDIFIER WITH OPTIONAL
FILTER ELIMINATOR



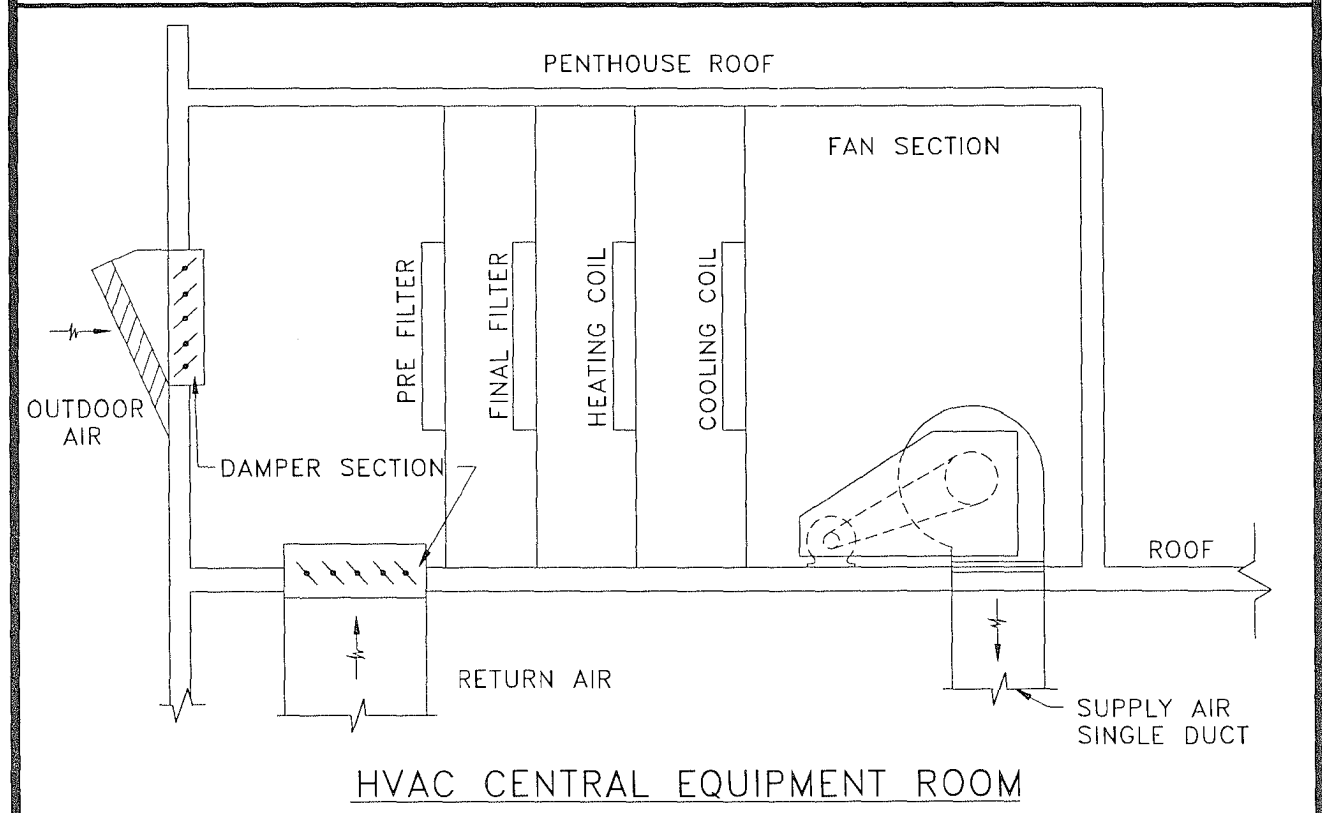
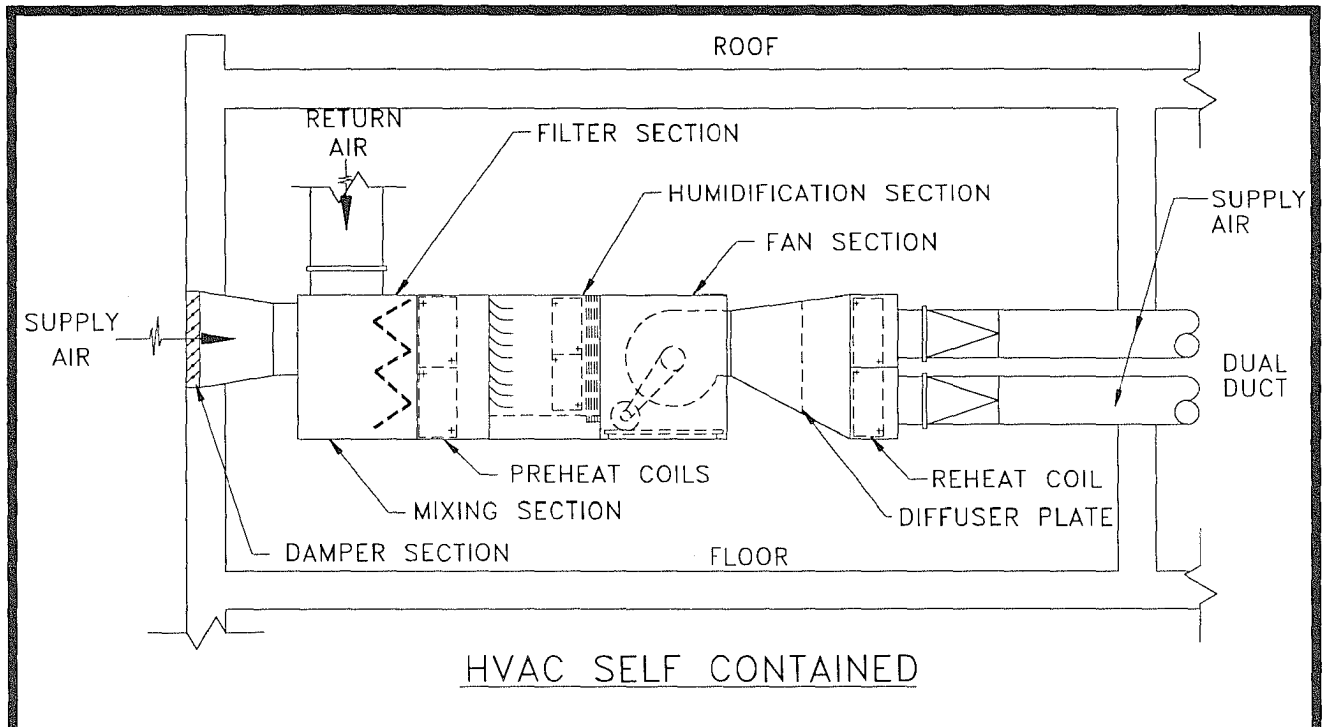
PNEUMATIC ATOMIZING HUMIDIFIER

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INDUSTRIAL HYGIENISTS

HVAC
HUMIDIFIER\DEHUMIDIFIER

DATE 02-97

FIGURE 8-15

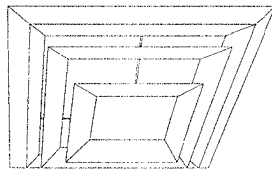


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INDUSTRIAL HYGIENISTS

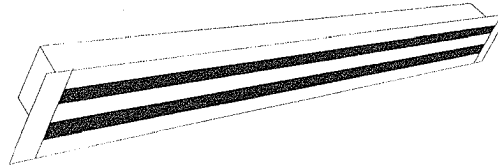
*HVAC SYSTEM
SELF CONTAINED
AND EQUIPMENT ROOM*

DATE 8-96

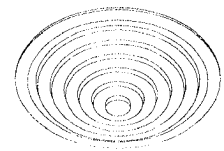
FIGURE 8-16



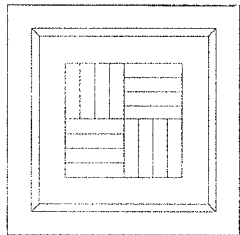
SQUARE



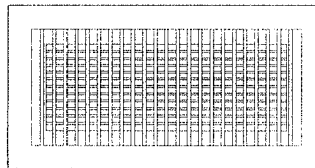
LINEAR



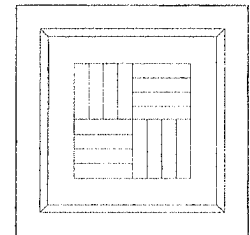
ROUND



PERFORATED

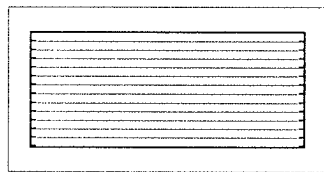


SIDE WALL

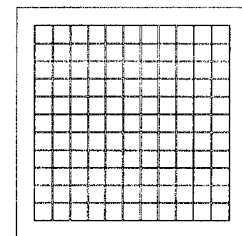


ADJUSTABLE BLADE

AIR SUPPLY DIFFUSERS



SIDE WALL



CEILING

RETURN AIR GRILLES

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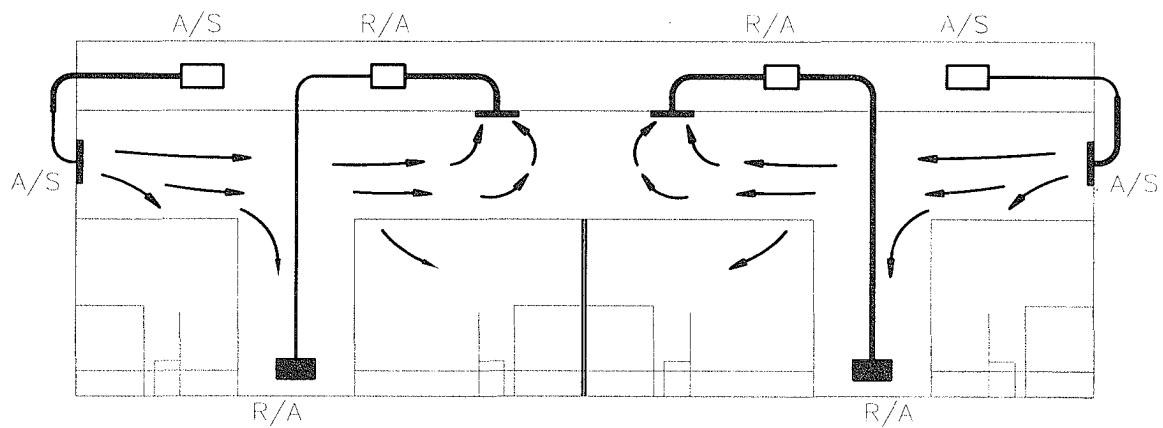
*AIR SUPPLY DIFFUSERS &
RETURN AIR GRILLES*

DATE

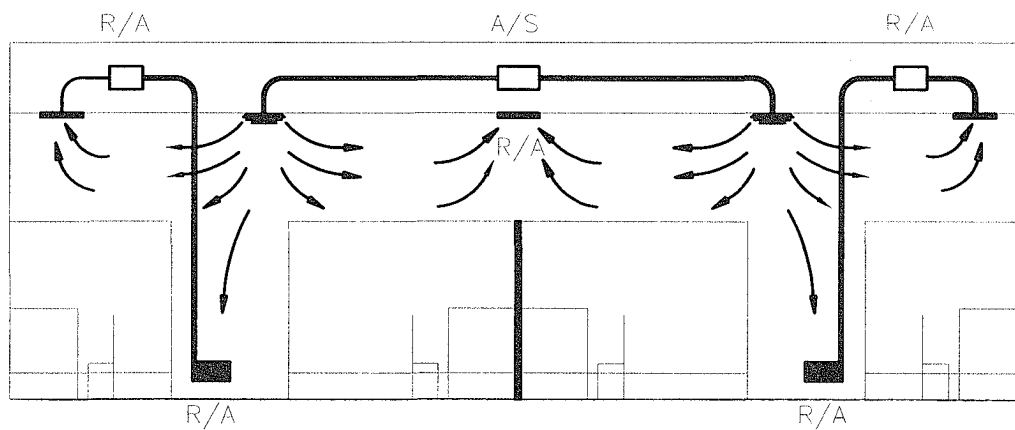
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FIGURE

8-17



SIDE WALL SUPPLY
GRILLES



CEILING SUPPLY
GRILLES

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*TYPICAL PARTITIONED
OFFICE AIR PATTERN*

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FIGURE

8-18

Chapter 9

TESTING OF VENTILATION SYSTEMS

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9.1 INTRODUCTION

Every ventilation system should be tested at the time of initial installation to verify the volumetric flow rate(s) and to obtain other information which can be compared with the original design data. Testing is necessary to verify the setting of blast gates, fire dampers, and other air flow control devices which may be a part of the system. Initial testing will provide a baseline for periodic maintenance checks and isolation of system failures should a malfunction occur. Many governmental codes require initial and periodic testing of exhaust systems for certain types of processes. Exhaust system test data are also useful as a basis for design of future installations where satisfactory air contaminant control is currently being achieved.

The tests described in this text pertain to ventilation systems only. Environmental tests should be conducted prior to and after installation to verify system performance. In these cases, the services of a qualified industrial hygienist may be required.

9.1.1 Initial Test: The Pitot tube and manometer are the standard for initial field testing of equipment when used as described. However, other instruments may be used. As noted later, all instruments must be calibrated. Identify on the test sheet the instruments and procedures used. A sample survey form is located at the end of the chapter (Figure 9-17). The following steps outline the recommended procedure and the minimum data necessary for a thorough initial ventilation test.

1. Review the system specifications and drawings to determine the relative location and sizes of ducts, fittings, and associated system components. Where possible, pertinent prints should be carried to the test site.
2. Inspect the system to determine that its installation is in accordance with the specifications and drawings. Check such items as fan rotation, belt slippage, and damper settings.
3. Include a drawing of the system as installed. Select and identify test locations.
4. Measure the volumetric flow rate, fan static pressure, fan speed, motor speed, motor amperes, and the temperature of the air in the system. Also determine pressure drops across all components such as coils, fittings, and air cleaning equipment.

Fan speed may be measured directly at the end of the fan shaft using a revolution counter and a watch. A tachometer or stroboscopic measuring device also may be used.

The operating amperage is obtained with an ammeter. The readings taken on each lead on the three-phase current should be averaged and compared to the rating on the motor name plate at the operating voltage to determine if the motor is operating within its rated

range.

For some tests, moisture content of the air in the system and/or ambient barometric pressure should be obtained also. See example calculations in Section 9.6.

5. Record the test data and design specifications on the data sheet. Calculate test results following the format on the data sheets.
6. Compare the test data with design specifications. Determine if alterations or adjustments of the system are necessary to meet specifications, codes, or standards.
7. If alterations or adjustments are made, retest the system, and record the final test data. On the drawing, note any physical changes that were made in the system.

9.1.2 Periodic Test: The performance of a system should be checked periodically. If there have been no alterations to the system, this can be done by static pressure measurements and close visual inspection. Measurements also can be made continuously by means of an operating console or other remote readout system.

The following is the recommended procedure with suggestions as to types of measurements needed to perform the periodic tests:

1. Refer to the initial data sheet for test locations.
2. Inspect the system for physical damage (broken, corroded, collapsed duct, etc.) and proper operation of components (fan, damper, air cleaner, controls, burner, etc.).
3. Measure static pressure at the same locations used in the initial test.
4. Compare measured static pressures with initial test. From these comparisons, determine if the system is performing at initial levels.
5. Make and record any correction required.
6. Recheck the system to verify performance if corrections have been made.

Whenever alterations have been made to the system, a new initial test is necessary following the procedures outlined under Section 9.1.1, "Initial Test."

9.2 MEASUREMENTS OF VOLUMETRIC FLOW RATE

The most important measurement in testing of systems is the measurement of the volumetric flow rate in cfm. This should be done before balancing the system is attempted. The commonly used instruments are of the velocity measuring type rather than quantity meters. Therefore, it is necessary to obtain not only the average air velocity through an opening or duct, but also the net cross-sectional area at the point of measurement. The volumetric flow rate then can be determined from the equation:

$$Q = VA$$

[9.1]

TABLE 9-1. Characteristics of Flow Instruments

Instrument	Range, fpm	Hole Size (for ducts)	Range, Temp*	Dust, Fume Difficulty	Calibration Requirements	Ruggedness	General Usefulness and Comments
PITOT TUBES with inclined manometer							
Standard	600 – up	3/8"	Wide	Some	None	Good	Good except at low velocities
Small Size	600 – up	3/16"	Wide	Yes	Once	Good	Good except at low velocities
Double	500 – up	3/4"	Wide	Small	Once	Good	Dirty air stream
SWINGING VANE ANEMOMETERS							
	25 – 10,000	1/2 – 1"	Medium	Some	Frequent	Fair	Good
ROTATING VANE ANEMOMETERS							
Mechanical	30 – 10,000	Not for duct use	Narrow	Yes	Frequent	Poor	Special; limited use
Electronic	25 – 200	Not for duct use	Narrow	Yes	Frequent	Poor	Special; can record; direct reading
	25 – 500						
	25 – 2000						
	25 – 5000						

*Temperature range: Narrow, 20–150 F; Medium, 20–300 F; Wide, 0–800 F

where:

Q = volumetric flow rate, (cfm)

V = average linear velocity, feet per minute (fpm)

A = cross-sectional area of duct or hood at the measurement location, ft^2

9.2.1 Air Velocity Instruments: The volumetric flow rate of an exhaust system can be determined by the use of various types of field instruments which measure air velocity directly. Typically, these instruments are used at exhaust and discharge openings or, depending on size and accessibility, inside a duct. The field technique is based on measuring air velocities at a number of points in a plane and averaging the results. The average velocity is used in Equation 9.1 to determine the volumetric flow rate. Due to the difficulty of measuring the area of an irregularly shaped cross-section and the rapid change in velocity as air approaches an exhaust opening, measurements obtained should be considered an approximation of the true air flow. All instruments should be handled and used in strict compliance with the recommendations and directions of the manufacturers. Table 9-1 lists some characteristics of typical air velocity instruments designed for field use.

Rotating Vane Anemometer (Figure 9-1): This instrument is accurate and can be used to determine air flow through large supply and exhaust openings. Where possible, the cross-sectional area of the instrument should not exceed 5.0% of the cross-sectional area of the duct or hood opening. The standard instrument consists of a propeller or revolving vane connected through a gear train to a set of recording dials that read the linear feet of air passing in a measured length of time. It is made in various sizes; 3", 4", and 6" are the most common. It

gives average flow for the time of the test (usually one minute). The instrument requires frequent calibration and the use of a calibration card or curve to determine actual velocity. The instrument may be used for either pressure or suction measurements using the correction coefficients listed by the manufacturer. The standard instrument has a useful range of 200–3000 fpm; specially built models will read lower velocities.

Direct-recording and direct-reading rotating vane anemometers are available. These instruments record and meter

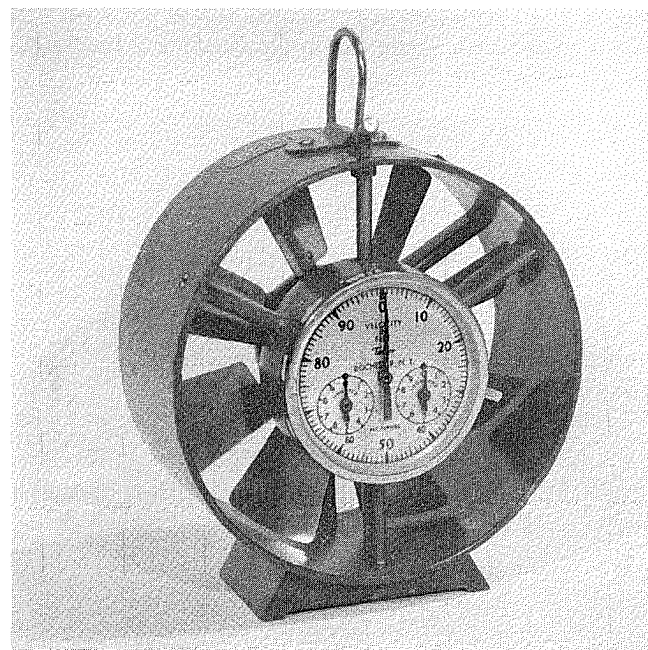


FIGURE 9-1. Rotating vane anemometer

electrical pulses developed by a capacitance or inductive transducer. The impulses are fed to the indicator unit where they are integrated to operate a conventional meter dial. Readings as low as 25 fpm can be measured and recorded.

The standard 4" rotating vane anemometer is unsuited for measurement in ducts less than 20" in diameter as it has too large a finite area and its equivalent cross-sectional area is difficult to compute. The conventional meter is not a direct-reading velocity meter and must be timed. It is fragile and care must be used in dusty or corrosive atmospheres. Newer units of 1" diameter which can be used in ducts as small as 5" in diameter are available.

Swinging Vane Anemometer (Figure 9-2): This instrument is extensively used in field measurements because of its portability, wide-scale range, and instantaneous reading features. Where accurate readings are desired, the correction coefficients in Table 9-2 should be applied. The instrument has wide application and, by a variety of fittings, can be used to check static pressures and a wide range of linear velocities. The minimum velocity is 50 fpm unless specially adapted for a lower range. The instrument is fairly rugged and accuracy is suitable for most field checks. Uses of the swinging vane anemometer and its various fittings are illustrated in Figure 9-3.

Before using, check the meter for zero setting by holding it horizontal and covering both ports so that no air can flow through. If the pointer does not come to rest at zero, an adjustment must be made to correct the starting point. The meter should be used in an upright position and, when using fittings, it must be held *out of the air stream* so that the air flows freely into the opening. The length and inside diameter of the connecting tubing will affect the calibration of the meter. When replacement is required, use only connecting tubing of the same length and inside diameter as that originally supplied with the meter.



FIGURE 9-2. Swinging vane anemometer

TABLE 9-2. Correction Factors for the Swinging Vane and Thermal Anemometers

Grille Openings	Correction Factor C_F (percent)
Pressure	
More than 4 in wide and up to 600 in ² area, free opening 70% or more of gross area, no directional vanes. Use free-open area.	93
Suction	
Square punched grille (use free-open area)	88
Bar grille (use gross area)	78
Strip grille (use gross area)	73
Free open, no grille	No correction

Where temperatures of an air stream vary more than 30 F from the standard temperature of 70 F and/or if the altitude is greater than 1,000 feet, it is advisable to make a correction for temperature and pressure. Corrections for change in density from variations in altitude and temperature can be made by using the actual gas density (ρ) shown in Equation 9.9 in the following equation:

$$V_c = V_r \sqrt{\frac{0.075}{\rho}} \quad [9.2]$$

where:

V_c = corrected velocity, fpm

V_r = velocity reading of instrument, fpm

ρ = actual gas density, lbm/ft³

Use at Supply Openings: On large (at least 3 ft²) supply openings where the instrument itself will not block the opening seriously and where the velocities are low, the instrument itself may be held in the air stream with the air impinging directly in the left port. When the opening is smaller than 3 ft² and/or where the velocities are above the "No Jet" scale, appropriate fittings must be used.

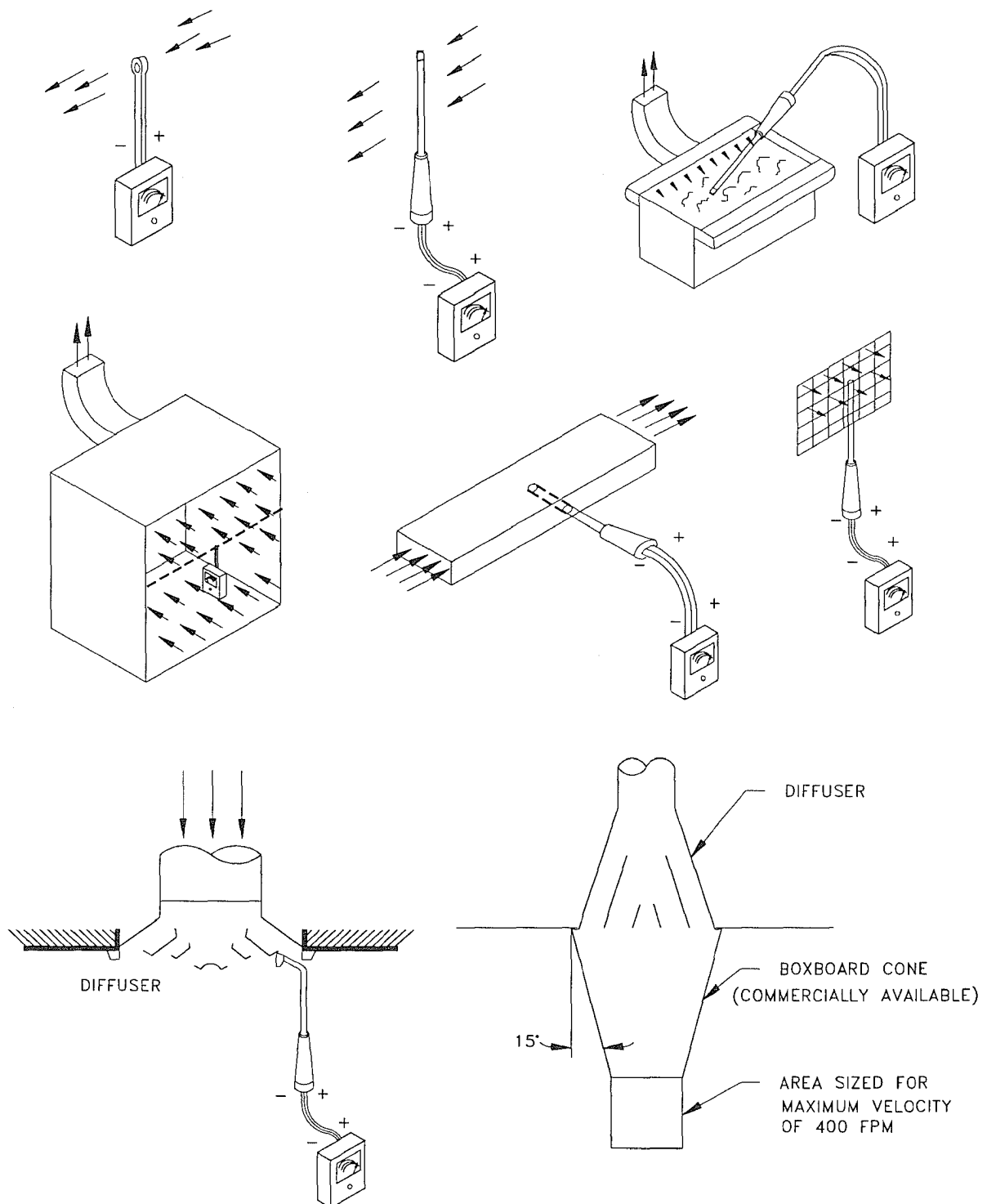
Because the velocity and static gradient in front of an exhaust opening is steep, the finned opening of the fitting must be held flush with the exhaust opening. If the opening is covered by a grille, hold the fin directly against the grille and use the correction coefficients listed in Table 9-2 when computing exhaust volumes.

$$Q = C_F VA \quad [9.3]$$

where:

C_F = correction coefficient in percent of scale reading.

While it can be used to measure air velocities, static pressure, and total pressure in ducts, it has several disadvantages. Used in place of a Pitot tube for velocity or total pressure measurements, it necessitates a much larger hole in the duct, often difficult and impractical to provide. When the velocities



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FIGURE

9-3

are high, there may be no appreciable errors at the high end of the scale and the instrument tends to read low on the discharge side of the fan and high on the inlet side.

The presence of dust, moisture, or corrosive material in the atmosphere presents a problem since the air passes through the instrument. In those instruments calibrated for use with a filter (the filter must always be used), the filter itself is a source of error because as the filter becomes plugged, its resistance increases and thus alters the amount of air passed to the swinging vane. The instrument requires periodic calibration and adjustment.

Thermal Anemometer (Figure 9-4): This type of instrument employs the principle that the amount of heat removed by an air stream passing a heated object is related to the velocity of the air stream. Since heat transfer to the air is a function of the number of molecules of air moving by a fixed monitoring point, the sensing element can be calibrated as a mass flow meter as well as a velocity recorder. Commercial instruments use a probe which consists of two integral sensors: a velocity sensor and a temperature sensor. The velocity sensor operates at a constant temperature — typically about 75 F above ambient conditions. Heating energy is supplied and controlled electrically by a battery-powered amplifier in the electronics circuit. The electrical current required to maintain the probe temperature in conjunction with the temperature sensor will provide an electrical signal which is proportional to the air velocity and is displayed on either a digital or analog meter. Additional features often include time integration of fluctuating readings and air temperature at the probe. Displays are available in either English or S.I. units.

The velocity sensor should be used with care in normal field use and is insensitive to mild particulate contamination. The probe can be used directly to measure air velocity in open spaces at air exhaust and supply air openings. Attachments

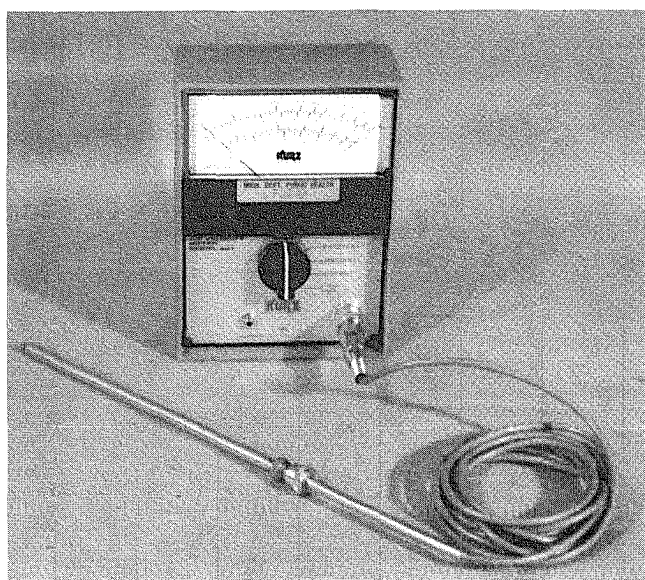


FIGURE 9-4. Thermal Anemometer

are available to measure velocity pressure. Due to the small diameter of the probe, measurements can be made directly inside of ducts using the measurement techniques described later for Pitot traverses. When used at supply or exhaust openings covered by grilles, the correction coefficient listed in Table 9.2 should be used.

Battery charging and maintenance is extremely important and the battery voltage must be checked prior to instrument use. The correction coefficients for this instrument are the same as a swinging vane anemometer (see Table 9.2). Instruments of this type require both initial and periodic calibration.

Smoke Tubes: Low velocity measurements may be made by timing the travel of smoke clouds through a known distance. Smoke trail observations are limited to velocities less than 150 fpm since high air velocities diffuse the smoke too rapidly. Commercially available, smoke tubes and candles are useful in the observation of flow patterns surrounding exhaust or supply openings. They also can be used for checking air movement and direction in plant space.

The visible plume is corrosive and should be used with care near sensitive processes or food preparation. Smoke candles are incendiary and thus cannot be used in flammable atmospheres. They should not be hand-held. Alternative methods of observing air flow patterns include the use of soap bubbles, water vapor cooled by dry ice (CO₂), and heated vegetable oil.

Tracer Gas: The principle of dilution sometimes is used to determine rate of air flow. A tracer gas is metered continuously into one or more intake ports (hood or duct openings) along with the entering air stream. After thorough mixing and system equilibrium has been established, air samples are collected at some point downstream — usually at or near the effluent point — and the concentration of the tracer gas in the exit stream is determined. The rate of air flow is readily calculated from the degree of dilution noted in the exit and feed gas concentrations (rate of air flow equals rate of feed divided by tracer gas concentrations).^(9,5)

The tracer gas usually is selected on the basis of the following: 1) ease of collection and analysis, 2) not present naturally in the process being studied, 3) not absorbed chemically or physically in the duct system, 4) non-reactive with other constituents of the gas stream, and 5) non-toxic, non-explosive, and non-odorous. Some frequently used tracer gases are sulfur hexafluoride and carbon dioxide.

9.3 CALIBRATION OF AIR MEASURING INSTRUMENTS

Direct-reading meters need regular calibration because they can be easily impaired by shock (dropping, jarring), dust, high temperatures, and corrosive atmospheres. Meters should be calibrated regularly and must be calibrated if they will not adjust to zero properly or if they have been subjected to rough handling and adverse atmospheres.

9.3.1 Design of a Calibrating Wind Tunnel: A typical

calibrating wind tunnel for testing air flow meters must have the following components:

1. *A satisfactory test section.* This is the section where the sensing probe or instrument is placed; it must be uniform in air flow both across the air stream and in line with the air flow. A section with a pronounced vena contracta and turbulence will not give satisfactory results.
2. *A satisfactory means of precisely metering the air flow.* The meter on this system must be accurate and with large enough scale graduations so that the volumetric flow rate is indicated within $\pm 1\%$. For convenience and time saving, a fixed single reading meter such as a venturi meter or orifice meter is preferable to a multi-point traverse type instrument such as a Pitot tube.
3. *A means of regulating and effecting air flow through the tunnel.* For usual calibrations of instruments used on heating, ventilating, and industrial exhaust systems, test velocities from approximately 50 to 8000 fpm are needed. Air flow regulation must be such that there is no disturbance in the test section. The regulating device must be easily and precisely set to the desired velocities. The fan must have sufficient capacity to develop the maximum velocity in the test section against the static pressure of the entire system.

To provide a satisfactory uniform flow in the test section, a bell-shaped streamline entry is necessary (Figure 9-5). There are various designs for this entry. One type is the elliptical approach in which curvature is similar to a one-quarter section of an ellipse in which the semi-major axis of the ellipse is equal to the duct diameter to which the entry is placed and the semi-minor axis is two-thirds of the semi-major axis. This type of entry can be made on a spinning lathe.

Actually, any type of smooth curved, bell-shaped entry which directs the air into the duct over a 180° angle should be satisfactory. A readily available entry is a tuba or Sousaphone bell. This bell entry should be connected to a 5.5" diameter smooth, seamless plastic tube. Ridges, small burrs, or obstructions should be filed so a smooth connection between horn and tube results.

For calibrating larger instruments such as the lower velocity swinging vane anemometer (Alnor velometer) and the rotating vane anemometer, a large rectangular test section of transparent plastic at least 2.5 ft² in cross-sectional area can be constructed with curved air foil inlets as shown in Figure 9-6. A fine mesh screen placed deep in the enclosure will assist in providing a uniform air flow in the test section.

A sharp-edged orifice, venturi meter, or a flow nozzle can be used as a metering device. Of these, the sharp-edged orifice has more resistance to flow but is more easily constructed, and it can be designed to be readily interchangeable for several

orifice sizes. The orifice can be mounted between two flanged sections sealed with gaskets as shown in Figure 9-5. Each orifice should be calibrated using a standard Pitot tube and manometer prior to use. For velocity measurements below 2,000 fpm, a micromanometer should be used.^(9.6)

Table 9-3 lists calculations for three sizes of orifices: 1.400", 2.625" and 4.900" diameters. When the orifices are placed in a 7" diameter duct and made to the precise dimensions given, no calibration is needed and the tabulated data in the Table will give volumetric flow rates within $\pm 5\%$ over the range of values shown for standard air density.

A centrifugal fan with sufficient capacity to exhaust 1,100 cfm at 10 "wg static pressure is needed for a wind tunnel with a 5.5" diameter test section using an orifice meter. Radial and backwardly inclined blade centrifugal fans are available with the required characteristics. The air flow can be changed with an adjustable damper at the discharge, a variable speed motor, or an adjustable drive on the fan.

The air flow for a sharp-edged orifice with pipe taps located 1" on either side of the orifice can be computed from the following equation for 2"-14" diameter ducts:

$$Q = 6KD^2 \sqrt{\frac{h}{\rho}} \quad [9.4]$$

where:

- Q = volumetric flow rate, cfm
- K = coefficient of air flow
- D = orifice diameter, inches
- h = pressure drop across orifice, "wg
- ρ = density, lb/m/ft³

The coefficient, K, is affected by the Reynolds number — a dimensionless value expressing flow conditions in a duct. The following equation gives a simplified method of calculating Reynolds number for standard air:

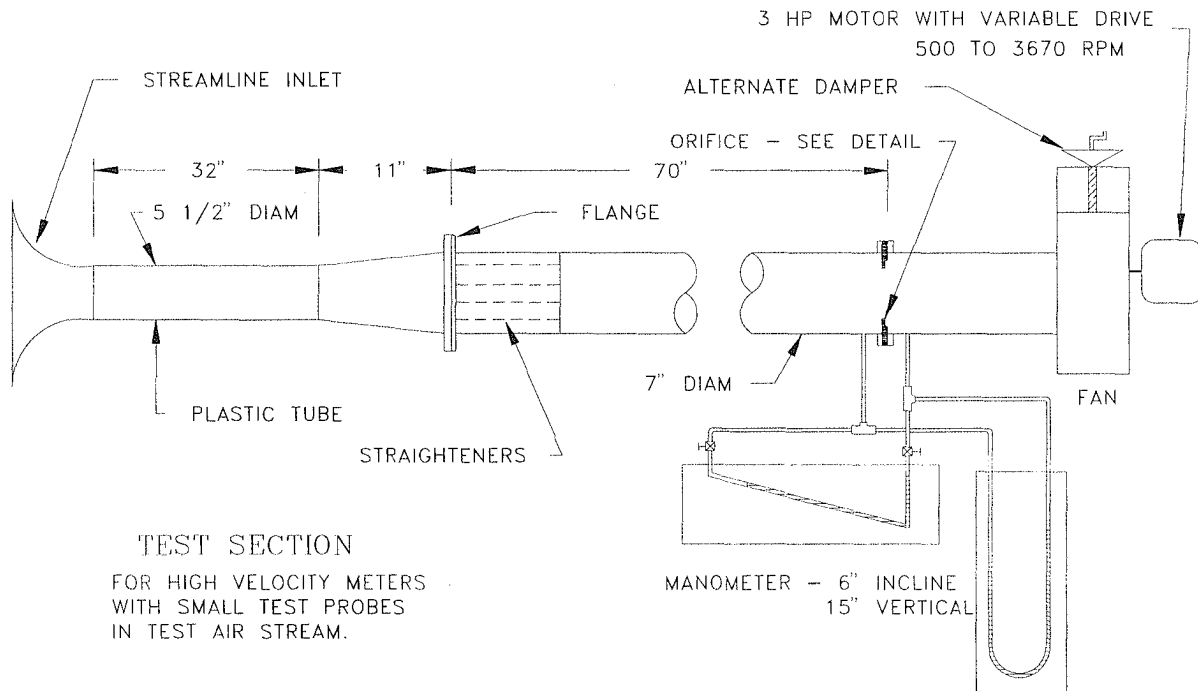
$$R = 8.4DV \quad [9.5]$$

where:

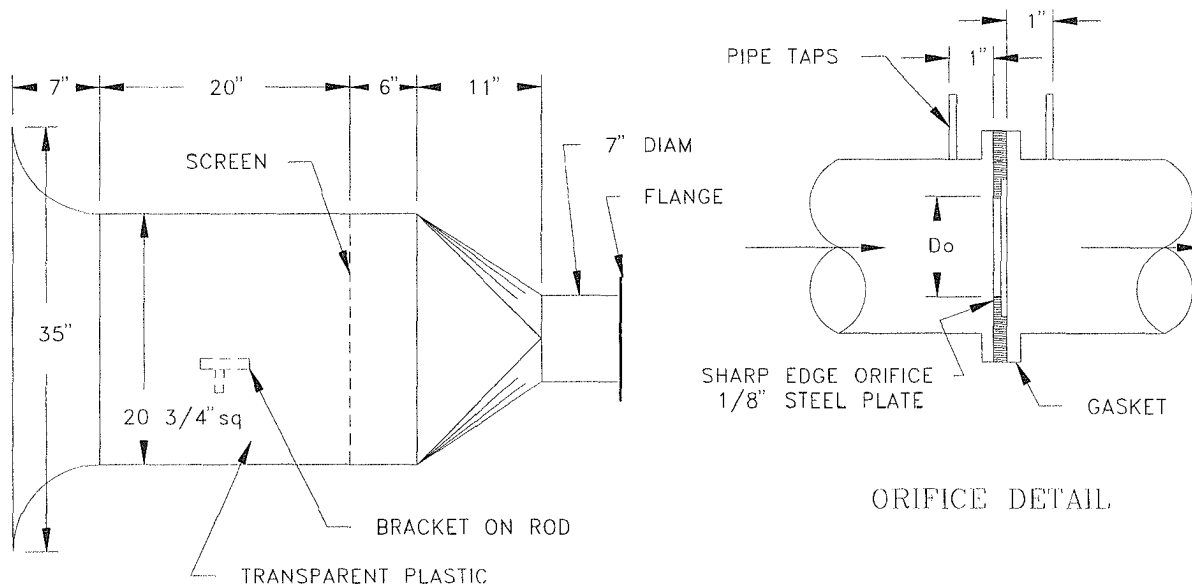
- R = Reynolds number, dimensionless
- V = velocity of air through orifice, fpm

The coefficient, K, can be selected from Table 9-4.^(9.3)

9.3.2 Use of Calibrating Wind Tunnel: Air velocity measuring instruments must be calibrated in the manner in which they are to be used in the field. Swinging vane and rotating vane anemometers are placed in the appropriate test section on a suitable support and the air velocity varied through the operating range of interest. Heated thermocouple instruments are calibrated in the same manner. Special Pitot tubes and duct probes of direct-reading instruments are placed through a suitable port in the circular duct section and the air velocity varied through the operating range of interest. Heated thermo-



CALIBRATION WIND TUNNEL



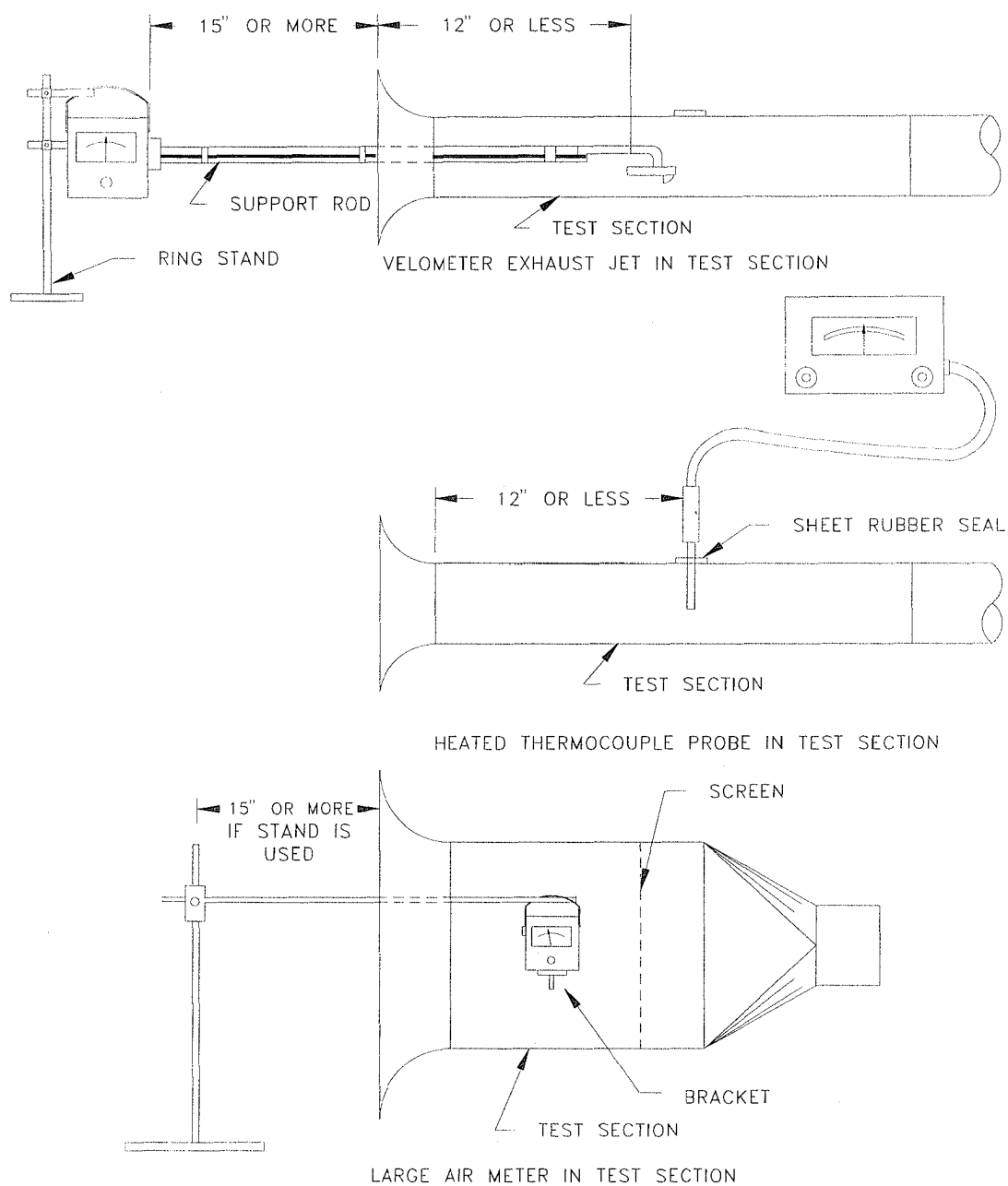
TEST SECTION
FOR LOW VELOCITY METERS WITH
LARGE AREA IN TEST AIR STREAM.

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FIGURE 9-5



KEEP TEST SECTION ENTRANCE
CLEAR OF OBSTRUCTIONS AND
FREE OF DRAFTS

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FIGURE

9-6

TABLE 9-3. Orifice Flow Rate (scfm) Versus Pressure Differential (in. of water)

ΔP "WC	ORIFICE SIZE			ΔP "WC	ORIFICE SIZE			ΔP "WC	ORIFICE SIZE		
	1.4"	2.625"	4.90"		1.4"	2.625"	4.90"		1.4"	2.625"	4.90"
0.02			57.1	1.22	28.7	101.4	410.3	4.10	52.3	185.3	746
0.04		18.7	78.8	1.24	28.9	102.3	413.6	4.20	52.9	187.5	755
0.06		22.8	95.3	1.26	29.2	103.1	416.9	4.30	53.5	189.7	763
0.08		26.2	109.2	1.28	29.4	103.9	420.1	4.40	54.1	191.9	772
0.10		29.3	121.5	1.30	29.6	104.7	423.4	4.50	54.7	194.0	781
0.12		32.1	132.6	1.32	29.8	105.5	426.5	4.60	55.3	196.2	789
0.14		34.6	142.8	1.34	30.1	106.3	429.7	4.70	55.9	198.3	797
0.16		37.0	152.3	1.36	30.3	107.1	432.9	4.80	56.5	200.4	806
0.18		39.2	161.2	1.38	30.5	107.9	436.0	4.90	57.1	202.4	814
0.20		41.3	169.6	1.40	30.7	108.6	439.1	5.00	57.6	204.4	822
0.22		43.3	177.6	1.42	30.9	109.4	442.2	5.10	58.2	206.5	830
0.24		45.2	185.2	1.44	31.2	110.2	445.2	5.20	58.8	208.5	838
0.26		47.0	192.6	1.46	31.4	110.9	448.3	5.30	59.3	210.4	846
0.28		48.8	199.6	1.48	31.6	111.7	451.3	5.40	59.9	212.4	854
0.30		50.5	206.5	1.50	31.8	112.4	454.3	5.50	60.4	214.3	862
0.32		52.1	213.0	1.52	32.0	113.2	457.2	5.60	61.0	216.3	869
0.34		53.7	219.4	1.54	32.2	113.9	460.2	5.70	61.5	218.2	877
0.36		55.3	225.6	1.56	32.4	114.6	463.1	5.80	62.0	220.0	884
0.38		56.8	231.6	1.58	32.6	115.4	466.0	5.90	62.6	221.9	892
0.40		58.3	237.5	1.60	32.8	116.1	468.9	6.00	63.1	223.8	899
0.42		59.7	243.2	1.62	33.0	116.8	471.8	6.10	63.6	225.6	907
0.44		61.1	248.8	1.64	33.2	117.5	474.7	6.20	64.1	227.4	914
0.46		62.4	254.3	1.66	33.4	118.2	477.5	6.30	64.6	229.2	921
0.48		63.8	259.6	1.68	33.6	118.9	480.3	6.40	65.1	231.0	928
0.50	18.5	65.1	264.9	1.70	33.8	119.6	483.1	6.50	65.6	232.8	935
0.52	18.8	66.4	270.0	1.72	34.0	120.3	485.9	6.60	66.1	234.6	942
0.54	19.2	67.6	275.0	1.74	34.2	121.0	488.7	6.70	66.6	236.3	949
0.56	19.5	68.9	280.0	1.76	34.4	121.7	491.5	6.80	67.1	238.1	956
0.58	19.9	70.1	284.8	1.78	34.6	122.4	494.2	6.90	67.6	239.8	963
0.60	20.2	71.3	289.6	1.80	34.8	123.1	496.9	7.00	68.1	241.5	970
0.62	20.6	72.4	294.3	1.82	35.0	123.8	499.7	7.10	68.5	243.2	977
0.64	20.9	73.6	298.9	1.84	35.2	124.4	502.4	7.20	69.0	244.9	984
0.66	21.2	74.7	303.4	1.86	35.4	125.1	505.0	7.30	69.5	246.5	990
0.68	21.5	75.8	307.9	1.88	35.5	125.8	507.7	7.40	69.9	248.2	997
0.70	21.8	76.9	312.3	1.90	35.7	126.4	510.4	7.50	70.4	249.9	1003
0.72	22.1	78.0	316.7	1.92	35.9	127.1	513.0	7.60	70.9	251.5	1010
0.74	22.4	79.1	320.9	1.94	36.1	127.8	515.6	7.70	71.3	253.1	1017
0.76	22.7	80.2	325.2	1.96	36.3	128.4	518.2	7.80	71.8	254.7	1023
0.78	23.0	81.2	329.3	1.98	36.5	129.1	520.8	7.90	72.2	256.4	1029
0.80	23.3	82.2	333.5	2.00	36.6	129.7	523.4	8.00	72.7	257.9	1036
0.82	23.6	83.2	337.5	2.10	37.5	132.9	536.2	8.10	73.1	259.5	1042
0.84	23.9	84.2	341.6	2.20	38.4	135.0	548.6	8.20	73.6	261.1	1048
0.86	24.1	85.2	345.5	2.30	39.3	139.0	560.8	8.30	74.0	262.7	1055
0.88	24.4	86.2	349.4	2.40	40.1	142.0	572.6	8.40	74.5	264.2	1061
0.90	24.7	87.2	353.3	2.50	40.9	144.9	584.3	8.50	74.9	265.8	1067
0.92	25.0	88.1	357.2	2.60	41.7	147.8	595.7	8.60	75.3	267.3	1073
0.94	25.2	89.1	361.0	2.70	42.5	150.6	606.9	8.70	75.7	268.8	1079
0.96	25.5	90.0	364.7	2.80	43.3	153.3	617.9	8.80	76.2	270.4	1085
0.98	25.8	91.0	368.4	2.90	44.0	156.0	628.6	8.90	76.6	271.9	1091
1.00	26.0	91.9	372.1	3.00	44.8	158.7	639.2	9.00	77.0	273.4	1097
1.02	26.3	92.8	375.7	3.10	45.5	161.3	649.6	9.10	77.4	274.9	1103
1.04	26.5	93.7	379.3	3.20	46.2	163.8	659.9	9.20	77.9	276.4	1109
1.06	26.8	94.6	382.9	3.30	46.9	166.4	670.0	9.30	78.3	277.8	1115
1.08	27.0	95.5	386.4	3.40	47.6	168.8	679.9	9.40	78.7	279.3	1121
1.10	27.3	96.3	390.0	3.50	48.3	171.3	689.7	9.50	79.1	280.8	1127
1.12	27.5	97.2	393.4	3.60	49.0	173.7	699.3	9.60	79.5	282.2	1132
1.14	27.8	98.1	396.9	3.70	49.7	170.1	708.8	9.70	79.9	283.6	1138
1.16	28.0	98.9	400.3	3.80	50.3	178.4	718.2	9.80	80.3	285.1	1144
1.18	28.2	99.8	403.7	3.90	51.0	180.7	727.5	9.90	80.7	286.5	1150
1.20	28.5	100.6	407.0	4.00	51.6	183.0	736.6	10.00	81.1	287.9	1155

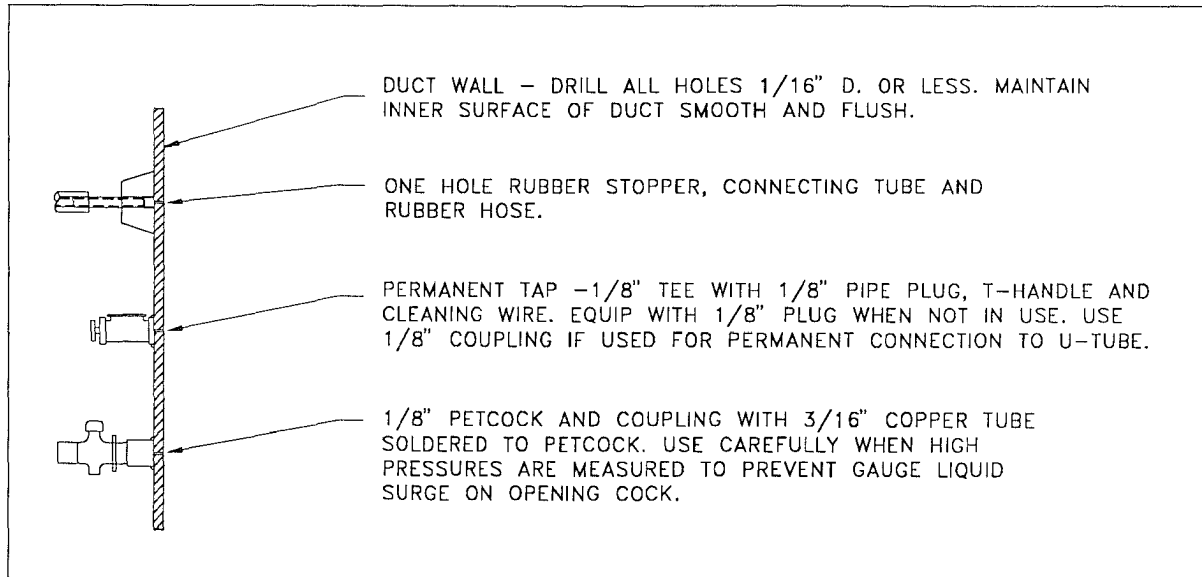


FIGURE 9-7. Static tap connections

couple instruments are calibrated in the same manner. Special Pitot tubes and duct probes of direct-reading instruments are placed through a suitable port in the circular duct section and calibrated throughout the operating range (Figure 9-6).^(9,9)

9.4 PRESSURE MEASUREMENT

At any point in an exhaust system, three air pressures exist which can be compared to the atmospheric pressure immediately surrounding the system. Typically, these pressures are measured in inches water gauge ("wg) and are related to each other as follows:

$$TP = SP + VP \quad [9.6]$$

where:

TP = total pressure, "wg

SP = static pressure, "wg

VP = velocity pressure, "wg

Static pressure is that pressure which tends to burst or collapse a duct and is positive when the pressure is above atmospheric and negative when below atmospheric. Velocity pressure is the pressure resulting from the movement of air and is always positive. Total pressure is the algebraic sum of the static pressure and velocity pressure and can be either positive or negative (see Figure 9-8).

9.4.1 Static Pressure is measured by a pressure measuring device, usually a simple U-tube manometer filled with oil, water, or other appropriate liquid and graduated in inches water gauge or similar reading pressure gauge. A vertical manometer is suitable for most static pressure measurements. The use of an inclined manometer will give increased accuracy and permits reading of lower values. For field measurement, one leg of the manometer is open to the atmosphere and

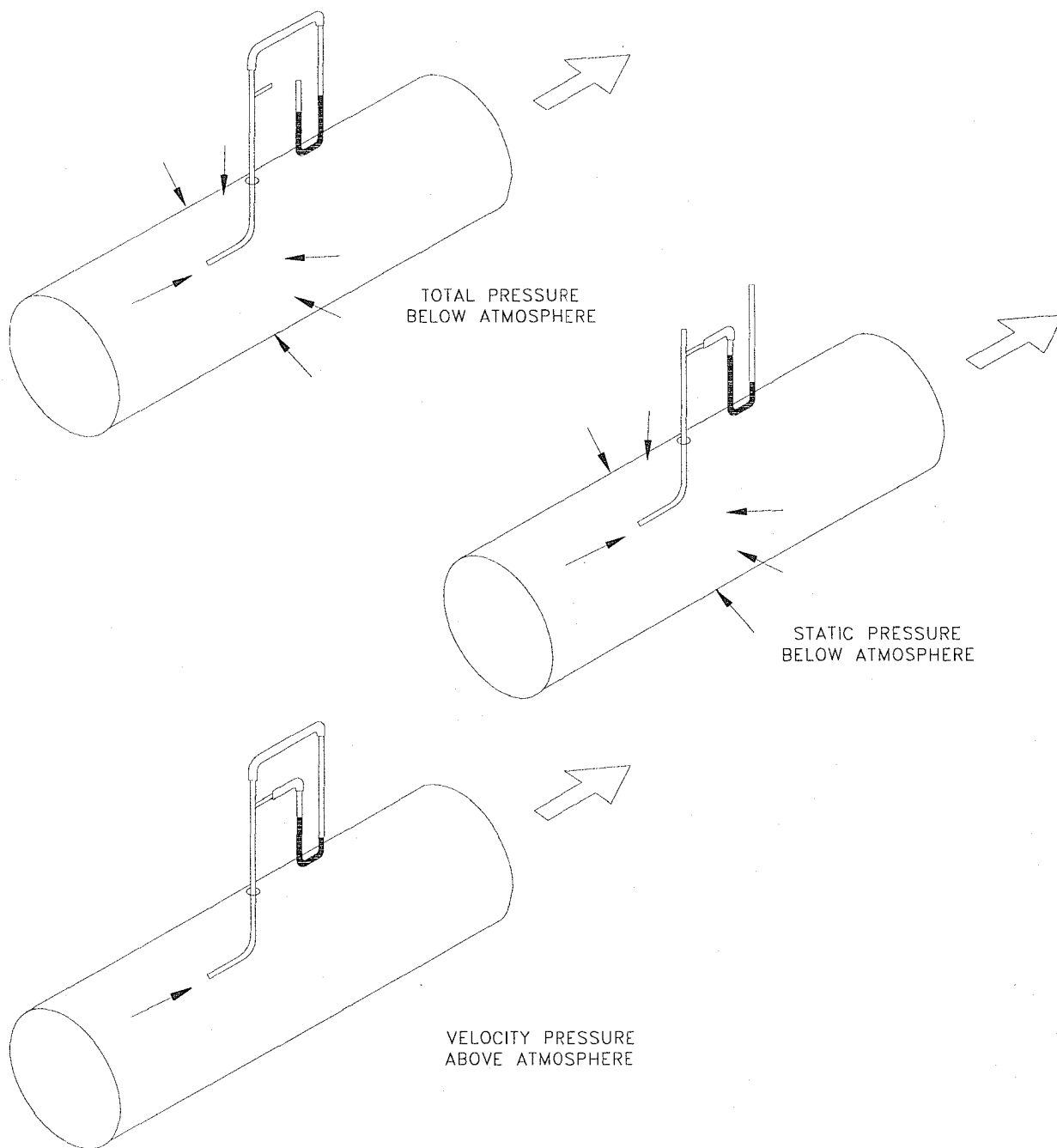
the other leg is connected with tubing held flush and tight against a small opening in the side of the pipe. Additional information concerning manometers and their construction can be found in References 9.1 and 9.2.

The location of the static pressure opening is usually not too important in obtaining a correct measurement except that one should avoid pressure measurement at the heel of an elbow or other location where static pressure may be incorrect because the direction of the velocity component is not parallel with the duct wall. It is usually advisable to drill 2–4 pressure holes at uniform distances around the duct in order to obtain an average and to detect any discrepancy in value.

The static pressure opening should be flush with the inner surface of the pipe wall and there should be no burrs or projections on the inner surface. The hole should be *drilled*, not punched. A 1/16"–1/8" hole is usually satisfactory since the size is not too important except for some types of instruments where air actually flows through the device (see Figure 9-7). The recommendations of the manufacturer concerning the size of the static pressure opening should be followed. A second method less likely to involve error is to use the static pressure element of a Pitot tube as shown in Figure 9-8. In use, the instrument must be pointed upstream and parallel to the duct for accurate measurement.

9.4.2 Hood Static Pressure: The hood static pressure method of estimating air flow into an exhaust hood or duct is based on the principle of the orifice; i.e., the inlet opening simulating an orifice. This method is quick, simple, and practical. It is a fairly accurate estimation of the volumetric air flow in branch exhaust ducts if the static pressure or suction measurement can be made at a point one to three duct diameters of straight duct downstream from the throat of the exhaust

$$\text{TOTAL PRESSURE} = \text{STATIC PRESSURE} + \text{VELOCITY PRESSURE}$$



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FIGURE 9-8

TABLE 9-4. Values of K in Equation 9.10 for Different Orifice Diameters to Duct-Diameter Ratios (d/D) and Different Reynolds Numbers*

d/D	Reynolds Number in Thousands						
	25	50	100	230	500	1000	10,000
0.100	0.605	0.601	0.598	0.597	0.596	0.595	0.595
0.200	0.607	0.603	0.600	0.599	0.598	0.597	0.597
0.300	0.611	0.606	0.603	0.603	0.601	0.600	0.600
0.400	0.621	0.615	0.611	0.610	0.609	0.608	0.608
0.450	0.631	0.624	0.619	0.617	0.615	0.615	0.615
0.500	0.644	0.634	0.628	0.626	0.624	0.623	0.623
0.550	0.663	0.649	0.641	0.637	0.635	0.634	0.634
0.600	0.686	0.668	0.658	0.653	0.650	0.649	0.649
0.650	0.717	0.695	0.680	0.674	0.670	0.668	0.667
0.700	0.755	0.723	0.707	0.699	0.694	0.692	0.691
0.750	0.826	0.773	0.747	0.734	0.726	0.723	0.721

*For duct diameters of 2" to 14" inclusive.

inlet and if an accurate analysis of the hood entry loss can be made.

This technique involves the measuring of hood static pressure by means of a U-tube manometer at one or more holes (preferably four, spaced 90° apart), one duct diameter downstream from the throat for all hoods having tapers, and three duct diameters from the throat for flanged or plain duct ends. The holes should be drilled 1/16"–1/8" in diameter or less; the holes should not be punched as inwardly projecting jagged edges of metal will disturb the air stream. The U-tube manometer is connected to each hole in turn by means of a thick-walled soft rubber tube and the difference in the height of the water columns is read in inches.

If an elbow intervenes between the hood and the suction-measurement location, the pressure loss caused by the elbow should be subtracted from the reading to indicate the suction produced by the hood and throat alone (see Chapter 5, Figure 5-14).

The values for hood entry loss coefficient (h_e) for various hood shapes are listed in Chapter 5, Figure 5-13. When the hood static pressure (SP_h) is known, the volumetric flow rate can be determined by the following equation:

$$Q = 1096A \sqrt{\frac{SP_h}{(1+h_e)\rho}} \quad [9.7]$$

where:

Q = volumetric flow rate, cfm

A = area of duct attached to hood, ft²

SP_h = U-tube average manometer reading, "wg

h_e = hood entry loss coefficient

ρ = actual gas density, lb/m/ft³

For standard air, Equation 9.7 becomes:

$$Q = 4005 A C_e \sqrt{SP_h} \quad [9.8]$$

As noted above, the coefficient of entry, C_e , can also be used in conjunction with SP_h to determine Q . To facilitate repeated measurements, it is sometimes convenient to post the value of C_e directly on the hood near the point where SP_h is measured.

9.4.3 Hood Static Pressure Interpretation: If the hood static pressure is known while a system is functioning properly, its continued effectiveness can be assured so long as the original value is not changed. Any change from the original measurement can only indicate a change in velocity in the branch and, consequently, a change in volumetric flow through the hood. This relationship will be true unless: 1) a hood design change has affected the entrance loss; 2) there are obstructions or accumulations in the hood or branch ahead of the point where the hood static pressure reading was taken; or 3) the system has been altered or added to. Depending on the location of the obstruction in the duct system, restrictions of the cross-sectional area will reduce the air flow although hood suction may increase or decrease.

Pressure readings vary as the square of the velocity or volumetric flow rate. To illustrate, an indicated reduction in static pressure readings of 30% would reflect a volumetric flow rate (or velocity) decrease of 6%.

A marked reduction in hood static pressure often can be traced to one or more of the following conditions:

1. Reduced performance of the exhaust fan caused by reduced shaft speed due to belt slippage, wear, or accumulation on rotor or casing that would obstruct air flow.
2. Reduced performance caused by defects in the exhaust piping such as an accumulation in branch or main ducts

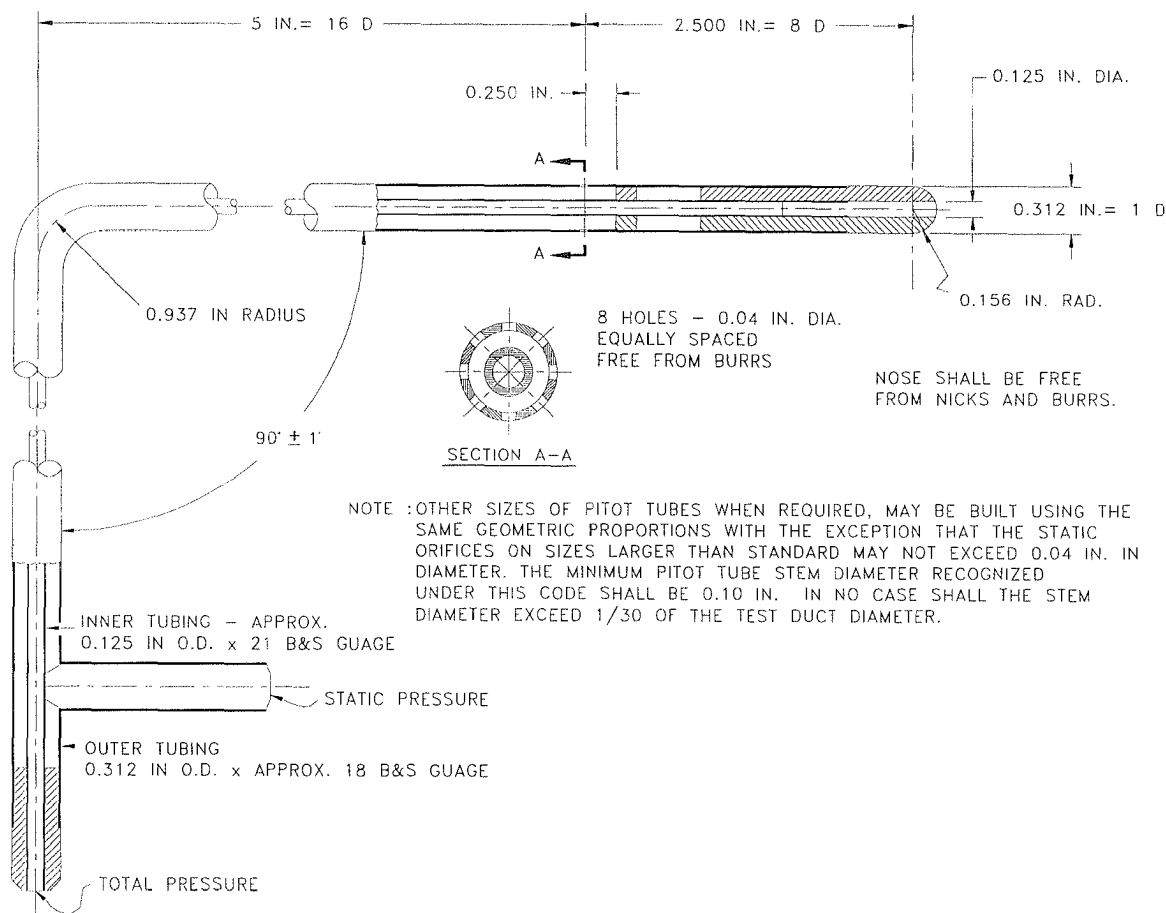


FIGURE 9-9. Standard Pitot tubes

due to insufficient conveying velocities, condensation of oil or water vapors on duct walls, adhesive characteristics of material exhausted, or leakage losses caused by loose clean-out doors, broken joints, holes worn in duct (most frequently in elbows), poor connection to exhaustor inlet, accumulations in ducts or on fan blades.

3. Reduced air flow rate also can be charged to additional exhaust duct openings added to the system (sometimes systems are designed for future connections and more air than required is handled by present branches until future connections are made) or change of setting of blast gates in branch lines. Blast gates adjust the air distribution between the various branches. Tampering with the blast gates can seriously affect such distribution and therefore they should be locked in place immediately after the system has been installed and its effectiveness verified. Fan volume control dampers also should be checked.
4. Reduced volumetric flow may be caused by increased pressure loss through the dust collector due to lack of maintenance, improper operation, wear, etc. These

effects will vary with the collector design. Refer to operation and maintenance instructions furnished with the collector or consult the equipment manufacturer.

9.4.4 Velocity Pressure: For measuring velocity pressure to determine air velocity, a standard Pitot tube may be used. A large volume of research and many applications have been devoted to the subject of flow measurements by this instrument, which was developed by Henry Pitot in 1734 while a student in Paris, France. A standard Pitot tube (see Figure 9-9) needs no calibration if carefully made and the accuracy of velocity pressure readings obtained are considered to be accurate at velocities above 600 fpm (see Table 9-1). For more details concerning specifications and application of the Pitot tube, see the "Standard Test Code" published by the American Society of Heating, Refrigerating and Air Conditioning Engineers and the Air Moving and Conditioning Association.^(9.1, 9.4)

The device consists of two concentric tubes — one measures the total or impact pressure existing in the air stream; the other measures the static pressure only. When the annular space and the center tube are connected across a manometer, the difference between the total pressure and the static pressure is indicated on the manometer. This difference is the

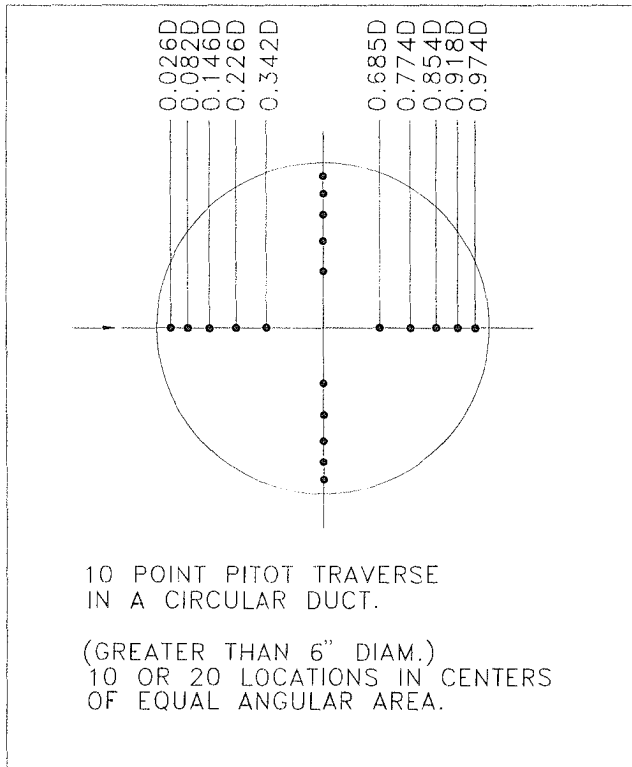


FIGURE 9-10A. 10-point Pitot traverse in a circular duct

velocity pressure.

The velocity pressure can be used to compute the velocity of the air stream if the density of the air is known. The following equation can be used:

$$V = 1096 \sqrt{\frac{VP}{\rho}} \quad [9.9]$$

where:

VP = velocity pressure, "wg
 ρ = actual gas density, lb/m/ft³

Where air is at standard conditions ($\rho = 0.075$ lb/m/ft³), Equation 9.9 becomes:

$$V = 4005 \sqrt{VP} \quad [9.10]$$

For example, if the temperature of the air stream varies more than 30 F from standard air (70 F and 29.92 "Hg) or the altitude of the site is more than 1,000 feet above or below sea level or the moisture content of the air is 0.02 lb/lb of dry air or greater, the actual gas density (ρ) must be used.

Velocity pressure versus velocity tables for standard air can be found in Chapter 5 (Tables 5-7A and 5-7B). These tables can be used for air at densities other than standard conditions by determining an equivalent velocity pressure.

$$VP_e = \frac{VP_m}{df} \quad [9.11]$$

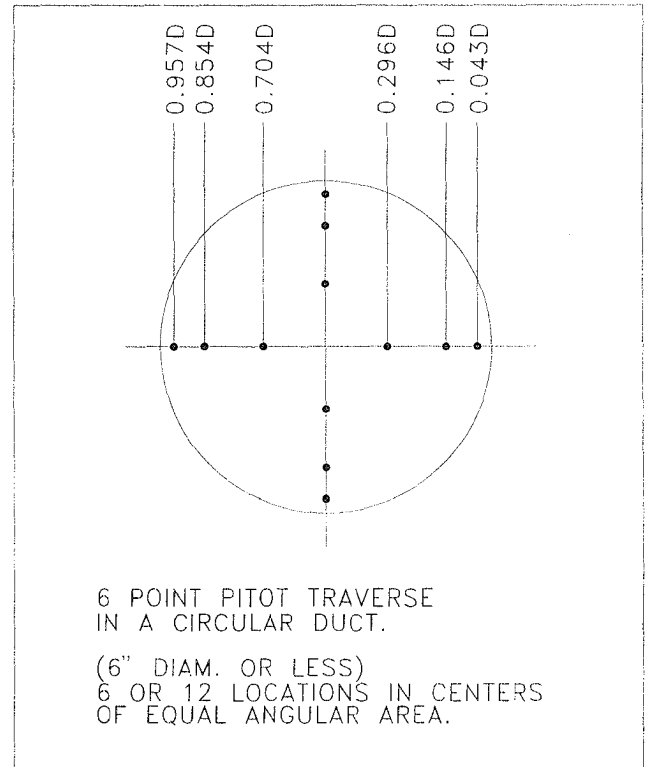


FIGURE 9-10B. 6-point Pitot traverse in a circular duct

where:

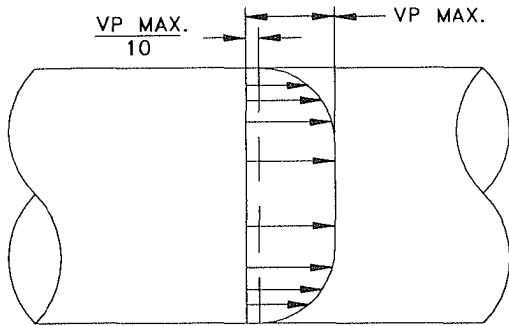
VP_e = equivalent velocity pressure, "wg
 VP_m = measured velocity pressure, "wg
 df = density factor coefficient

The equivalent VP then can be used in the velocity pressure versus velocity table selected to give the actual velocity at duct conditions.

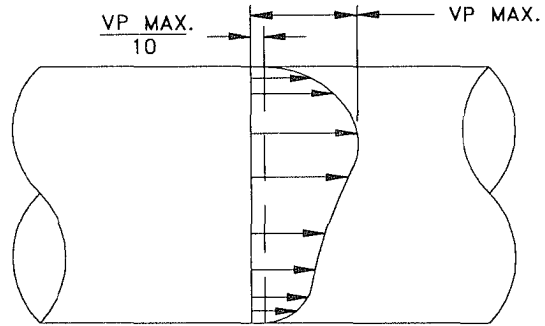
A number of techniques can be used to determine the volumetric flow rate at hood openings and at other points in an exhaust system using the fluid flow principles previously described. The method selected will depend on the degree of accuracy required, time available for testing, and the type of test data required. It is extremely important that measurements taken at the time of the tests include all necessary information to determine the gas density to permit the calculation of the actual velocity and volumetric flow rate.

9.5 PITOT TRAVERSE METHOD

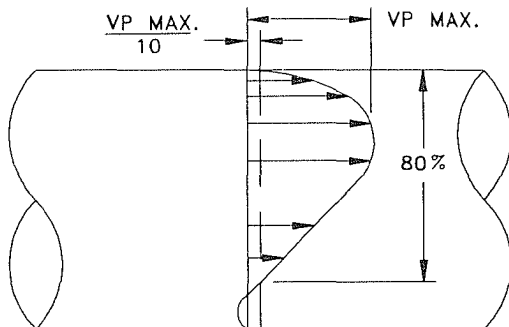
Because the air flow in the cross-section of a duct is not uniform, it is necessary to obtain an average by measuring VP at points in a number of equal areas in the cross-section. The usual method is to make two traverses across the diameter of the duct at right angles to each other. Readings are taken at the center of annular rings of equal area (see Figures 9-10A and 9-10B). Whenever possible, the traverse should be made 7 1/2 duct diameters or more downstream from any major air



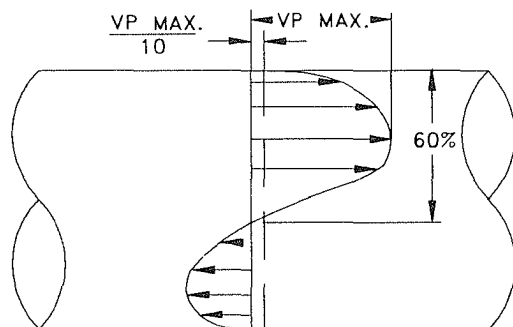
A: FULLY DEVELOPED VP DISTRIBUTION



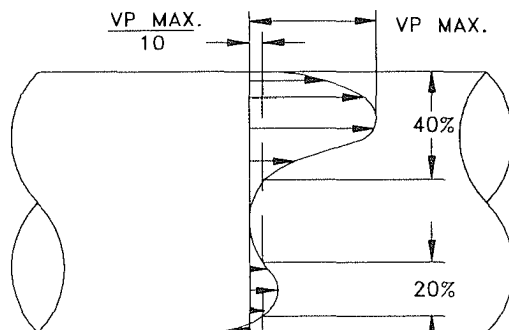
B: GOOD VP DISTRIBUTION. (ALSO SATISFACTORY FOR FLOW INTO FAN INLETS. BUT MAY BE UNSATISFACTORY FOR FLOW INTO INLET BOXES - MAY PRODUCE SWIRL IN BOXES.)



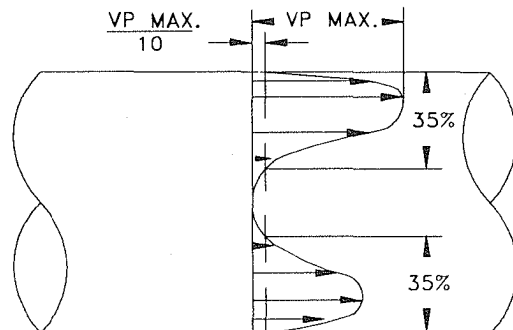
C: SATISFACTORY VP DISTRIBUTION - MORE THAN 75% OF VP READINGS GREATER THAN $\frac{VP \text{ MAX.}}{10}$



D: DO NOT USE! UNSATISFACTORY VP DISTRIBUTION - LESS THAN 75% OF VP READINGS GREATER THAN $\frac{VP \text{ MAX.}}{10}$



E: DO NOT USE! UNSATISFACTORY VP DISTRIBUTION - LESS THAN 75% OF VP READINGS GREATER THAN $\frac{VP \text{ MAX.}}{10}$



F: DO NOT USE! UNSATISFACTORY VP DISTRIBUTION - LESS THAN 75% OF VP READINGS GREATER THAN $\frac{VP \text{ MAX.}}{10}$

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FIGURE 9-11

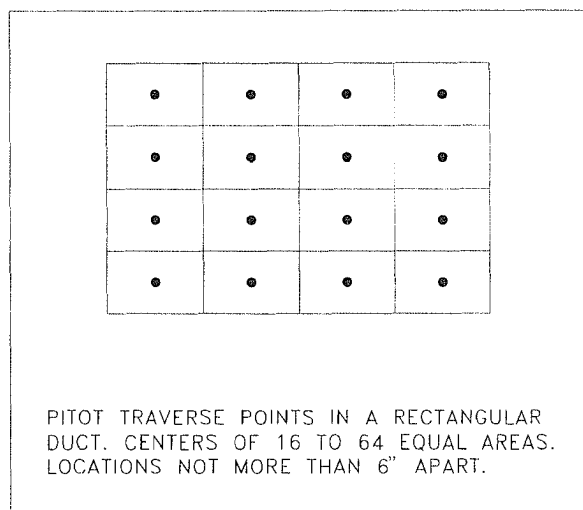


FIGURE 9-12. Pitot traverse points in a rectangular duct

disturbance such as an elbow, hood, branch entry, etc. Where measurements are made closer to disturbances, the results must be considered subject to some doubt and checked against a second location. If agreement within 10% of the two traverses is obtained, reasonable accuracy can be assumed and the average of the two readings used. Where the variation exceeds 10%, a third location should be selected and two air flows in the best agreement averaged and used. The use of a single centerline reading for obtaining average velocity is a very coarse approximation and is NOT recommended.

The reason for the uncertainty and variation in measurements is the non-uniformity of air flow after a disturbance. Figure 9-11 shows some air flow patterns that could develop after a disturbance and the resulting difficulties in obtaining good reliable measures are evident.

For round ducts 6" and smaller, at least 6 traverse points should be used. For round ducts larger than 6" diameter, at least 10 traverse points should be employed. The number of traverse locations on each diameter and the number of tra-

verse diameters required are determined by the need for accuracy and the symmetry of the measured values. Where uniform velocity pressure profiles exist, a single traverse along one diameter may be adequate. Where the values are moderately skewed, the use of two diameters is recommended. For greater accuracy, a third diameter should be used. Six, ten, and twenty point traverse points for various duct diameters are given in Tables 9-5, 9-6, and 9-7. To minimize errors, a Pitot tube smaller than the standard 5/16" O.D. should be used in ducts less than 12" in diameter.

For square or rectangular ducts, the procedure is to divide the cross-section into a number of equal rectangular areas and measure the velocity pressure at the center of each. The number of readings should not be less than 16. However, enough readings should be made so the greatest distance between centers is approximately 6" (see Figure 9-12).

The following data are essential and more detailed data may be taken if desired:

- The area of the duct at the traverse location.
- Velocity pressure at each point in the traverse.
- Temperature of the air stream at the time and location of the traverse.

The velocity pressure readings obtained are converted to velocities and the velocities, **not the velocity pressures**, are averaged. Where more convenient, *the square root of each of the velocity pressures may be averaged* and this value then converted to velocity (average). The measured air flow is then the average velocity multiplied by the cross-sectional area of the duct ($Q = VA$). Where conditions are not standard, see "Corrections for Non-Standard Conditions."

The Pitot tube cannot be used for measuring velocities less than 600 fpm in the field. It is susceptible to plugging in air streams with heavy dust and/or moisture loadings. A vibration free mounting is necessary if using a liquid manometer. See Reference 9.10 for special instrumentation which can be used to measure low velocities.

Modified Pitot Tubes: Modified Pitot tubes have been made in an effort to reduce plugging difficulties encountered in heavy dust streams or to increase manometer differentials enabling the measurement of lower velocities in the field. These are referred to as "S"-type (Staubscheide) tubes. They usually take the form of two relatively large impact openings, one facing upstream and the other facing downstream. Such tubes are useful when thick-walled ducts, such as boiler stacks, make it difficult or impossible to insert a conventional Pitot tube through any reasonably sized port opening. They require only initial calibration for all conditions.

Measurements made with an "S"-type Pitot tube cannot be used directly. The tube first must be calibrated against a standard Pitot tube and the velocity pressure measured corrected to the actual velocity pressure.

TABLE 9-5. Distance from Wall of Round Pipe to Point Reading (nearest 1/8 inch) for 6-Point Traverse

Duct DIA	R ₁ .043 DIA	R ₂ .146 DIA	R ₃ .296 DIA	R ₄ .704 DIA	R ₅ .854 DIA	R ₆ .957 DIA
3	1/8	1/2	7/8	2 1/8	2 1/2	2 7/8
3 1/2	1/8	1/2	1	2 1/2	3	3 3/8
4	1/8	5/8	1 1/8	2 7/8	3 3/8	3 7/8
4 1/2	1/4	5/8	1 3/8	3 1/8	3 7/8	4 1/4
5	1/4	3/4	1 1/2	3 1/2	4 1/4	4 3/4
5 1/2	1/4	3/4	1 5/8	3 7/8	4 3/4	5 1/4
6	1/4	7/8	1 3/4	4 1/4	5 1/8	5 3/4

TABLE 9-6. Distance from Wall of Round Pipe to Point of Reading (nearest 1/8 inch) for 10-Point Traverse

DUCT DIA	R ₁ 0.026DIA	R ₂ 0.082DIA	R ₃ 0.146DIA	R ₄ 0.226DIA	R ₅ 0.342DIA	R ₆ 0.658DIA	R ₇ 0.774DIA	R ₈ 0.854DIA	R ₉ 0.918DIA	R ₁₀ 0.974DIA
4	1/8	3/8	5/8	7/8	1 3/8	2 5/8	3 1/8	3 3/8	3 5/8	3 7/8
4 1/2	1/8	3/8	5/8	1	1 1/2	3	3 1/2	3 7/8	4 1/8	4 3/8
5	1/8	3/8	3/4	1 1/8	1 3/4	3 1/4	3 7/8	4 1/4	4 5/8	4 7/8
5 1/2	1/8	1/2	3/4	1 1/4	1 7/8	3 5/8	4 1/4	4 3/4	5	5 3/8
6	1/8	1/2	7/8	1 3/8	2	4	4 5/8	5 1/8	5 1/2	5 7/8
7	1/8	5/8	1	1 5/8	2 3/8	4 5/8	5 3/8	6	6 3/8	6 7/8
8	1/4	5/8	1 1/8	1 3/4	2 3/4	5 1/4	6 1/4	6 7/8	7 3/8	7 3/4
9	1/4	3/4	1 1/4	2	3 1/8	5 7/8	7	7 3/4	8 1/4	8 3/4
10	1/4	7/8	1 1/2	2 1/4	3 3/8	6 5/8	7 3/4	8 1/2	9 1/8	9 3/4
11	1/4	7/8	1 5/8	2 1/2	3 3/4	7 1/4	8 1/2	9 3/8	10 1/8	10 3/4
12	3/8	1	1 3/4	2 3/4	4 1/8	7 7/8	9 1/4	10 1/4	11	11 5/8
13	3/8	1	1 7/8	2 7/8	4 1/2	8 1/2	10 1/8	11 1/8	12	12 5/8
14	3/8	1 1/8	2	3 1/8	4 3/4	9 1/4	10 7/8	12	12 7/8	13 5/8
15	3/8	1 1/4	2 1/4	3 3/8	5 1/8	9 7/8	11 5/8	12 3/4	13 3/4	14 5/8
16	3/8	1 1/4	2 3/8	3 5/8	5 1/2	10 1/2	12 3/8	13 5/8	14 3/4	15 5/8
17	1/2	1 3/8	2 1/2	3 7/8	5 3/4	11 1/4	13 1/8	14 1/2	15 5/8	16 1/2
18	1/2	1 1/2	2 5/8	4 1/8	6 1/8	11 7/8	13 7/8	15 3/8	16 1/2	17 1/2
19	1/2	1 1/2	2 3/4	4 1/4	6 1/2	12 1/2	14 3/4	16 1/4	17 1/2	18 1/2
20	1/2	1 5/8	2 7/8	4 1/2	6 7/8	13 1/8	15 1/2	17 1/8	18 3/8	19 1/2
22	5/8	1 3/4	3 1/4	5	7 1/2	14 1/2	17	18 3/4	20 1/4	21 3/8
24	5/8	2	3 1/2	5 1/2	8 1/4	15 3/4	18 1/2	20 1/2	22	23 3/8
26	5/8	2 1/8	3 3/4	5 7/8	8 7/8	17 1/8	20 1/8	22 1/4	23 7/8	25 3/8
28	3/4	2 1/4	4 1/8	6 3/8	9 5/8	18 3/8	21 5/8	23 7/8	25 3/4	27 1/4
30	3/4	2 1/2	4 3/8	6 3/4	10 1/4	19 3/4	23 1/4	25 5/8	27 1/2	29 1/4
32	7/8	2 5/8	4 5/8	7 1/4	11	21	24 3/4	27 3/8	29 3/8	31 1/8
34	7/8	2 3/4	5	7 3/4	11 5/8	22 3/8	26 1/4	29	31 1/4	33 1/8
36	1	3	5 1/4	8 1/8	12 3/8	23 5/8	27 7/8	30 3/4	33	35
38	1	3 1/8	5 1/2	8 5/8	13	25	29 3/8	32 1/2	34 7/8	37
40	1	3 1/4	5 7/8	9	13 5/8	26 3/8	31	34 1/8	36 3/4	39
42	1 1/8	3 3/8	6 1/8	9 1/2	14 3/8	27 5/8	32 1/2	35 7/8	38 5/8	40 7/8
44	1 1/8	3 5/8	6 3/8	10	15	29	34	37 5/8	40 3/8	42 7/8
46	1 1/4	3 3/4	6 3/4	10 3/8	15 3/4	30 1/4	35 5/8	39 1/4	42 1/4	44 3/4
48	1 1/4	4	7	10 7/8	16 3/8	31 5/8	37 1/8	41	44	46 3/4

TABLE 9-7. Distance from Wall of Round Pipe to Point of Reading (nearest 1/8 inch) for 20 Point Traverse

Duct dia	R_1 0.013D	R_2 0.039D	R_3 0.067D	R_4 0.097D	R_5 0.129D	R_6 0.165D	R_7 0.204D	R_8 0.250D	R_9 0.306D	R_{10} 0.388D
	R_{11} 0.612D	R_{12} 0.694D	R_{13} 0.750D	R_{14} 0.796D	R_{15} 0.835D	R_{16} 0.871D	R_{17} 0.903D	R_{18} 0.933D	R_{19} 0.961D	R_{20} 0.987D
40	1/2 24 1/2	1 1/2 27 3/4	2 5/8 30	3 7/8 31 7/8	5 1/8 33 3/8	6 5/8 34 7/8	8 1/8 36 1/8	10 37 3/8	12 1/4 38 1/2	15 1/2 39 1/2
42	1/2 25 3/4	1 5/8 29 1/8	2 7/8 31 1/2	4 1/8 33 3/8	5 3/8 35 1/8	6 7/8 36 5/8	8 5/8 37 7/8	10 1/2 39 1/8	12 7/8 40 3/8	16 1/4 41 1/2
44	1/2 26 7/8	1 3/4 30 1/2	3 33	4 1/4 35	5 5/8 36 3/4	7 1/4 38 3/8	9 39 3/4	11 41	13 1/2 42 1/4	17 1/8 43 1/2
46	5/8 28 1/8	1 3/4 31 7/8	3 1/8 34 1/2	4 1/2 36 5/8	6 38 3/8	7 5/8 40	9 3/8 41 1/2	11 1/2 42 7/8	14 1/8 44 1/4	17 7/8 45 3/8
48	5/8 29 3/8	1 7/8 33 1/4	3 1/4 36	4 5/8 38 1/4	6 1/4 40 1/8	7 7/8 41 3/4	9 3/4 43 3/8	12 44 3/4	14 3/4 46 1/8	18 5/8 47 3/8
50	5/8 30 5/8	2 34 5/8	3 3/8 37 1/2	4 7/8 39 3/4	6 1/2 41 3/4	8 1/4 43 1/2	10 1/4 45 1/8	12 1/2 46 5/8	15 3/8 48	19 3/8 49 3/8
52	5/8 31 7/8	2 36 1/8	3 1/2 39	5 41 3/8	6 3/4 43 1/2	8 1/2 45 1/4	10 5/8 47	13 48 1/2	15 7/8 50	20 1/8 51 3/8
54	5/8 33	2 1/8 37 1/2	3 5/8 40 1/2	5 1/4 43	7 45 1/8	8 7/8 47	11 48 3/4	13 1/2 50 3/8	16 1/2 51 7/8	21 53 3/8
56	3/4 34 1/4	2 1/8 38 7/8	3 3/4 42	5 3/8 44 5/8	7 1/4 46 3/4	9 1/4 48 3/4	11 3/8 50 5/8	14 52 1/4	17 1/8 53 7/8	21 3/4 55 1/4
58	3/4 35 1/2	2 1/4 40 1/4	3 7/8 43 1/2	5 5/8 46 1/8	7 1/2 48 1/2	9 1/2 50 1/2	11 7/8 52 3/8	14 1/2 54 1/8	17 3/4 55 3/4	22 1/2 57 1/4
60	3/4 36 3/4	2 3/8 41 5/8	4 45	5 7/8 47 3/4	7 3/4 50 1/8	9 7/8 52 1/4	12 1/4 54 1/8	15 56	18 3/8 57 5/8	23 1/4 59 1/4
62	3/4 37 7/8	2 3/8 43	4 1/8 46 1/2	6 49 3/8	8 51 3/4	10 1/4 54	12 5/8 56	15 1/2 57 7/8	19 59 5/8	24 1/8 61 1/4
64	3/4 39 1/8	2 1/2 44 3/8	4 1/4 48	6 1/4 50 7/8	8 1/4 53 1/2	10 1/2 55 3/4	13 1/8 57 3/4	16 59 3/4	19 5/8 61 1/2	24 7/8 63 1/4
66	7/8 40 3/8	2 5/8 45 3/4	4 3/8 49 1/2	6 3/8 52 1/2	8 1/2 55 1/8	10 7/8 57 1/2	13 1/2 59 5/8	16 1/2 61 5/8	20 1/4 63 3/8	25 5/8 65 1/8
68	7/8 41 5/8	2 5/8 47 1/8	4 1/2 51	6 5/8 54 1/8	8 3/4 56 3/4	11 1/4 59 1/4	13 7/8 61 3/8	17 63 1/2	20 7/8 65 3/8	26 3/8 67 1/8
70	7/8 42 7/8	2 3/4 48 1/2	4 3/4 52 1/2	6 3/4 55 3/4	9 58 1/2	11 1/2 61	14 1/4 63 1/4	17 1/2 65 1/4	21 1/2 67 1/4	27 1/8 69 1/8
72	7/8 44	2 3/4 50	4 7/8 54	7 57 1/4	9 1/4 60 1/8	11 7/8 62 3/4	14 3/4 65	18 67 1/8	22 69 1/4	28 71 1/8
74	7/8 45 1/4	2 7/8 51 3/8	5 55 1/2	7 1/8 58 7/8	9 1/2 61 7/8	12 1/8 64 1/2	15 1/8 66 7/8	18 1/2 69	22 5/8 71 1/8	28 3/4 73 1/8
76	1 46 1/2	3 52 3/4	5 1/8 57	7 3/8 60 1/2	9 7/8 63 1/2	12 1/2 66 1/8	15 1/2 68 5/8	19 70 7/8	23 1/4 73	29 1/2 75
78	1 47 3/4	3 54 1/8	5 1/4 58 1/2	7 1/2 62 1/8	10 1/8 65 1/8	12 7/8 67 7/8	15 7/8 70 1/2	19 1/2 72 3/4	23 7/8 75	30 1/4 77
80	1 49	3 1/8 55 1/2	5 3/8 60	7 3/4 63 5/8	10 3/8 66 7/8	13 1/8 69 5/8	16 3/8 72 1/4	20 74 5/8	24 1/2 76 7/8	31 79

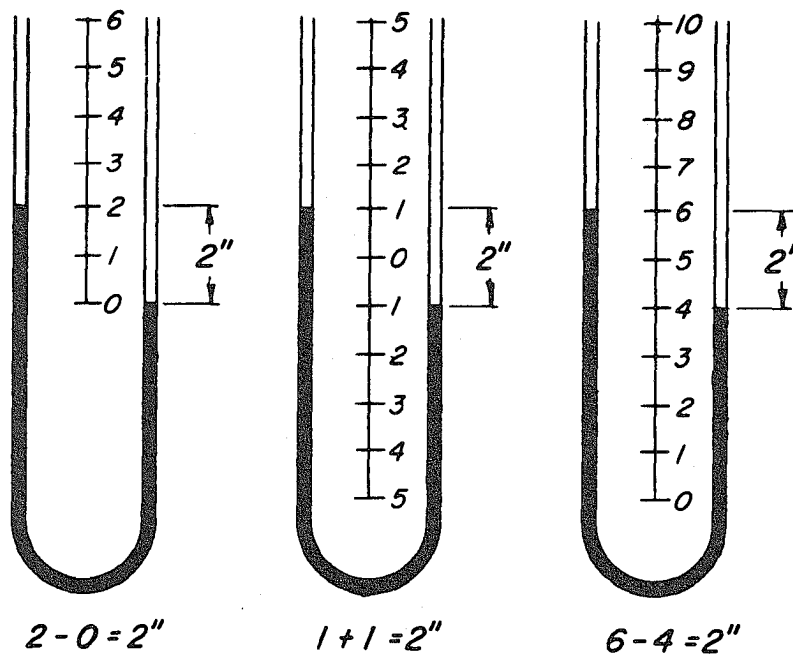


FIGURE 9-13. U-tube manometer

Other modified forms of the Pitot tube are the air foil pitometer, the Pitot venturi, and the air speed nozzle, to name a few.^(9.7, 9.8)

Pressure Sensors: Pressure sensors can be used in conjunction with the pitot tube to measure pressures existing within ventilation systems. These devices are described below.

U-Tube Manometer: The vertical U-tube (see Figure 9-13) is the simplest type of pressure gauge. Usually calibrated in inches water gauge, it is used with various fluid media such as alcohol, mercury, oil, water, kerosene and special manometer fluids. The U-tube may be used for either portable or stationary applications. Available commercial units offer a wide latitude in range, number of columns, and styles. Tubes are usually of all-plastic construction to minimize breakage. One leg may be replaced by a reservoir or well (well-type manometer) with the advantage of easier manometer reading.

Inclined Manometer (Figure 9-14): Increased sensitivity and scale magnification is realized by tilting one leg of the U-tube to form an inclined manometer or draft gauge. The inclined manometer gives increased accuracy and permits lower readings. In commercial versions, only one tube of the small bore is used and the other leg is replaced by a reservoir. The accuracy of the gauge is dependent on the slope of the tubes. Consequently, the base of the gauge must be leveled carefully and the mounting must be firm enough to permit accurate leveling. The better draft gauges are equipped with a built-in level, leveling adjustment and, in addition, a means of adjusting the scale to zero. Some models include over-pressure safety traps to prevent loss of fluid in the event of pressure surges beyond the manometer range.

A modification of the inclined manometer is the inclined-vertical gauge in which the indicator leg is bent or shaped to give both a vertical and inclined portion — the advantage is

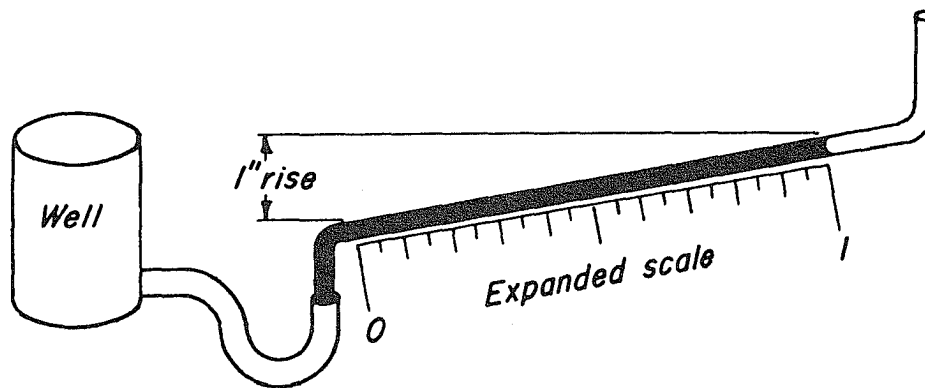


FIGURE 9-14. Inclined manometer

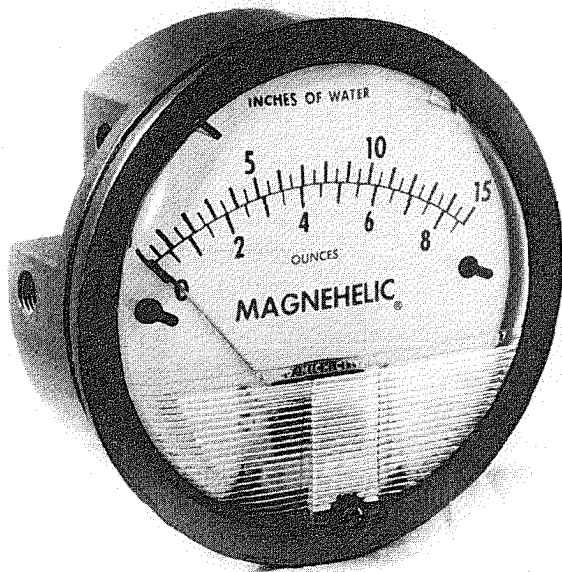


FIGURE 9-15. Aneroid gauge

smaller physical size for a given range while retaining the refined measurement afforded by the inclined manometer. As in the U-tube and inclined gauges, the commercial units available offer a wide choice in range, number of columns, and calibration units.

Aneroid Gauges: This type of gauge is used as a field instrument in ventilation studies for measuring static, velocity, or total pressure with a Pitot tube or for single tube static pressure measurements. A number of manufacturers offer gauges suitable for the measurement of the low pressures encountered in ventilation studies. Perhaps the best known of this type is the Magnehelic™ gauge (Figure 9-15). The principal advantages of this gauge can be listed as follows: easy to read, greater response than manometer types; very portable — small physical size and weight; absence of fluid means less maintenance; and mounting and use in any position is possible without loss of accuracy. Principal disadvantages are that the gauge is subject to mechanical failure, requires periodic calibration checks, and occasional recalibration.

Electronic Aneroid Gauges: Commercial instruments are now available which will measure and record static pressure as well as integrate velocity pressure directly to velocity using the pressure sensing principles of an aneroid gauge. This type of instrument can be connected directly to a standard Pitot tube and used in the same manner as a U-tube manometer. The instruments are light in weight, easily hand-held, and can be equipped with an electronic digital display or print recorder with measurement data in either English or S.I. units. Because they are battery powered, periodic servicing is required as is calibration.

9.6 CORRECTIONS FOR NON-STANDARD CONDITIONS

Air velocities sometimes are measured at conditions significantly different from standard. If these conditions are ignored, serious errors can be introduced in the determination of the actual duct velocity and the volumetric flow rate(s) in the system. Elevation, pressure, temperature, and moisture content all affect the density of the air stream. The actual density present in the system must be used in either Equation 9.2 or 9.9 to determine the actual velocity.

Correction for changes in elevation, duct pressure, and temperature can be made independently of each other with reasonable accuracy. The individual correction coefficients are multiplied together to determine the change from standard air density. The actual air density becomes:

$$\rho = 0.075 df \text{ and} \quad [9.12]$$

$$df = CF_e CF_p CF_t \quad [9.13]$$

where:

CF_e = correction for elevations outside the range of ± 1000 ft

CF_p = correction for local duct pressures greater than ± 20 "wg

CF_t = correction for temperatures outside the range of 40 to 100 F

One exception to this general rule is when elevations significantly different from sea level are coupled with high moisture content. Where this occurs, a psychrometric chart based upon the barometric pressure existing at the elevation of concern should be used. See Chapter 5 for an explanation of the determination of density using a psychrometric chart when moisture content and temperature are significantly different from standard.

The correction coefficient for elevation, CF_e , can be given by

$$CF_e = [1 - (6.73 \times 10^{-6})(z)]^{5.258} \quad [9.14]$$

where:

z = elevation, ft.

The correction coefficient for local duct pressure, CF_p , can be given by

$$CF_p = \frac{407 + SP}{407} \quad [9.15]$$

where:

SP = static pressure, "wg. (Note that the algebraic sign of SP is important.)

The correction coefficient for temperature, CF_t , can be given by

$$CF_t = \frac{530}{t + 460} \quad [9.16]$$

where:

t = dry-bulb temperature, F (Note: Algebraic sign of t must be used)

Density factors (df) for various altitudes, barometric pressures, and temperature conditions are shown on Table 9-8.

Example 1: A velocity pressure reading of 1.0 "wg was taken with a Pitot tube in a duct where the dry-bulb temperature is 300 F, the moisture content is negligible and the static pressure is -23.5 "wg. The system is installed at an elevation of 5000 feet. What would the density and actual velocity be at that point?

As the moisture content is unimportant, Equations 9.12 and 9.13 can be used directly to determine the density.

The individual correction coefficients can be found from Equations 9.14 through 9.16 as

$$CF_e = [1 - (6.73 \times 10^{-6})(5000)]^{5.258} = 0.84$$

$$CF_p = \frac{407 - 23.5}{407} = 0.94$$

$$CF_t = \frac{530}{300 + 460} = 0.70$$

Then the density at this condition would be

$$\rho = (0.075)(0.84)(0.94)(0.70) = 0.0415 \text{ lbm / ft}^3$$

and the velocity from Equation 9.9 would be

$$V = 1096 \sqrt{\frac{10}{0.0415}} = 5380 \text{ fpm}$$

Note that an error of 26% would result if standard density had been assumed.

Example 2: A swinging vane anemometer is used to determine the velocity in a duct at sea level where the dry bulb temperature is 250 F, the SP = -10 "wg and moisture is negligible. What is the actual duct velocity if the anemometer reading is 3150?

The temperature correction coefficient is 0.75 from Equation 9.16 and the density would be

$$\rho = (0.75)(0.075) = 0.0563 \text{ lbm / ft}^3$$

Therefore, the actual velocity in the duct would be

$$V_{\text{corrected}} = V_{\text{measured}} \sqrt{df}$$

$$V_{\text{corrected}} = 3150 \sqrt{\frac{0.075}{0.0563}} = (3150)(1.15) = 3636 \text{ fpm}$$

9.6.1 Example Traverse Calculations: Measurement of air velocity at non-standard conditions requires calculation of the true air velocity, accounting for difference in air density due to air temperature, humidity, and barometric pressure.

The following calculations illustrate the method of calculation and the effect of varying air density.

1. Standard Conditions:

Air Temp. = 79 F; Wet-Bulb Temp. = 50 F
Barometer = Std. (29.92 "Hg); 24" Duct Diameter

Pitot Traverse #1			Pitot Traverse #2 (⊥ to Traverse #1)		
Traverse Pt.	VP _M	V _s [*]	Traverse Pt.	VP _M	V _s [*]
1	0.22	1879	1	0.23	1921
2	0.28	2119	2	0.27	2081
3	0.32	2260	3	0.33	2301
4	0.33	2301	4	0.34	2335
5	0.34	2335	5	0.34	2335
6	0.35	2369	6	0.35	2369
7	0.33	2301	7	0.34	2335
8	0.32	2230	8	0.32	2260
9	0.30	2193	9	0.32	2230
10	0.24	1962	10	0.25	2003
21949			22170		

*Calculated from Equation 9.9 or Chapter 5, Table 5-7

$$\text{Average Velocity, } V_s = \frac{21949 + 22170}{20} = \frac{44119}{20}$$

$$= 2205.9 = 2206 \text{ fpm}$$

$$Q_s = VA = 2206 \times 3.142 = 6931.2 = 6931 \text{ scfm}$$

2. Elevated Temperature:

Air Temp. = 150 F; Wet-Bulb Temp. = 80 F
Barometer = Std.; 24" Outside Diameter Duct

Pitot Traverse #1			Pitot Traverse #2 (⊥ to Traverse #1)		
Traverse Pt.	VP _M	V _s [*]	Traverse Pt.	VP _M	V _s [*]
1	0.22	2015	1	0.23	2060
2	0.28	2275	2	0.27	2235
3	0.32	2430	3	0.33	2465
4	0.33	2470	4	0.34	2505
5	0.34	2505	5	0.34	2505
6	0.35	2540	6	0.35	2540
7	0.33	2470	7	0.34	2505
8	0.32	2395	8	0.32	2430
9	0.30	2355	9	0.32	2395
10	0.24	2105	10	0.25	2150
23560			23790		

*Calculated from Equation 9.9 or Chapter 5, Table 5-7

To determine the air velocity at standard conditions (V_s) for each VP_M , the density (ρ) can be calculated using Equations 9.12 and 9.16:

$$\rho = 0.075 \times (530 \div 610) = 0.065 \text{ lbm/ft}^3$$

Using Equation 9.11, each VP_M is multiplied by the ratio $0.075 \div 0.065$ and the resulting V_s values averaged.

$$\begin{aligned} \text{Average Velocity, } V_s &= \frac{23560 + 23790}{20} = \frac{47350}{20} \\ &= 2368 \text{ fpm} \end{aligned}$$

$$Q_s = VA = 2368 \times 3.142 = 7440.3 = 7440 \text{ scfm}$$

Short Method:

Find: "standard velocity" average from measured $VP_s = 2206$ fpm (From #1)

VP for 2206 fpm = 0.30 (Equation 9.10); at 150 F, density = $0.075 \div 0.87 = 0.065 \text{ lbm/ft}^3$

$$VP_s = VP_M \times (0.075 \div 0.065) = 0.346 = 0.35 \text{ "wg}$$

$$V_s = 2370 \text{ fpm}$$

3. Elevated Temperature and Moisture:

Air Temp = 150 F; Wet-Bulb Temp. = 140 F
Barometer = Std.; 24" Outside Diameter Duct

Pitot Traverse #1			Pitot Traverse #2 (\perp to Traverse #1)		
Traverse Pt.	VP_M	V_s^*	Traverse Pt.	VP_M	V_s^*
1	0.22	2100	1	0.23	2145
2	0.28	2370	2	0.27	2325
3	0.32	2530	3	0.33	2570
4	0.33	2570	4	0.34	2610
5	0.34	2610	5	0.34	2610
6	0.35	2645	6	0.35	2645
7	0.33	2570	7	0.34	2610
8	0.32	2490	8	0.32	2530
9	0.30	2450	9	0.32	2490
10	0.24	2190	10	0.25	2235
		24525			24770

*Calculated from Equation 9.9 or Chapter 5, Table 5-7

To determine the air velocity at standard conditions (V_s) for each VP_M , the air density (ρ) can be calculated using the psychrometric charts found in Chapter 5.

$$\rho = 0.075 \times 0.80 \text{ (density coefficient - mixture)}$$

$$= 0.06 \text{ lb / m / ft}^3$$

Using Equation 9.11, each VP_M is multiplied by the ratio $0.075 \div 0.06$ and the resulting V_s values averaged.

$$\begin{aligned} \text{Average velocity, } V_s &= \frac{24525 + 24770}{20} \\ &= \frac{49295}{20} = 2460 \\ &= 2465 \text{ fpm} \end{aligned}$$

(V_s may be found also by the Short Method found in #2.)

$$Q_{\text{actual}} = V_s A = 2465 \times 3.142$$

$$= 7745 \text{ cfm of air and water mixture}$$

$$\text{Weight of mixture} = Q_s \times 0.075 \times d$$

$$= 7745 \times 0.075 \times 0.80 = 465 \text{ lb}$$

From psychrometric charts, weight of water in mixture = 0.15 lb. H_2O /lb dry air.

Weight of dry air

$$= \frac{(\text{weight of mixture})}{(\text{weight of dry air} + \text{moisture per lbm dry air})}$$

$$= \frac{465}{1.15} = 404 \text{ lbm}$$

$$Q = \frac{404 \text{ lbm/min}}{\text{Std density}} = \frac{404}{0.075} = 5387 \text{ scfm}$$

Alternate Method:

From Chapter 5, humid volume = 19.3 ft^3 of mixture/lb dry air (Interpolate).

$$\text{Weight of dry air} = \frac{Q_s}{19.3} = \frac{7745}{19.3} = 403 \text{ lb}$$

$$Q_s = \frac{403 \text{ lbm}}{0.075} = 5373 \text{ scfm}$$

4. High or Low Altitudes:

$Q_s = V_s \times A$ where V_s can be obtained from Equations 9.2 and 9.9 in conjunction with Table 9.8 or Equations 9.12 and 9.13.

9.7 CHECK-OUT PROCEDURES

The following procedure may be used on systems (see Figure 9-16) that were designed to balance without the aid of blast gates. It is intended as an initial verification of the design computations and contractors' construction in new systems, but it may be used also for existing systems when design calculations are available or can be recomputed. It does not detect poor choices of design criteria such as low conveying or capture velocities, and consequently, will not reveal inade-

TABLE 9-8. Air Density Correction Factor, df

		ALTITUDE RELATIVE TO SEA LEVEL, ft															
		− 5000	− 4000	− 3000	− 2000	− 1000	0	1000	2000	3000	4000	5000	6000	7000	8000	9000	10000
		Barometric Pressure															
°Hg		35.74	34.51	33.31	32.15	31.02	29.92	28.86	27.82	26.82	25.84	24.89	23.98	23.09	22.22	21.39	20.57
°w		486.74	469.97	453.67	437.84	422.45	407.50	392.98	378.89	365.21	351.93	339.04	326.54	314.42	302.66	291.26	280.21
Temp., F	Density Factor, df																
− 40	1.51	1.46	1.40	1.36	1.31	1.26	1.22	1.17	1.13	1.09	1.05	1.01	0.97	0.94	0.90	0.87	
0	1.38	1.33	1.28	1.24	1.19	1.15	1.11	1.07	1.03	1.00	0.96	0.92	0.89	0.86	0.82	0.79	
40	1.27	1.22	1.18	1.14	1.10	1.06	1.02	0.99	0.95	0.92	0.88	0.85	0.82	0.79	0.76	0.73	
70	1.19	1.15	1.11	1.07	1.04	1.00	0.96	0.93	0.90	0.86	0.83	0.80	0.77	0.74	0.71	0.69	
100	1.13	1.09	1.05	1.02	0.98	0.95	0.91	0.88	0.85	0.82	0.79	0.76	0.73	0.70	0.68	0.65	
150	1.04	1.00	0.97	0.93	0.90	0.87	0.84	0.81	0.78	0.75	0.72	0.70	0.67	0.65	0.62	0.60	
200	0.96	0.93	0.89	0.86	0.83	0.80	0.77	0.75	0.72	0.69	0.67	0.64	0.62	0.60	0.57	0.55	
250	0.89	0.86	0.83	0.80	0.77	0.75	0.72	0.69	0.67	0.64	0.62	0.60	0.58	0.55	0.53	0.51	
300	0.83	0.80	0.78	0.75	0.72	0.70	0.67	0.65	0.62	0.60	0.58	0.56	0.54	0.52	0.50	0.48	
350	0.78	0.75	0.73	0.70	0.68	0.65	0.63	0.61	0.59	0.57	0.54	0.52	0.50	0.49	0.47	0.45	
400	0.74	0.71	0.69	0.66	0.64	0.62	0.59	0.57	0.55	0.53	0.51	0.49	0.48	0.46	0.44	0.42	
450	0.70	0.67	0.65	0.63	0.60	0.58	0.56	0.54	0.52	0.50	0.48	0.47	0.45	0.43	0.42	0.40	
500	0.66	0.64	0.61	0.59	0.57	0.55	0.53	0.51	0.49	0.48	0.46	0.44	0.43	0.41	0.39	0.38	
550	0.63	0.61	0.58	0.56	0.54	0.52	0.51	0.49	0.47	0.45	0.44	0.42	0.40	0.39	0.38	0.36	
600	0.60	0.58	0.56	0.54	0.52	0.50	0.48	0.46	0.45	0.43	0.42	0.40	0.39	0.37	0.36	0.34	
700	0.55	0.53	0.51	0.49	0.47	0.46	0.44	0.42	0.41	0.39	0.38	0.37	0.35	0.34	0.33	0.31	
800	0.50	0.49	0.47	0.45	0.44	0.42	0.41	0.39	0.38	0.36	0.35	0.34	0.32	0.31	0.30	0.29	
900	0.47	0.45	0.43	0.42	0.40	0.39	0.38	0.36	0.35	0.34	0.32	0.31	0.30	0.29	0.28	0.27	
1000	0.43	0.42	0.40	0.39	0.38	0.36	0.35	0.34	0.33	0.31	0.30	0.29	0.28	0.27	0.26	0.25	

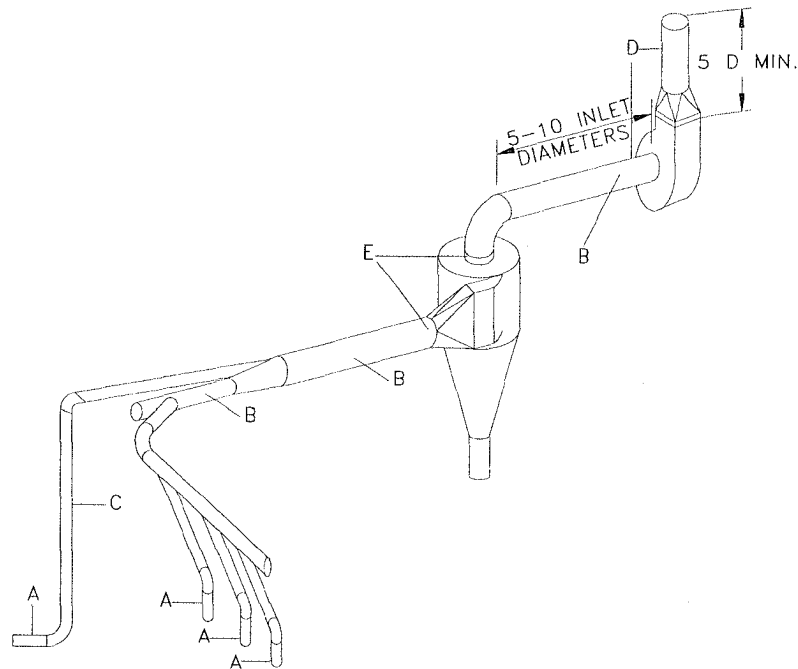
quate control due to this type of error. Agreement with design within $\pm 10\%$ is considered acceptable.

1. Determine volumetric flow in duct with a pitot traverse. If volumetric flow matches design, go to Step 4; otherwise, continue with 1a.
 - a. Check fan size against plan;
 - b. Check fan speed and direction of rotation against design;
 - c. Check fan inlet and outlet configuration against plan.
2. If a discrepancy is found and corrected, return to Step 1. If not, measure fan inlet and outlet static pressures and compute the fan static pressure. Using fan table, check flow, fan static pressure and fan speed (RPM). If agreement is acceptable although at some other operating point than specified, fan is satisfactory, and trouble is elsewhere in the system. Go to Step 3.
3. If fan *inlet* static pressure is greater (more negative) than calculated in the design, proceed to Step 4. If fan *outlet* static pressure is greater (more positive) than design, proceed to Step 8.
4. Measure hood static pressure on each hood and check against design. If correct, go to Step 10; otherwise, continue with Step 4a.
 - a. Check size and design of hoods and slots against plan;

b. Examine each hood for obstructions.

5. After all hood construction errors and obstructions have been corrected, if hood static pressures are correct, return to Step 1; if too low, proceed to Step 6.
6. Measure static pressure at various junctions in ducts and compare with design calculations. If too high at a junction, proceed upstream until static pressures are too low and isolate the trouble. In an area where the loss exceeds design:
 - a. Check angle of entries to junctions against plan;
 - b. Check radii of elbows against plan;
 - c. Check duct diameters against plan;
 - d. Check duct for obstructions.

If the static pressure is too low, proceed downstream and locate the trouble.
7. After correcting all construction details which deviate from specifications, return to Step 1.
8. Measure pressure differential across air cleaning device and check against manufacturer's data. If loss is excessive, make necessary corrections and return to Step 1. If loss is less than anticipated, proceed to Step 8a.
 - a. Check ducts, elbows, and entries as in Step 6a and 6d.



POINT	MEASUREMENT	LOCATION OF MEASUREMENT	MEASUREMENT USE
A	HOOD STATIC PRESSURE	DISTANCE FROM HOOD - 3 PIPE ϕ 'S-FLANGED OR PLAIN HOOD 1 ϕ -TAPERED HOOD	1. ESTIMATE FLOW: $Q=4005C_eA\sqrt{SP_h}$ 2. CHECK POINT FOR HOOD AND SYSTEM PERFORMANCE.
B	VELOCITY AND STATIC PRESSURE	BRANCH AND MAINS-PREFERABLY 7.5 ϕ 'S STRAIGHT RUN DOWNSTREAM FROM NEAREST AIR DISTURBANCE (EL, ENTRY, ETC..)	1. TRANSPORT VELOCITY 2. EXHAUST VOLUME: $Q=VA$ 3. SP AS SYSTEM CHECK POINT
C	CENTERLINE VP	SMALL DUCTS LOCATION AS ABOVE. CENTERLINE VELOCITY READING ONLY.	ROUND DUCT ONLY. USE ON SMALL DUCTS WHERE TRANSVERSE IMPRACTICAL OR WHERE APPROXIMATE VOLUME WANTED.
D	STATIC, VELOCITY AND TOTAL PRESSURES	INLET AND OUTLET OF FAN-ANY TWO OF THREE READINGS AT EACH LOCATION	1. FAN STATIC AND TOTAL PRESSURES $FSP = SP_0 - SP_i - VP_i$ $TP = SP_0 - SP_i + VP_0 - VP_i$ 2. MOTOR SIZE OR GFM ESTIMATE $CFM \times TP$ $BHP = \frac{6356 \times ME \text{ OF FAN}}{1000}$ 3. SP AS SYSTEM CHECK POINT
E	STATIC PRESSURE	INLET AND OUTLET OF COLLECTOR DIFFERENTIAL PRESSURE	1. COMPARE PRESSURE DROP WITH NORMAL OPERATING RANGE 2. CHECKPOINTS FOR MAINTENANCE. READINGS ABOVE OR BELOW NORMAL INDICATE PLUGGING, WEAR OR DAMAGE TO COLLECTOR ELEMENTS, NEED OF CLEANING

IN ADDITION TO THE ABOVE, FACE VELOCITY (HOOD FACE) AND CAPTURE VELOCITY (POINT OF CONTAMINANT DISPERSION) MEASUREMENTS ARE USUALLY MADE TO DEFINE HOOD PERFORMANCE. OBSERVATION OF AIR FLOWS SURROUNDING EXHAUST OPENINGS MAY BE VISUALLY AUGMENTED BY USE OF SMOKE GENERATORS, TRAILS, AND STREAMERS.

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SAMPLE SYSTEM

DATE

1-88

FIGURE

9-16

PLANT _____ DEPT. _____ DATE _____
 OPERATION EXHAUSTED _____ BY _____

LINE SKETCH SHOWING POINTS OF MEASUREMENT

DATE SYSTEM INSTALLED _____

HOOD AND TRANSPORT VELOCITY

POINT	DUCT		VP	SP	FPM	CFM	REMARKS
	D	AREA (Tbl. 5-5)	IN. H ₂ O	IN. H ₂ O	(Tbl. 9-1)	Q = VA	

PITOT TRAVERSE
 PITOT READINGS— SEE TABLES 9-1 TO 9-4

FAN
 TYPE _____
 SIZE _____

POINTS	VP	VEL.	VP	VEL.	VP	VEL.
1						
2						
3						
4						
5						
6						
7						
8						
9						
10						
TOTAL VEL.						
AVERAGE VEL.						
CFM						

POINT	DIA.	SP	VP	TP	CFM
INLET					
OUTLET					

FAN SP _____ (SEE SECTION 6)
 MOTOR
 NAME _____ SIZE _____
 HP _____ E _____ L _____ W _____
 COLLECTOR
 TYPE & SIZE _____

POINT	DIA.	SP	Δ SP
INLET			
OUTLET			

NOTES _____

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SURVEY FORM

DATE 1-88

FIGURE 9-17

- b. Check system discharge type and dimensions against plans.
9. If errors are found, correct, and return to Step 1. If no errors can be detected, recheck design against plan, recalculate, and return to Step 1 with new expected design parameters.
10. Measure control velocities at all hoods where possible. If control is inadequate, redesign, or modify hood.
11. The above process should be repeated until all defects are corrected and hood static pressures and control velocities are in reasonable agreement with design. The actual hood static pressures should then be recorded for use in periodic system checks. A file should be prepared containing the following documents:
 - System plan
 - Design calculations
 - Fan rating table
 - Hood static pressures after field measurement
 - Maintenance schedule
 - Periodic hood static pressure measurement log
 - Periodic maintenance log

9.7.1 Difficulties Encountered in Field Measurement:

The general procedures and instrumentation for the measurement of air flow have been previously discussed. However, special problems connected with air flow necessitate a somewhat more detailed discussion.

Some of these special problems are as follows:

1. Measurement of air flow in highly contaminated air which may contain corrosive gases, dusts, fumes, mists, or products of combustion.
2. Measurement of air flow at high temperatures.
3. Measurement of air flow in high concentrations of water vapor and mist.
4. Measurement of air flow where the velocity is very low (see Reference 9.11).
5. Measurement of air flow in locations of turbulence and non-uniform air flow; e.g., discharge of cupolas, locations near bends, enlargements or discharges from exhaust fans.
6. Measurement of air flow in connection with isokinetic sampling when the velocity is constantly changing.

Selection of Instruments: The selection of the proper instrument will depend on the range of air flow to which the instrument is sensitive; its vulnerability to high temperatures, corrosive gases, and contaminated atmospheres; its portability and ruggedness and the size of the measuring probe relative

to the available sampling port. A brief summary of the characteristics of a few of the instruments which can be used is given in Table 9.1.

In many cases, conditions for air flow measurement are so severe that it is difficult to select an instrument. Generally speaking, the Pitot tube is the most serviceable instrument; it has no moving parts, is rugged, and will stand high temperatures and corrosive atmospheres when it is made of stainless steel. It is subject to plugging, however, when it is used in a dusty atmosphere. It cannot be used for measurement of low velocities. A special design of Pitot tube can be used for dusty atmospheres. In many cases, it is difficult to set up an inclined manometer in the field because many readings are made from ladders, scaffolds and difficult places. This greatly limits the lower range of the Pitot tube. A mechanical gauge can be used in place of a manometer. A mechanical gauge is estimated to be accurate to 0.02 "wg with proper calibration.

For lower velocities, the swinging-vane anemometer previously described can be used if conditions are not too severe. The instrument can be purchased with a special dust filter which allows its use in light dust loadings. It can be used in temperatures up to 100 F if the jet is exposed to the high temperature gases only for a very short period of time (30 seconds or less). It cannot be used in corrosive gases. If the very low velocity jet is used, a hole over 1" in diameter must be cut into the duct or stack.

For very low velocities, anemometers utilizing the heated thermocouple principle can be used under special conditions. In most cases, these anemometers cannot be used in temperatures above 300 F. Contact the manufacturer to determine to what degree the thermocouple probe will withstand corrosive gases.

In sampling work where a match of velocities in the sampling nozzle and air stream under changing velocities is required, the null method is sometimes used. This method uses two static tubes or inverted impact tubes, one located within the sampling nozzle and the other in the air stream. Each is connected to a leg of the manometer; the sampling rate is adjusted until the manometer reading is zero.

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Chapter 10

SPECIFIC OPERATIONS

The following illustrations of hoods for specific operations are intended as guides for design purposes and apply to usual or typical operations. In most cases, they are taken from designs used in actual installations of successful local exhaust ventilation systems. All conditions of operation cannot be categorized, and because of special conditions (i.e., cross-drafts, motion, differences in temperature, or use of other means of contaminant suppression), modifications may be in

order.

Unless it is specifically stated, the design data are not to be applied indiscriminately to materials of high toxicity, e.g., beryllium and radioactive materials. Thus the designer may require higher or lower air flow rates or other modifications because of the peculiarities of the process in order to adequately control the air contaminant.

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10.10 CLEANROOMS

U.S. Federal Standard 209E^(10.10.1) establishes standard classes of air cleanliness for airborne particulate levels in cleanrooms and clean zones. This standard is issued by the General Services Administration of the United States. While nominally a publication for use by federal agencies, FED-STD-209E has been adopted by American industry. It prescribes methods for class verification and monitoring of air cleanliness. It also addresses certain other factors that affect control of airborne contaminants.

FED-STD-209E does not address the physical, chemical, radiological, or viable nature of airborne contaminants. It also does not address the occupational health concerns of employees working in cleanroom environments.

A cleanroom class is the statistically allowable number of particles, greater than or equal to 0.5 micrometers in size, per cubic foot of air. Cleanroom classes are shown in Table 10.10.1.

In order to meet the class limits, a high efficiency particulate air (HEPA) or ultra low penetration air (ULPA) filter is required. A HEPA filter is a disposable, extended-media, dry-type filter in a rigid frame with a minimum particle collecting efficiency of 99.97% for 0.3 micrometer, thermally generated dioctylphthalate (DOP), or specified alternate, aerosol particles at a maximum clean resistance of 1.0 "wg when tested at rated air flow capacity. An ULPA filter is a disposable, extended-media, dry-type filter in a rigid frame with a minimum particle collecting efficiency of 99.999% for particulate diameters between 0.1 and 0.2 micrometers in size.

Military specifications^(10.10.2) and publications^(10.10.3) by the Institute of Environmental Sciences (IES) define HEPA and ULPA filter construction. Filters having an efficiency even higher than an ULPA filter are available from some companies specializing in cleanrooms and air filtration.

TABLE 10.10.1. Class limits in particles per cubic foot of size equal to or greater than particle sizes shown.*

Room Class	Measured Particle Size (Micrometers)				
	0.1	0.2	0.3	0.5	5.0
1	35	7.5	3	1	NA
10	350	75	30	10	NA
100	NA**	750	300	100	NA
1000	NA	NA	NA	1000	7
10,000	NA	NA	NA	10,000	70
100,000	NA	NA	NA	100,000	700

*The class limit particle concentrations shown are defined for class purposes only and do not necessarily represent the size distribution to be found in any particular situation.

**Not applicable.

The primary design considerations for cleanrooms are the supply air flow rate, the air flow patterns within the cleanroom, the method for recirculating the air from the cleanroom and the filter efficiency.

Air is supplied to the cleanroom by an air handling system containing the components needed for heating, cooling, and humidity control. Noise is readily transmitted to the cleanroom so very slow fan speeds, vibration isolation and noise control devices are important design considerations. The air circulation system will also contain two or three stages of prefiltration. This allows the final filters in the cleanroom ceiling to remain in place for very long periods of time. A final filter life of ten years or more is typical for Class 100 and better cleanrooms.

Air from the supply system enters the cleanroom through either a ducted module or a pressurized plenum. VS-10-01 shows the ducted module arrangement. Ducted modules containing HEPA or ULPA filters are connected to the main air supply duct by flexible branch ducts. The modules usually contain an internal baffle for balancing the air exhaust which must be at a uniform velocity across the face of the filter. The ducted modules are mounted in a T-bar grid and sealed with either solid gaskets or a liquid gel sealant. The ducted modules, because of long filter life, usually are considered to be throwaway items; however, some arrangements do permit the replacement of filters from within the cleanroom. A ducted module system offers maximum flexibility for cleanroom modification.

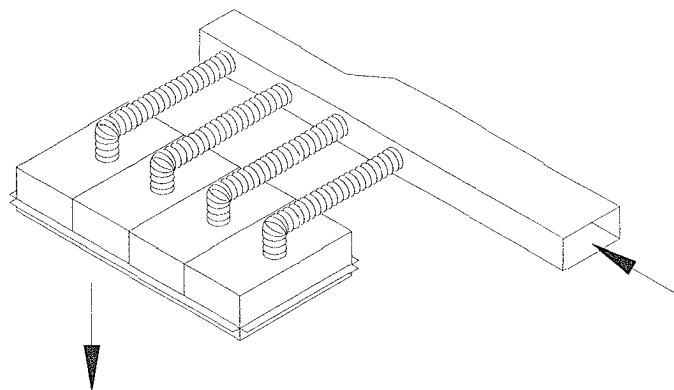
VS-10-02 shows a pressurized plenum arrangement. A heavy-duty grid system is suspended from the ceiling with suspension rods and the HEPA or ULPA filters sealed in the grid with liquid gel or solid gaskets. The entire plenum is pressurized by the air supply system to allow a uniform flow of air through the filters to the cleanroom below. A pressurized plenum system will usually cost less than a ducted module system for large cleanrooms.

VS-10-03 shows raised floor and low sidewall arrangements. Air is returned through a utility chase to the cleanroom supply air system. To provide better particulate control, the raised floor arrangement is preferred. The low sidewall return should not be used for vertical downflow cleanrooms more than 14 feet wide in order not to disrupt laminar flow at the work area.

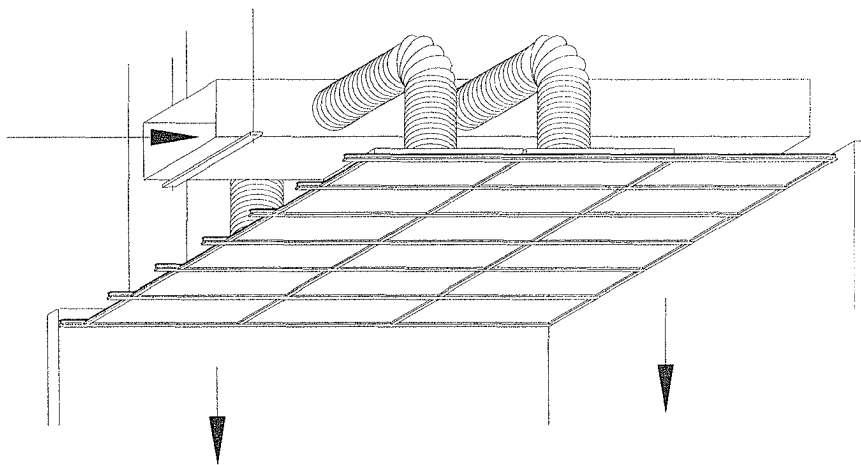
IES-RP-CC-006-84^(10.10.4) contains testing methods for characterizing the performance of cleanrooms. It defines terms having special meaning and describes test procedures to assure proper cleanroom operation. Uniform air flow is defined as unidirectional with all velocity readings within 20% of the average velocity of the work area. The air velocity at the work area is generally about 100 fpm; however, design conditions may require velocities of 10 fpm or lower.

REFERENCES

- 10.10.1 U.S. General Services Administration: FED-STD-209E, Federal Standard, Cleanroom and Work Station Requirements, Controlled Environment. Federal Supply Service, General Services Administration, Washington, DC (June 15, 1988).
- 10.10.2 U.S. Army: MIL-F-51068(EA), Specification Filters, Particulate, High-Efficiency, Fire Resistant, Biological Use, General Specifications For. Commander, U.S. Army Armament Research and Development Command, Aberdeen Proving Ground, MD (October 4, 1982).
- 10.10.3 Institute of Environmental Sciences: IES-RP-CC-001.3, HEPA and ULPA Filters. IES, Mount Prospect, IL.
- 10.10.4 Institute of Environmental Sciences: IES-RP-CC-006, Testing Cleanrooms. IES, Mount Prospect, IL.



Air is supplied to the ducted modules from the air distribution duct through flexible branch ducts, which are secured at the bottom ends by clamps. A damper (not shown) on the inside of the collar allows balancing of the air flowing from the module to the cleanroom.



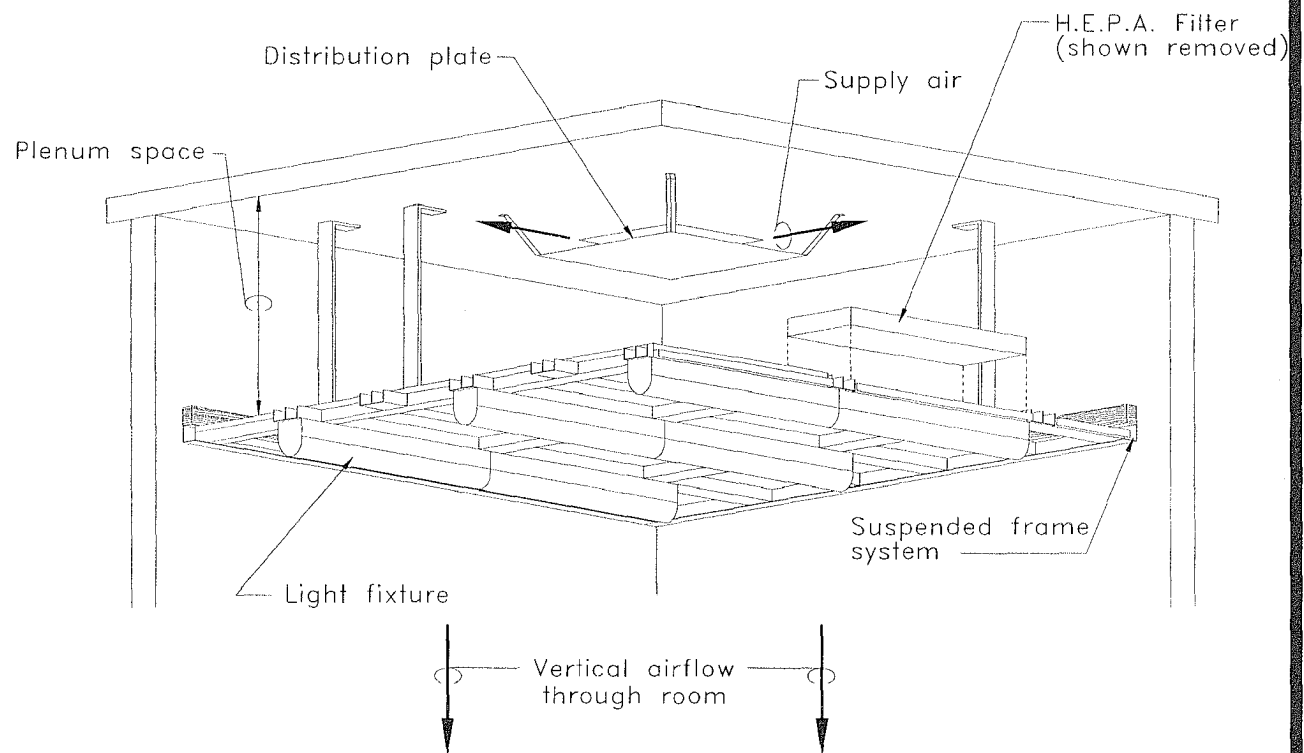
Ducted modules are mounted in 2' x 4' grids and sealed with gaskets or a liquid gel sealant. Tear drop or recessed lighting (shown) provides illumination.

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*CLEANROOM
DUCTED MODULE*

DATE 12-90

FIGURE VS-10-01



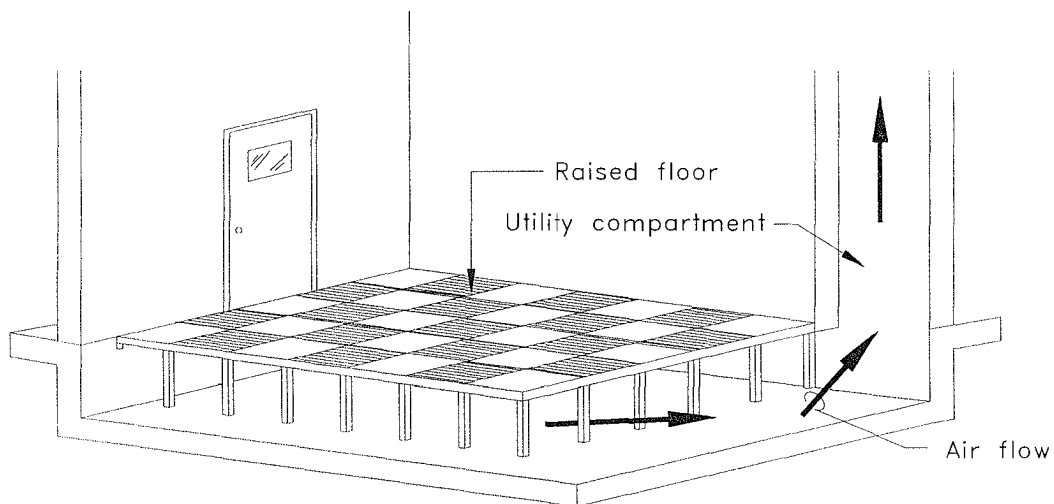
Supply air enters a pressurized plenum and strikes a distribution plate. Each 2' x 4' opening in the support structure contains a HEPA filter. Filters are sealed around the perimeter with gaskets or a fluidic sealant. The framing structure is supported from the plenum ceiling by suspension rods. Tear drop lighting is shown.

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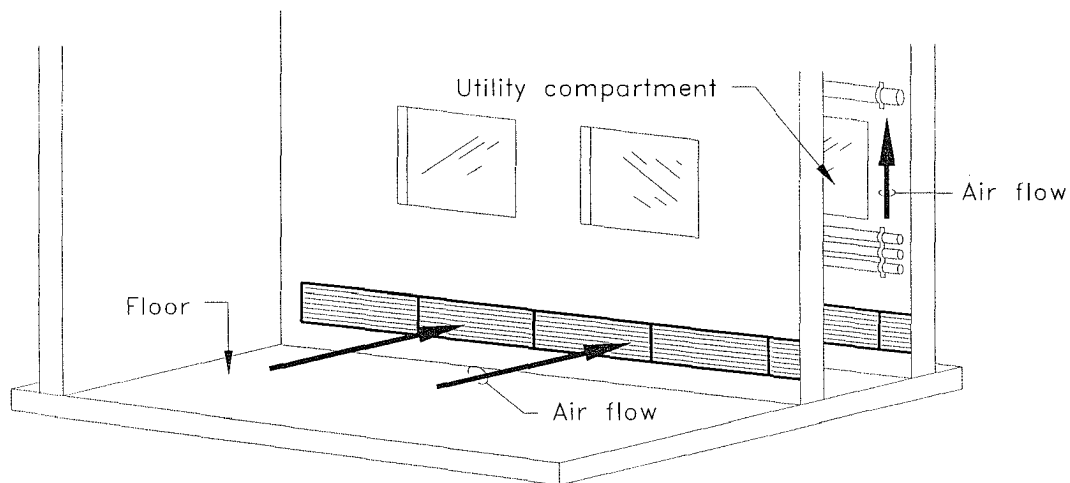
*CLEANROOM
PRESSURIZED PLENUM*

DATE 12-90

FIGURE VS-10-02



Raised floor with depressed slab. Air is returned through a utility compartment to the air supply system.



Low sidewall grille return through a utility compartment to the air supply system. Room width is limited to 14 feet if laminar air flow is to be achieved. The distance from the top of the grille to the floor should not exceed 18 inches.

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*CLEANROOM RETURN
AIR ARRANGEMENTS*

DATE

12-90

FIGURE

VS-10-03

10.15 FILLING OPERATIONS

Filling operations have special considerations that should be addressed when designing hoods. An enclosed space is not empty but rather is filled with air. When material enters the space, it forces the air out which in turn can carry some of the material with it. Also, additional air can be entrained by the material stream entering the enclosed space. This effect is a function of the size of the particles and the distance the material must fall. These two effects must be considered when designing hoods for material handling situations.

If there are any openings in the walls of the container which is being filled, some "splashing" of the material can occur. This can lead to loss of material through cracks and openings in the receiving vessel. The design of the ventilation system should take this effect into account.

The proper choice of exhaust flow rate is critical. If too little air is exhausted, the air displaced by the falling material may exceed the exhaust rate and the contaminant may not be adequately controlled. If too much air is exhausted, excess material could be entrained into the exhaust air stream. As this material often is the product, excess product loss could occur.

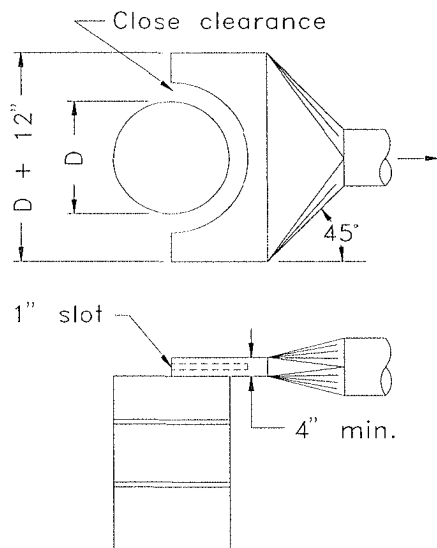
VS-15-01 illustrates four different ways of controlling barrel or drum filling operations. VS-15-02 illustrates bag filling and weighing. VS-15-03 depicts a bag tube packer. VS-15-10 and VS-15-11 depict a weighing hood where dry materials are removed from a bulk pack and weighed into smaller bags.^(10,15,3) These smaller bags are then packed into a

container. Bags containing toxic materials can be opened within an enclosing hood such as shown on VS-15-20.^(10,15,4)

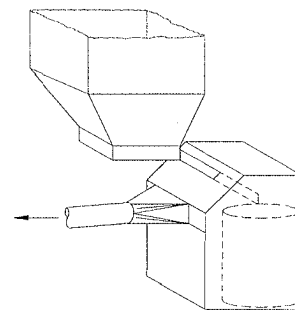
VS-15-30 shows how to extract a toxic liquid from a process line or vessel for analysis and VS-15-21 shows a possibility of controlling leaks around rotating shafts that enter containers.^(10,15,5)

REFERENCES

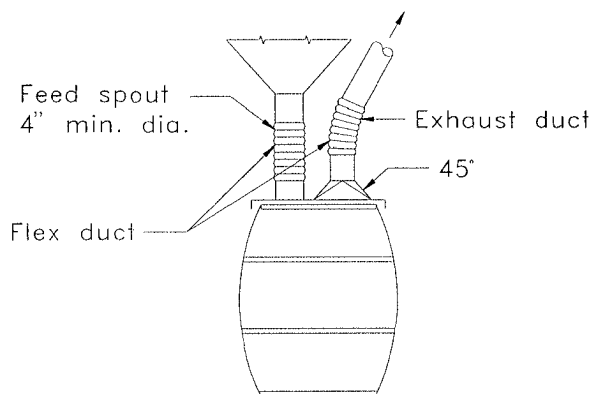
- 10.15.1 Hama, G.M.: Ventilation Control of Dust from Bagging Operations. Heating and Ventilating, p. 91 (April, 1948).
- 10.15.2 Cooper, T.C.: Control Technology for a Dry Chemical Bagging and Filling Operations. Monsanto Agricultural Products Co., Cincinnati, OH (1983).
- 10.15.3 Gressel, M.G.; Fischback, T.J.: Workstation Design Improvements for the Reduction of Dust Exposures During Weighing of Chemical Powders. Applied Industrial Hygiene 4:227-233 (1989).
- 10.15.4 Goldfield, J.; Brandt, F.E.: Dust Control Techniques in the Asbestos Industry. A paper presented at the American Industrial Hygiene Conference, Miami Beach, FL (May 12-17, 1974).
- 10.15.5 Langner, R.R.: How to Control Carcinogens in Chemical Production. Occupational Health and Safety (March-April, 1977).



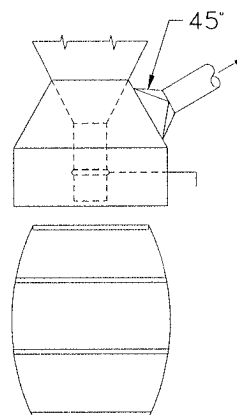
$Q = 100 \text{ cfm/ft}^2$ barrel top (minimum)
 Minimum duct velocity = 3500 fpm
 $h_e = 1.78 VP_s + 0.25 VP_d$



$Q = 150 \text{ cfm/ft}^2$ of open face area
 Minimum duct velocity = 3500 fpm
 $h_e = 0.25 VP_d$ (45° taper)



$Q = 50 \text{ cfm} \times \text{drum diam. (ft)}$
 Minimum duct velocity = 3500 fpm
 $h_e = 0.25 VP_d$



$Q = 300-400 \text{ cfm}$
 Minimum duct velocity = 3500 fpm
 $h_e = 0.25 VP_d$

Note 1: Air displaced by material feed rate may require higher exhaust flow rates.

Note 2: Excessive air flow can cause loss of product.

Reference: 10.15.1

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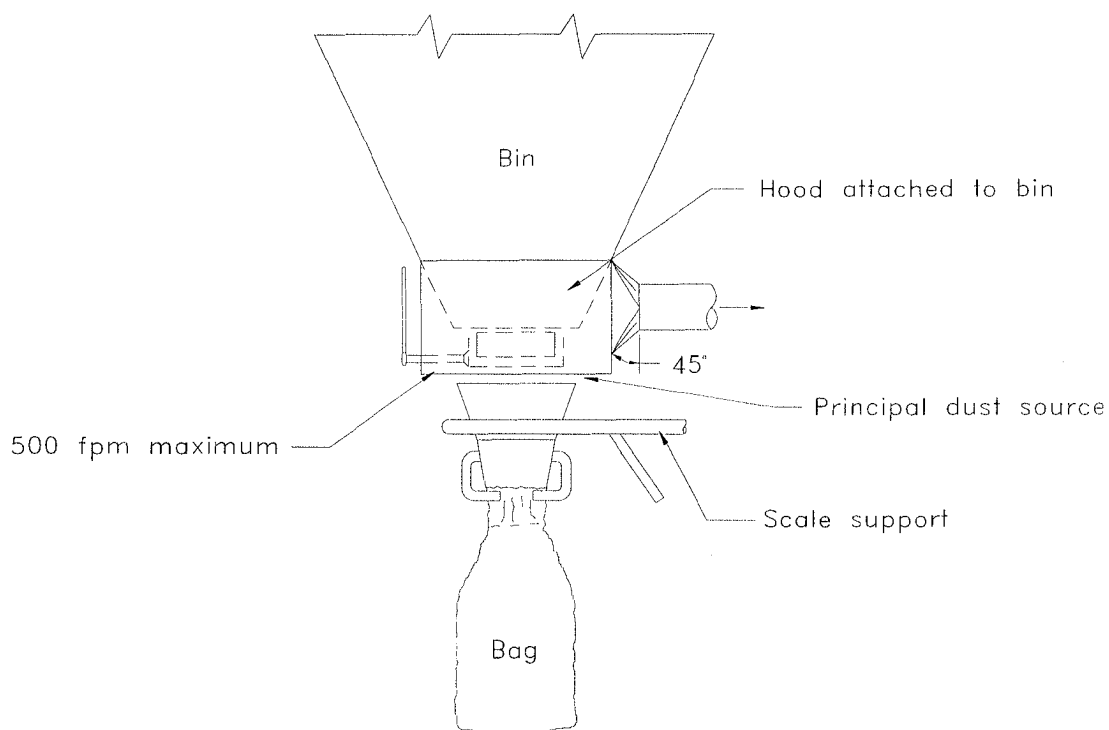
BARREL FILLING

DATE

1-91

FIGURE

VS-15-01



$Q = 400-500 \text{ cfm} - \text{non-toxic dust}$
 $1000-1500 \text{ cfm} - \text{toxic dust}$

Minimum duct velocity = 3500 fpm
 $h_e = 0.25 \text{ VP}_d$

Note: Care must be taken such that too much air is not used, as valuable product will be pulled into the exhaust system.

Reference: 10.15.2

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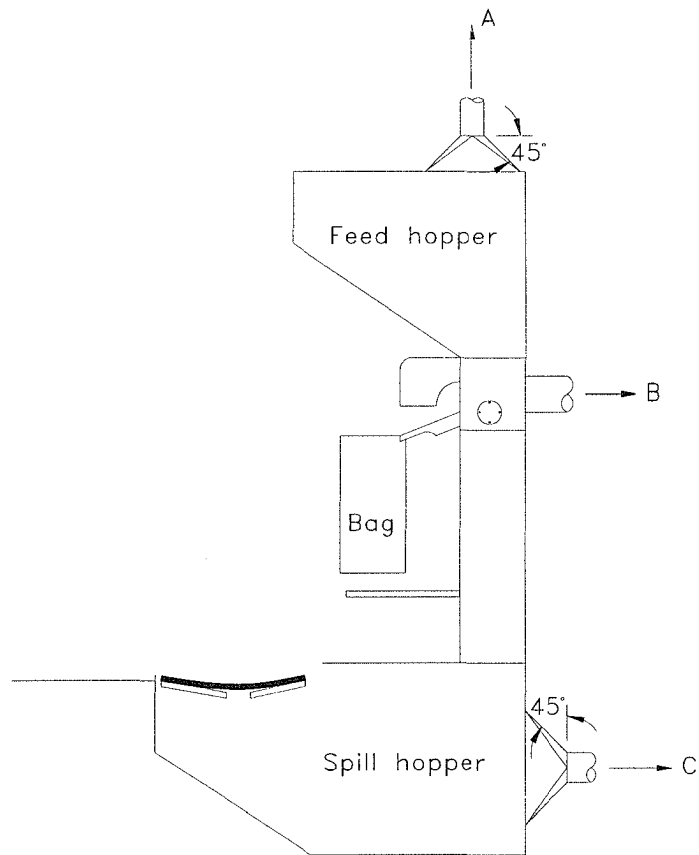
BAG FILLING

DATE

1-91

FIGURE

VS-15-02



$Q = 500 \text{ cfm per filling tube}$
 $= 500 \text{ cfm at Feed hopper}$
 $= 950 \text{ cfm at Spill hopper}$

Minimum duct velocity = 3500 fpm

$h_e = 0.25 VP_d$ for take-off at A and C
 $1.0 VP_d$ for take-off at B

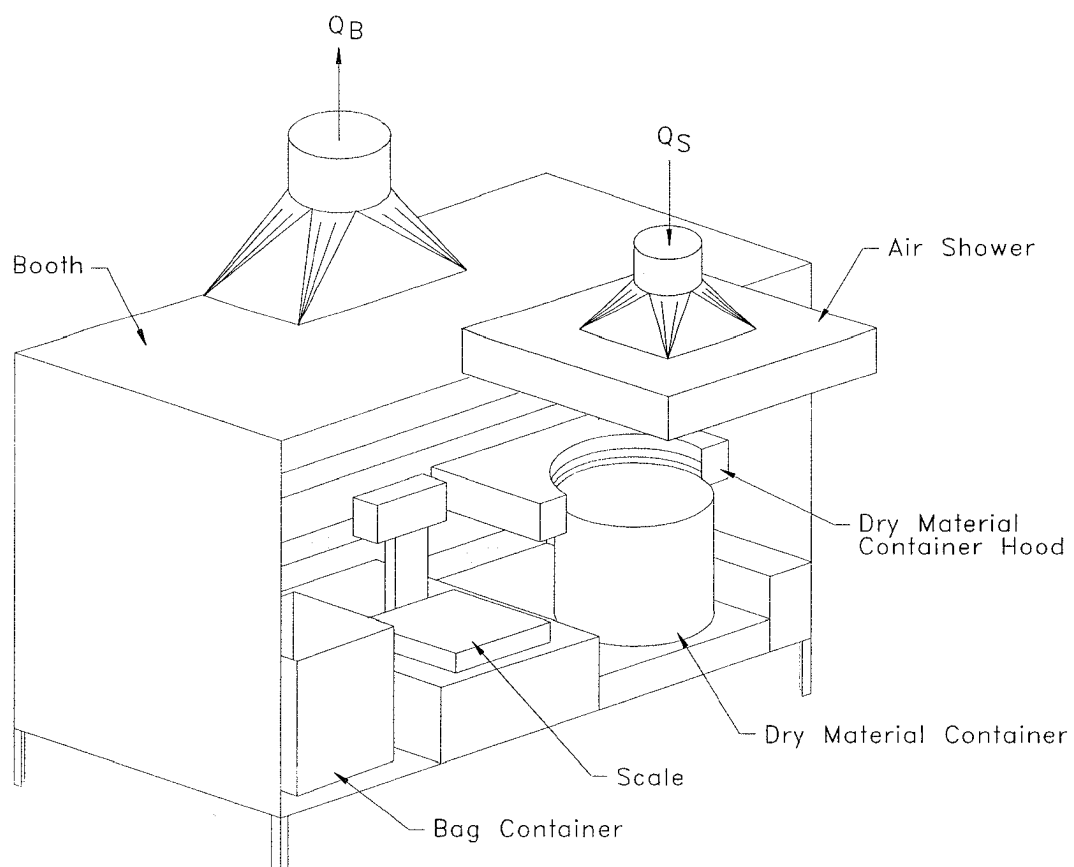
Reference: 10.15.2

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BAG TUBE PACKER

DATE 1-91

FIGURE VS-15-03



NOTE: See VS-15-11 for design details

Reference 10.15.3

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*WEIGH HOOD ASSEMBLY
DRY MATERIAL*

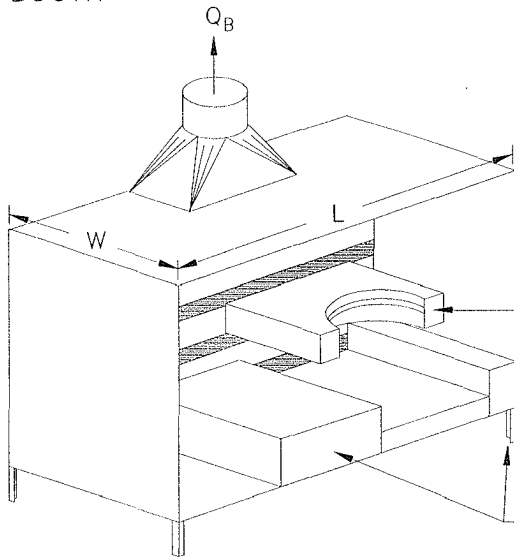
DATE

1-91

FIGURE

VS-15-10

BOOTH



$$Q_B = 50 \text{ cfm/ft}^2 \text{ of face open area.}$$

L and W to fit operation

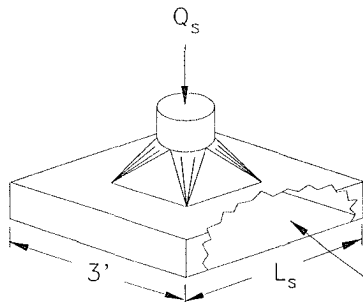
Minimum duct velocity = 3500 fpm

$$h_e = 1.78 VP_s + 0.25 VP_d$$

Dry material container hood is extension of booth slot

Configure to fit equipment.

AIR SHOWER



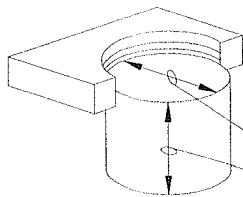
$$Q_s = 100 L_s \text{ cfm}$$

$L_s = 3$ feet. (Can be longer if required to fit workstation but do not exceed $1/2$ booth length; L)

0.25" pegboard or equivalent, 20 percent maximum open area.

DRY MATERIAL CONTAINER HOOD

Hood is extension of booth slot. An additional takeoff(s) may be used if required for hood air flow distribution.



Airflow and hood slot design per VS-15-01

12" to 24" Diameter

24" Maximum

Reference 10.15.3

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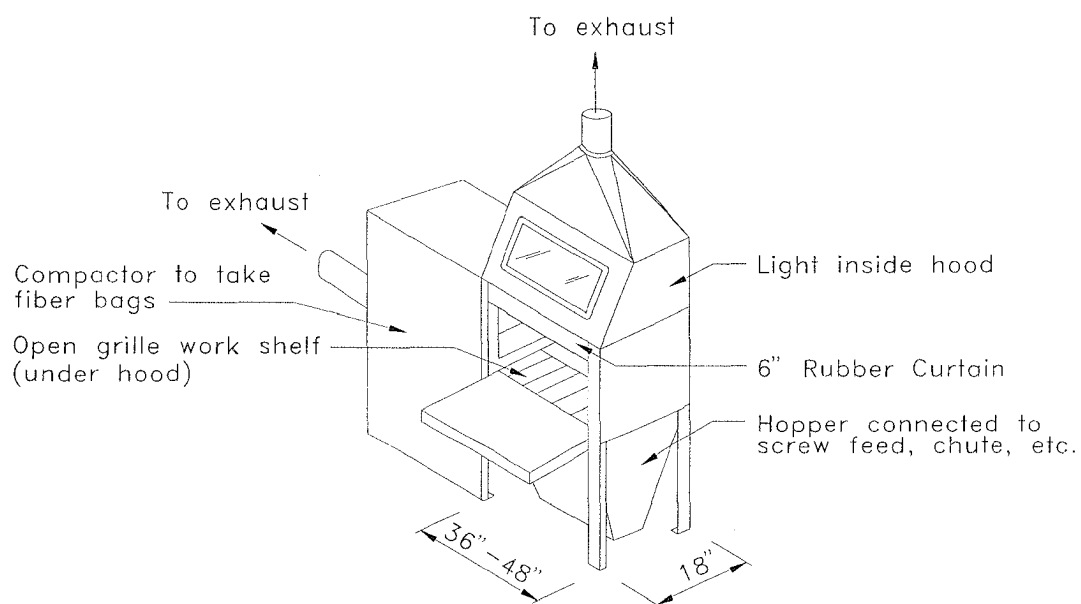
WEIGH HOOD DETAILS
DRY MATERIAL

DATE

2-91

FIGURE

VS-15-11



Q = minimum 250 cfm/ft² of open area
 Minimum duct velocity = 3500 fpm
 $h_e = 0.25 VP_d$

Reference: 10.15.4

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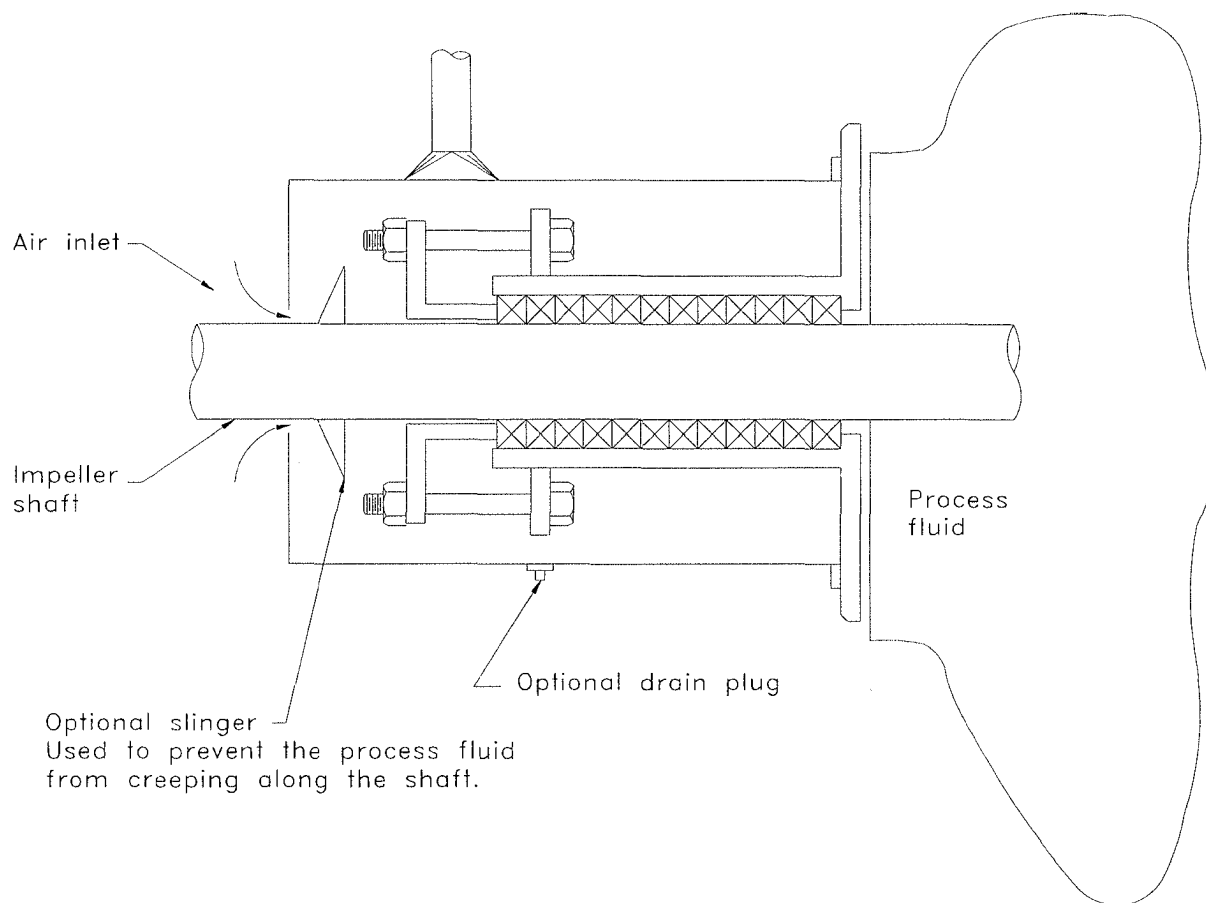
*TOXIC MATERIAL
BAG OPENING*

DATE

1-91

FIGURE

VS-15-20



$$Q = 500 \text{ cfm/ft}^2 \text{ of open area} \\ \text{(typically 10-40 cfm)}$$

Note: Sufficient air must be provided to dilute flammable gases and/or vapors to below 25% of LEL. See Chapter 2.

$$\text{Duct velocity} = 2000 \text{ fpm}$$

$$h_e = 1.78VP_s + 0.25VP_d$$

Note: Similar hood is appropriate for unions

Reference: 10.15.5

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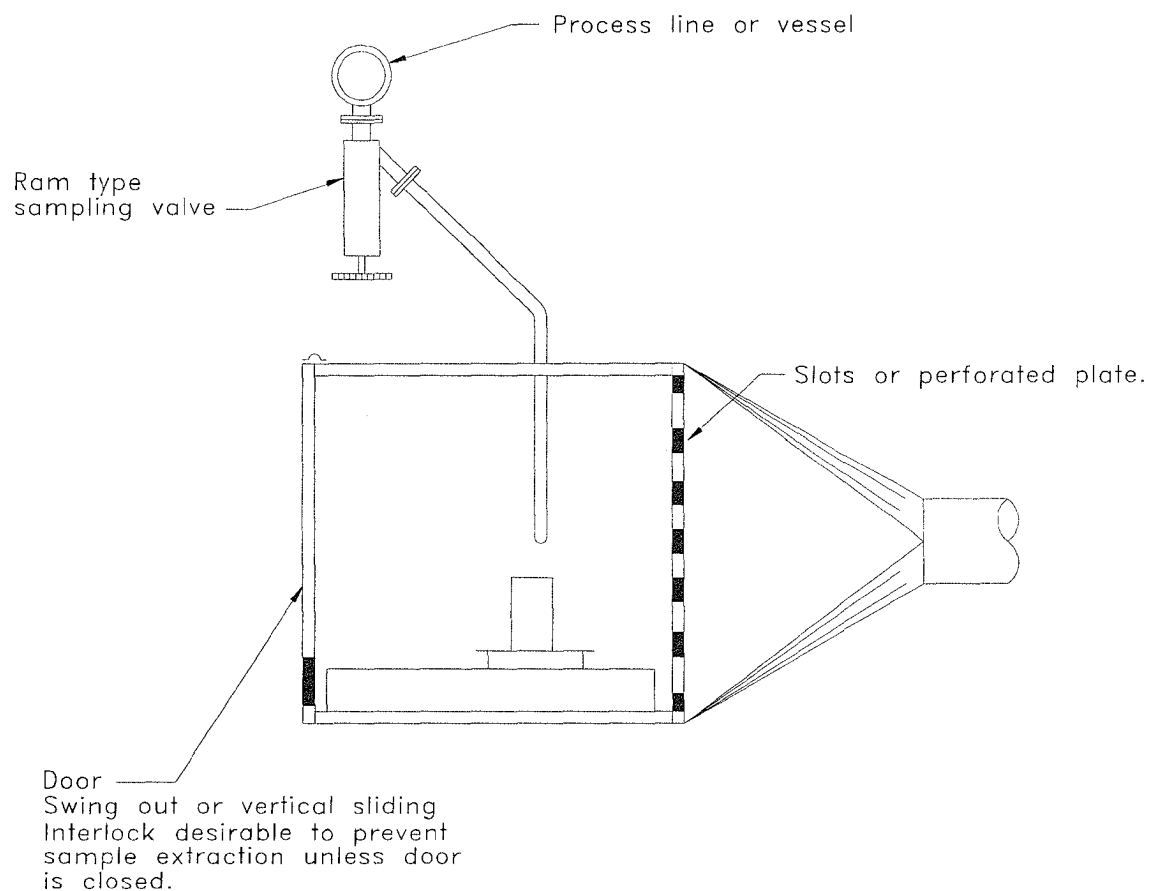
SHAFT SEAL ENCLOSURE

DATE

1-91

FIGURE

VS-15-21



$$Q = 125 \text{ cfm/ft}^2 \text{ of open area (door area) minimum}$$

$$\text{Duct velocity} = 2000 \text{ fpm}$$

$$h_e = 1.78 \text{ VP}_s + 0.50 \text{ VP}_d$$

NOTE: Sufficient air must be provided when door closed to dilute flammable gases and or vapors to 25 % of LEL. See Chapter 2

Reference: 10.15.5

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SAMPLING BOX

DATE

1-91

FIGURE

VS-15-30

10.20 FOUNDRY OPERATIONS

Foundry operations include many operations common to other industries. Some of these operations are covered in the following subsections of this chapter:

- 10.45 Machining
- 10.50 Material Transport
- 10.55 Metal Melting
- 10.60 Mixing
- 10.80 Surface Cleaning
- 10.90 Welding and Cutting

This subsection addresses operations that are more unique to the foundry industry: casting shakeout and core making.

10.20.1 Casting Shakeout: Foundry shakeout ventilation rates depend on the type of enclosure and the temperature of the sand and castings. The enclosing shakeout hood (VS-20-01) requires the smallest air flow rate. The side draft shakeout hood (VS-20-02) requires additional air flow rates but provides improved access for casting and sand delivery and for casting removal. The downdraft shakeout (VS-20-03) is the least effective in controlling contaminant and requires the

highest ventilation rates. It is not recommended for hot castings. The shakeout hopper below the shakeout table requires additional exhaust ventilation equivalent to 10% of the shakeout hood exhaust rate.

Particular attention should be paid to the conveyor removing sand from the shakeout. This conveyor requires hoods and ventilation as described in Section 10.50.

Rotary tumble mills used for shakeout should be treated as an enclosing hood with a minimum inward velocity of 150 fpm through any opening.

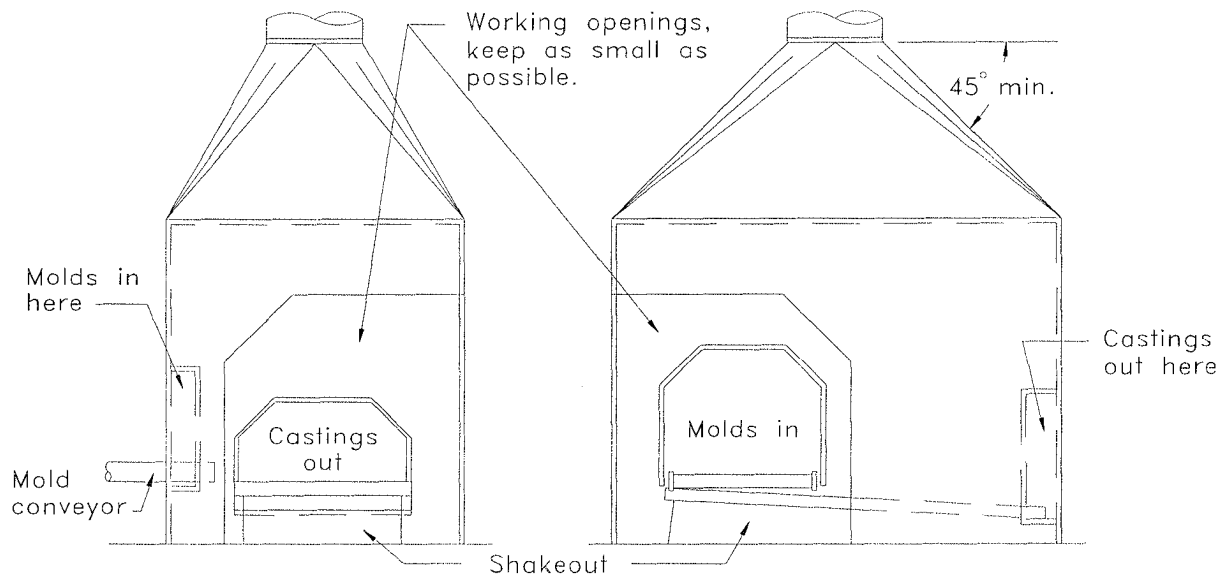
10.20.2 Core Making: Core making machines require ventilation to control reactive vapors and gases such as amines and isocyanates that are used in the core making process. A minimum capture velocity of 75 fpm is required. However, a ventilation rate as high as 250 cfm/ft² of opening may be necessary for adequate control of contaminant emissions. When cores are cured in ovens, adequate ventilation control of the oven is required.

REFERENCES

- 10.20.1 American Foundrymen's Society, Inc.: Foundry Ventilation Manual. AFS, Des Plaines, IL (1985).

ENCLOSING HOOD

Provides best control with least flow rate
 Minimum duct velocity = 4000 fpm
 $h_e = 0.25 VP_d$



Shakeout exhaust, minimum*

Type of hood	Hot castings	Cool castings
Enclosing ** VS-20-01	200 cfm/ft ² opening At least 200 cfm/ft ² grate area	200 cfm/ft ² opening At least 150 cfm/ft ² grate area
Two sides and 1/3 top area enclosed ** VS-20-02	300 cfm/ft ² grate area	275 cfm/ft ² grate area
Side hood (as shown or equivalent) ** VS-20-02	400-500 cfm/ft ² grate area	350-400 cfm/ft ² grate area
Double side hood ** VS-20-02	400 cfm/ft ² grate area	300 cfm/ft ² grate area

- *Choose higher values when
- (1) Castings are quite hot
 - (2) Sand to metal ratio is low
 - (3) Cross-drafts are high

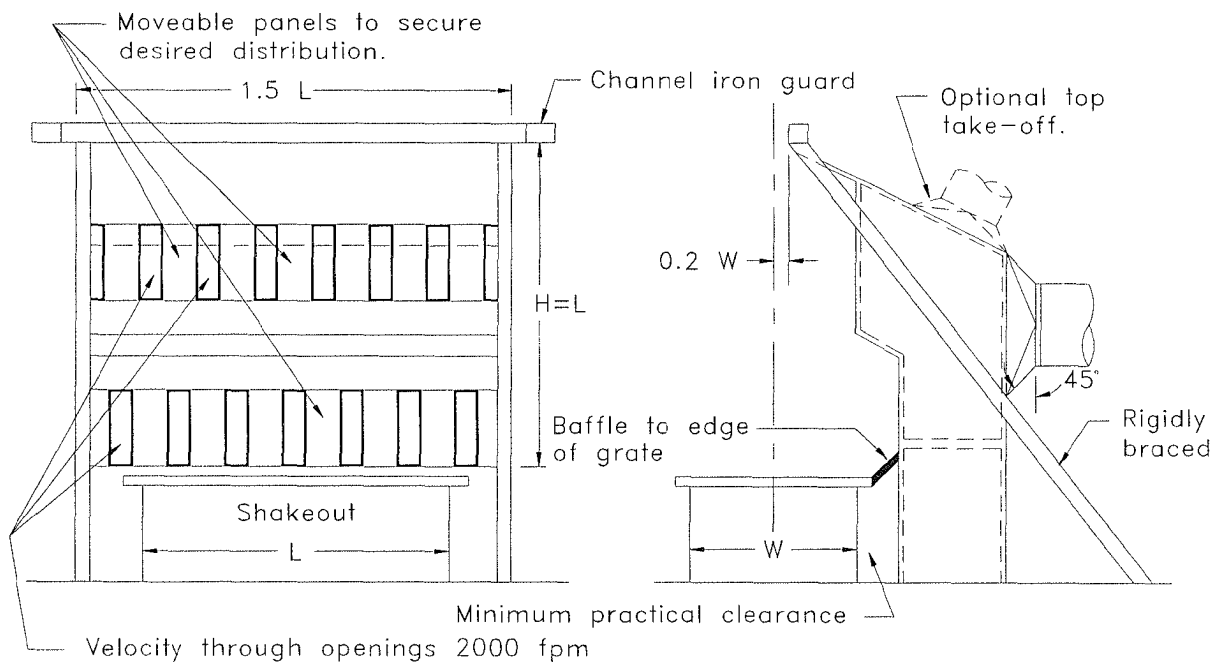
** Shakeout hoppers require an additional 10% exhaust.

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FOUNDRY SHAKEOUT
 ENCLOSING

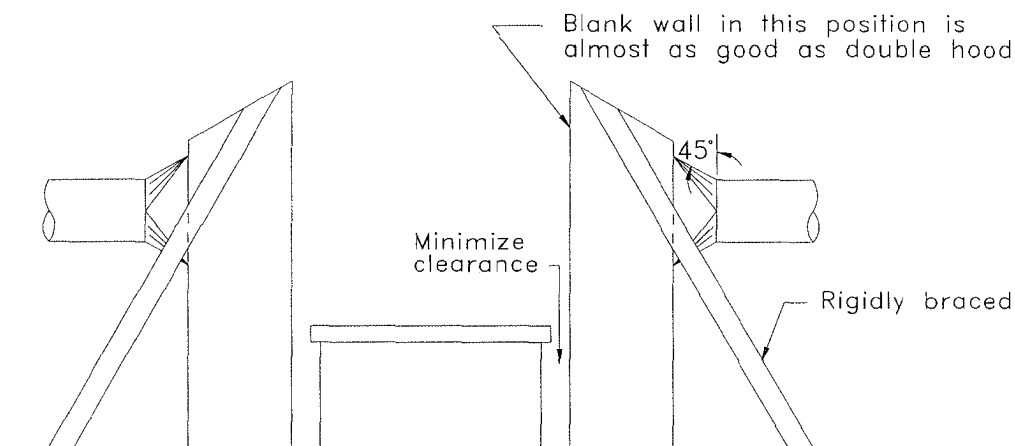
DATE 10-90

FIGURE VS-20-01



SIDE-DRAFT HOOD

Minimum duct velocity = 4000 fpm.
 $h_e = 1.78 VP_s + 0.25 VP_d$



DOUBLE SIDE-DRAFT

Proportions same as single side-draft hood except for overhang.

Minimum duct velocity = 4000 fpm
 Slots sized for 2000 fpm
 $h_e = 1.78 VP_s + 0.25 VP_d$

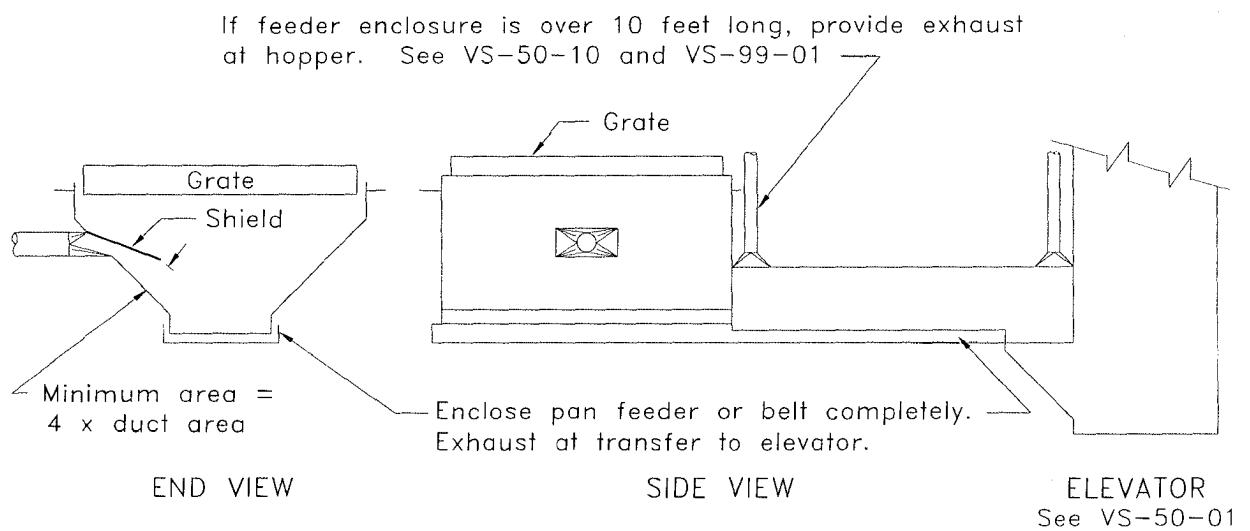
See VS-20-01 for exhaust rates

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FOUNDRY SHAKEOUT
 SIDE DRAFT

DATE 10-90

FIGURE VS-20-02



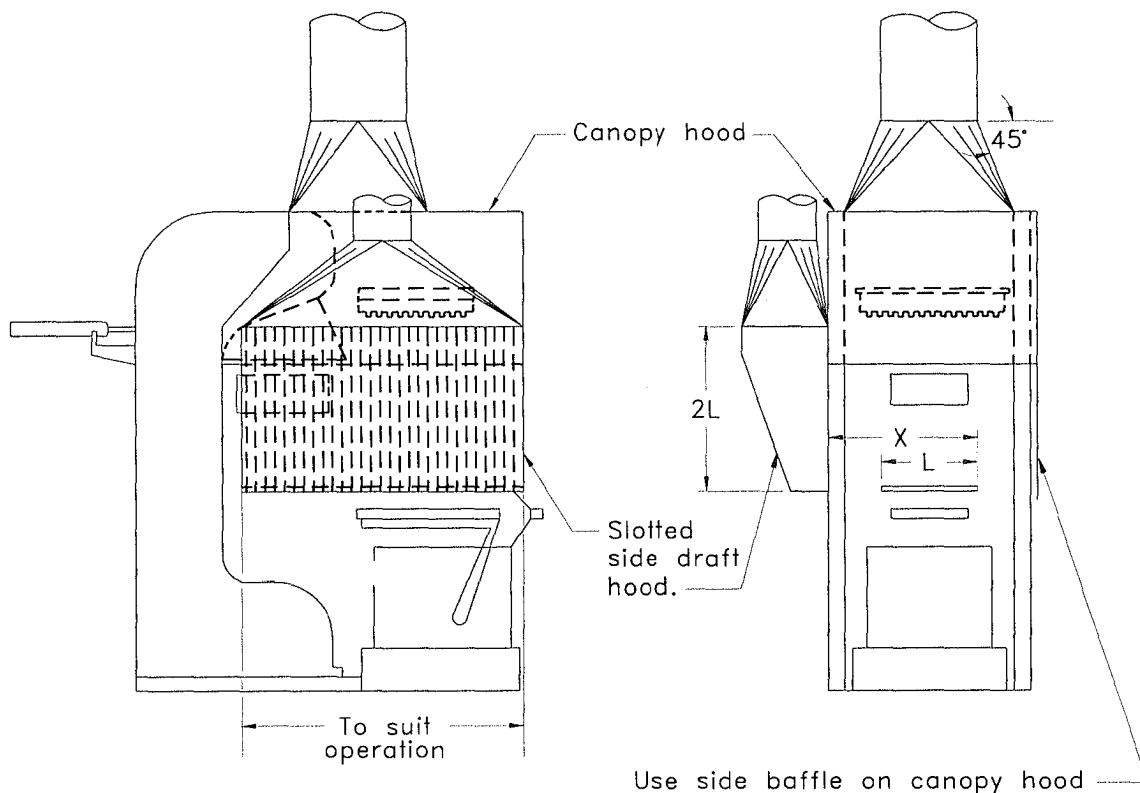
HOPPER EXHAUST DETAIL

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FOUNDRY SHAKEOUT

DATE 11-90

FIGURE VS-20-03



Canopy hood: $Q = 250 \text{ cfm/ft}^2 \text{ canopy} - \text{single unit}$
 $150 \text{ cfm/ft}^2 \text{ canopy} - \text{double unit}$
 $h_e = 0.25 VP_d$

Note: Slotted side draft hoods required to remove smoke as hot cores emerge from machine.
 Minimum capture velocity = 150 fpm

Side draft hood: $Q = 150(10X^2 + A)$ where A equals hood area
 $h_e = 1.78 VP_s + 0.25 VP_d$

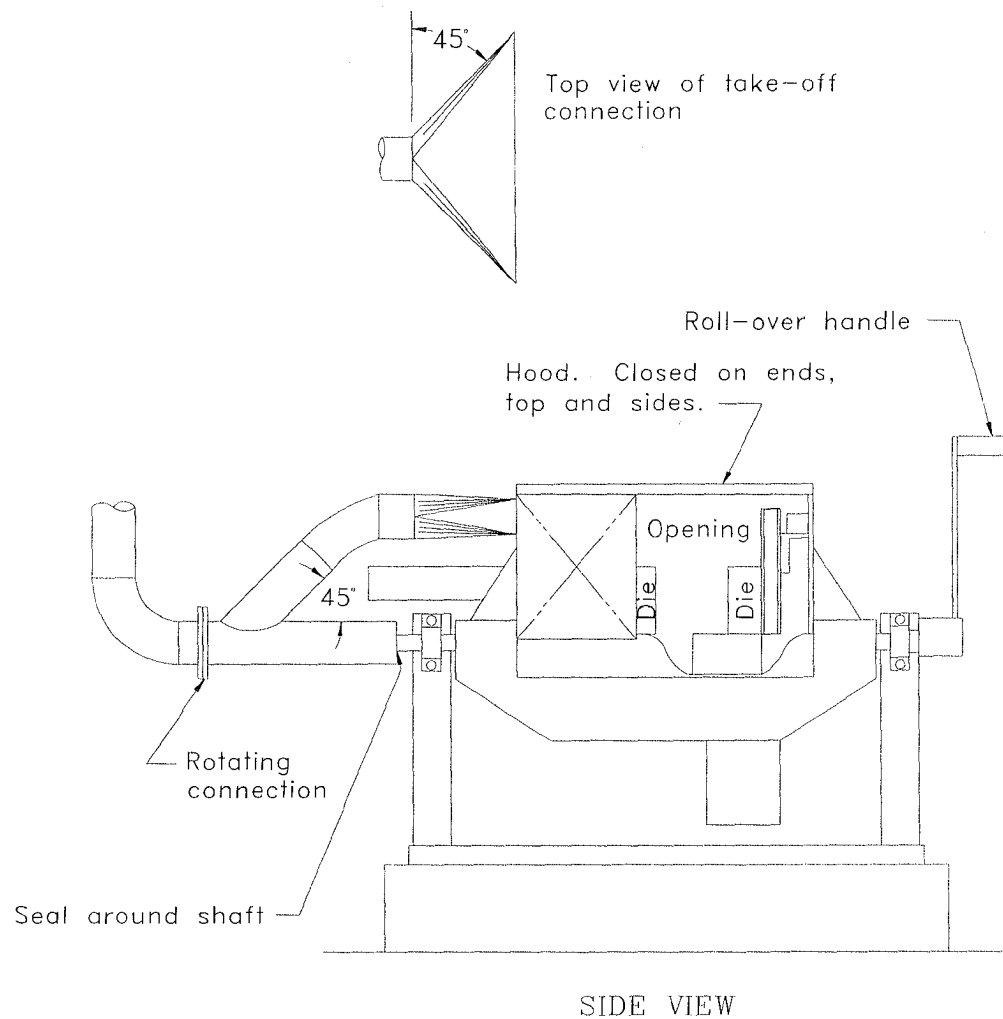
Note: Conveyor or cooling area require ventilation for large cores. Scrap conveyor or tote boxes may also require additional ventilation.
 Minimum duct velocity = 3500 fpm

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SHELL CORE MAKING

DATE 10-90

FIGURE VS-20-10



$$Q = 200 \text{ cfm/ft}^2 \text{ of open face area}$$

$$\text{Minimum duct velocity} = 3500 \text{ fpm}$$

$$h_e = 0.25 VP_d$$

Note: Elbow and rotating connection losses not included.

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CORE MAKING MACHINE
SMALL ROLL-OVER TYPE

DATE 10-90

FIGURE VS-20-11

10.25 GAS TREATMENT

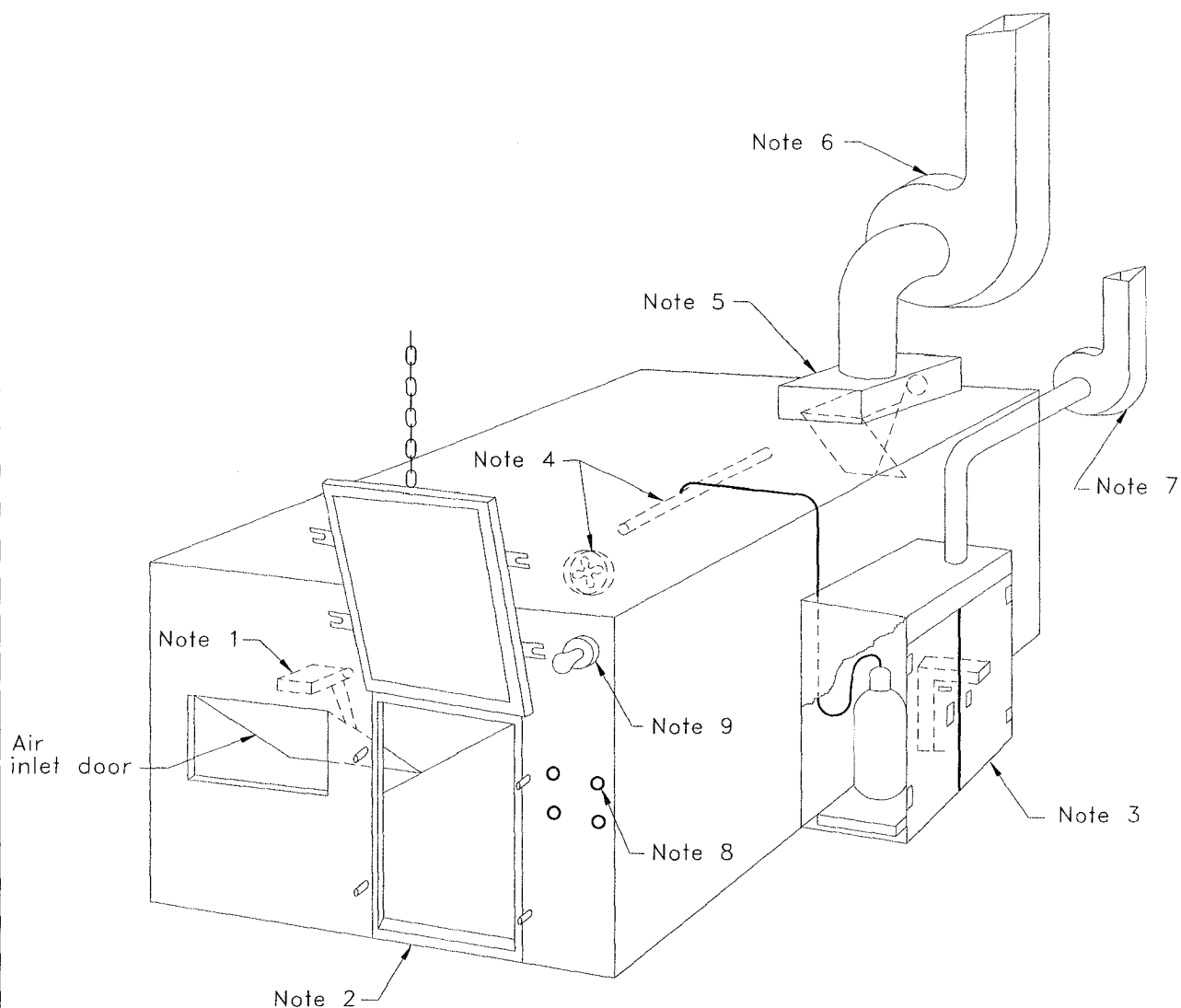
The handling of gas cylinders for industrial operations requires special attention. In addition to the potential safety problems associated with transportation and use of compressed gas cylinders, the gas inside the cylinders can escape through leaky valves and fittings. During connection and disconnection of the gas lines, due to the operating pressures, gas can be released.

This section of VS-prints illustrates uses of toxic gases

during fumigation (VS-25-01 and -02) and during ethylene oxide sterilization (VS-25-10, -11 and -12).

REFERENCES

- 10.25.1 Mortimer, V.D.; Kercher, S.L.; O'Brien, D.M.: Effective Controls for Ethylene Oxide — A Case Study. *Applied Industrial Hygiene* 1(1):15–20 (1986).
- 10.25.2 Hama, G.M.: Ventilation for Fumigation Booths. *Air Engineering* (December 1964).



Ventilation Rates

Allow 60 minute purge time

Ventilation rate must be 20 air changes per hour or greater.

Design must provide:

- (a) 500 fpm velocity or greater through air inlet door when large access door is closed
- and
- (b) at least 100 fpm through all openings when large access door is open.

Reference: 10.25.2

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FUMIGATION BOOTH

DATE

1-91

FIGURE

VS-25-01

NOTES:

1. Provide an air inlet with automatic damper closure; damper must be interlocked with fan circuit to open only when fan is turned on. Size opening for a minimum velocity of 500 fpm. Air inlet must be located so purge air sweeps entire booth.
2. Loading door must be opened only when booth has been completely purged. Provide gaskets, screw clamps, and brackets for applying uniform pressure for a gas-tight fit.
3. Provide ventilated cabinet for gas cylinders in use and being stored. Fan must be on continuously and exhaust approximately 500 cfm to produce a negative pressure in the cabinet when the doors are closed.
4. Provide nozzle openings for introducing fumigant gas. A circulating cabinet fan should also be provided for obtaining good mixture of fumigant gas.
5. Mechanical fan damper must be provided that closes tightly when fan is shut off during fumigation and opens when fan is turned on. Damper controls should be interlocked with fan controls.
6. Fan for ventilating fumigation booth must be sized to dilute air to safe limit in required time. Use vertical, outside, discharge stack away from windows, doors, and air intakes.
7. Fumigant gas cylinder cabinet fan must run continuously.
8. Control switches for fan and lights and an air flow switch-actuated pilot light are recommended.
9. Red warning light to indicate booth is under fumigation as a protection against careless entry is recommended.
10. To facilitate penetration of fumigant gas and subsequent airing out, mattresses should be loaded with separators to allow free air space around each mattress.
11. Fumigants with no odor-warning properties should be used together with an odor-indicating chemical.
12. Where toxic fumigants are used, a leak test should be made on the booth. The booth first should be tested by lighting several large smoke candles in it with doors and dampers closed. Leaks can be noted by the presence of smoke at the point of escape. Where highly diffusible toxic gases are used, an additional test should be made with the booth under charge, at doors and dampers, with a sensitive detecting meter or sampling device.

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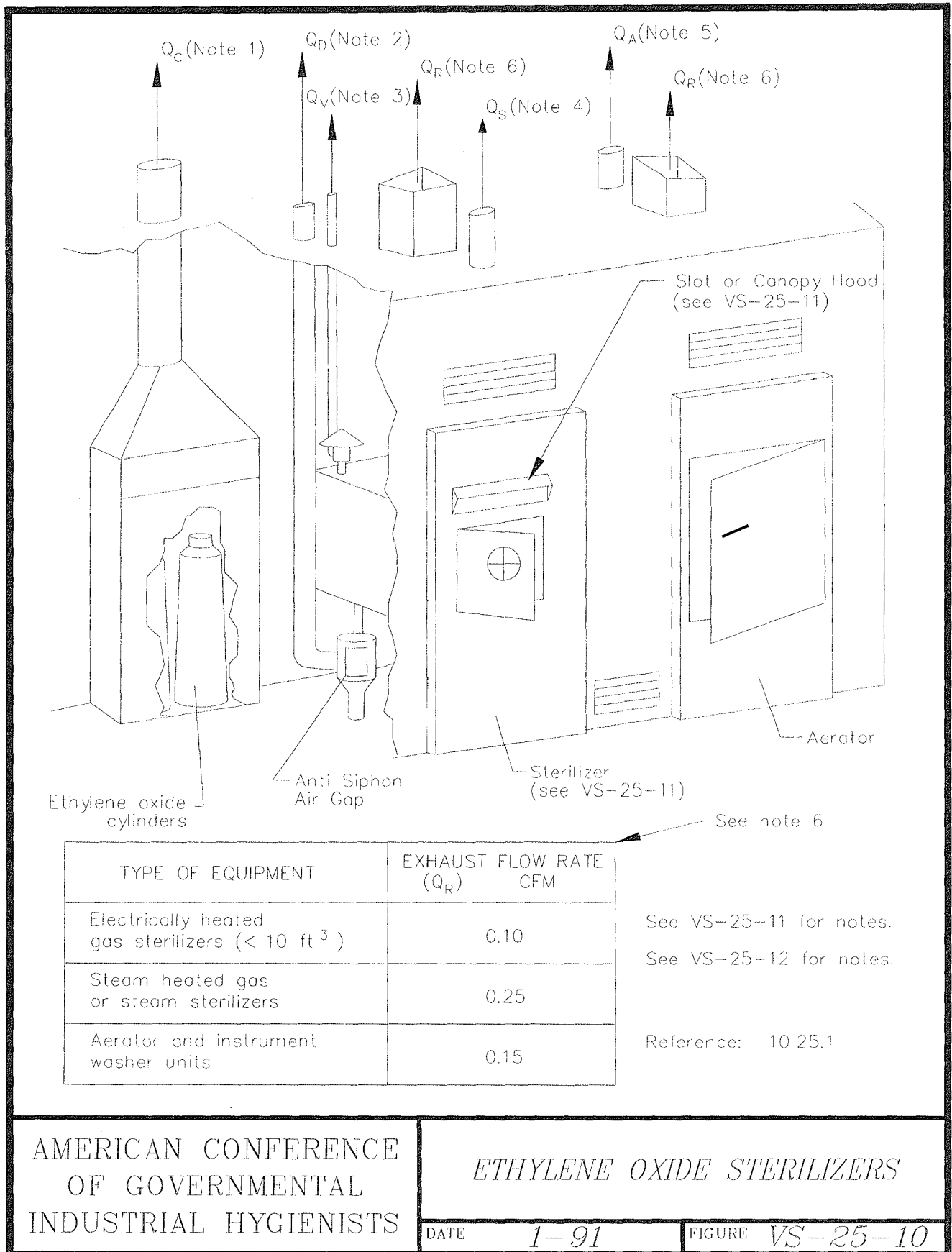
FUMIGATION BOOTH NOTES

DATE

1-91

FIGURE

VS-25-02



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ETHYLENE OXIDE STERILIZERS

DATE

1-91

FIGURE

VS-25-10

NOTES:

1. The ethylene oxide (EtO) supply cylinders should be placed in a ventilated cabinet or a partially enclosed hood with an exhaust rate, Q_C , of at least 100 cfm/ft² of open area.
2. The anti-syphon air gap in the sterilizer evacuation drain line should be enclosed and ventilated. The enclosure should have one or two openings to allow air, Q_D , to enter and to prevent liquid, which might back up from the drain, from reaching the sterilizer evacuation line. In lieu of a greater value specified by the sterilizer/vacuum pump manufacturer, Q_D should be approximately 50 cfm and the openings sized to maintain approximately a 600 fpm face velocity.
3. The overpressure relief valve should be vented to carry EtO out of the building if it should ever open. With a sealed line connecting the valve with the ventilation duct, there will be no ventilation volume, Q_V , except when the valve opens. Consult the sterilizer manufacturer for the proper size of this line; too much resistance could interfere with proper venting of the chamber.
4. A hood should be placed above the sterilizer door to remove EtO rising from the chamber when the sterilizer door is "cracked" open a few inches for approximately 15 minutes before the sterilized items are removed from the chamber. See VS-25-12 for a discussion of the exhaust volume, Q_S , requirements.
5. If an aerator is installed, its door should be hinged, and it should be placed beside the sterilizer so that the doors of the gas sterilizer and aerator open away from each other to facilitate transferring the sterilized items. Consult the manufacturer for the required air flow, Q_A .
6. The room behind the wall enclosing the sterilizer(s) and other equipment should be exhausted adequately to handle the air driven to the ceiling by the thermal gradients caused by the heated equipment. The ideal arrangement would be to have a properly sized vent above each piece of heated equipment. The total Q_R should be the sum of the values for each piece of heated equipment (see VS-25-10) plus 100 cfm/ft² of open area for transfer vents placed in the upper portion of the room. However, federal hospital standards specify that, for a recess room containing a gas sterilizer, the volume exhausted in one hour should be at least ten times the room volume. Transfer vents placed in the lower portion of the room will help the influx of air to supply the thermal air currents and would not add to the total exhaust requirement.
7. All air that could contain EtO should be exhausted through a ventilation system which does not have vents in any other rooms. The discharge of the fan on the roof should be located so that the exhausted air will not re-enter the building or expose people outside the building. This ventilation system should have a flow sensor/alarm to warn if it is not functioning properly. If there is the possibility of lint in the exhausted air, use a differential pressure sensor or some other type that will not be clogged or stuck open by the accumulation of lint.

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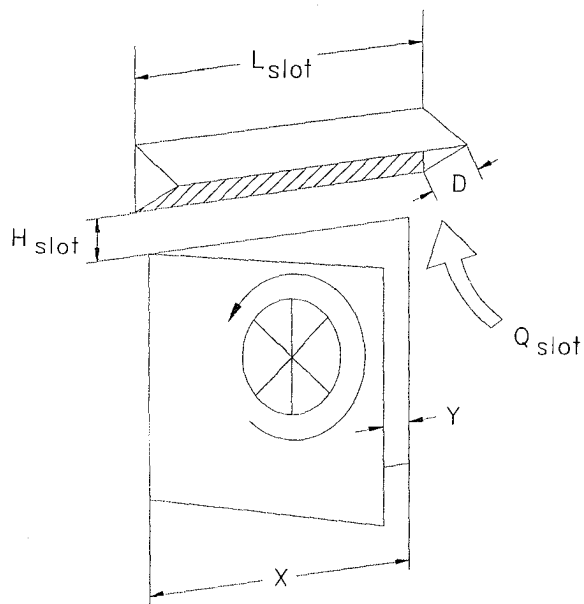
*ETHYLENE OXIDE STERILIZER
NOTES*

DATE

1-91

FIGURE

VS-25-11



SLOT HOOD

$$H_{\text{slot}} \leq 3"$$

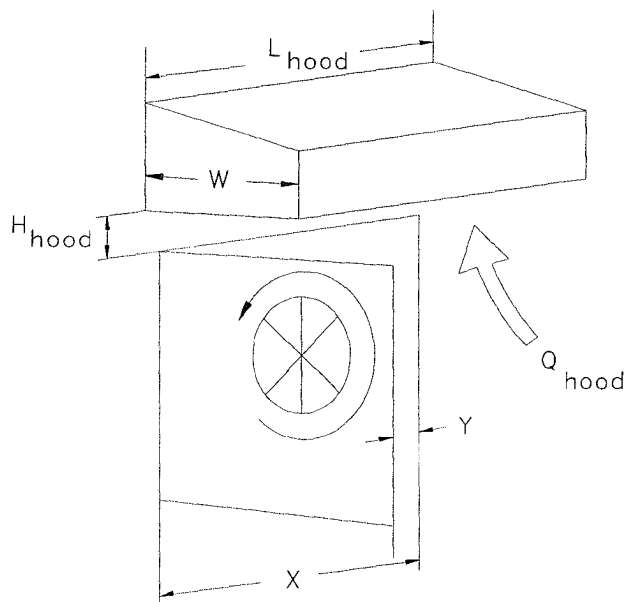
$$L_{\text{slot}} = X + 0.66H_{\text{slot}}$$

$$D = 1"$$

slot face to be at 45°-90°
angle with plane of enclosure

$$Q_{\text{slot}} = 75 \text{ cfm/ft slot length}$$

$$Y = 2" \text{ for airing out sterilizer chamber.}$$



CANOPY HOOD

$$Q_{\text{hood}} = 100 L_{\text{hood}} W$$

$$H_{\text{hood}} = 12" \text{ to } 24"$$

$$W = \frac{H_{\text{hood}}}{3}$$

$$L_{\text{hood}} = X + 0.66H_{\text{hood}}$$

$$Y = 2" \text{ for airing out sterilizer chamber.}$$

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ETHYLENE OXIDE STERILIZER
HOOD DETAILS

DATE 02-91

FIGURE VS-25-12

10.30 KITCHEN EQUIPMENT

The purpose of an exhaust system for kitchen equipment is to control heat, humidity and grease vapor released into the space by cooking or dishwashing equipment. A secondary consideration is the control of combustion products associated with the heat source which may be vented separately or through the hood itself.

National Fire Protection Association (NFPA) Standard 96^(10.30.1) describes grease filter construction as well as hood construction necessary to maintain hood integrity in the event of a fire. Welded seam construction is preferred and sometimes required by public health authorities to assure cleanability and ease of maintenance. The National Sanitation Foundation Standard No. 4^(10.30.2) also lists hood construction requirements for cleanability and integrity in the cooking and food zones within the hood. In all cases, the local health authorities having jurisdiction should be consulted for construction requirements prior to hood fabrication.

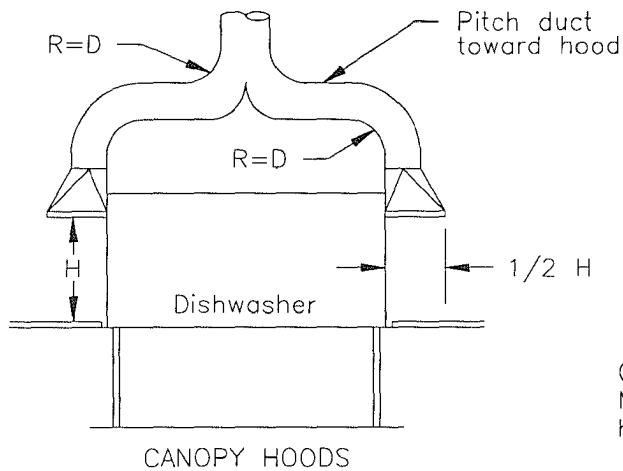
Fire is a primary concern with all cooking equipment. Each hood will require some type of fire suppression consistent

with local fire code requirements. The system selected must not compromise sanitation or endanger workers due to location or system activation. Hood or duct penetrations by fire suppression piping, etc., must be sealed to prevent short circuiting of air or loss of fire arrestance.

For high temperatures situations such as exposed flames or charcoal, the grease filters must be sufficiently removed from the heat source to prevent ignition. Fan selection may require use of high temperature fan components and consideration of the effect of change in air density.

REFERENCES

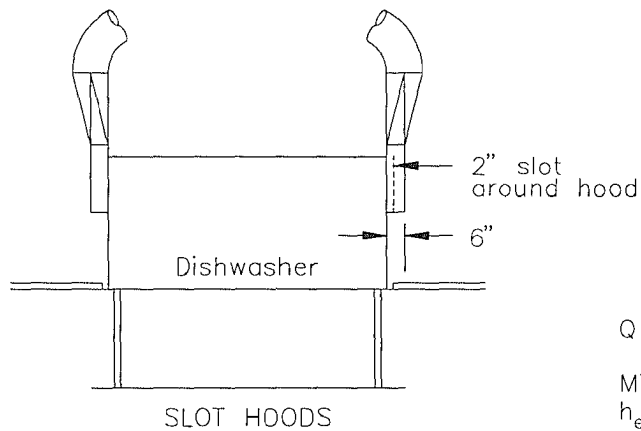
- 10.30.1 National Fire Protection Association: Standard for the Installation of Equipment for the Removal of Smoke and Grease-Laden Vapors from Commercial Cooking Equipment, (Standard 96). NFPA, Quincy, MA (1987).
- 10.30.2 National Sanitation Foundation: Commercial Cooking and Hot Food Storage Equipment, Standard No. 4. NSF, Ann Arbor, MI (1986).



$$Q = 250 \text{ cfm/ft}^2 \text{ of door area— each end}$$

$$\text{Minimum duct velocity} = 1000 - 3000 \text{ fpm}$$

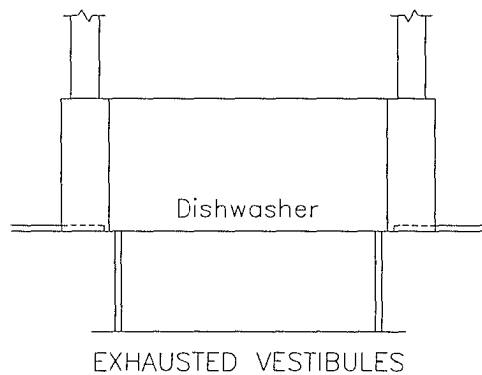
$$h_e = 0.25 VP_d$$



$$Q = 150 \text{ cfm/ft}^2 \text{ of door area (150WH) each end}$$

$$\text{Minimum duct velocity} = 1000 - 3000 \text{ fpm}$$

$$h_e = 1.00 VP_s + 0.25 VP_d$$



$$Q = 150 \text{ cfm/ft}^2 \text{ of entrance and exit area}$$

$$\text{Minimum duct velocity} = 1000 - 3000 \text{ fpm}$$

$$h_e = 0.50 VP_d$$

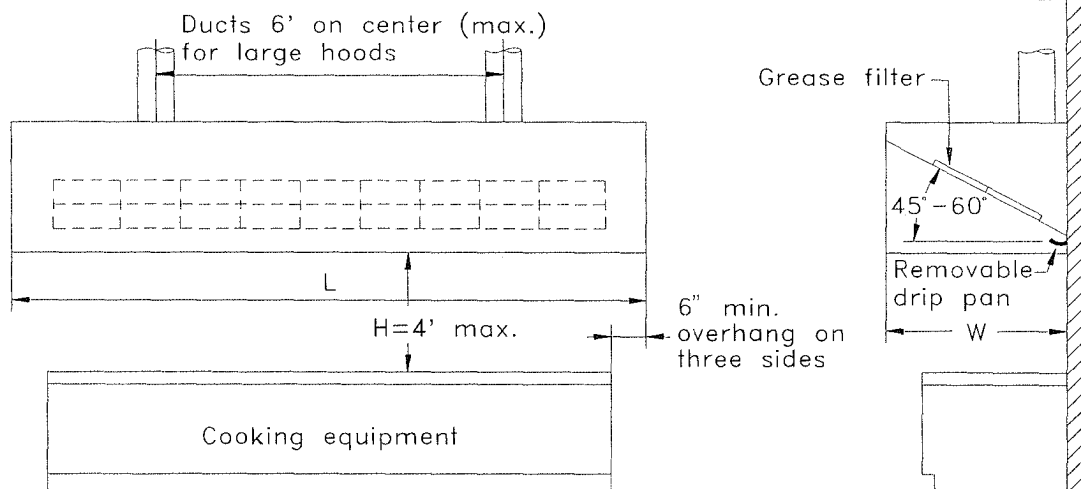
Note: If direct exhaust connections are provided from dishwasher body, cap these connections and use external hoods.

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DISHWASHER VENTILATION

DATE 10-90

FIGURE VS-30-01



HOOD AGAINST WALL

$Q = 80 \text{ cfm/ft}^2$ of hood area (80 WL)

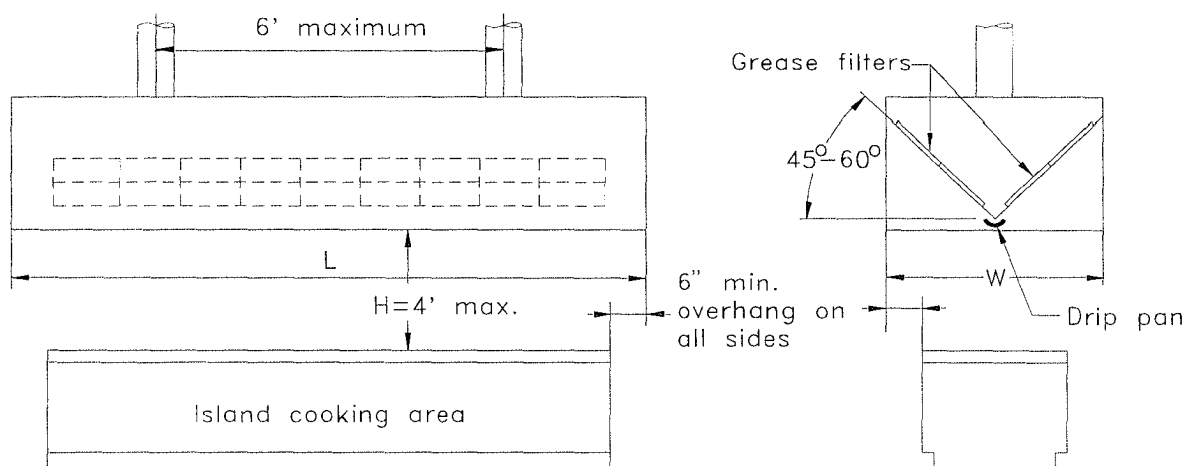
Not less than 50 cfm/ft^2 of face area (50 PH)

$P = \text{perimeter of hood} = 2W + L$

Duct velocity = 1000 - 4000 fpm, to suit conditions

$h_e = (\text{filter resistance} + 0.1'') + 0.50 VP_d$ (straight take off)

$h_e = (\text{filter resistance} + 0.1'') + 0.25 VP_d$ (tapered take off)



ISLAND TYPE HOOD

$P = \text{perimeter of hood}$
 $= 2W + 2L$

$Q = 125 \text{ cfm/ft}^2$ of hood area (125 WL)

Not less than 50 cfm/ft^2 of face area (50 PH)

Minimum duct velocity = 1000 - 4000 fpm, to suit conditions

$h_e = (\text{filter resistance} + 0.1'') + 0.50 VP_d$ (straight take off)

$h_e = (\text{filter resistance} + 0.1'') + 0.25 VP_d$ (tapered take off)

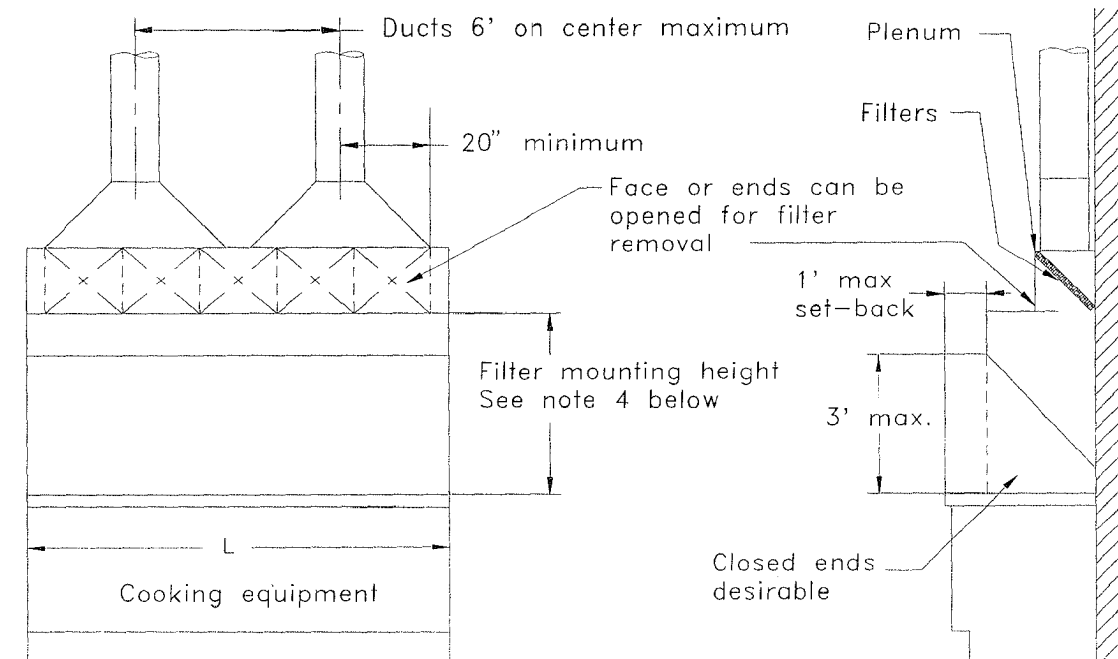
Note: See VS-30-11 for information about filters and fans for range hoods.

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KITCHEN RANGE HOODS

DATE 10-90

FIGURE VS-30-10



LOW SIDE WALL HOOD

$Q = 200 \text{ cfm/lineal ft of cooking surface (200L)}$

Minimum duct velocity = 1000 - 4000 fpm, to suit conditions

$h_e = (\text{filter resistance} + 0.1'') + 0.50 VP_d \text{ (straight take off)}$

$h_e = (\text{filter resistance} + 0.1'') + 0.25 VP_d \text{ (tapered take off)}$

NOTES FOR KITCHEN HOODS

Filters: 1. Select practical filter size.

2. Determine number of filters required from manufacturer's data.
(Usually: 2 cfm maximum exhaust for each square inch of filter area.)

3. Install at 45°-60° to horizontal. Never horizontal.

4. Filter mounting height (Reference 10.30.1)

a. No exposed cooking flame --- 18" minimum to lowest edge of filter.

b. Charcoal and similar fires --- 4' minimum to lowest edge of filter.

5. Shield filters from direct radiant heat.

6. Provide removable grease drip pan.

7. Clean pan and filters regularly.

Fan: 1. Use upblast discharge fan. Downblast is not recommended.

2. Select fan for design Q and SP resistance of filters and duct.

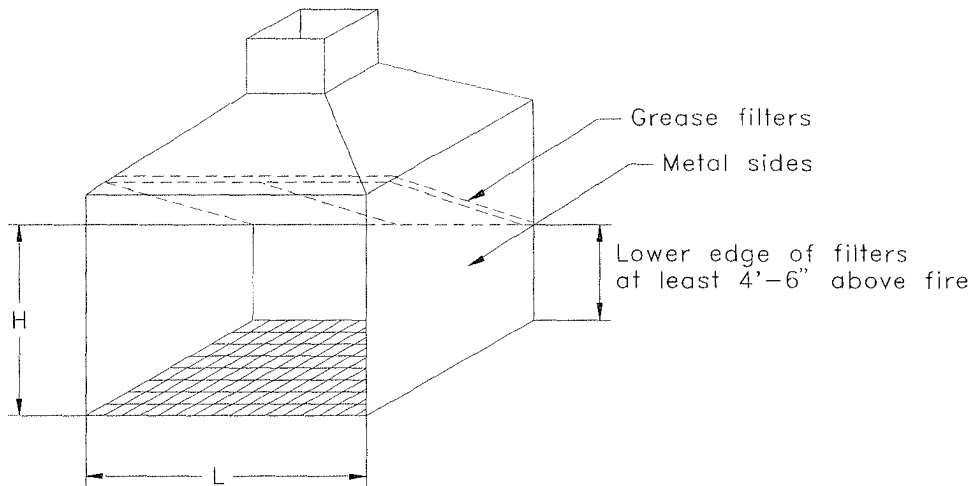
3. Adjust fan specification for expected exhaust air temperature.

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KITCHEN RANGE HOOD

DATE 10-90

FIGURE VS-30-11



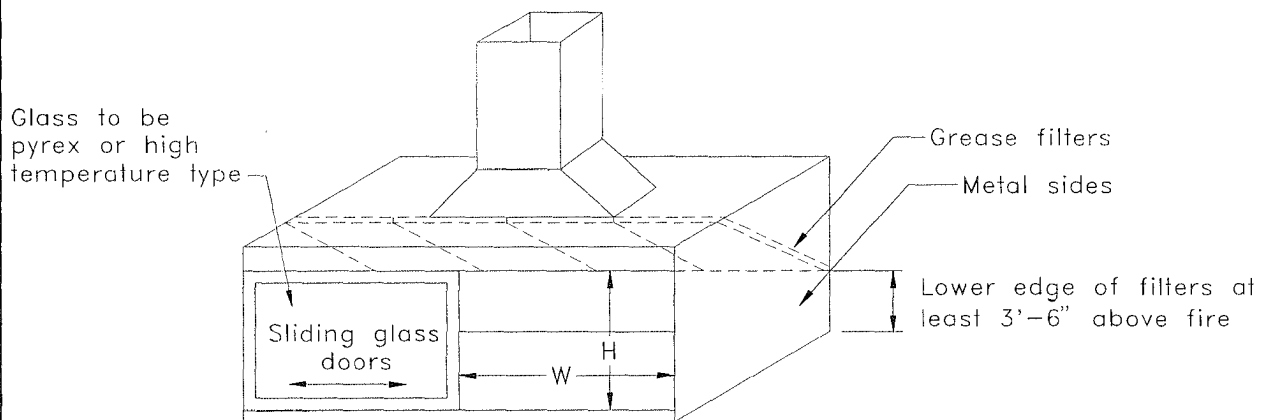
CHARCOAL BROILER

$$Q = 100 LH$$

Minimum duct velocity = 1000 - 3000 fpm

$$h_e = (\text{filter resistance} + 0.1'') + 0.50 \text{ VP (straight take off)}$$

$$h_e = (\text{filter resistance} + 0.1'') + 0.25 \text{ VP (tapered take off)}$$



BARBEQUE PITS

Notes: 1. If hood is more than 12 feet long use multiple takeoffs 6 feet on center.

2. See VS-30-11 for information about filters and fans for range hoods.

$$Q = 100 WH \text{ (maximum open door area, ft}^2\text{)}$$

Minimum duct velocity = 1000 - 3000 fpm

$$h_e = (\text{filter resistance} + 0.1'') + 0.50 \text{ VP (straight take off)}$$

$$h_e = (\text{filter resistance} + 0.1'') + 0.25 \text{ VP (tapered take off)}$$

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CHARCOAL BROILER AND
BARBEQUE PIT VENTILATION

DATE

10-90

FIGURE

VS-30-12

10.35 LABORATORY VENTILATION

The primary method of contaminant control within the laboratory is exhaust ventilation and, in particular, laboratory hoods. This section presents information on laboratory hoods but expands to other types of ventilation control such as biological safety cabinets, clean benches, and other local exhaust systems found in the laboratory.

10.35.1 Laboratory Hoods: In most cases, laboratory hoods will be purchased from manufacturers specializing in the design and construction of laboratory hoods. VS-35-01 shows a typical laboratory hood design. VS-35-02 describes general-use laboratory hoods and VS-35-03 describes perchloric acid hoods. VS-35-04 describes work practices for laboratory hoods.

Several features are essential to the proper performance of the hood. The most important aspect of the hood is the aerodynamic entry characteristics. For the hood to adequately control contaminants, the entry must be smooth. This usually is achieved with an airfoil sill at the leading edge of the workbench. Often, beveled jambs at the side wall entry will improve the air flow.

In many cases, good performance correlates with uniform face velocity. To achieve a uniform face velocity, many hood manufacturers provide adjustable slots in the plenum at the back of the hood. Although the adjustment will allow for unusual conditions such as large hot plates for sample digestions, inappropriate adjustment of the slots can have a detrimental effect on hood performance.^(10.35.1)

Supply Air Distribution: For typical operation of a laboratory hood, the worker stands at the face of the hood and manipulates the apparatus in the hood. The indraft at the hood face creates eddy currents around the worker's body which can drag contaminants in the hood along the worker's body and up to the breathing zone. The higher the face velocity, the greater the eddy currents. For this reason, higher face velocities do not result in greater protection as might be supposed.

Room air currents have a large effect on the performance of the hood. Thus, the design of the room air supply distribution system is as important in securing good hood performance as the face velocity of the hood. American Society of Heating, Refrigeration and Air Conditioning (ASHRAE) research project RP-70 results, reported by Caplan and Knutson,^(10.35.2) conclude in part:

1. Lower breathing zone concentrations can be attained at 50 cfm/ft² face velocities with good air supply distribution than at 150 cfm/ft² with poor air distribution. With a good air supply system and tracer gas released at 8 liters per minute inside the hood, breathing zone concentrations can be kept below 0.1 ppm and usually below 0.05 ppm.
2. The terminal throw velocity of supply air jets should

be no more than one-half the hood face velocity; such terminal throw velocities are far less than conventional practice.

3. Perforated ceiling panels provide a better supply system than grilles or ceiling diffusers in that the system design criteria are simpler and easier to apply, and precise adjustment of the fixtures is not required.

For the reasons described, an increased hood face velocity may be self-defeating because the increased air volume handled through the room makes the low-velocity distribution of supply air more difficult.

Selection of Hood Face Velocity: The interaction of supply air distribution and hood face velocity makes any blanket specification of hood face velocity inappropriate. Higher hood face velocities will be wasteful of energy and may provide no better or even poorer worker protection. The ANSI/ASHRAE Hood Performance Test^(10.35.3) may be used as a specification. The specified performance should be required of both the hood manufacturer and the designer of the room air supply system.

The specification takes the form:

AUyyy, Alyyy, or AMyyy

where:

AU identifies an "as used" test

AI identifies an "as installed" test

AM identifies an "as manufactured" test

yyy = control level, ppm, at the breathing zone of the worker.

Any well-designed airfoil hood, properly balanced, can achieve < 0.10 ppm control level when the supply air distribution is good. Therefore, it would seem appropriate that the "AM" requirements would be < 0.10 ppm. The "AU" requirement involves the design of the room supply system and the toxicity of the materials handled in the hood. The "AU" specification would be tailored to suit the needs of the laboratory room location.

For projected new buildings, it is frequently necessary to estimate the cost of air conditioning early — before the detailed design and equipment specifications are available. For that early estimating, the guidelines listed in Table 10.35.1 can be used.

10.35.2 Biological Safety Cabinets: Biological safety cabinets (BSCs) are classified as Class I; Class II, Types A, B1, B2 and B3; and Class III.

Class I BSC provides personnel and environmental protection but does not protect the product. The front panel can be open, allowing room air to enter the cabinet, sweep the inner surfaces and exhaust out the duct. A front closure panel with glove ports may be installed. If gloves are installed, air is

TABLE 10.35.1. Laboratory Hood Ventilation Rates

Condition	cfm/ft ² Open Hood Face
1. Ceiling panels properly located with average panel face velocity < 40 fpm. ^(10.35.2) Horizontal sliding sash hoods. No equipment in hood closer than 12 inches to face of hood. Hoods located away from doors and traffic ways.*	60
2. Same as 1 above; some traffic past hoods. No equipment in hoods closer than 6 inches to face of hood. Hoods located away from doors and traffic ways.*	80
3. Ceiling panels properly located with average panel face velocity < 60 fpm. ^(10.35.2) or ceiling diffusers properly located; no diffuser immediately in front of hoods; quadrant facing hood blocked; terminal throw velocity < 60 fpm. No equipment in hood closer than 6 inches to face of hood. Hoods located away from doors or traffic ways.*	80
4. Same as 3 above; some traffic past hood. No equipment in hood closer than 6 inches to face of hood.	100
5. Wall grilles are possible but not recommended for advance planning of new facilities	

*Hoods near doors are acceptable if 1) there is a second safe egress from the room; 2) traffic past hood is low; and 3) door is normally closed.

drawn through a secondary opening equipped with a roughing filter. A laboratory hood, as shown in VS-35-20, could be considered a Class I BSC if the exhausted air is passed through HEPA filters prior to release to the atmosphere.

Class II BSCs provide personnel, product, and environmental protection. Class II cabinets differ in the proportion of air recirculated within the cabinet; velocity of air flow to the work surface; where the exhausted air is discharged; and whether the contaminated air plenum is under positive pressure. A Type A cabinet (VS-35-10) may discharge the exhausted air, after HEPA filtration, directly into the room. Type A cabinets which discharge into the work area are not recommended for use with gases or vapors. A primary application is for sterile packaging. Care is required while decontaminating the cabinet.

Type B hoods (VS-35-11) discharge the exhaust but may recirculate within the cabinet. Type B1 cabinets recirculate about 30% of the air within the BSC and typically exhaust the remainder outside the laboratory (i.e., exhaust air is not discharged back into the room). The contaminated plenum is under negative pressure. Type B2 cabinets are referred to as "total exhaust" cabinets as the contaminated air is exhausted to the atmosphere after HEPA filtration without recirculation in the cabinet or return to the laboratory room air. Type B3 BSCs have HEPA filtered downflow air that is a portion of the mixed downflow and inflow air from a common exhaust plenum.

Class III BSCs (VS-35-20) provide the highest level of protection to personnel and the environment. The cabinet is totally enclosed with operations conducted through attached gloves. See "National Sanitation Foundation Standard No. 49"^(10.35.4) for descriptions and requirements of the various classes of BSCs.

10.35.3 Clean Benches: Clean benches can be divided into laminar flow and exhausted clean benches.

Laminar flow clean benches provide product protection only. In a laminar flow clean bench, room air is HEPA-filtered, directed across the work area and discharged back to

the room. Air may be directed horizontally as depicted in VS-35-30 or vertically as in VS-35-31. Neither of these hoods provide worker protection. Workers using the Horizontal Laminar Flow Clean Bench are exposed to the product as the air sweeps across the product into the worker's face. Workers' arms or other objects protruding into the Vertical Laminar Flow Clean Bench opening may cause contaminated air to spill into the room. Personal protective equipment or general ventilation should be provided as needed.

Other types of clean benches incorporate the same general principles of biological safety cabinets and utilize HEPA filtered laminar flow within the hood to provide product protection and exhaust sufficient air to ensure flow into the hood at the face to provide operator protection.

10.35.4 Laboratory Equipment: Some laboratory equipment such as evaporation hoods (VS-35-40), discharge from instruments such as ICP or AA and some ovens (VS-35-41) require local exhaust ventilation to adequately control contaminant releases. Often, specially designed ventilation specific to the operation provides better control than using a laboratory hood to control these releases.

REFERENCES

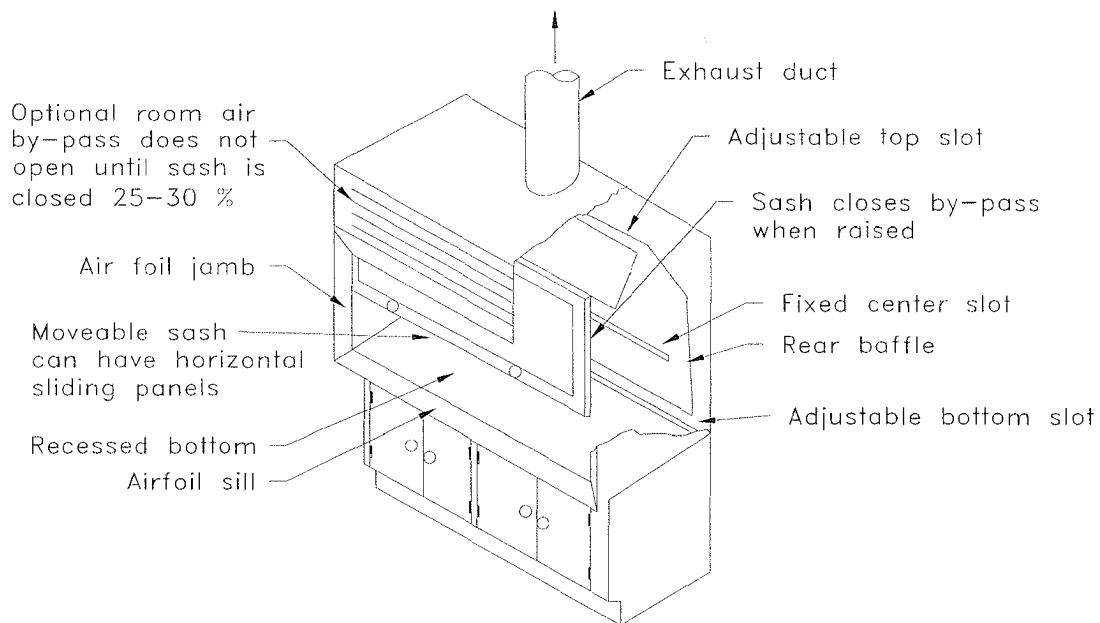
- 10.35.1 Knutson, G.W.: Effect of Slot Position on Laboratory Fume Hood Performance. Heating, Piping and Air Conditioning (February, 1984).
- 10.35.2 Caplan, K.J.; Knutson, G.W.: Influence of Room Air Supply on Laboratory Hoods. American Industrial Hygiene Association Journal 43(10):738-746 (1982).
- 10.35.3 American Society of Heating, Refrigerating and Air Conditioning Engineers: ANSI/ASHRAE Standard 110-1995, Method of Testing the Performance of Laboratory Fume Hoods. ASHRAE, Atlanta, GA (1995).
- 10.35.4 National Sanitation Foundation: Standard 49, Class II (Laminar Flow) Biohazard Cabinetry. NSF, Ann

Arbor, MI (1987).

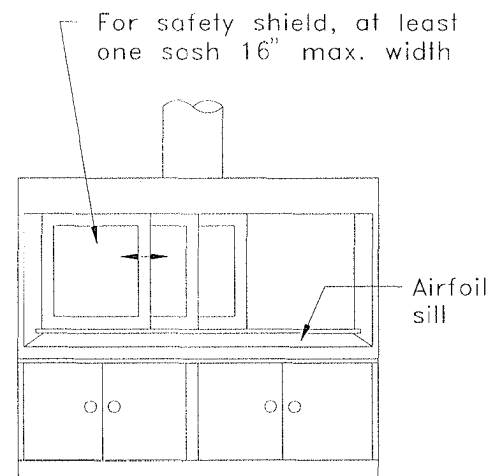
- 10.35.5 U. S. Air Force: Technical Order 00-25-203: Standards and Guidelines for Design and Operation of Clean Rooms and Clean Work Stations. Office of Technical Services, Department of Commerce,

Washington, DC (July 1963).

- 10.35.6 Harris, W.P.; Christofano, E.E.; Lippman, M.: Combination Hot Plate and Hood for Multiple Beaker Evaporation. American Industrial Hygiene Association Journal 22(4) (August 1961).



VERTICAL SASH AIRFOIL HOOD


HORIZONTAL SASH
AIRFOIL HOOD

$Q = 80-100 \text{ cfm/ft}^2$ full open area
depending on quality of supply
air distribution and uniformity
of face velocity

$$h_e = 0.5 VP_d$$

Duct velocity = 1000-2000 fpm to
suit conditions

Design specifications:

General use laboratory hoods—See VS-35-02

Perchloric acid —See VS-35-03

"Auxiliary Air" or "Compensating" hoods
furnish some replacement air at hood face,
design varies with vendor.

Work practices — See VS-35-04

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TYPICAL LABORATORY HOOD

DATE 02-91

FIGURE VS-35-01

GENERAL USE LABORATORY HOODS

- A. Provide uniform exhaust air distribution in hood. Adjust baffles and air flow for ≤ 10 percent variation in point-to-point face velocity with sash in maximum open position.
- B. Locate hood away from heavy traffic aisles and doorways. Hoods near doors are acceptable if: 1) there is a second safe means of egress from room, 2) traffic past hood is low, and 3) door is normally closed.
- C. Use corrosion-resistant materials suitable for expected use.
- D. Provide air cleaning on exhaust air if necessary and adequate stack height to minimize re-entry of contaminants or to comply with air pollution regulations.
- E. Avoid sharp corners at jambs and sill. Tapered or round hood inlets are desirable. An airfoil shroud at sill is important.
- F. Provide filters for radioactive materials in greater than "exempt" quantities.
- G. By-pass opening in hood is desirable to avoid excessive indraft under partially closed sash condition. Opening to be baffled to prevent splash from eruption in hood as shown in VS-35-01.
- H. Provide tempered or conditioned replacement air to laboratory. Replacement air volume to be selected for desired air balance with adjoining spaces.
 - I. In order to reduce air flow volumes, local exhaust hood should be considered instead of laboratory bench hoods for fixed setups.
- J. For air conservation, use horizontal sliding sash with airfoil sill.
- K. All bench hoods should have a recessed work surface and airfoil sill.

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*GENERAL USE LABORATORY
HOODS*

DATE 02-91

FIGURE VS-35-02

PERCHLORIC ACID HOODS

Perchloric acid is extremely dangerous because it is a very strong oxidizer. When the acid reacts with organic material, an explosive reaction product may be formed.

1. Do not use perchloric acid in a hood designed for other purposes. Identify perchloric acid hoods with large warning signs.
2. Provide exhaust ventilation and room supply air with minimal challenge to the hood.
3. Utilize local exhaust ventilation within the hood to minimize condensation of vapors inside the hood.
4. Locate all utility controls outside the hood.
5. Materials of construction for this type of hood and duct must be nonreactive, acid resistant, and relatively impervious. AVOID ORGANIC MATERIALS unless known to be safe. Stainless steel type 316 with welded joints is preferred. Unplasticized polyvinyl chloride or an inorganic ceramic coating, such as porcelain, is acceptable.
6. Ease of cleanliness is paramount. Use stainless steel with accessible rounded corners and all-welded construction.
7. The work surface should be water tight with a minimum of 0.5-inch dished front and sides and an integral trough at the rear to collect the washdown water.
8. Design washdown facilities into the hood and duct. Use daily or more often to thoroughly clean perchloric acid from the exhaust system surfaces.
9. Each perchloric acid hood should have an individual exhaust system. Slope horizontal runs to drain. Avoid sharp turns.
10. Construct the hood and duct to allow easy visual inspection.
11. Where required, use a high-efficiency (greater than 80%) wet collector constructed for perchloric acid service. Locate as close to the hood as possible to minimize the accumulation of perchloric acid in the exhaust duct.
12. Use only an acid-resistant metallic fan protected by an inorganic coating or an air injector.
13. Lubricate the fan with a fluorocarbon-type grease.
14. Locate the fan outside the building.
15. The exhaust discharge must terminate out-of-doors, preferably using a vertical discharge cap that extends well above the roof eddy zone. See Figure 5.30.

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PERCHLORIC ACID HOOD NOTES

DATE

02-91

FIGURE

VS-35-03

WORK PRACTICES FOR LABORATORY HOODS

No large, open-face hood with a low face velocity can provide complete safety for a worker standing at the face against all events that may occur in the hood. The hood may not adequately protect the worker from volatile or otherwise airborne contaminants with a TLV in the low part-per-billion range. For more ordinary exposures, a properly designed hood in a properly ventilated room can provide adequate protection. However, certain work practices are necessary for the hood to perform capably. The following work practices are generally required; more stringent practices may be necessary in some circumstances.

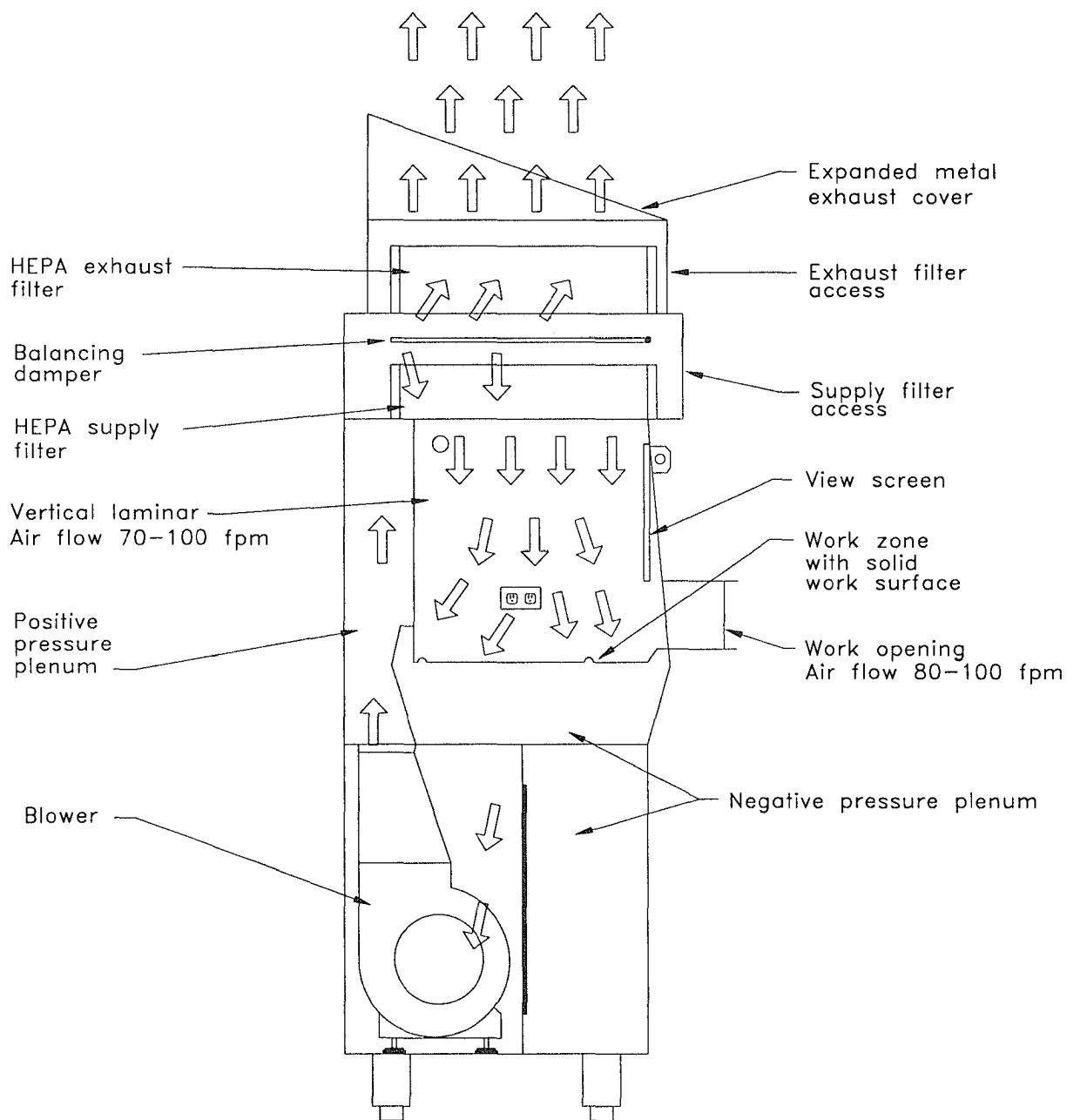
1. Conduct all operations that may generate air contaminants at or above the appropriate TLV inside a hood.
2. Keep all apparatus at least 6 inches back from the face of the hood. A stripe on the bench surface is a good reminder.
3. Do not put your head in the hood when contaminants are being generated.
4. Do not use the hood as a waste disposal mechanism except for very small quantities of volatile materials.
5. Do not store chemicals or apparatus in the hood. Store hazardous chemicals in an approved safety cabinet.
6. Keep the hood sash closed as much as possible.
7. Keep the slots in the hood baffle free of obstruction by apparatus or containers.
8. Minimize foot traffic past the face of the hood.
9. Keep laboratory doors closed (exception: some laboratory designs require lab doors to be open).
10. Do not remove hood sash or panels except when necessary for apparatus set-up; replace sash or panels before operating.
11. Do not place electrical receptacles or other spark sources inside the hood when flammable liquids or gases are present. No permanent electrical receptacles are permitted in the hood.
12. Use an appropriate barricade if there is a chance of explosion or eruption.
13. Provide adequate maintenance for the hood exhaust system and the building supply system. Use static pressure gauges on the hood throat, across any filters in the exhaust system, or other appropriate indicators to ensure that exhaust flow is appropriate.
14. If hood sash is supposed to be partially closed for the operation, the hood should be so labeled and the appropriate closure point clearly indicated.

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*WORK PRACTICES FOR
LABORATORY HOODS*

DATE 02-91

FIGURE VS-35-04



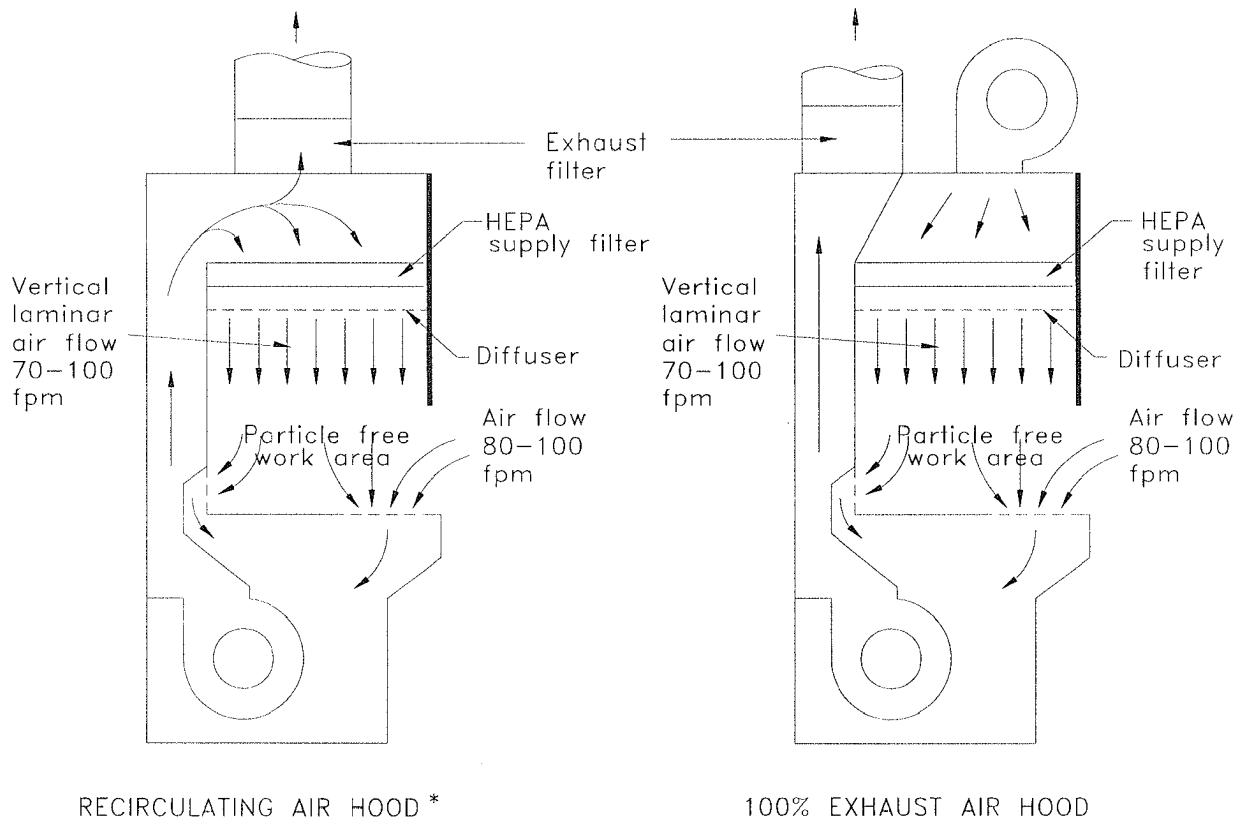
CLASS II TYPE A

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*BIOLOGICAL SAFETY CABINET
CLASS II, TYPE A*

DATE 02-91

FIGURE VS-35-10



* Recirculating Air Hoods are not recommended for use with gases or vapors.

Note: See "National Sanitation Foundation Standard 49" (10.35.4) for requirements and definitions of classes.

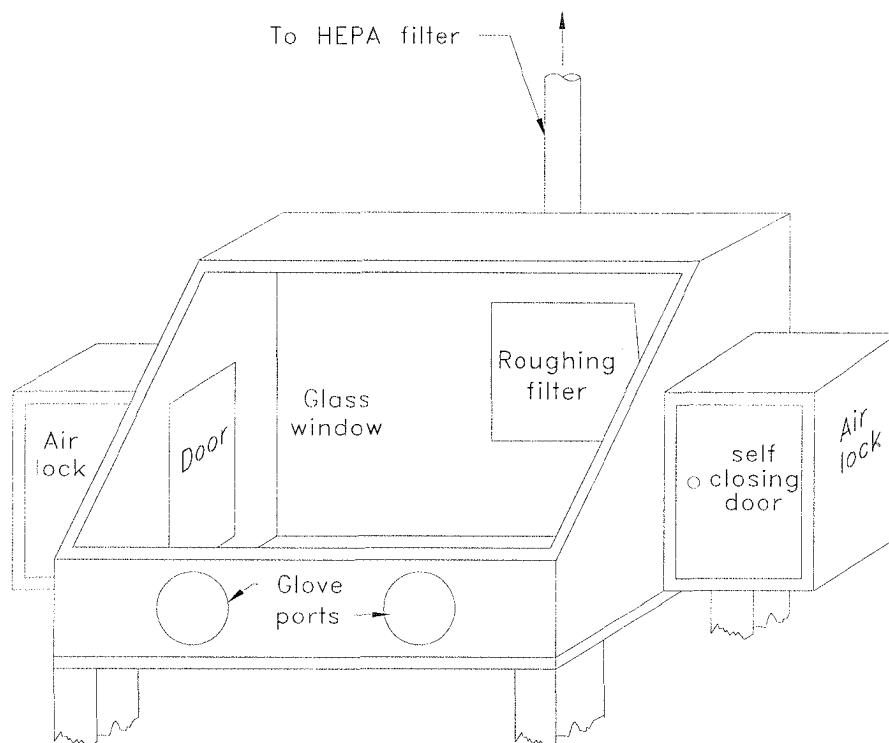
For product protection only, see VS-35-30 and VS-35-31.

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BIOLOGICAL SAFETY CABINET
CLASS II, TYPE B

DATE 02-91

FIGURE VS-35-11



$Q = 50 \text{ cfm/ft}^2$ of open door area and 0.25" wg SP
on a closed system.

$h_e = 0.50 \text{ VP}_d$

Duct velocity = 2000–4000 fpm

Filters : 1. Inlet air filters in doors.

2. Roughing filter at exhaust connection to hood.

3. HEPA filter

All facilities totally enclosed in hood. Exterior controls may be advisable.

Arm length rubber gloves are sealed to glove port rings.

Strippable plastic on interior and air cleaner on exhaust outlet may be
used to facilitate decontamination of the system.

Filter units may be installed in the doors to allow the air flow necessary
for burners etc.

For filters, see Chapter 4.

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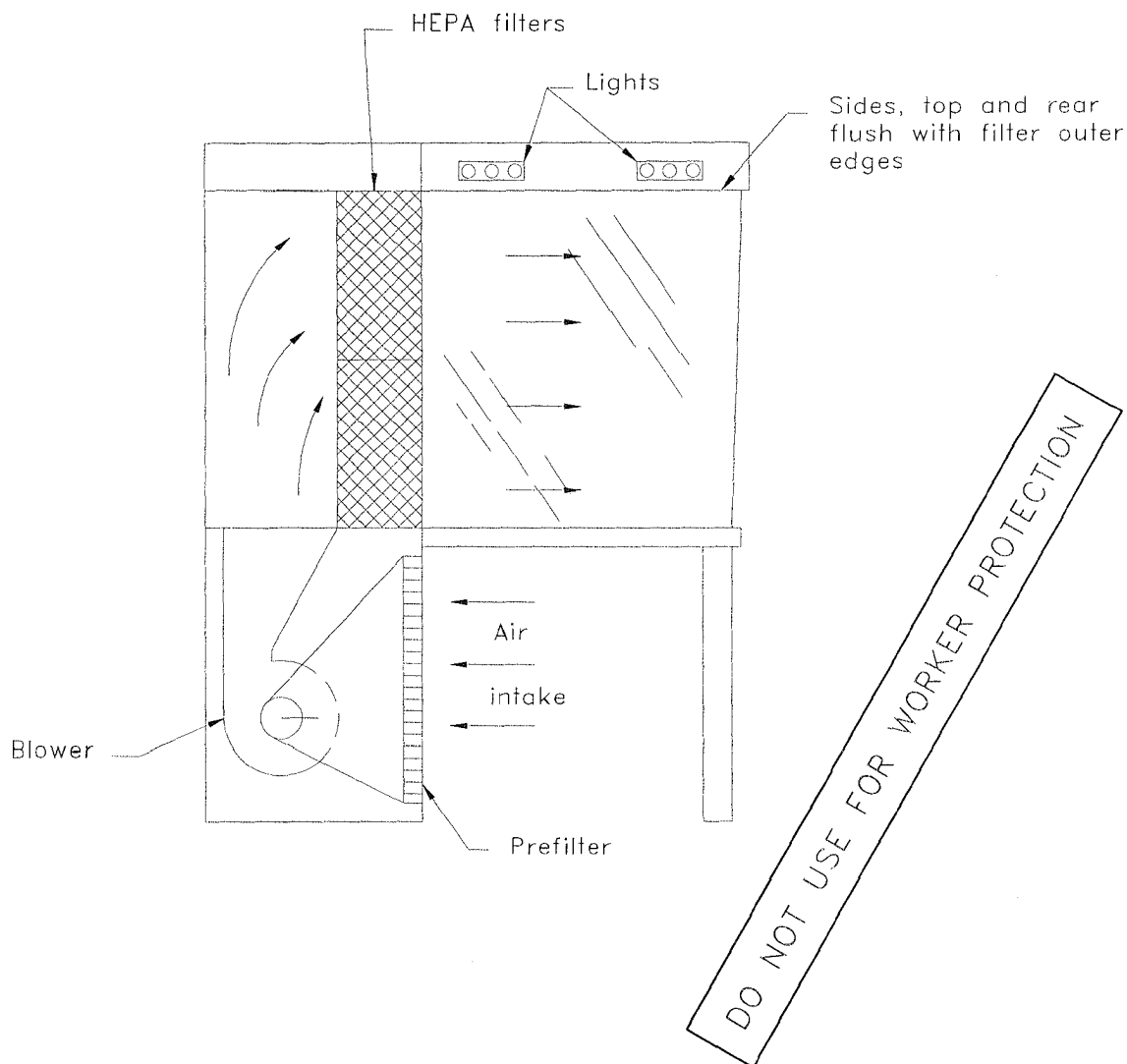
*DRY BOX OR GLOVE HOOD
FOR HIGH TOXICITY & RADIOACTIVE
MATERIALS*

DATE

02-91

FIGURE

VS-35-20



Face velocity = $90 \text{ fpm} \pm 20 \text{ fpm}$

Note : Total power input must be considered as part of air conditioning load.

This hood does not provide protection for the operator.

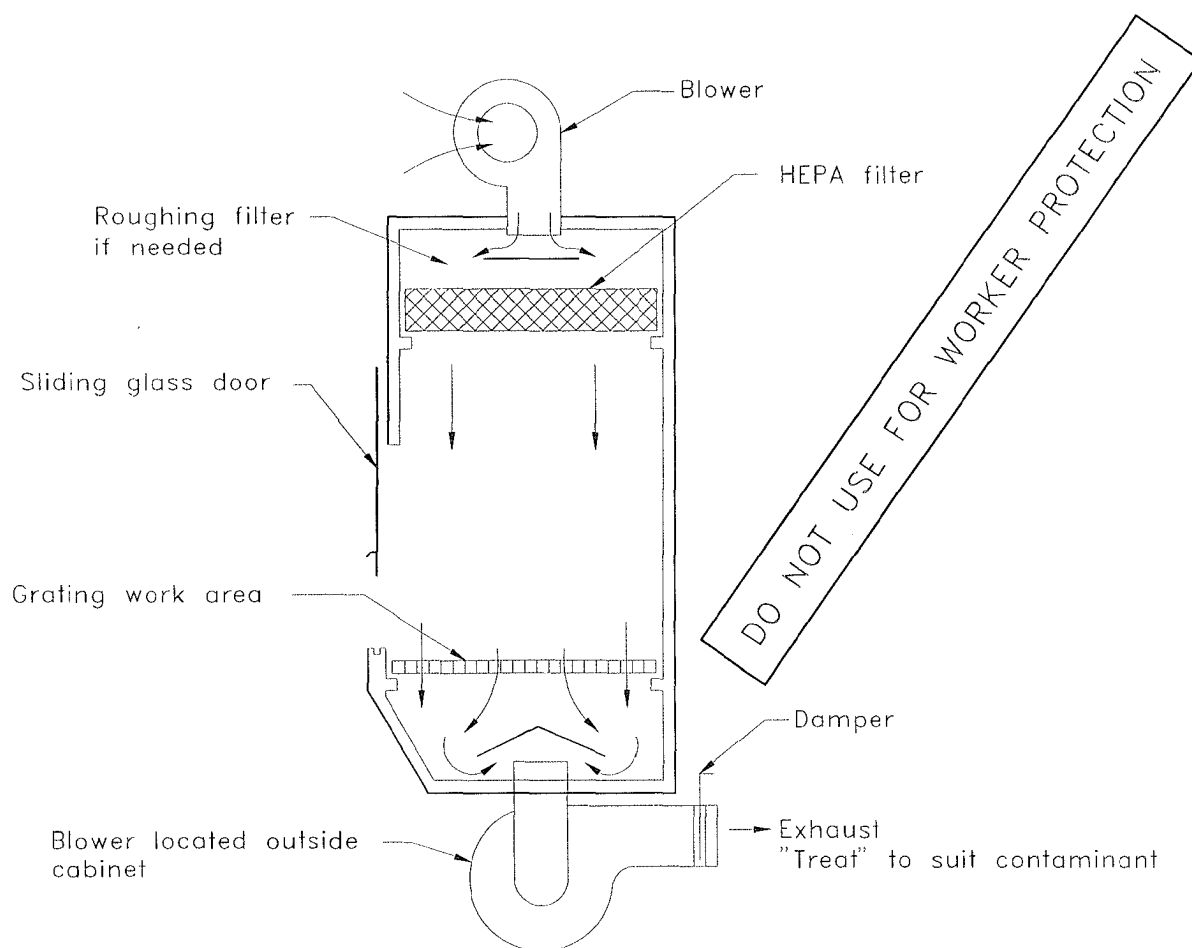
Reference 10.35.5

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HORIZONTAL LAMINAR FLOW
CLEAN BENCH
(PRODUCT PROTECTION ONLY)

DATE 02-91

FIGURE VS-35-30



Vertical velocity = 90 fpm with average minimum
uniformity ± 20 fpm
Duct velocity = 2000 - 4000 fpm to suit conditions

Clean station for control of air particles

Notes: Supply and exhaust should be maintained equal by
flow meter control techniques.

This hood does not provide protection for the operator.

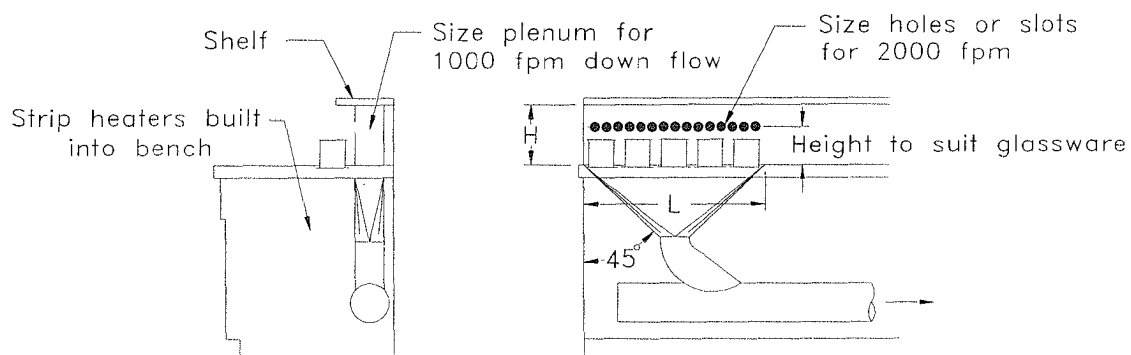
Do not use with toxic material.

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VERTICAL LAMINAR FLOW
CLEAN BENCH
(PRODUCT PROTECTION ONLY)

DATE 92-91

FIGURE VS-35-31

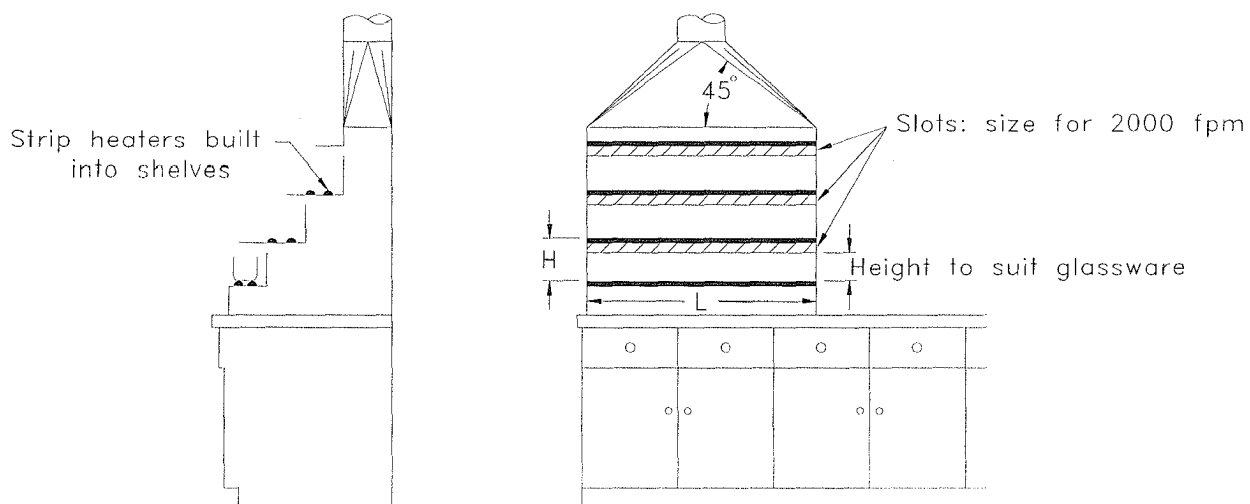


EVAPORATION BENCH

$Q = 20 \text{ cfm/foot of hood or } 50 \text{ HL}$

Minimum duct velocity = 2000 fpm

$$h_e = 1.78VP_s + 0.25VP_d$$



EVAPORATION HOOD

$Q = 20 \text{ cfm/foot of shelf or } 50 \text{ HL for each shelf}$

Minimum duct velocity = 2000 fpm

$$h_e = 1.78VP_s + 0.25VP_d$$

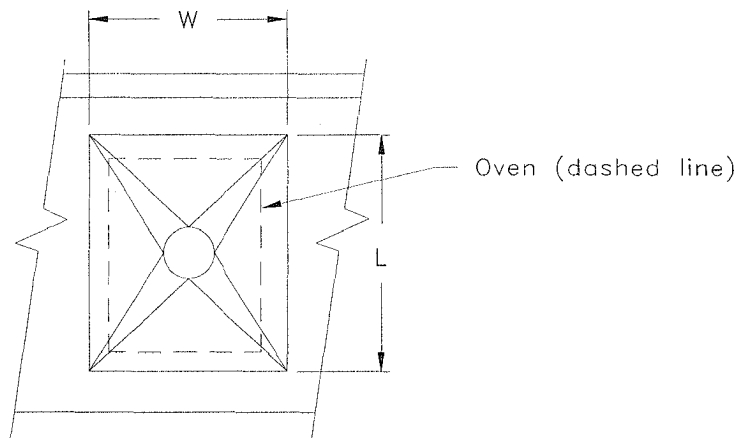
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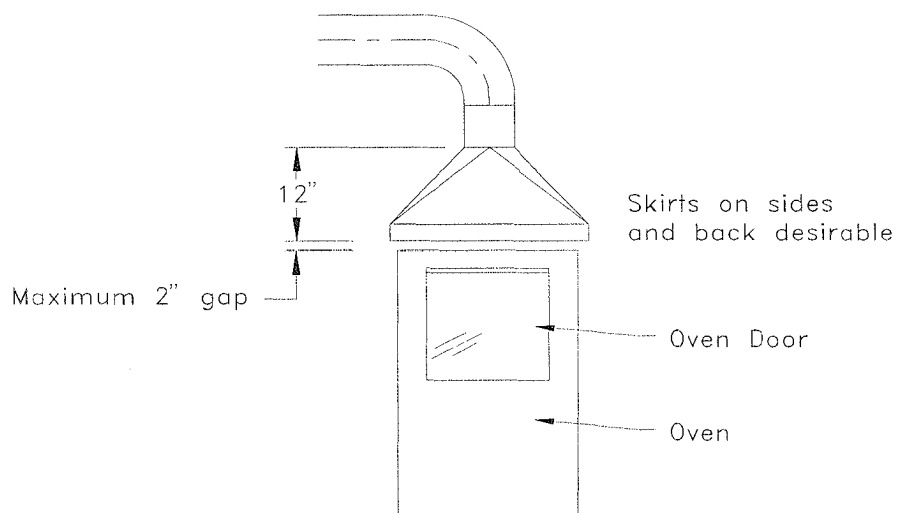
SPECIALIZED LABORATORY
HOOD DESIGNS

DATE 02-91

FIGURE VS-35-40



Top View



$$Q = 200 - 400 \text{ cfm}$$

$$\text{Minimum duct velocity } 1000 - 3000 \text{ fpm}$$

$$h_e = 0.25 \text{ VP}_d$$

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OVEN EXHAUST

DATE 02-91

FIGURE VS-35-41

10.40 LOW VOLUME-HIGH VELOCITY EXHAUST SYSTEMS

The low volume-high velocity (LVHV) exhaust system is a unique application of exhaust which uses small volumes of air at high velocities to control dust from portable hand tools and machining operations. Control is achieved by exhausting the air directly at the point of dust generation using close-fitting, custom-made hoods. Capture velocities are relatively high but the exhaust volume is low due to the small distance required. For flexibility, small diameter, light-weight plastic hoses are used with portable tools. This results in very high duct velocities but allows the application of local exhaust ventilation to portable tools which otherwise would require larger flow rates and large duct sizes when controlled by conventional exhaust methods. The resulting additional benefit is the reduction of replacement air requirements.

This technique has found a variety of applications although its use is not common. Rock drilling dust has been controlled by using hollow core drill steel with suitable exhaust holes in the drill bits. Air is exhausted either by a multi-stage turbine of the size generally used in industrial vacuum cleaners or, in the case of one manufacturer,^(10.40.1) by the exhaust air from the pneumatic tool which operates a Venturi to withdraw air from the drill. Some applications use flexible connections to a central vacuum system to aid in the control of graphite dust at conventional machining operations. One- to two-inch diameter flexible hose was used with simple exhaust hoods mounted directly at the cutting tool. In a similar application for the machining of beryllium,^(10.40.2) a central vacuum system utilizing 1.5-inch I.D. flexible hoses was employed. The exhaust hoods were made of lucite or transparent material and were tailor-made to surround the cutting tools and much of the work. Exhaust flow rates vary from 120–150 cfm with inlet velocities of 11,000–14,000 fpm. In another application,^(10.40.3) a portable orbital sanding machine has been fitted with a small exhaust duct surrounding the edge of the plate. A fitting has been provided to connect this to the flexible hose of a standard domestic vacuum cleaner.

VS-40-01 to VS-40-06 illustrate a custom-made line of exhaust hoods available.^(10.40.4) The required air flow rates range from 60 cfm for pneumatic chisels to 380 cfm for swing grinders. Due to the high entering velocities involved, static pressures are in the range of 7" to 14" of mercury (95 to 290 "wg). This high pressure is necessary to create the high capture velocities at the source to control the dust. However, there are disadvantages associated with high velocities: 1) small metal parts can be sucked into the hood; 2) coolants may

be disturbed; and 3) very high noise levels may be produced.

10.40.1 Design—Calculations: With the exception of the proprietary system mentioned which can be purchased as a "package," the design calculations for these systems are largely empirical and little performance data are available for the user. In normal ventilation practice, air is considered to be incompressible since static pressures vary only slightly from atmospheric pressure. However, in LVHV systems the extreme pressures required introduce problems of air density, compressibility, and viscosity which are not easily solved. Also, pressure drop data for small diameter pipe, especially flexible tubing, is not commonly available. For practical purposes, the turbine exhaustor should be selected for the maximum simultaneous exhaust flow rate required. Resistance in the pipe should be kept as low as possible; flexible tubing of less than 1- to 1.5-inch diameter should be limited to 10 feet or less. In most applications, this is not a severe problem.

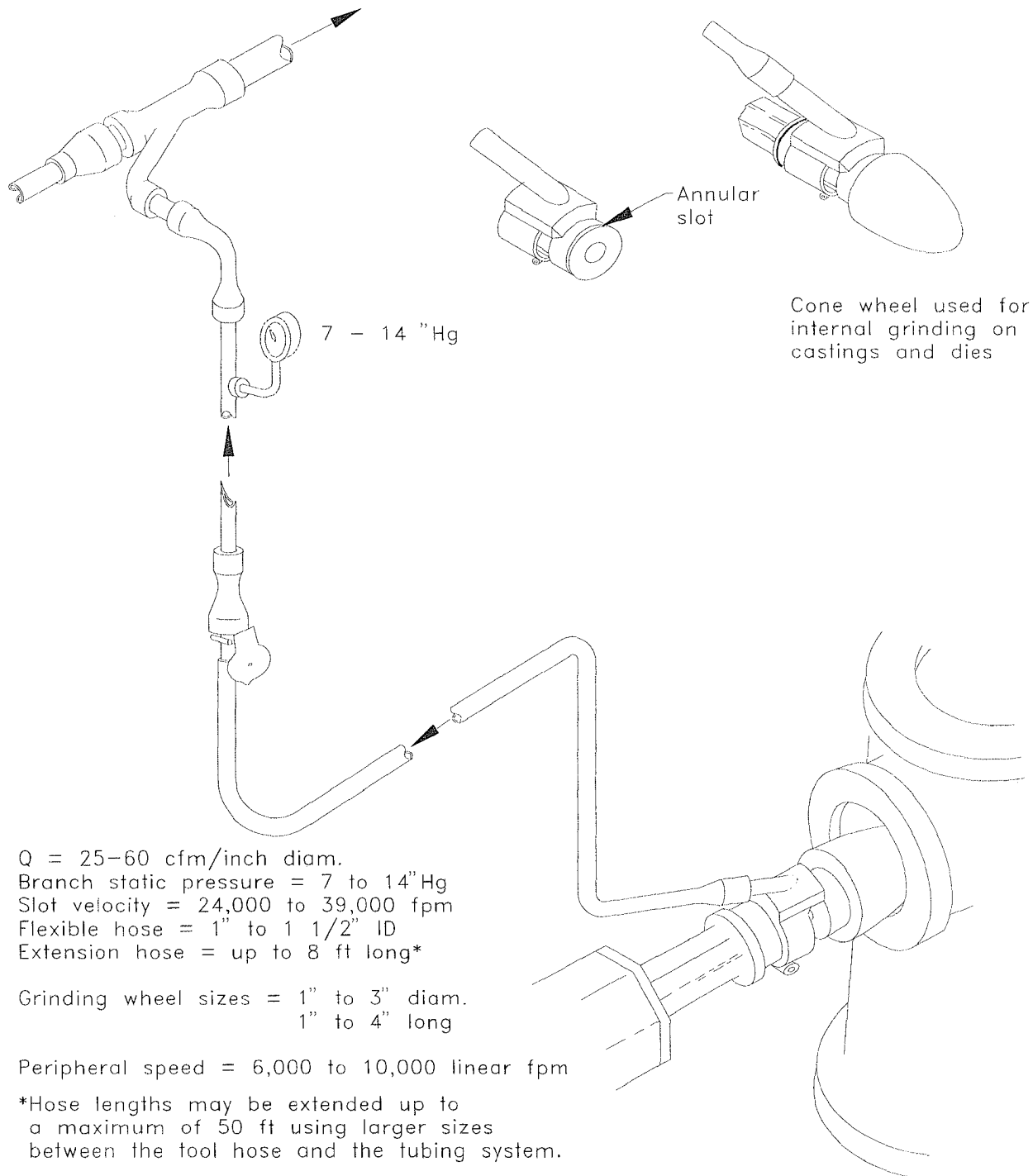
The main consideration in piping for such systems is to provide smooth internal configuration so as to reduce pressure loss at the high velocities involved and to minimize abrasion. Ordinary pipe with threaded fittings is to be avoided because the lip of the pipe or male fitting, being of smaller diameter than the female thread, presents a discontinuity which increases pressure loss and may be a point of rapid abrasion.

For dust exhaust systems, a good dust collector and primary separator should be mounted ahead of the exhaustor to minimize erosion of the precision blades and subsequent loss in performance. Final balance of the system can be achieved by varying the length and diameters of the small flexible hoses.

It must be emphasized that although data are empirical, LVHV systems require the same careful design as the more conventional ones. Abrupt changes of direction, expansions, and contractions must be avoided, and care must always be taken to minimize pressure losses.

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- 10.40.3 Master Power, Inc., Westminster, MD.
- 10.40.4 Hoffman Air and Filtration Div., Clarkson Industries, Inc., New York.



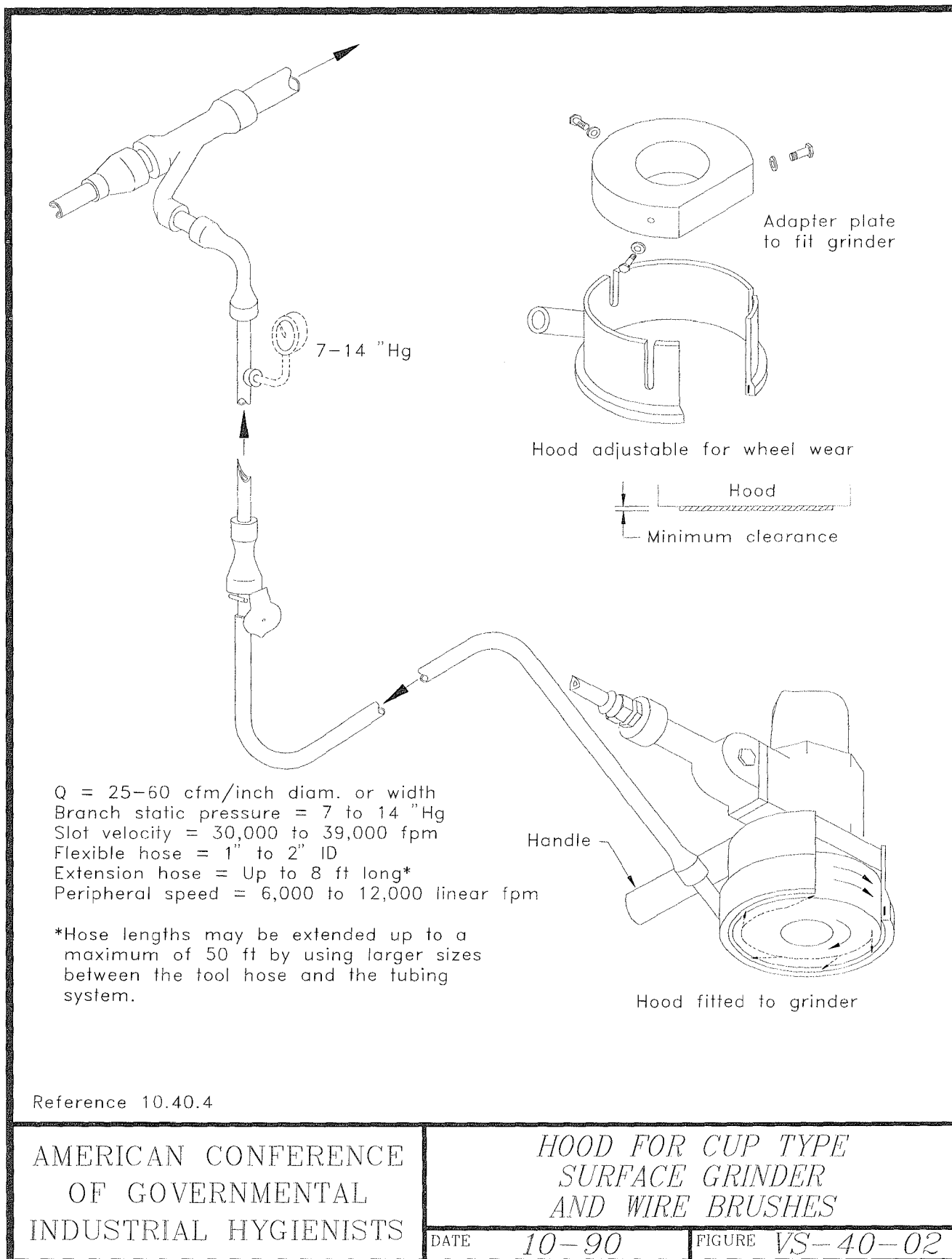
Reference 10.40.4

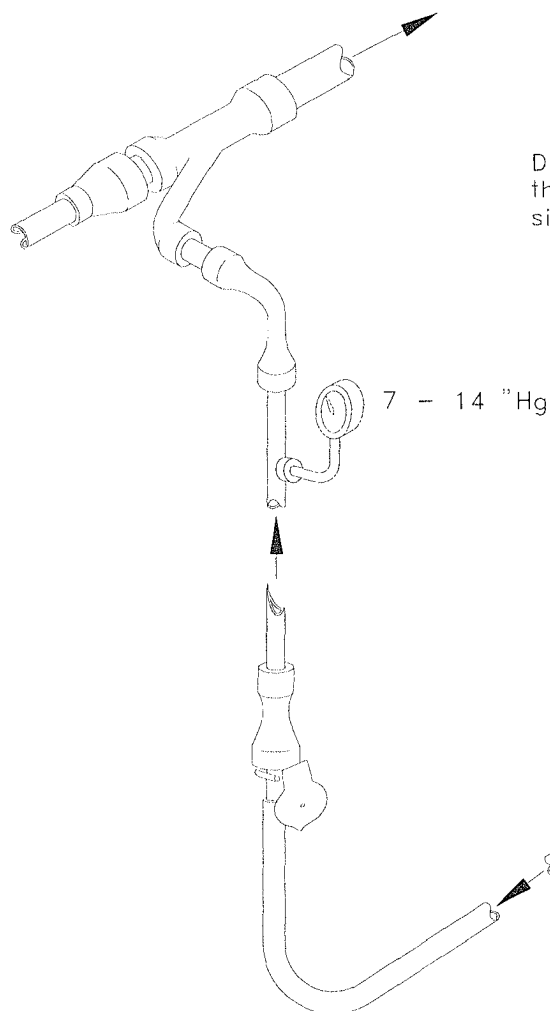
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EXTRACTOR HEAD FOR CONE
 WHEELS AND MOUNTED POINTS

DATE 10-90

FIGURE VS-40-01





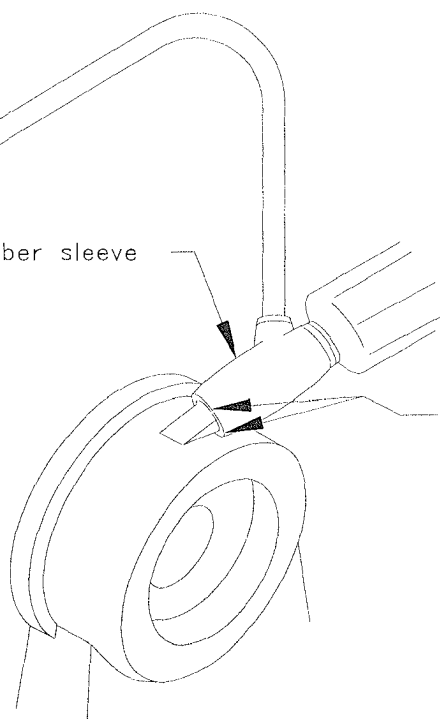
Dust is extracted through slots built into the rubber sleeve; slots should be on both sides of the cutting edge of the chisel

Q = 25-60 cfm/inch diam.
 Branch static pressure = 7 to 14 "Hg
 Slot velocity = 24,000 to 39,000 fpm
 Flexible hose = 1" to 1 1/2" ID
 Extension hose = Up to 8 ft long*
 Chisel sizes = 13/16" octagonal
 7/8" octagonal
 7/8" hexagonal

*Hose lengths may be extended up to a maximum of 50 ft by using larger sizes between the tool and the tubing system.

Rubber sleeve

Slots



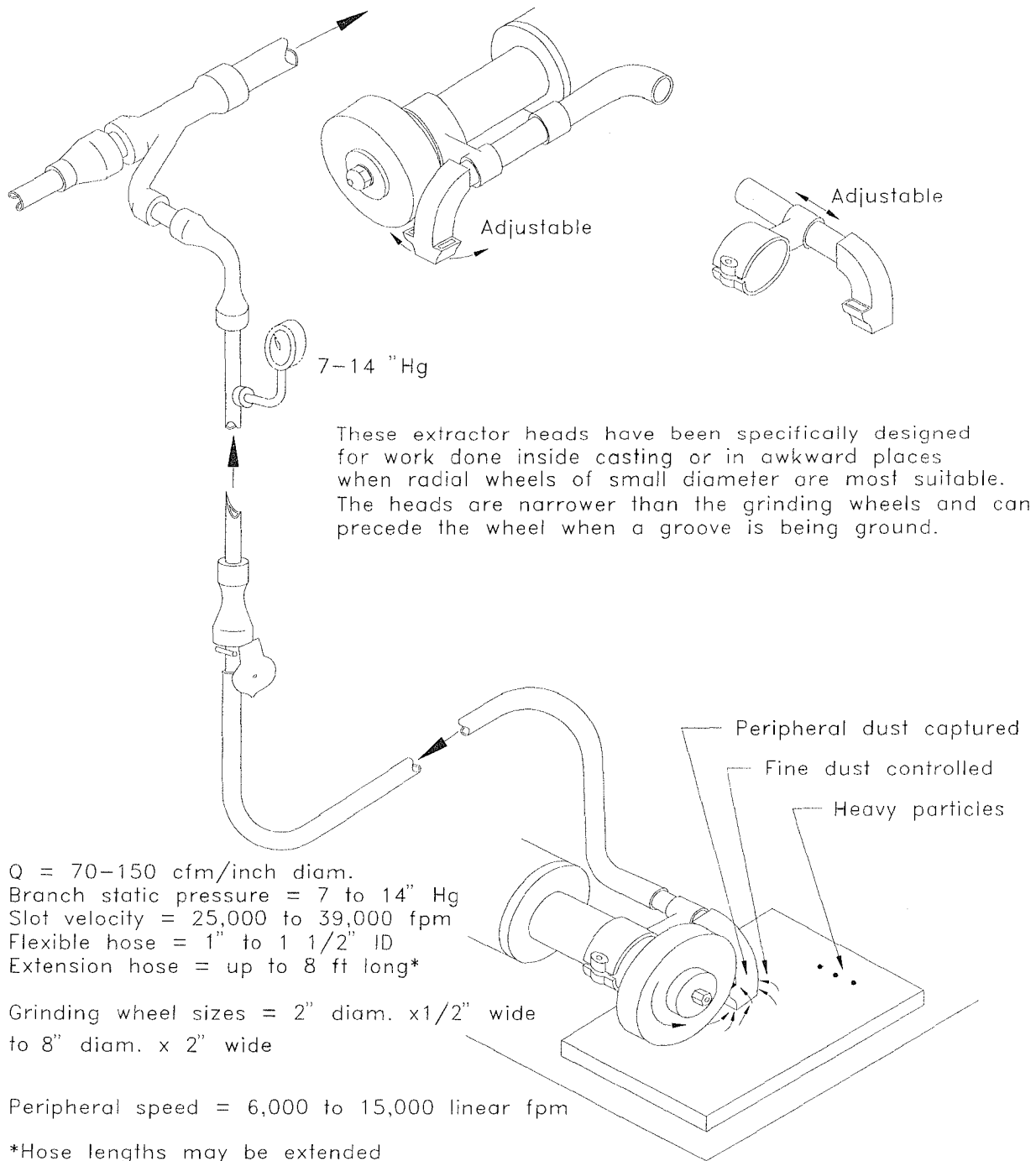
Reference 10.40.4

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PNEUMATIC CHISEL SLEEVE

DATE 10-90

FIGURE VS-40-03



Q = 70-150 cfm/inch diam.
 Branch static pressure = 7 to 14" Hg
 Slot velocity = 25,000 to 39,000 fpm
 Flexible hose = 1" to 1 1/2" ID
 Extension hose = up to 8 ft long*

Grinding wheel sizes = 2" diam. x 1/2" wide
 to 8" diam. x 2" wide

Peripheral speed = 6,000 to 15,000 linear fpm

*Hose lengths may be extended
 up to a maximum of 50 ft by
 using larger sizes between the
 tool hose and the tubing system.

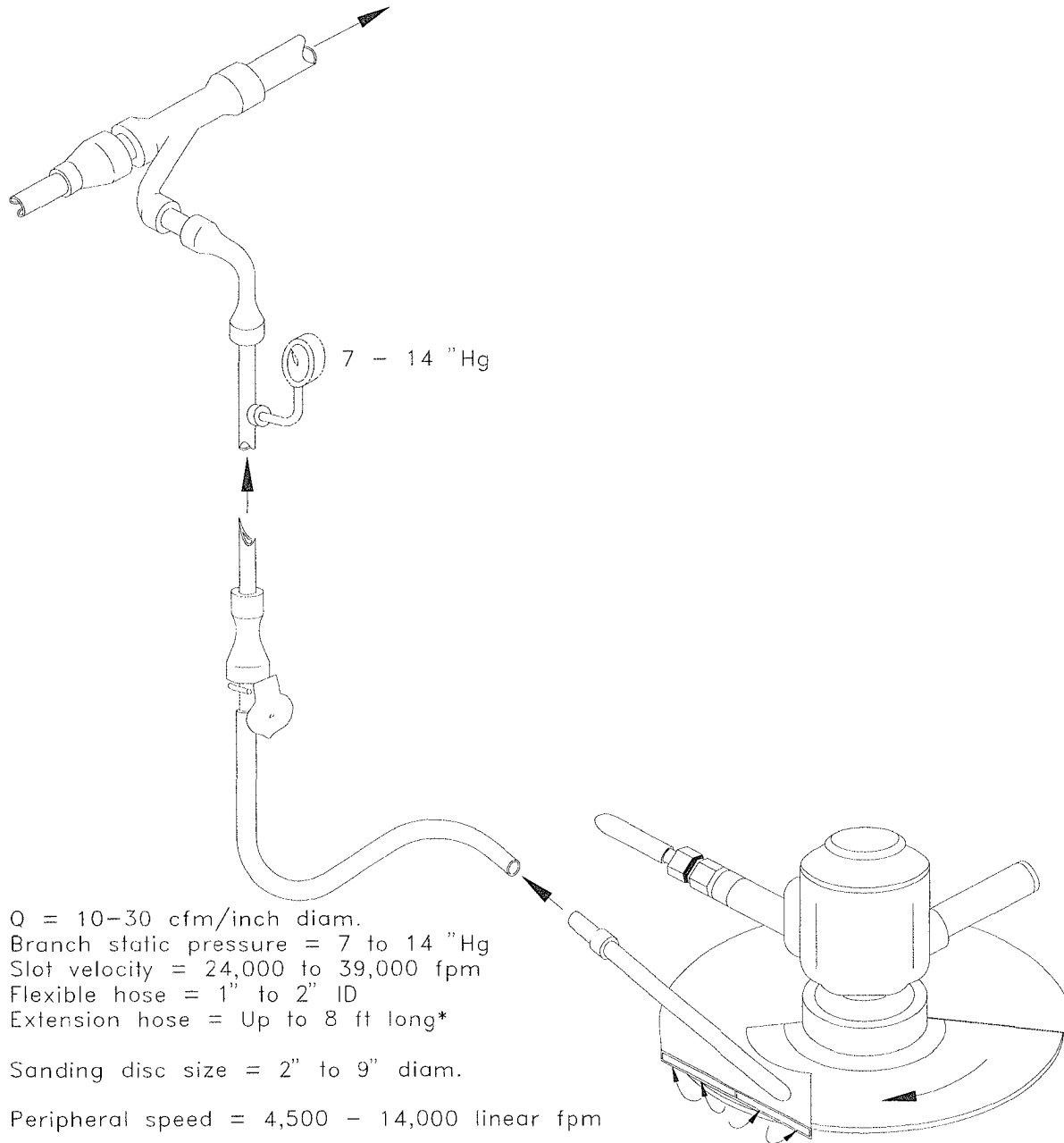
Reference 10.40.4

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EXTRACTOR HEAD FOR SMALL
 RADIAL GRINDERS

DATE 10-90

FIGURE VS-40-04



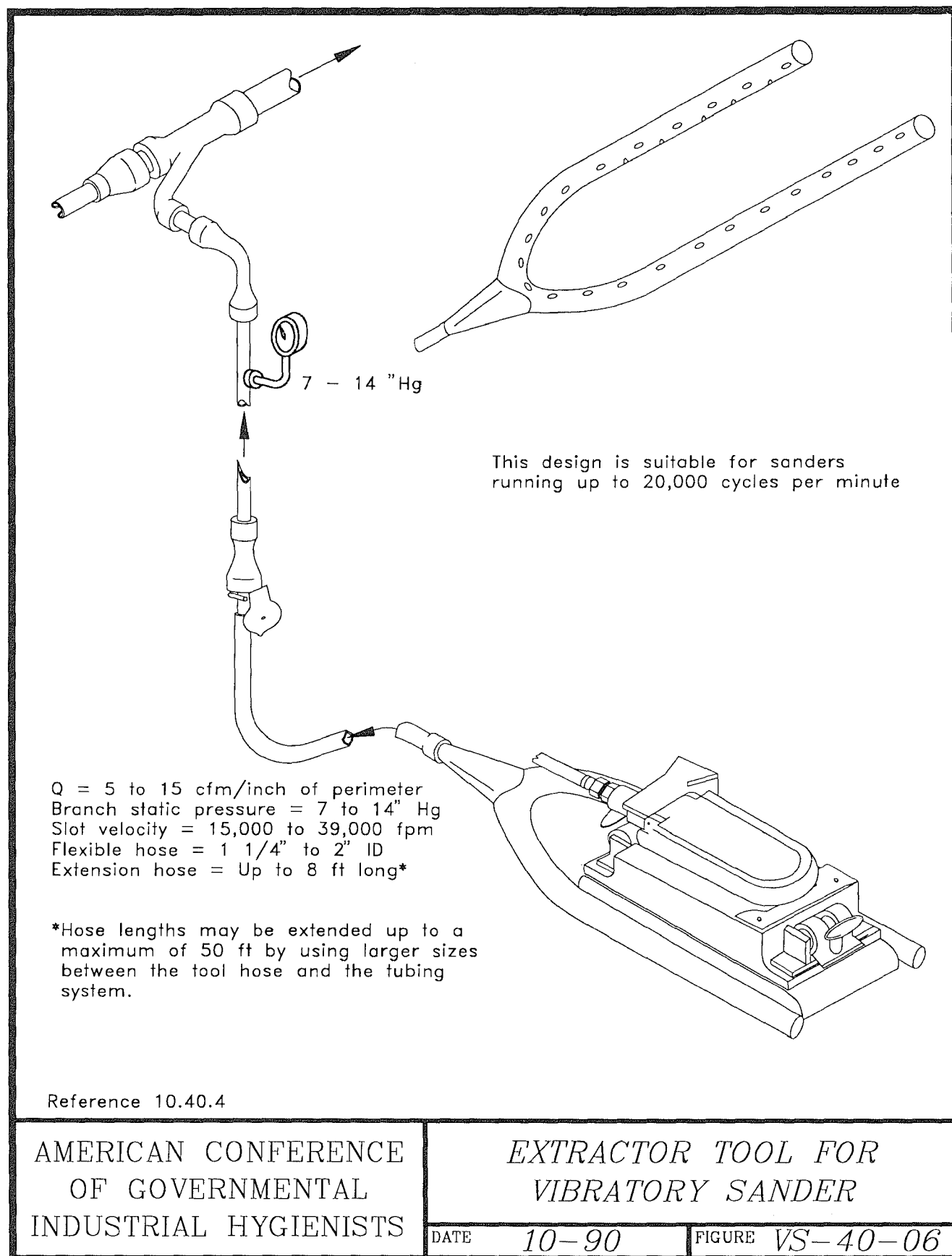
Reference 10.40.4

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EXTRACTOR HOOD FOR
 DISC SANDER

DATE 10-90

FIGURE VS-40-05

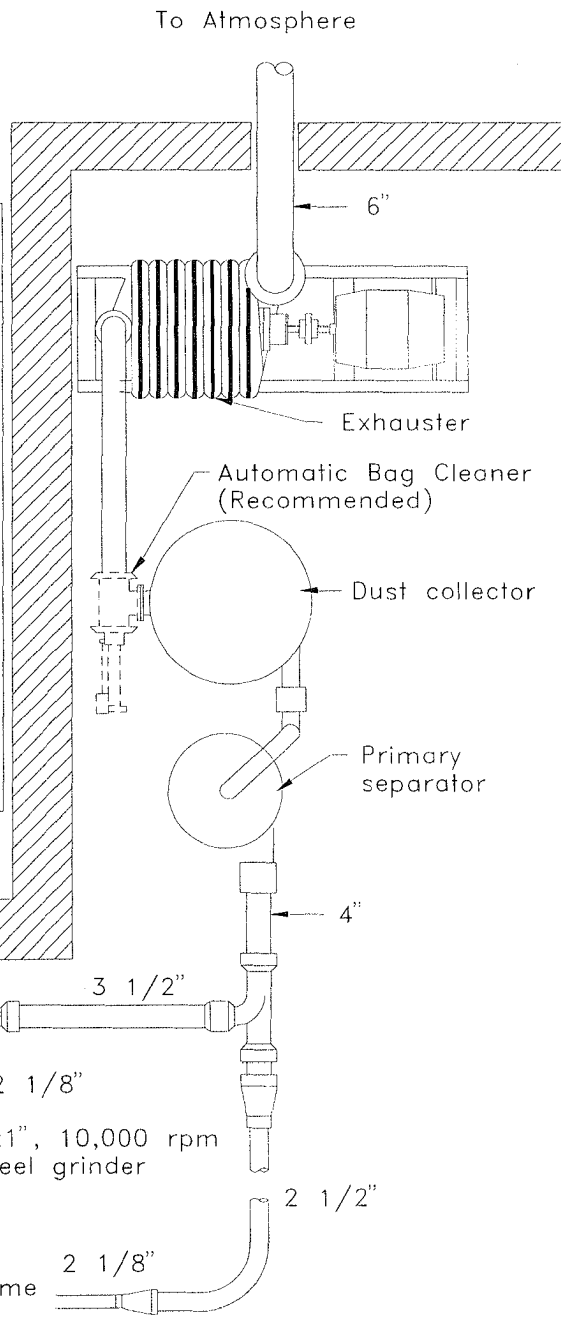


System Notes

Bell and socket, smooth-flow type tubing and fittings should be used throughout the system.

When system is used for vacuum cleaning of abrasive materials, Schedule No. 40 pipe and cast iron drainage fittings, or heavier, should be used in place of tubing.

	cfm	I.D. Plastic Hose Size (inches)
Disk sanders 3-9 inch diam.	60-175	1-1.5
Vibratory pad sander - 4"x9"	100	1.25
Router, 1/8"-1"	80-100	1-1.25
Belt sander 3"-4000 fpm	70	1
Pneumatic chisel	60	1
Radial wheel grinder	70	1
Surface die grinder, 1/4"	60	1
Cone wheel grinder	90	1.25
Cup stone grinder, 4"	100	1.25
Cup type brush, 6"	150	1.5
Radial wire brush, 6"	90	1.25
Hand wire brush 3" x 7"	60	1
Rip out knife	175	1.5
Rip out cast cutter	150	1.5
Saber saw	120	1.5
Swing frame grinder 2" x 18"	380	2.5
Saw abrasive 3"	100	1.25



Typical Layout

Reference 10.40.4

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TYPICAL SYSTEM
LOW VOLUME-HIGH VELOCITY

DATE 10-90

FIGURE VS-40-20

10.45 MACHINING

The primary purpose of metal cutting machines is to finish rough parts formed by other processes to specific dimensions. Finishing and shaping may be accomplished by a variety of cutting tools such as saws, broaches, and chisel-shaped tool bits held in fixtures with fixed or movable drives. Cutting is accomplished by creating relative motion between the tool edge and the material blank. Chips of varying sizes are produced; chip size depends on the material being cut, feed rate of the tool, and relative speed or feed between the tool and the metal being shaped.

Non-traditional methods of metal cutting and shaping include electrochemical, electrodischarge, wire electrodischarge, and laser beam machining. With the exception of the laser beam, each of the processes utilizes a circulating oil or water-base dielectric to facilitate molecular erosion as well as to remove process heat and particulate debris. The rate of metal removal is controlled carefully by regulating the flow of electrical current between the shaped anode or wire and the workpiece. The laser beam is used in a dry environment and metal cutting is accomplished by vaporizing the workpiece along the cutting edge with a focused beam of high energy light. The process is flexible and a variety of metallic and non-metallic materials can be shaped by this means.^(10.45.1)

It is estimated that up to 97% of the work involved in conventional metal cutting results in heat. The rate of heat removal must be controlled carefully in order to protect both the cutting tool and the metallurgy of the work being cut. Where convection or radiant cooling is insufficient, a cutting fluid can be used to reduce friction, carry away generated heat, and, more commonly, flush away metal chips produced by the cutting process. Cutting fluids include straight-chained and synthetic mineral oils as well as soluble oil emulsions in water. A variety of water soluble lubri-coolants (1–5% mixture of lubricants, emulsifiers, rust inhibitors, and other chemicals in water) are used commonly, particularly for high speed metal working machines. In some applications the lubri-coolant mixture is applied as a mist by using a small volume of liquid in a high velocity air stream. In the more usual situation, liquid is applied by flooding the tool in the cutting zone to flush away cutting debris. The latter type system requires a low pressure pump with valves; filters; settling chamber to separate the fluid from the chips; and a reservoir which permits recirculation. Where liquids cannot be used, low temperature nitrogen or carbon dioxide gas can be used as a cooling media for both the tool and the cutting surface as well as a means of dispersing particulate debris.

The hazards created by skin exposure to the water lubri-coolant mixtures, particulates and oil mist/vapor produced in the transfer of heat is best handled with engineering controls—primarily ventilation. An additional health concern is the fact that soluble oil emulsions provide a breeding ground for bacteria and, therefore, it is common practice to add biocides to prevent odor generation and decomposition of the oil mixture. Biocides and

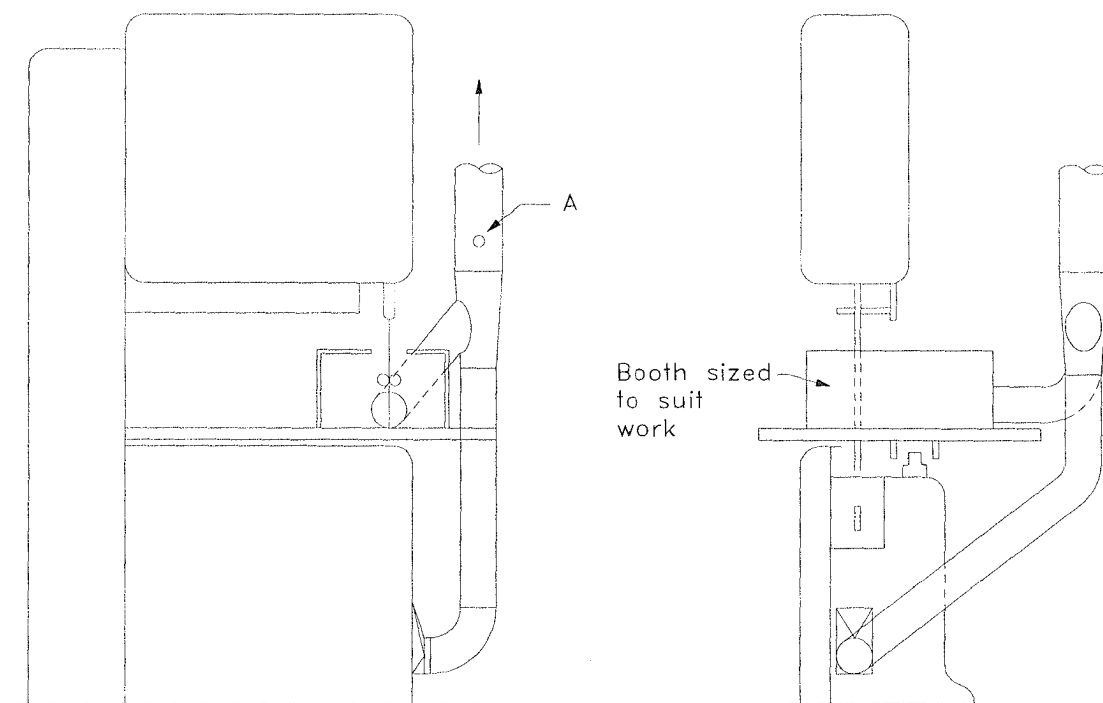
other additives may be primary skin irritants or cause hypersensitive dermatitis. It is for these reasons that mist, vapors, and particulates must be controlled adequately.^(10.45.2)

Mist and vapors from machining operations can be controlled by a combination of machine enclosure and local exhaust ventilation. Exhaust hoods and enclosures should be designed so the machine can be serviced easily and the operation observed when required. Hood sides should act as splash guards since an indraft of air will not stop liquid directly thrown from rotating parts. All components should be robust and rigidly supported. To facilitate maintenance, service and tool adjustment, portions of the hood enclosure which are not permanently fixed should be designed for easy removal. Thought should be given also to the use of sliding, hinged, or bellows-connected panels in locations where frequent access is required. All windowed openings must be shatter-proof with appropriate internal lighting. All non-fixed panels should be designed with overlapping, drip-proof edges. The use of gaskets or seals on abutting panels is not recommended. Ventilation rates vary; however, a minimum of 100 fpm indraft usually is required to prevent vapor and mist from exiting the enclosure. A typical machine enclosure will require a volumetric flow rate of from 400–500 cfm minute. Additional air may be required to control heat generated within the enclosure as well as to maintain adequate vision. Where coolant flumes are used for chip transport, additional exhaust ventilation is required to control air entrainment. Baffles above the liquid level are beneficial and flumes should be enclosed to the extent possible.

Local ventilation control is the preferred method—particularly in machine environments which are temperature controlled with refrigerated air conditioning systems. In more open workrooms, the use of dilution ventilation may be adequate to control air contaminants. For further information on dilution ventilation, see Chapter 2 of this manual.

REFERENCES

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- 10.45.2 O'Brien, Dennis; Frede, John C.: Guidelines for the Control of Exposure to Metal Working Fluids. National Institute for Occupational Safety and Health (February 1978).
- 10.45.3 Schulte, H.F.; Hyatt, E.C.; Smith, Jr., F.S.: Exhaust Ventilation for Machine Tools Used on Materials of High Toxicity. American Medical Association Archives of Industrial Hygiene and Occupational Medicine, Vol. 5, No. 21 (January 1952).
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Q at booth = 225 cfm/ft^2 open area
 Q at bottom = 350 cfm

Minimum duct velocity = 4000 fpm

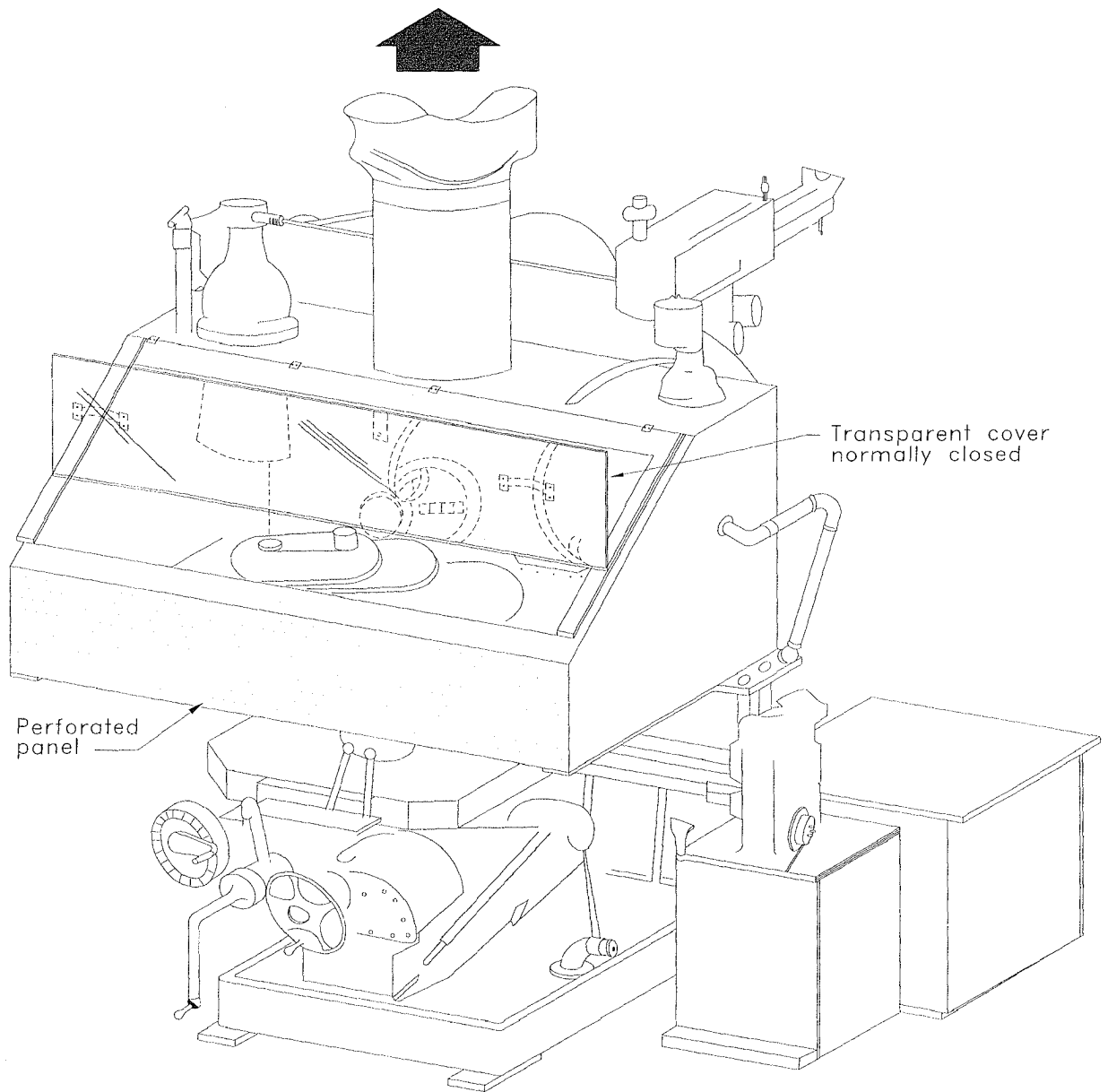
$h_e = 1.75 \text{ VP}_d$, at point A

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METAL CUTTING BANDSAW

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FIGURE VS-45-01



$Q=300 \text{ cfm/ft}^2$ of open area
Minimum duct velocity = 3500 fpm
 $h_e = 0.35 \text{ VP}_d$

Reference 10.45.4

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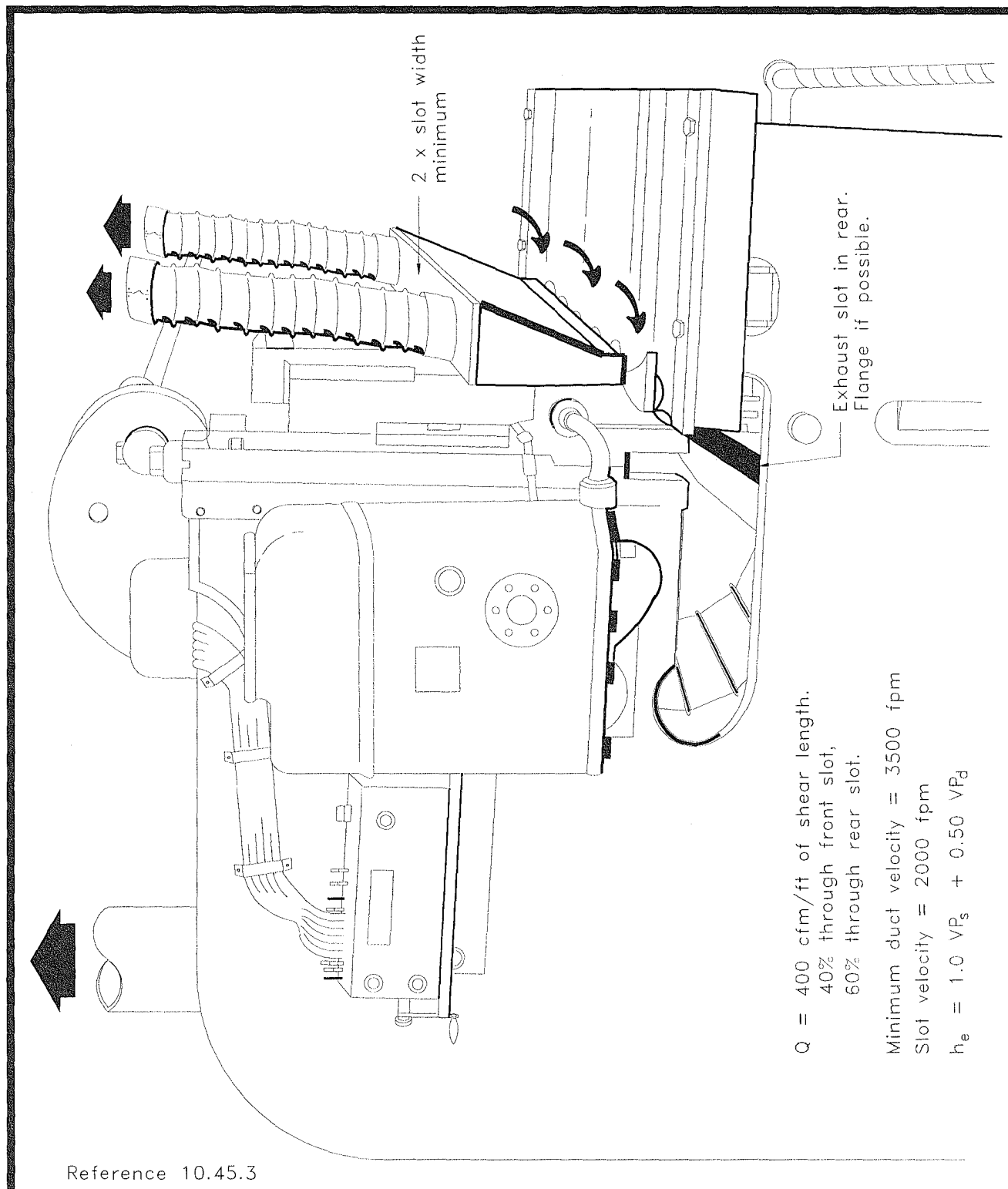
HIGH TOXICITY MATERIALS
MILLING MACHINE HOOD

DATE

11-90

FIGURE

VS-45-02



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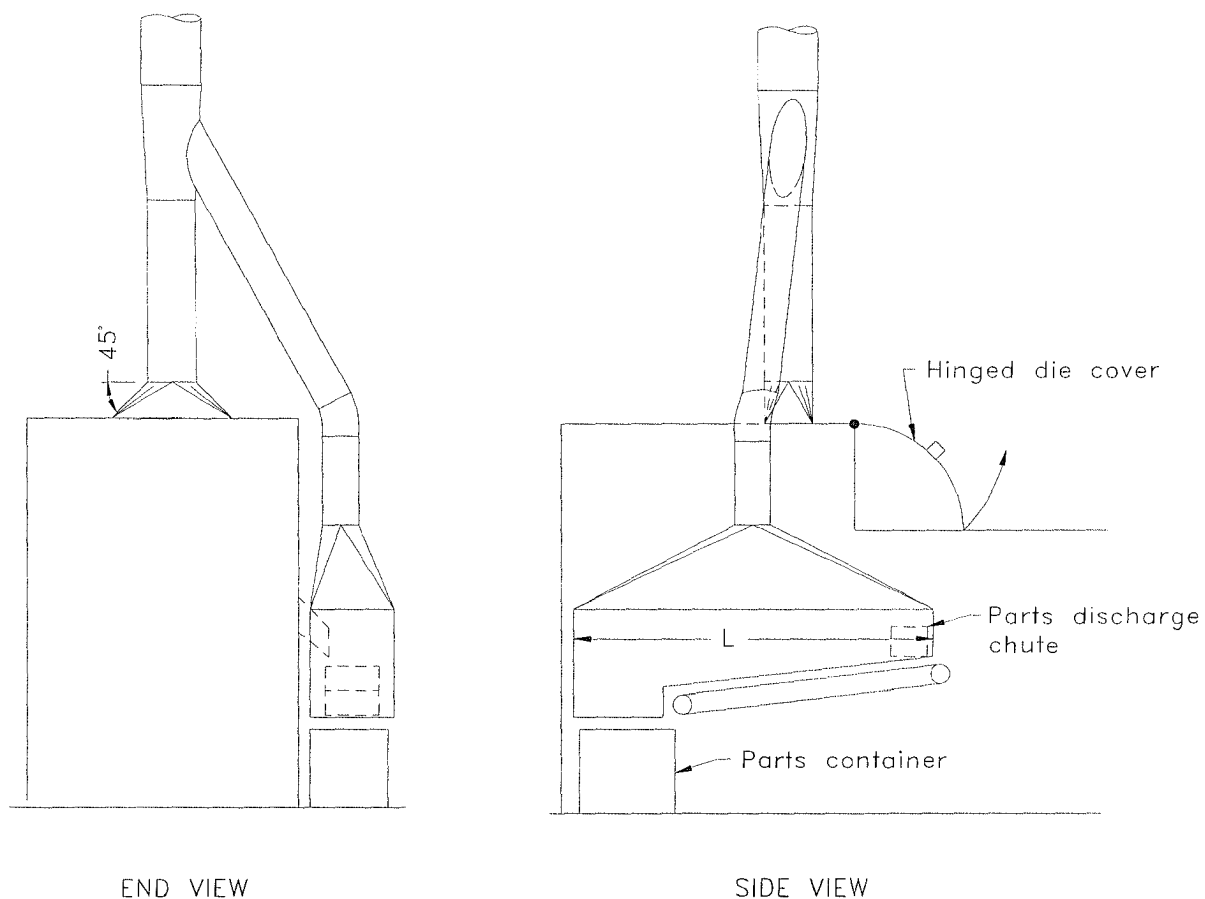
METAL SHEARS
HIGH TOXICITY MATERIALS

DATE

11-90

FIGURE

VS-45-03



Cold header :

$$Q = 750 \text{ cfm/ft}^2 \text{ of die opening}$$

$$\text{Minimum duct velocity} = 3500 \text{ fpm}$$

$$h_e = 1.0 VP_s + 0.25 VP_d$$

Parts discharge and container :

$$Q = 100 \text{ cfm/ft of hood length}$$

$$\text{Minimum duct velocity} = 3500 \text{ fpm}$$

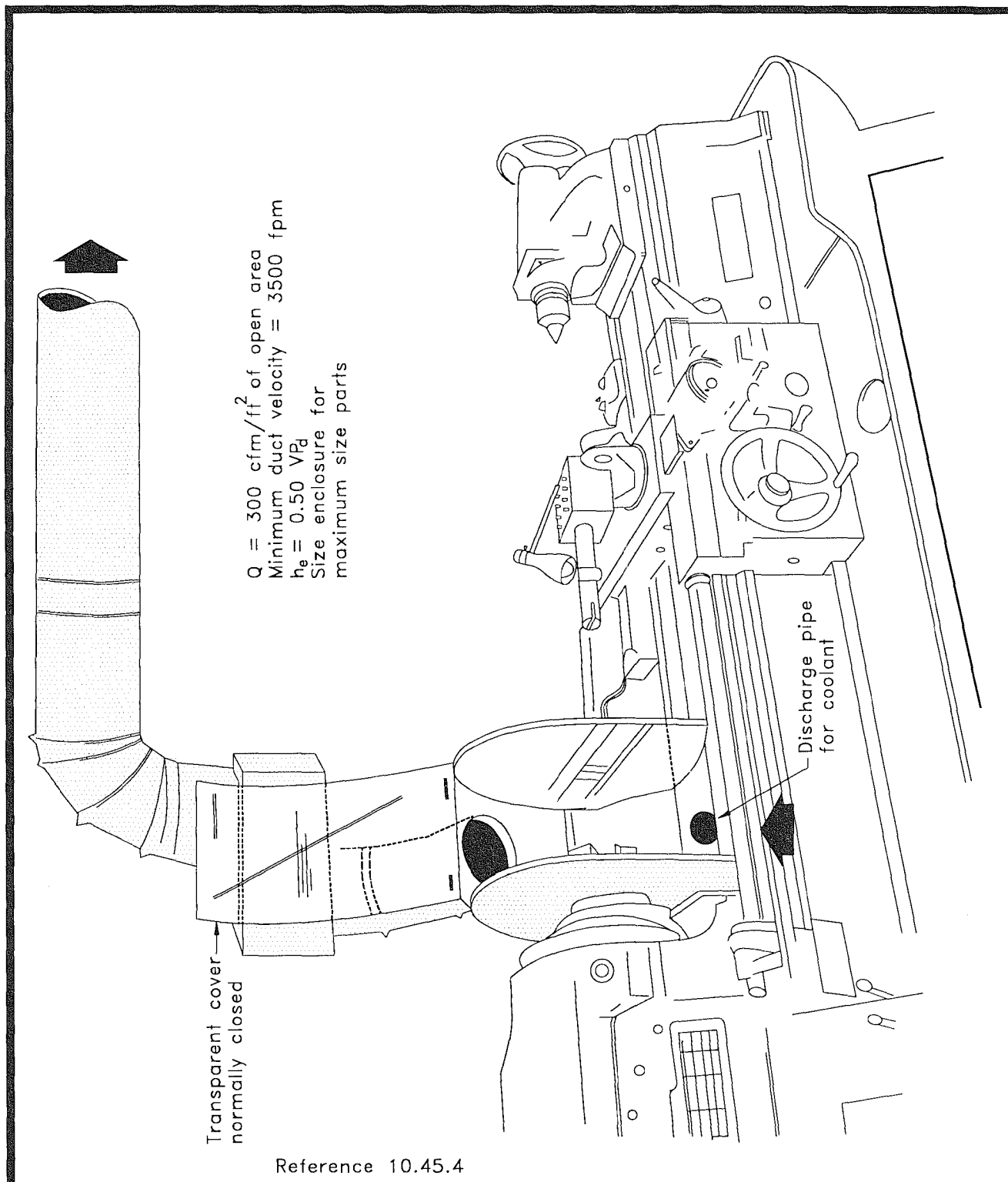
$$h_e = 0.25 VP_d$$

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COLD HEADING MACHINE
VENTILATION

DATE 11-90

FIGURE VS-45-04



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LATHE HOOD

DATE 11-90

FIGURE VS-45-05

10.50 MATERIAL TRANSPORT

Ventilation of material transport systems generally requires the use of an exhausted enclosure because of the motion and quantity of material involved. If the enclosure were perfectly air tight there would be no need for exhaust. However, there usually are cracks and other leak points in addition to the openings necessary for personnel and material access.

For enclosures where there is little motion and low material quantity, the exhaust rate is the product of the total openings in square feet and some velocity between 50 and 200 fpm. However, in some cases the inward flow of material and entrained air can overwhelm the exhaust flow rate calculated on the basis of enclosure openings. In such cases the material flow rate, the dustiness of the material, and the height of fall in transferring from one surface to another must be considered in the system design.^(10.50.1, 10.50.2) Other design factors include:

1. The rate of air induction into the space.
2. The location of cracks or other openings in relation to the "splash" or agitation of material during transfer.
3. The need to avoid excessive product withdrawal.
4. Adequate air flow for dilution of interior concentrations for visibility or safety from explosions.

10.50.1 Bucket Elevators: Air motion caused by the bucket moving within the elevator is not significant. The motion of buckets in one direction is offset by the opposite flow. Consequently, an exhaust rate of 100 cfm/ft² of elevator cross-section is adequate for most elevator applications (see VS-50-01 for details). Additional ventilation is required as materials enter and leave the elevator (see VS-50-10, VS-50-20 and VS-50-21). Handling hot material often causes significant thermal buoyancy which requires increased exhaust ventilation to overcome this challenge.

10.50.2 Conveyors: Dust from the operation of belt conveyors originates mainly at the tail pulley where material is received and at the head pulley where material is discharged. The exhaust requirement at the head pulley is generally small because air is induced downward and away from this transfer point. An exhaust rate of 150–200 cfm/ft² of opening often is adequate.

At the tail pulley, the exhaust requirements are determined by the amount of air induced by the delivery chute. An exhaust of 350 cfm/ft² of belt width often is adequate where the

material does not fall more than 3 feet. The exhaust point should be located at least twice the belt width away from the point where the material hits the belt. Where the material falls more than 3 feet, additional exhaust is required (see VS-50-20 for details). Note that very dry or dusty material may require flowrates 1.5 to 2.0 times these values.

Belt conveyors should be covered and exhausted at 30 foot intervals at a rate of 350 cfm/ft of belt width. Vibrating feeders should be exhausted at a rate of 500 cfm/ft of feeder width. Rubber or canvas flexible seals should be provided from the feeder sides and end to the hopper sides and end.

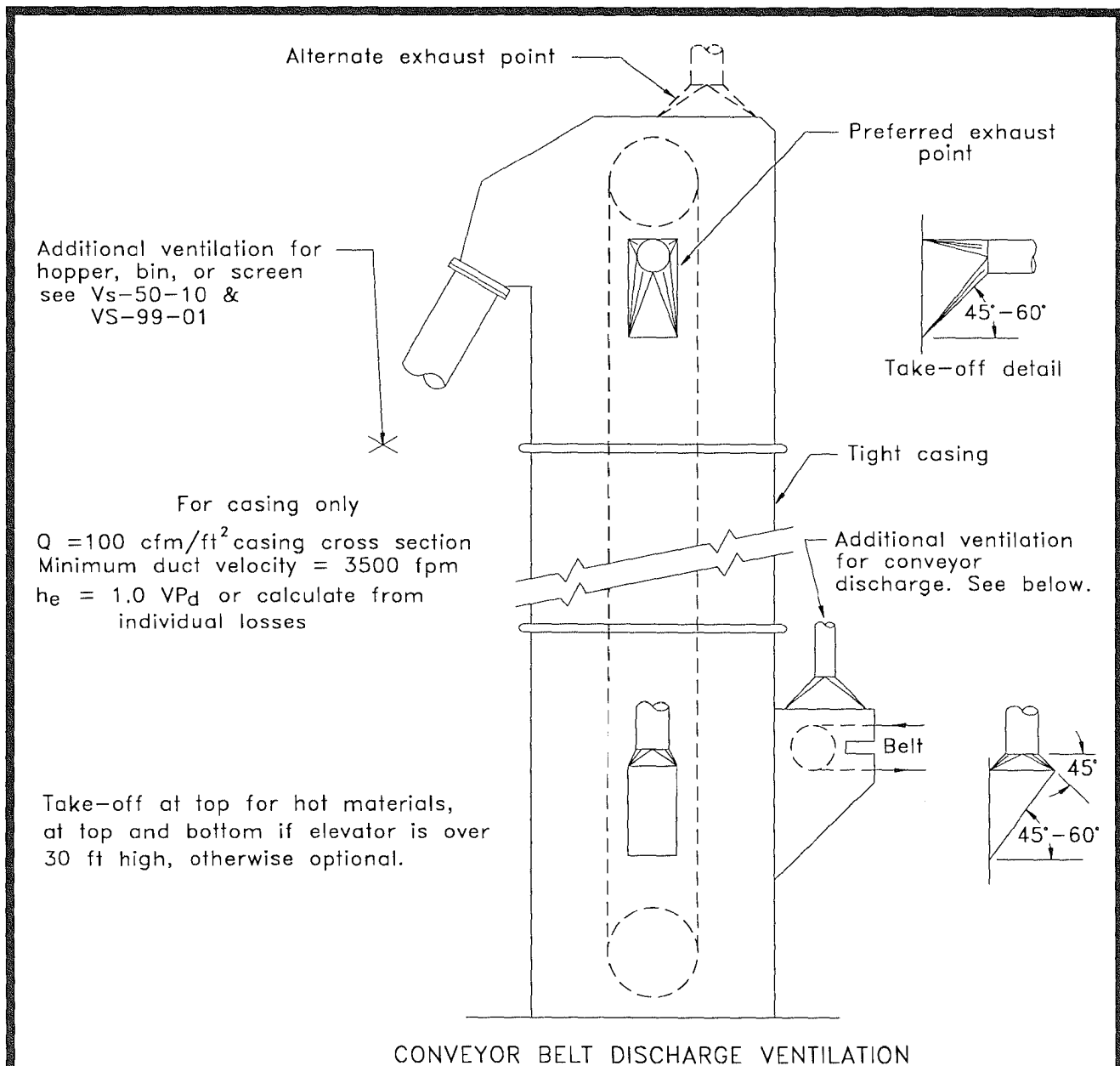
The conveying of toxic material requires additional care in enclosure design to ensure that no air leaks out and that sufficient access is available for inspection and cleanout. The head pulley should be equipped with a scraper or brush (see VS-50-21).

10.50.3 Bin and Hopper Ventilation: For the mechanical loading of bins and hoppers, the exhaust rates previously listed for belt conveyors are appropriate. An exhaust rate of 150 cfm/ft² of hopper is adequate for manual loading operations. The enclosure should cover as much of the hopper opening as possible.

10.50.4 Loading and Unloading: For loading and unloading operations, a ventilation rate of 150–200 cfm/ft² of enclosure opening is adequate *provided the enclosure is large enough to accommodate the "splash" effect*. The entrance to enclosure for truck dumps should be covered with flaps to minimize ventilation requirements. Rotary or bottom car dumps generally are exhausted at the rate of 50–100 cfm/ft² of hopper area.

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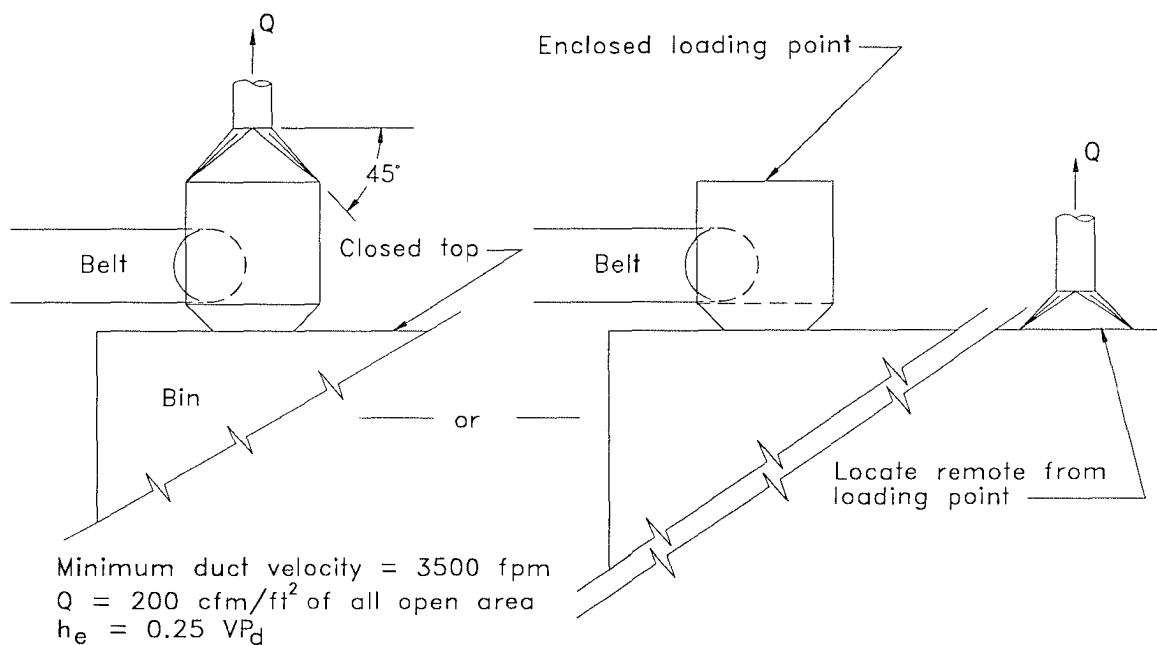
BELT SPEED	FLOWRATE
Less than 200 fpm	350 cfm/ft of belt width. Not less than 150 cfm/ft ² of opening.
Over 200 fpm	500 cfm/ft of belt width. Not less than 200 cfm/ft ² of opening.

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BUCKET ELEVATOR
VENTILATION

DATE 1-91

FIGURE VS-50-01



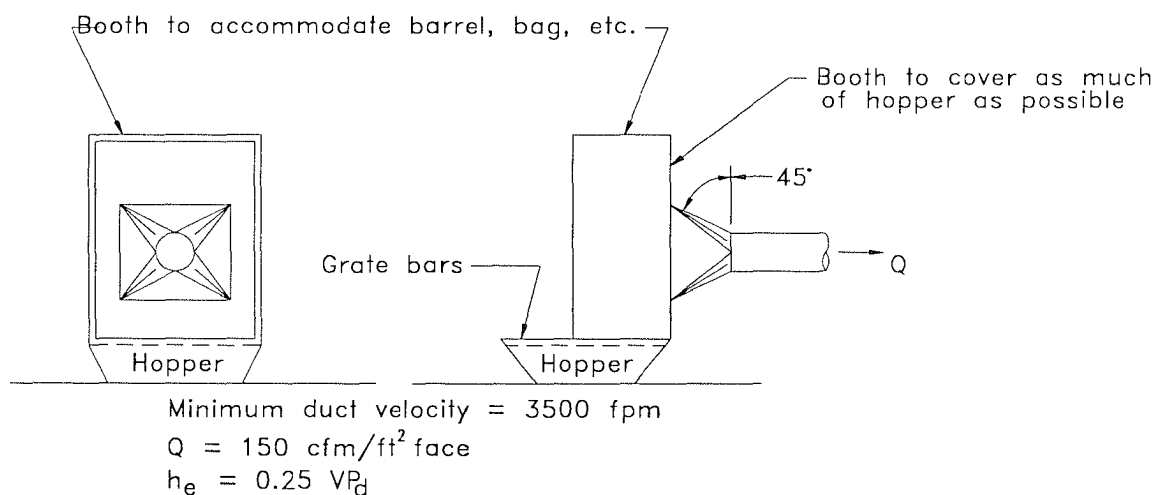
MECHANICAL LOADING

Belt speed
 Less than 200 fpm ----

Flowrate
 350 cfm/ft of belt width.
 Not less than 150 cfm/ft² of opening.

Over 200 fpm ----

500 cfm/ft of belt width.
 Not less than 200 cfm/ft² of opening.



MANUAL LOADING

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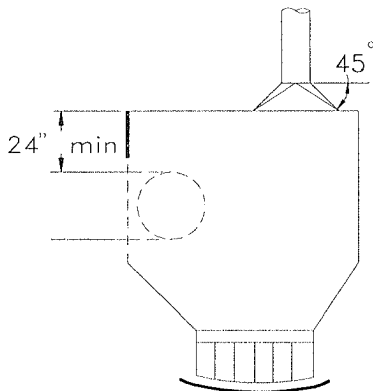
BIN & HOPPER VENTILATION

DATE

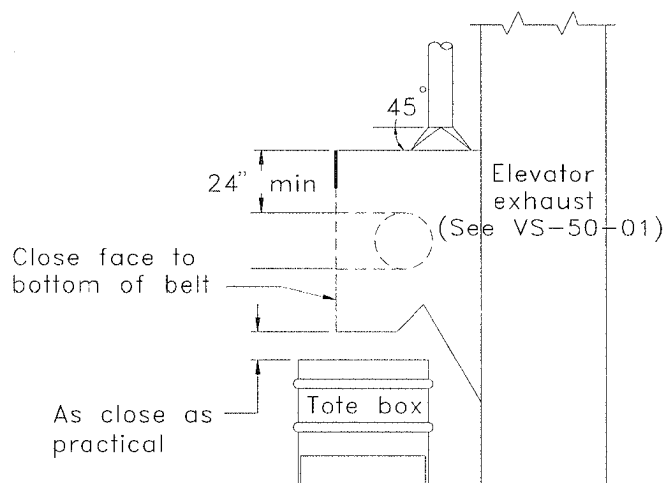
1-91

FIGURE

VS-50-10



1. Conveyor transfer less than 3' fall. For greater fall, provide additional exhaust at lower belt.
See 3 below.
 $h_e = 0.25 VP_d$

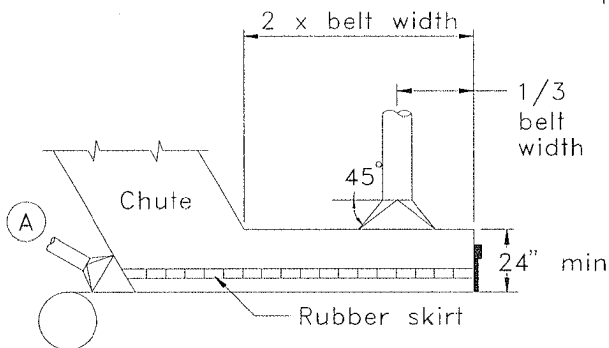


2. Conveyor to elevator with magnetic separator.
 $h_e = 0.25 VP_d$

DESIGN DATA

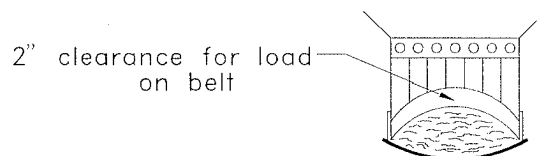
Transfer points:

Enclose to provide 150 - 200 fpm indraft at all openings. (Underground mining tunnel ventilation will interfere with conveyor exhaust systems.)



3. Chute to belt transfer and conveyor transfer, greater than 3' fall. Use additional exhaust at (A) for dusty material as follows:
Belt width 12"-36", $Q=700$ cfm
Belt width above 36", $Q=1000$ cfm
 $h_e = 0.25 VP_d$

Note: Dry, very dusty materials may require exhaust flowrates 1.5 to 2.0 times stated values.



DETAIL OF BELT OPENING

$Q = 350$ cfm/ft belt width for belt speeds under 200 fpm. (minimum)
 $= 500$ cfm/ft belt width for belt speeds over 200 fpm and for magnetic separators. (minimum)
Minimum duct velocity = 3500 fpm
 $h_e = 0.25 VP_d$

Conveyor belts:

Cover belt between transfer points
Exhaust at transfer points
Exhaust additional 350 cfm/ft. of belt width at 30' intervals. Use 45 tapered connections.

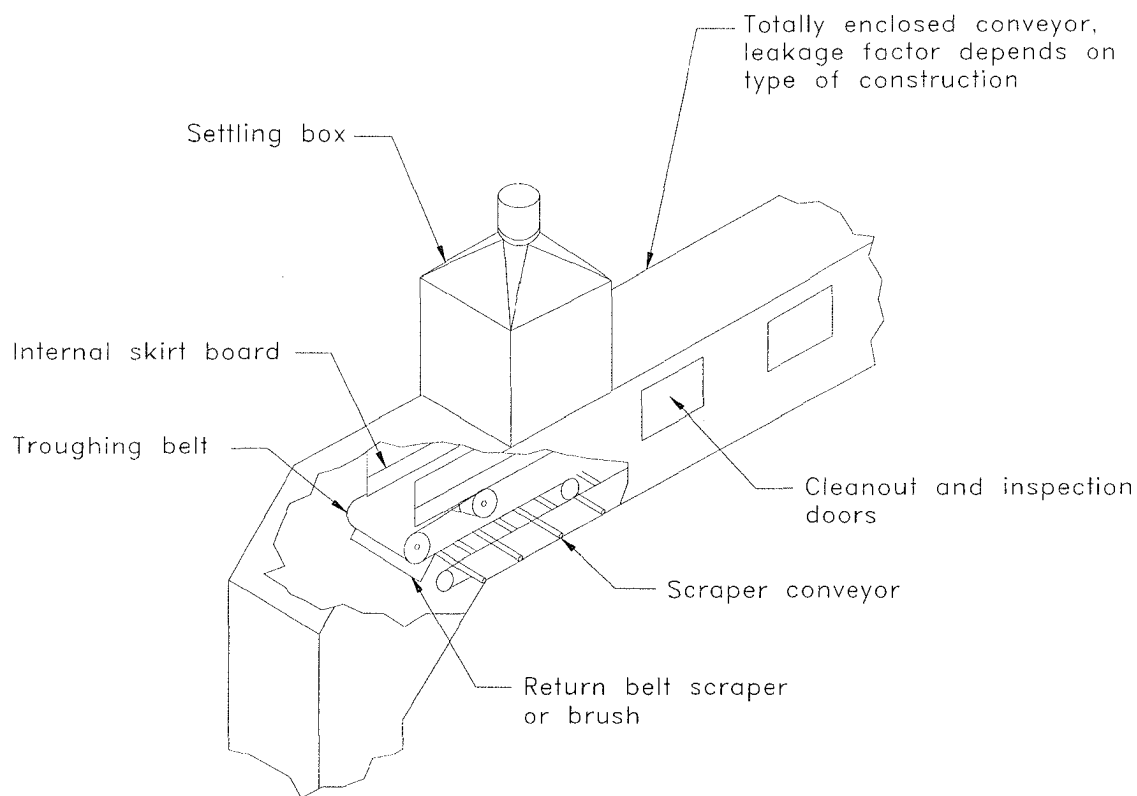
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CONVEYOR BELT VENTILATION

DATE

1-91

FIGURE VS-50-20



$$Q = 250 \text{ cfm/ft}^2 \text{ of open area}$$

$$\text{Minimum duct velocity} = 3500 \text{ fpm}$$

$$h_e = 0.4 \text{ VP}_d$$

Reference: 10.50.3

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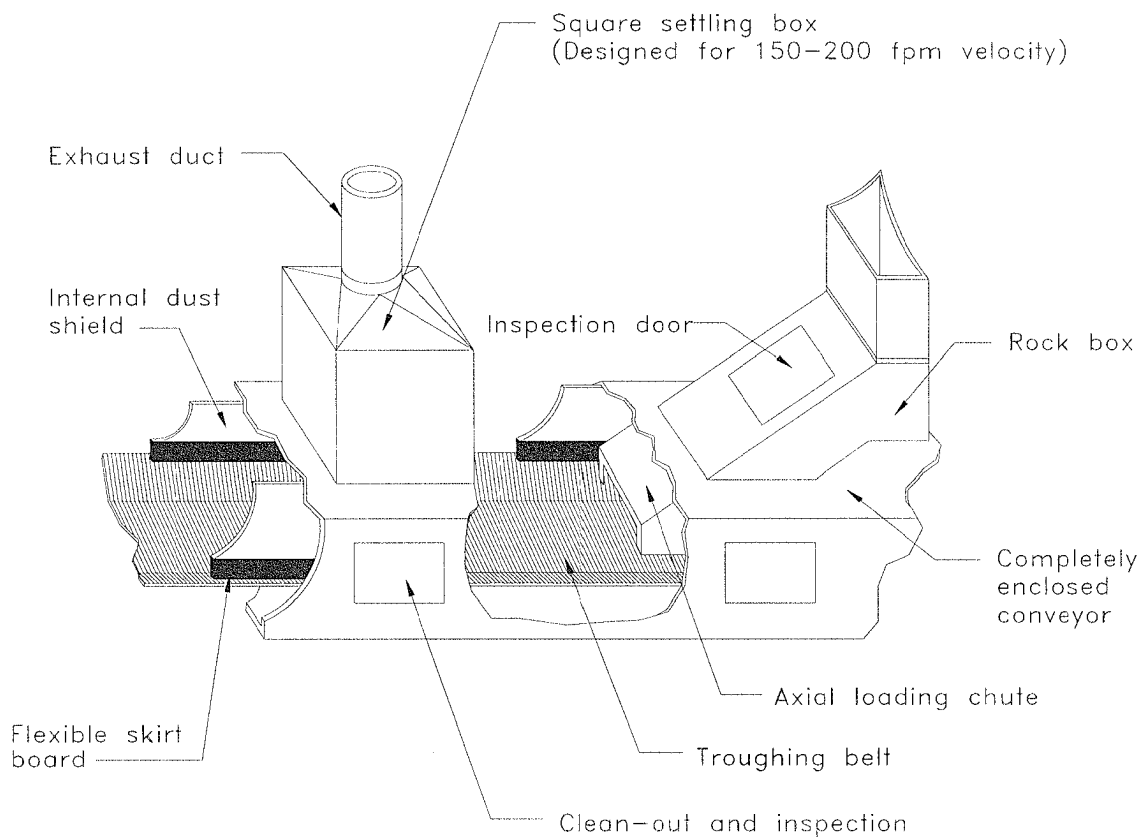
*TOXIC MATERIAL BELT
CONVEYING HEAD PULLEY*

DATE

1-91

FIGURE

VS-50-21



$$Q = 250 \text{ cfm/ft}^2 \text{ of open area}$$

$$\text{Minimum duct velocity} = 3500 \text{ fpm}$$

$$h_e = 0.4 \text{ VP}_d$$

Reference: 10.50.3

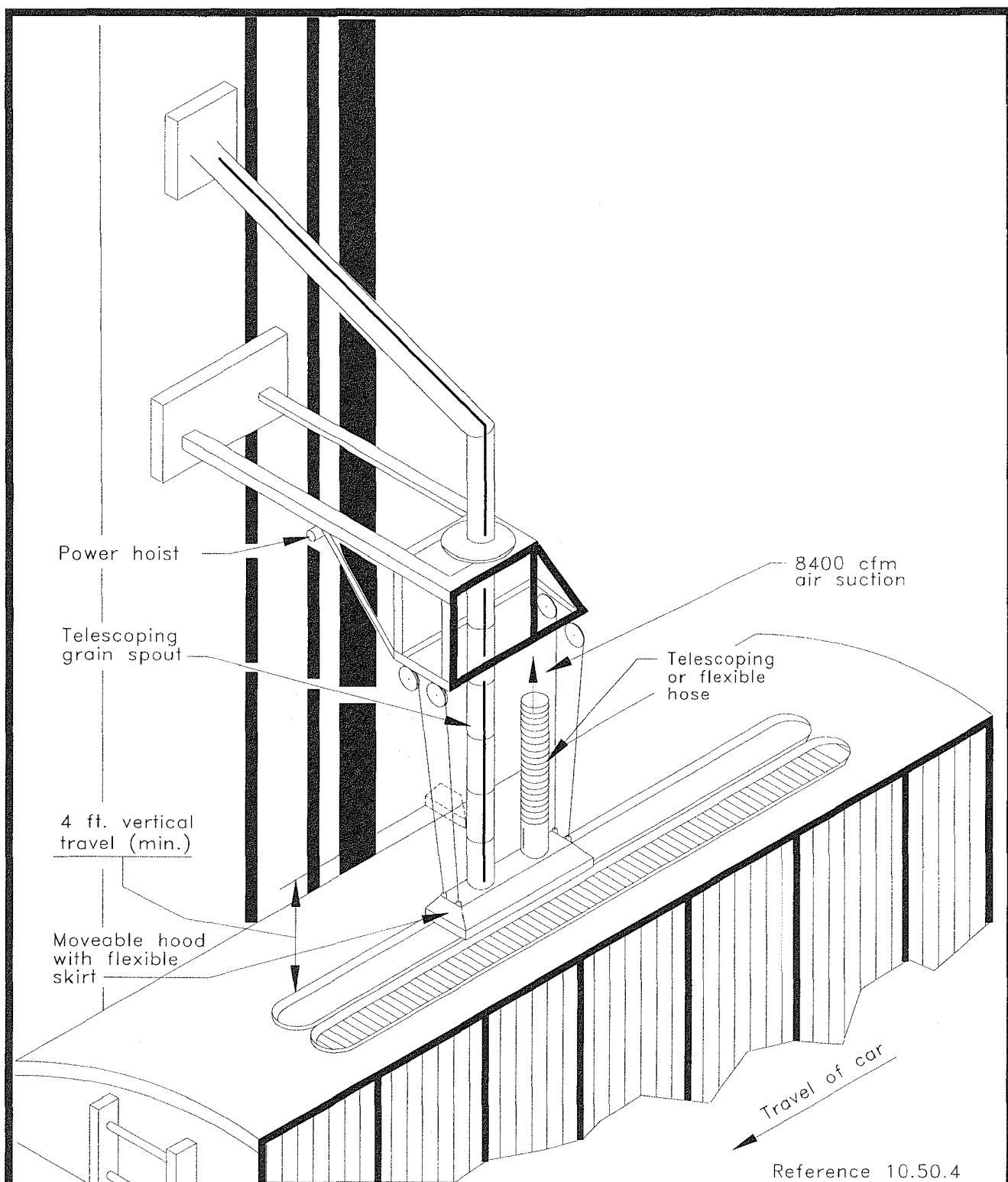
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*TOXIC MATERIAL
CONVEYOR BELT LOADING*

DATE

1-91

FIGURE VS-50-22



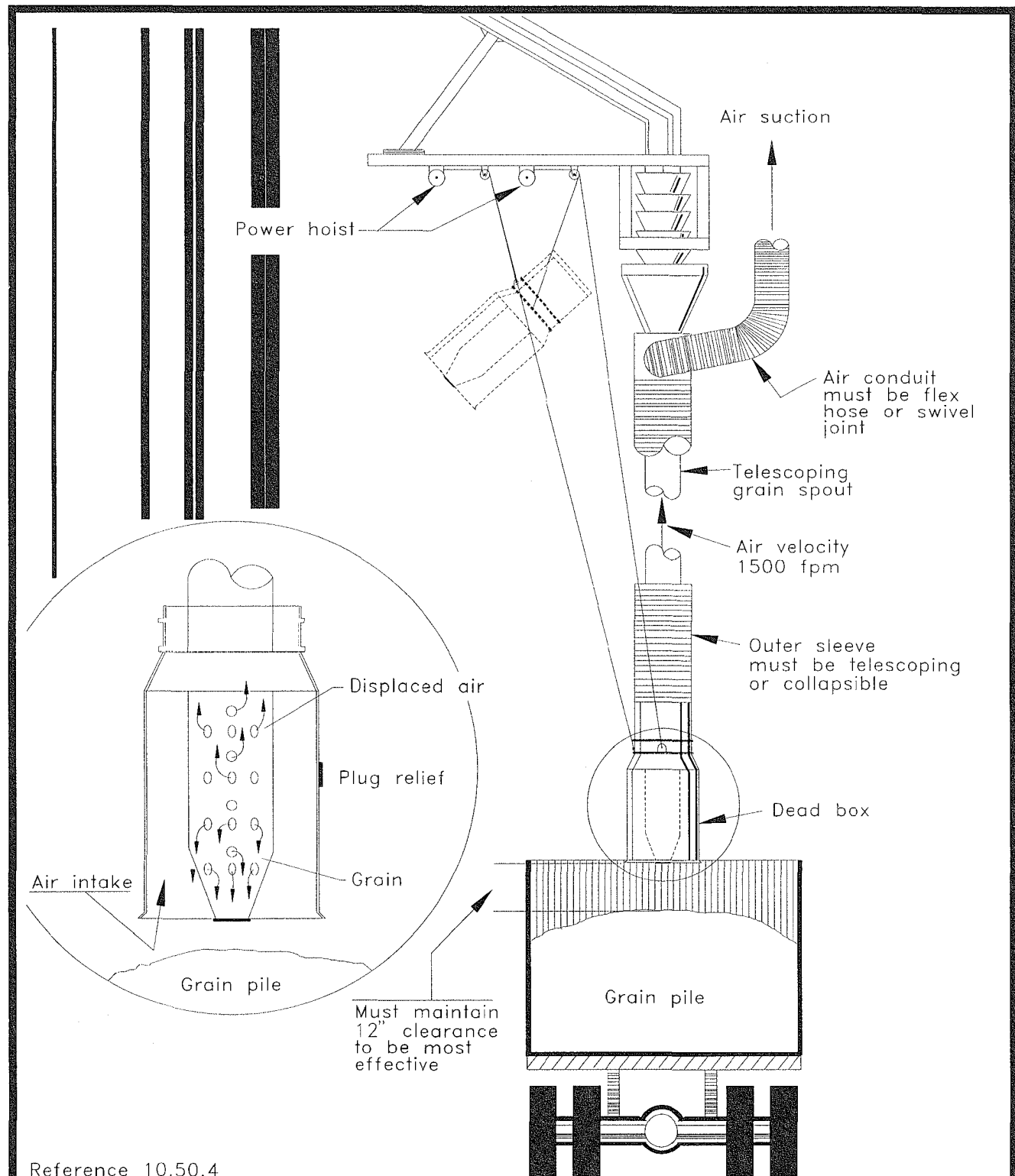
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RAIL LOADING

DATE

2-91

FIGURE VS-50-30



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TRUCK LOADING

DATE 2-91

FIGURE VS-50-31

10.55 METAL MELTING FURNACES

This set of VS-prints describes hood designs for a variety of metal melting furnaces, including electric induction, carbon arc, convention, and crucible, which use natural gas or electric resistance elements as the heat source. Exhaust ventilation usually is required to control specific oxides associated with the metal being melted or contaminants carried in the scrap charge. In some cases, a single hood will suffice for charging, melting, and pouring. In other cases, a separate hood, remote from the primary melter, may be required for charging because of the nature of the charge or the large open area necessary at this phase of the operation. This is true particularly of electric arc furnaces which are completely open for charging. After charging, the port exhaust can be used to achieve control during the remainder of the melting and pouring cycles.

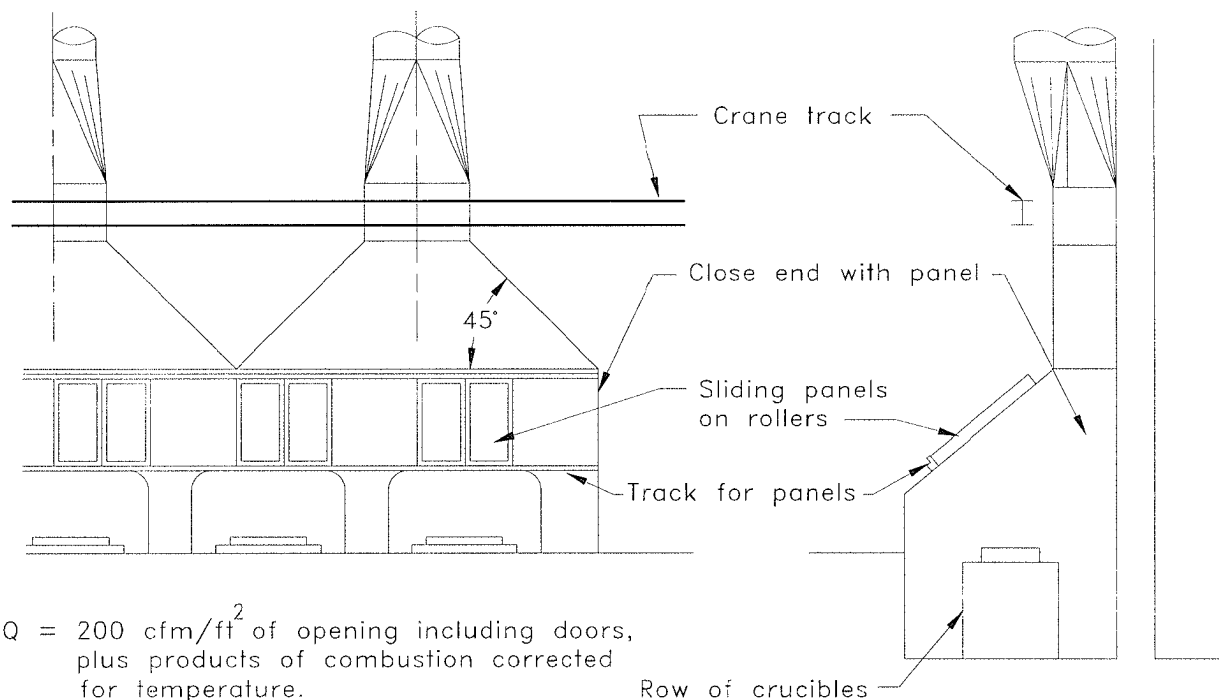
All metal melting will produce a slag which must be removed prior to pouring. This activity may produce a significant release of oxides and may require a separate exhaust system for oxide and/or dross control. Where metal purification is performed directly within the furnace or melting vessel,

such as the addition of oxygen or chlorine, additional exhaust may be required to contain the rapidly generated plume.

All systems must be designed to include the increase in air temperature under operating conditions to insure an adequate air flow into the hood. The air temperature rise is usually relatively low except where metal inoculants or oxidizers are added to the molten charge. During this phase of metal melting, a significant temperature rise will occur and it is customary to provide a large hood in which gas expansion can take place.

REFERENCES

- 10.55.1 American Air Filter Co.: Rotoclone Dust Control (January 1946).
- 10.55.2 Kane, J.M.: Foundry Ventilation. The Foundry (February and March 1946).
- 10.55.3 Kane, J.M.: The Application of Local Exhaust Ventilation to Electric Melting Furnaces. In: Transactions of the American Foundrymen's Association, Vol. 52, p. 1351 (1945).

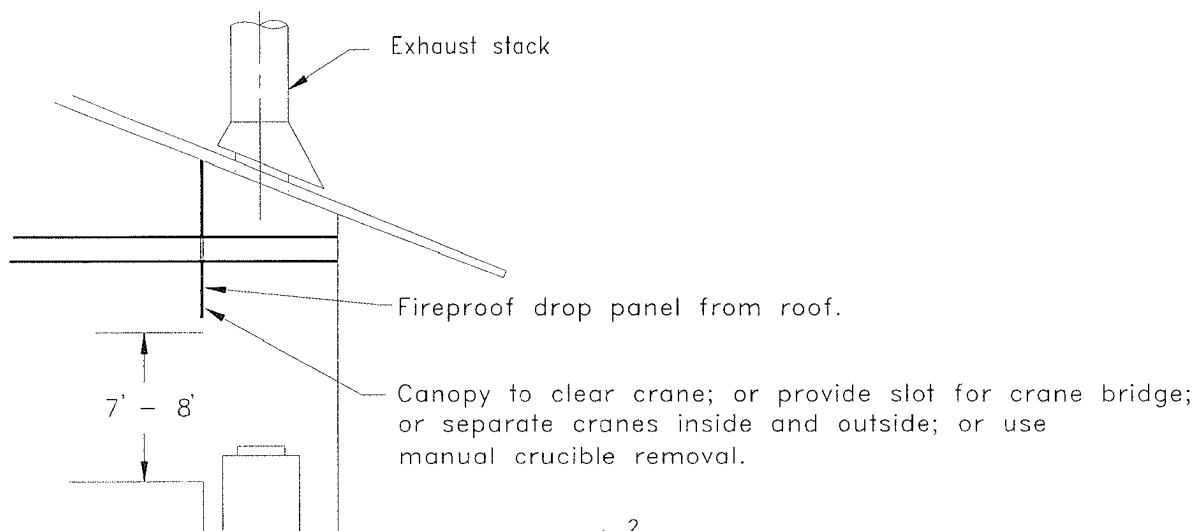


$Q = 200 \text{ cfm/ft}^2$ of opening including doors,
plus products of combustion corrected
for temperature.

Minimum duct velocity = 3500 fpm

$h_e = 0.5 V_D^2$

Note: Same principle of sliding or swinging
doors is applied to individual furnace
enclosures.



$Q = 200 \text{ cfm/ft}^2$ of total opening, minimum, plus
products of combustion corrected for temperature.

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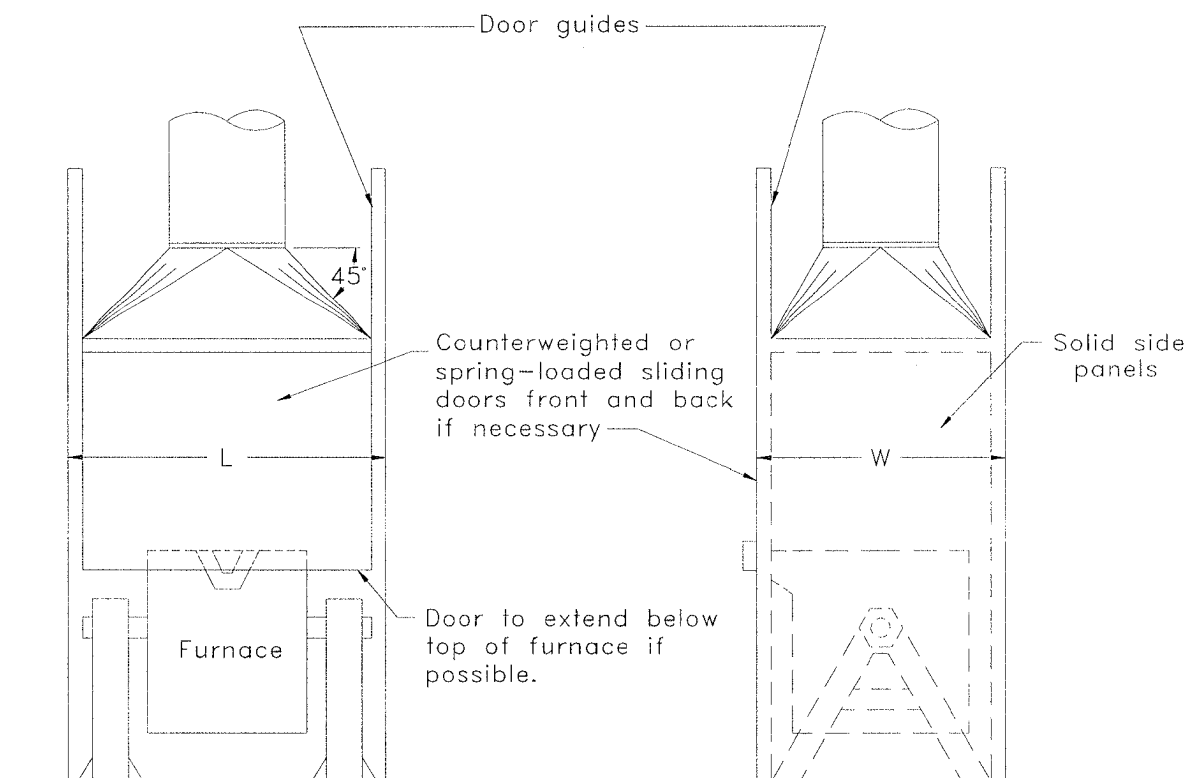
MELTING FURNACE
CRUCIBLE, NON-TILT

DATE

02-91

FIGURE

VS-55-01



$Q = 200 LW$; but not less than
 200 cfm/ft^2 of all openings
 with doors open. Correct for products
 of combustion and temperature.
 Minimum duct velocity = 3500 fpm
 $h_e = 0.25 VP_d$

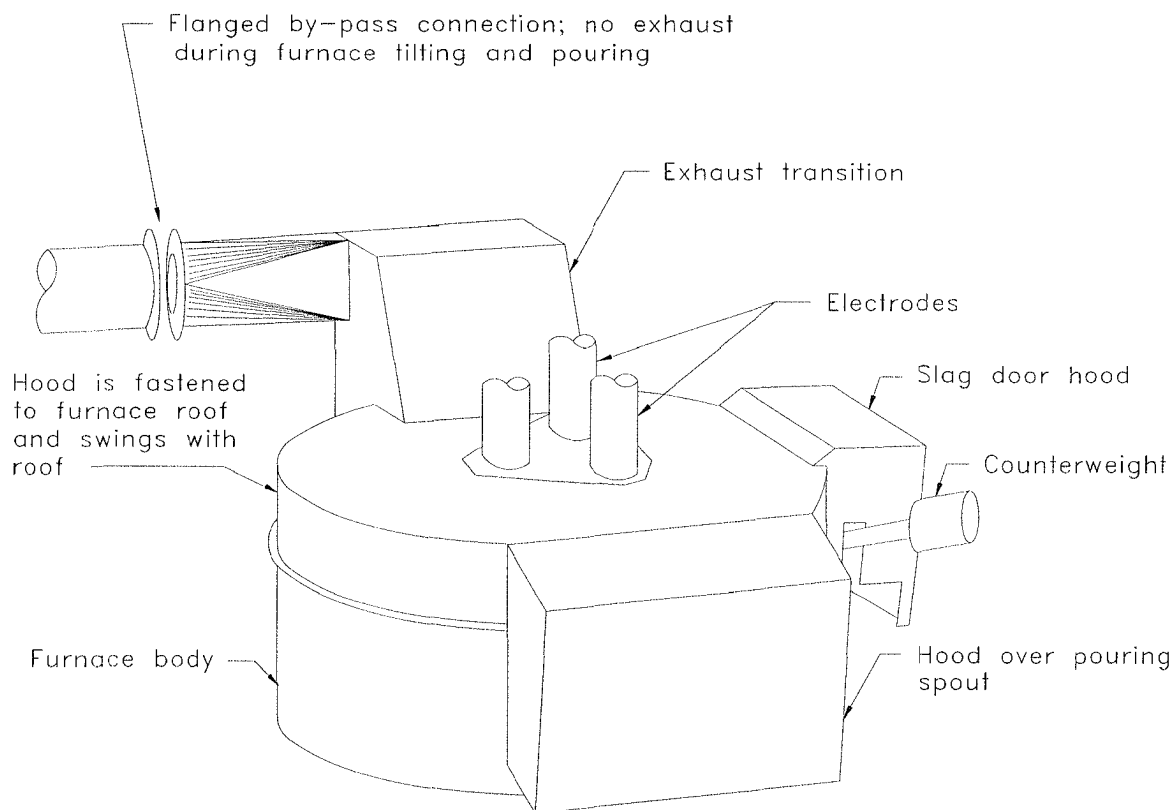
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MELTING FURNACE
 TILTING

DATE

02-91

FIGURE VS-55-02



Close Capture:

For Q, SP and operating temperature, consult manufacturer
 Approximate exhaust rate = 2500 scfm/ton of charge

Alternate designs:

1. Some exhaust designs utilize direct furnace roof tap. For details consult manufacturer.
2. Canopy hoods require large exhaust and are not recommended. Canopy hoods can be used as secondary hoods to capture fugitive emissions.

References 10.55.1, 10.55.2, 10.55.3

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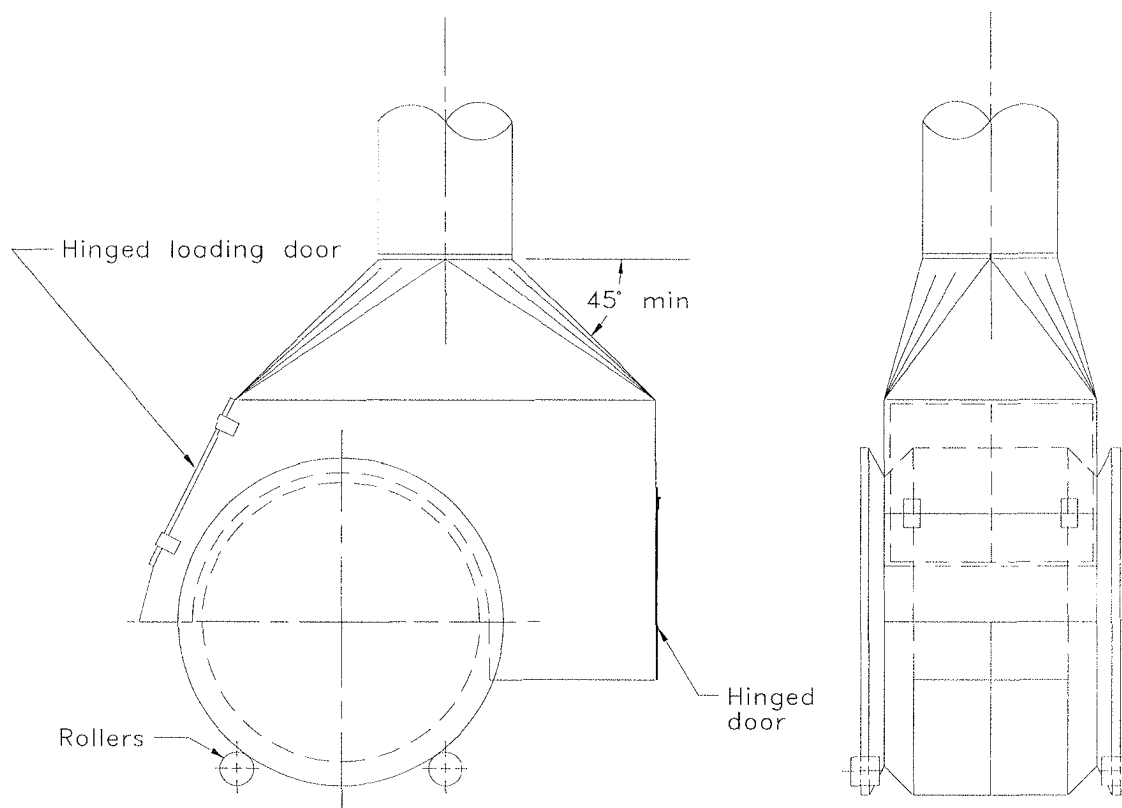
*MELTING FURNACE
 ELECTRIC, TOP ELECTRODE*

DATE

02-91

FIGURE

VS-55-03



$$Q = 400 \text{ cfm/ft}^2 \text{ of opening}$$

$$\text{Minimum duct velocity} = 3500 \text{ fpm}$$

$$h_e = 1.78 VP_s + 0.25 VP_d$$

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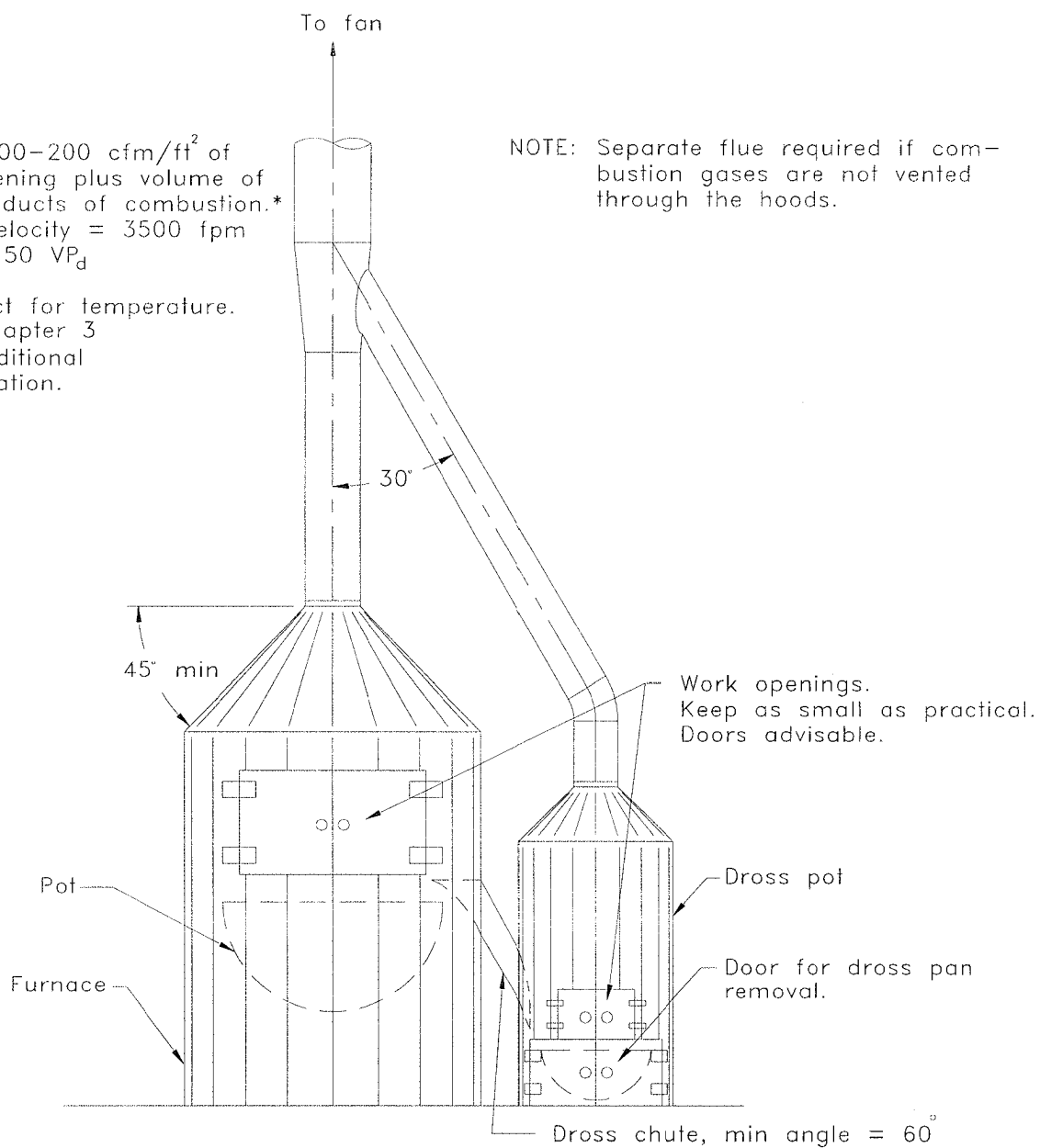
MELTING FURNACE
ELECTRIC ROCKING

DATE

02-91

FIGURE

VS-55-04



STATIONARY FURNACE OR MELTING POT

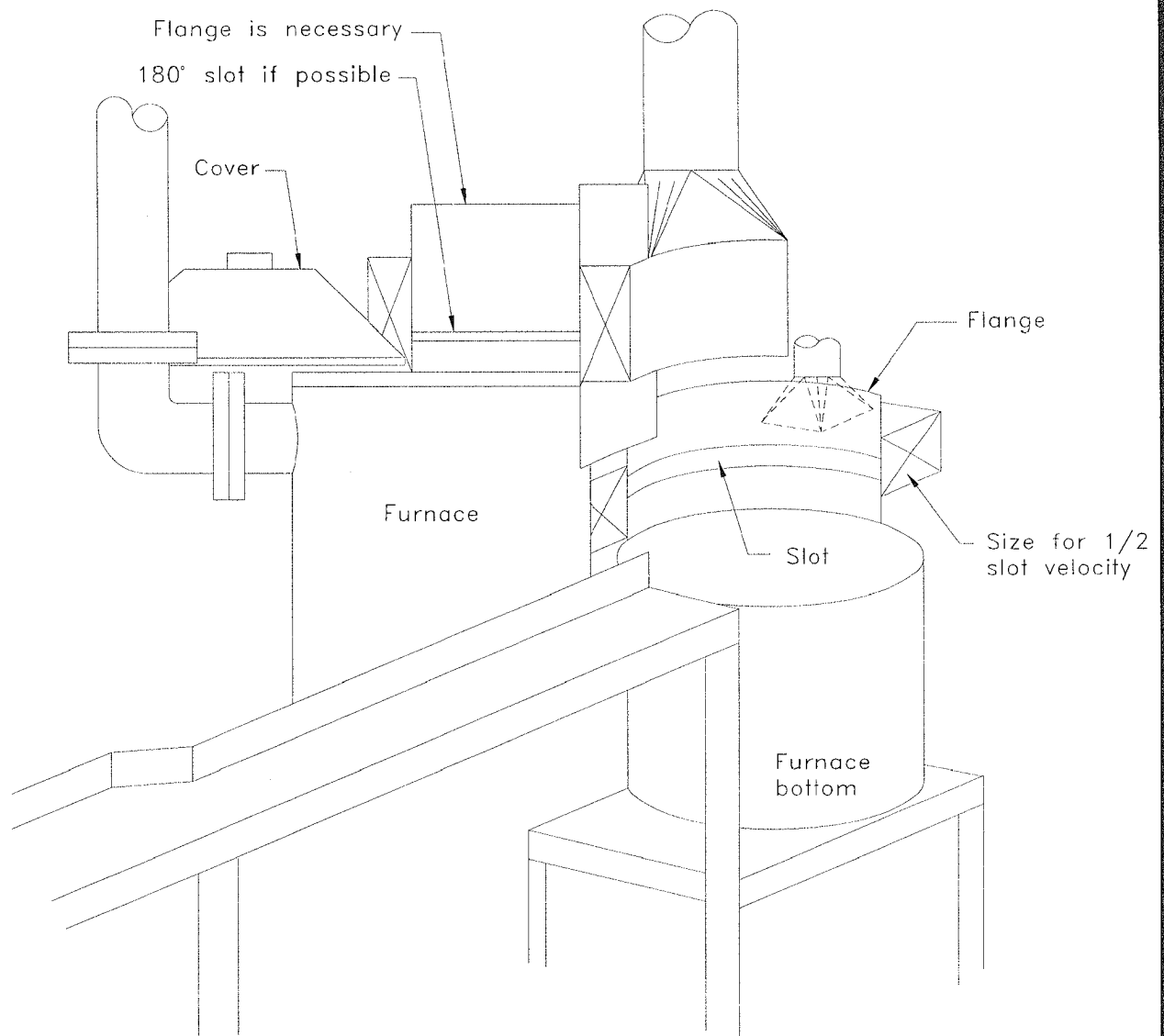
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MELTING POT & FURNACE

DATE

02-91

FIGURE VS-55-05



$Q = 175 \text{ cfm/ft}^2$ of furnace top with curved slot and flanges.

Slot velocity = 2000 fpm

Minimum duct velocity = 3500 fpm

Entry losses = $1.78 VP_s + 0.25 VP_d$

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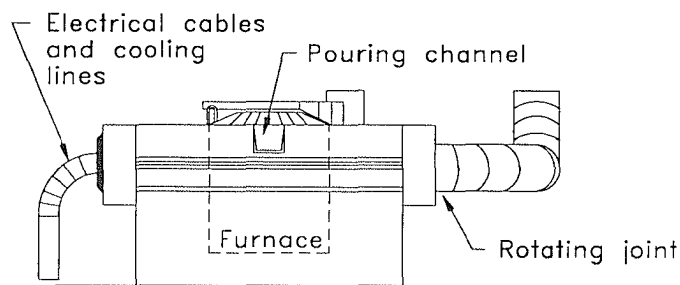
CRUCIBLE MELTING FURNACE
HIGH TOXICITY MATERIAL

DATE

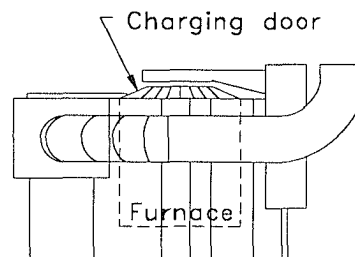
02-91

FIGURE

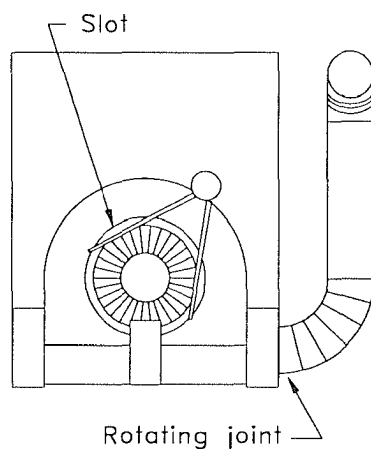
VS-55-06



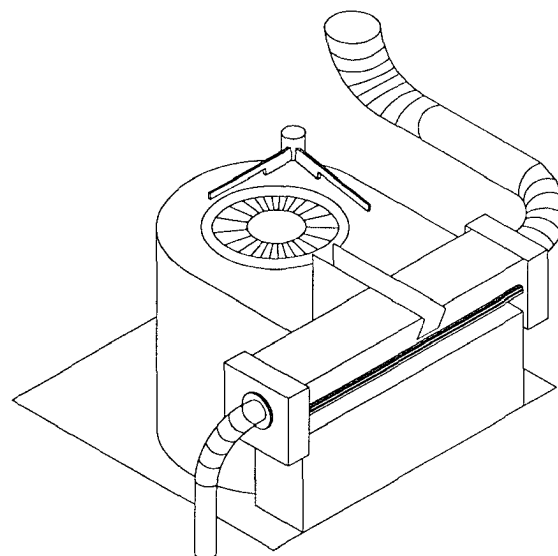
FRONT VIEW



SIDE VIEW



TOP VIEW



ISOMETRIC VIEW

$Q = 350 \text{ cfm/ft.}^2$ open area. Correct for temperature and combustion products.

Entry loss = $1.78 \text{ VP}_s + 0.5 \text{ VP}_d$

Slot velocity = 2000 fpm

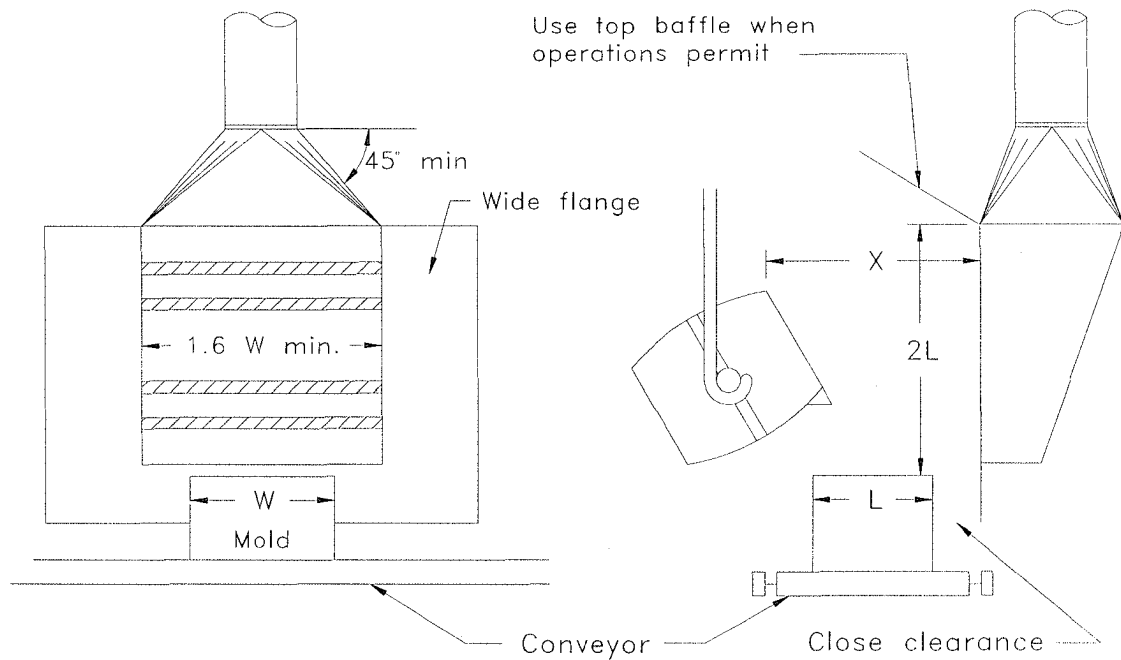
Minimum duct velocity = 3500 fpm

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INDUCTION MELTING FURNACE
—TILTING

DATE 02-91

FIGURE VS-55-07



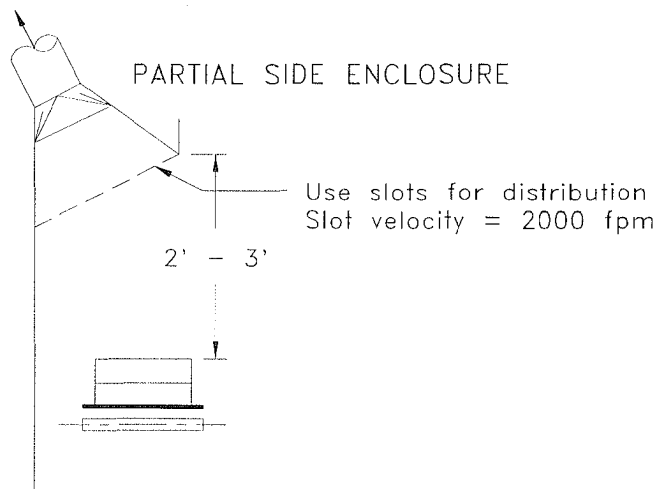
SMALL MOLDS

$Q = 200 (10 X^2 + A)$ where A equals hood area.

Minimum duct velocity = 3500 fpm

$h_e = 1.78 VP_s + 0.25 VP_d$

Use slots for uniform distribution, size slots for 2000 fpm



Note:

For large molds and ladles provide large - draft hood similar to shakeout.

$Q = 400 \text{ cfm/ft}^2$ working area.

$Q = 200 - 300 \text{ cfm/ft}$ of hood length.

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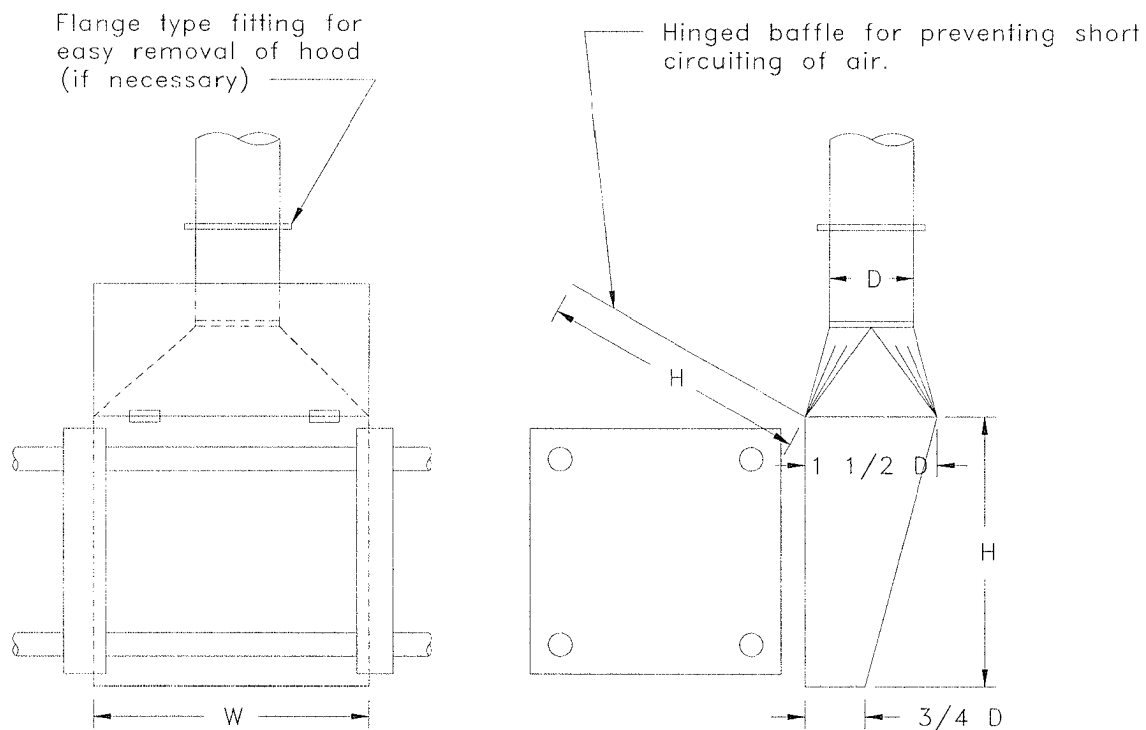
POURING STATION

DATE

02-91

FIGURE

VS-55-10



Note: Place hood as close to machine as possible. If more than 4 inches from back of machine, hinged side baffles should be used.

Note: Products of combustion require separate flue or may be vented into hood.

$$Q = 300WH$$

$$h_e = 0.25 VP_d$$

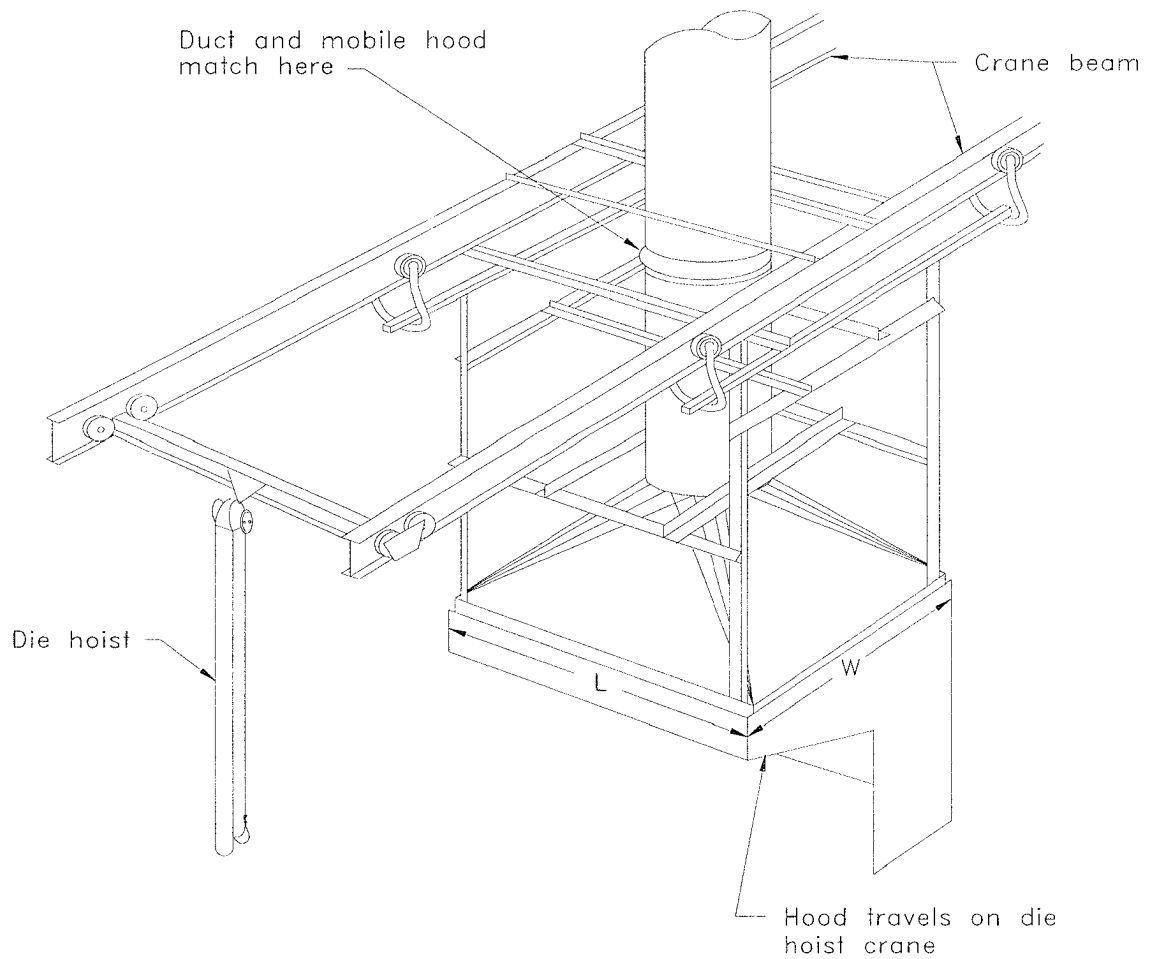
Minimum duct velocity = 2000 fpm.

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*FIXED POSITION
DIE CASTING HOOD*

DATE 02-91

FIGURE VS-55-20



$$Q = 300WL$$

Minimum duct velocity = 2000 fpm

$$h_e = 0.25 VP_d$$

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MOBILE HOOD
DIE CASTING

DATE

02-91

FIGURE

VS-55-21

10.60 MIXING

Mixing operations combine a large variety of materials, usually without significant chemical reactions. This section includes categories of mixing operations.

10.60.1 *Mixing and Mulling:* Mixers and mullers require exhaust ventilation to provide a minimum velocity of 150 fpm through all openings. Additional ventilation may be required when flammable solvents are used. The dilution ventilation rates should maintain concentrations within the muller below 25% of the Lower Explosive Limit (LEL). Some codes or standards may require ventilation rates which ensure the concentration of flammable vapor is maintained below 20% of the LEL.

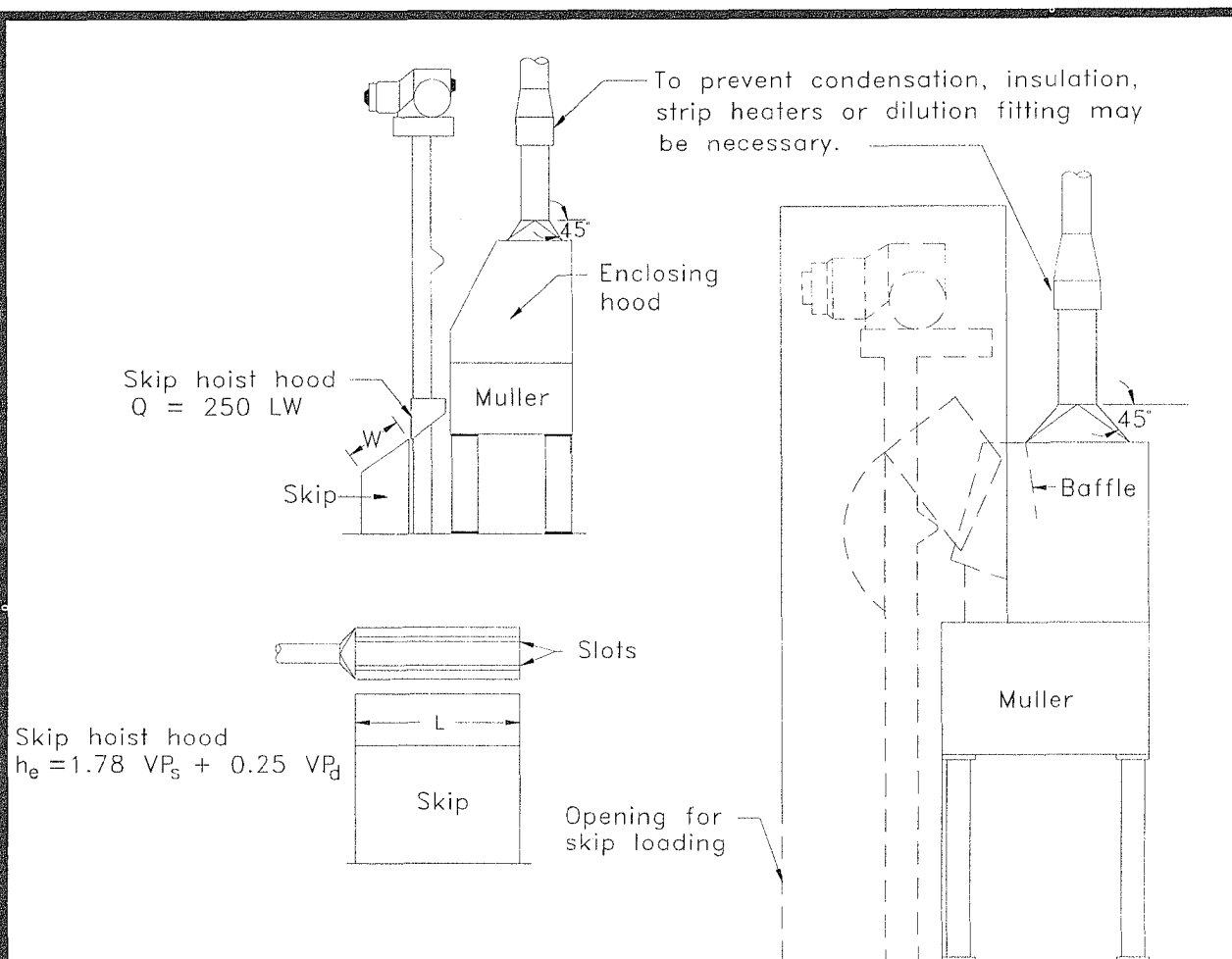
10.60.2 *Roll Mixing:* The machinery shown in this subsection is used to mix and blend quantities of viscid materials, such as rubber and plastic, with additives that are dry powers or liquids. Emissions of gases and vapors may evolve due to

chemical reactions or may be caused by the elevated temperature of the mixed materials. Particulate emissions can occur during additions as well as during mechanical blending.

The roller mill ventilation shown in VS-60-12 encloses the roller mill to the maximum extent possible except for a front opening of sufficient height and width to permit operator access and material entry/removal. An air curtain directed upward towards the top of the enclosure provides a barrier to contaminant escape but still permits operator access and material entry/removal. The design values provided are critical for proper operation.

REFERENCES

- 10.60.1 Hampl, V.; Johnston, O.E.; Murdock, D.M.: Application of an Air Curtain Exhaust System at a Milling Process. American Industrial Hygiene Association Journal, 49(4):167-175 (1988).



$Q = 150 \text{ cfm/ft}^2$ through all openings but not less than:

Muller diam. feet	Exhaust, cfm
4	750
6	900
7	1050
8	1200
10	1575

Minimum duct velocity = 4000 fpm

$h_e = 0.25 VP_d$

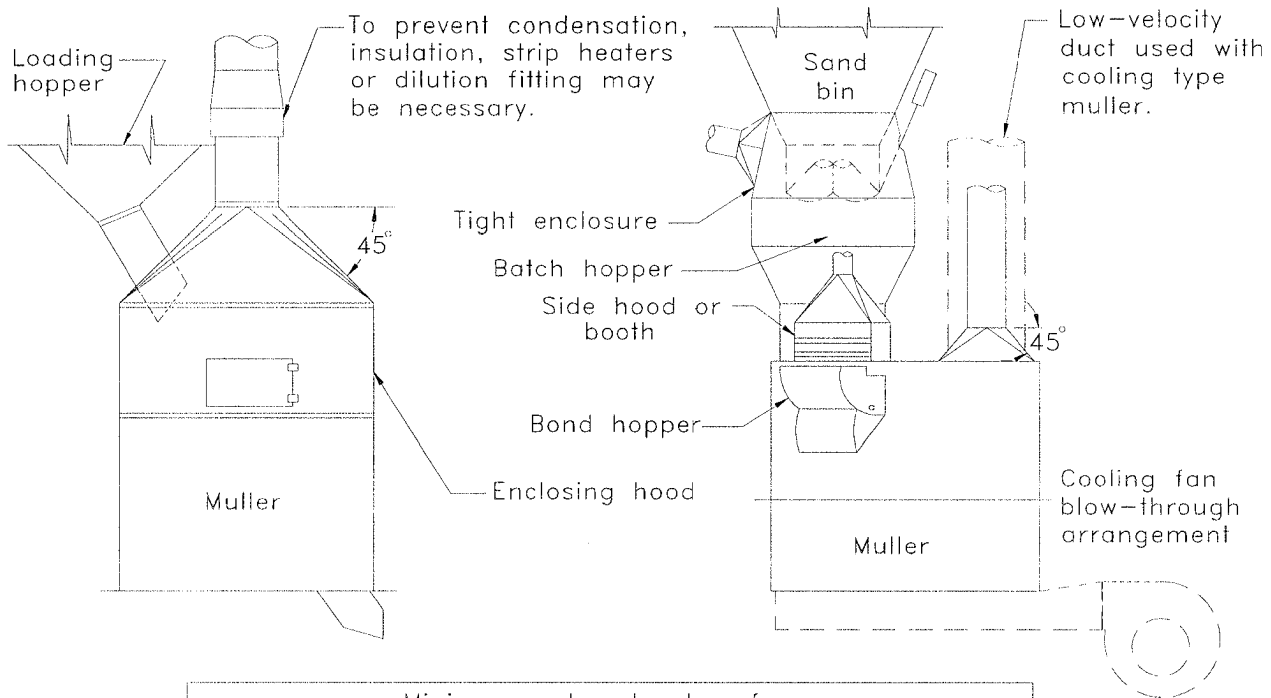
- Notes:
1. Other types of mixers: enclose as much as possible and provide 150 cfm/ft² of remaining openings
 2. When flammable solvents are used in mixer, calculate minimum exhaust rate for dilution to 25% of the LEL. See Chapter 2
 3. For air cooled mullers see VS-60-02

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MIXER AND MULLER HOOD

DATE 11-90

FIGURE VS-60-01



Minimum exhaust rate, cfm			
Location	Muller type		
	No cooling	Blow-through cooling	Draw-through cooling
Batch hopper	Note 1	600	Note 1
Bond hopper	600	600	600
Muller:	Note 2	Note 3	Note 3
4' diameter	750	"	"
6' diameter	900	"	"
7' diameter	1050	"	"
8' diameter	1200	"	"
10' diameter	1575	"	"

Minimum duct velocity = 4500 fpm

$h_e = 0.25 \text{ VP}$

Notes:

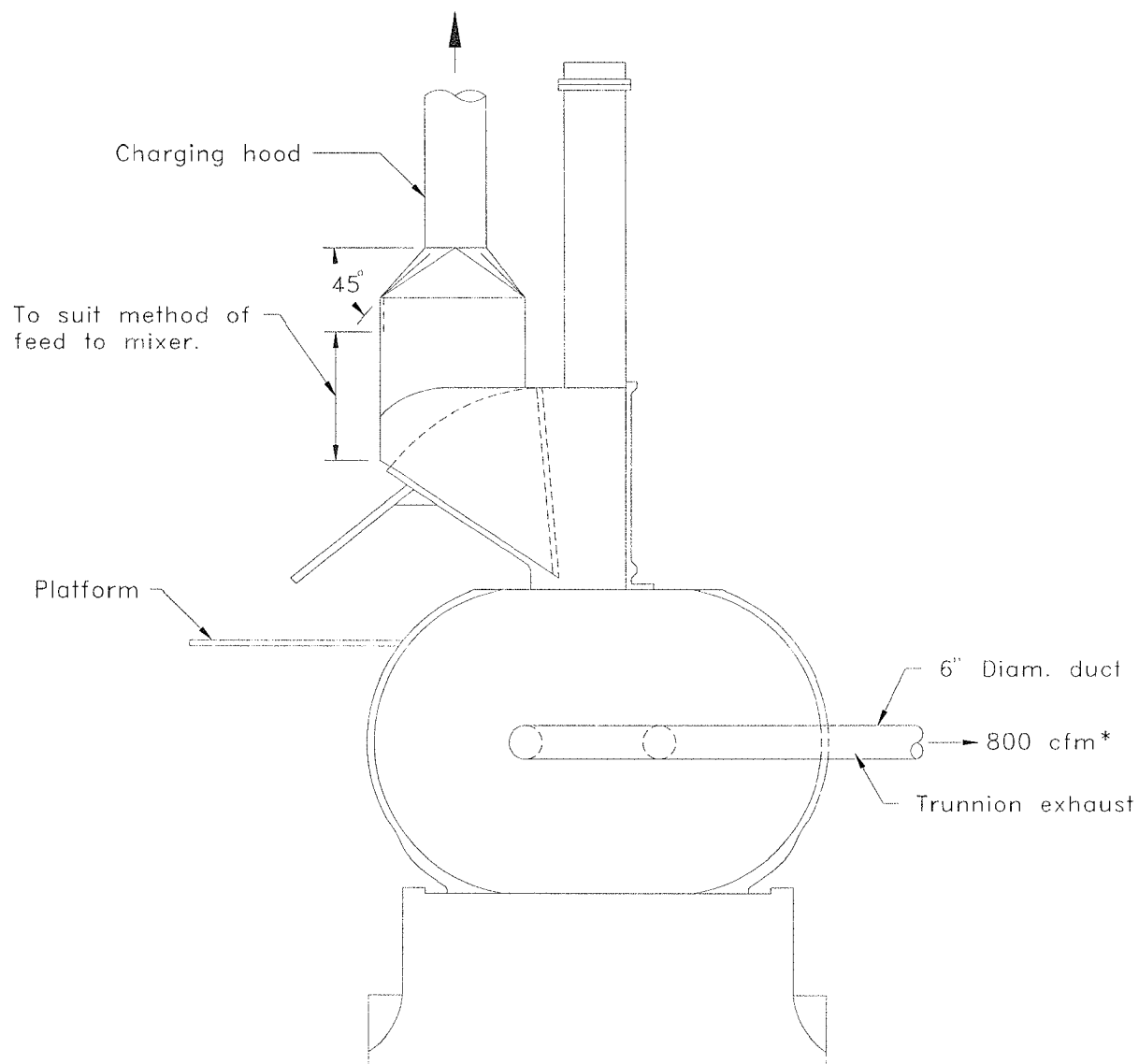
1. Batch hopper requires separate exhaust with blow-through cooling. With other fan arrangement (muller under suction), separate exhaust may not be required. (If skip hoist is used, see VS-60-01)
2. Maintain 150 fpm velocity through all openings in muller hood. Exhaust flow rates shown are the minimum for control.
3. Cooling mullers do not require additional exhaust if maintained in dust tight condition. Blow-through fan must be off during loading. If muller is not dust tight, exhaust as in note 2 plus cooling air flow rate.
4. When flammable solvents are used in mixer, calculate minimum exhaust flow rate for dilution to 25% of the LEL. See Chapter 2.

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AIR COOLED
MIXER AND MULLER

DATE 11-90

FIGURE VS-60-02



$Q = 200-300 \text{ cfm/ft}^2$ open face area.
 500 cfm/ft^2 of belt width if belt feeder used.
 Duct velocity = 3500 fpm minimum.
 $h_e = 0.25 VP_d$ at hood
 $= 1.0 VP_d$ at trunnion

* Consult mixer manufacturer for specific recommendations.

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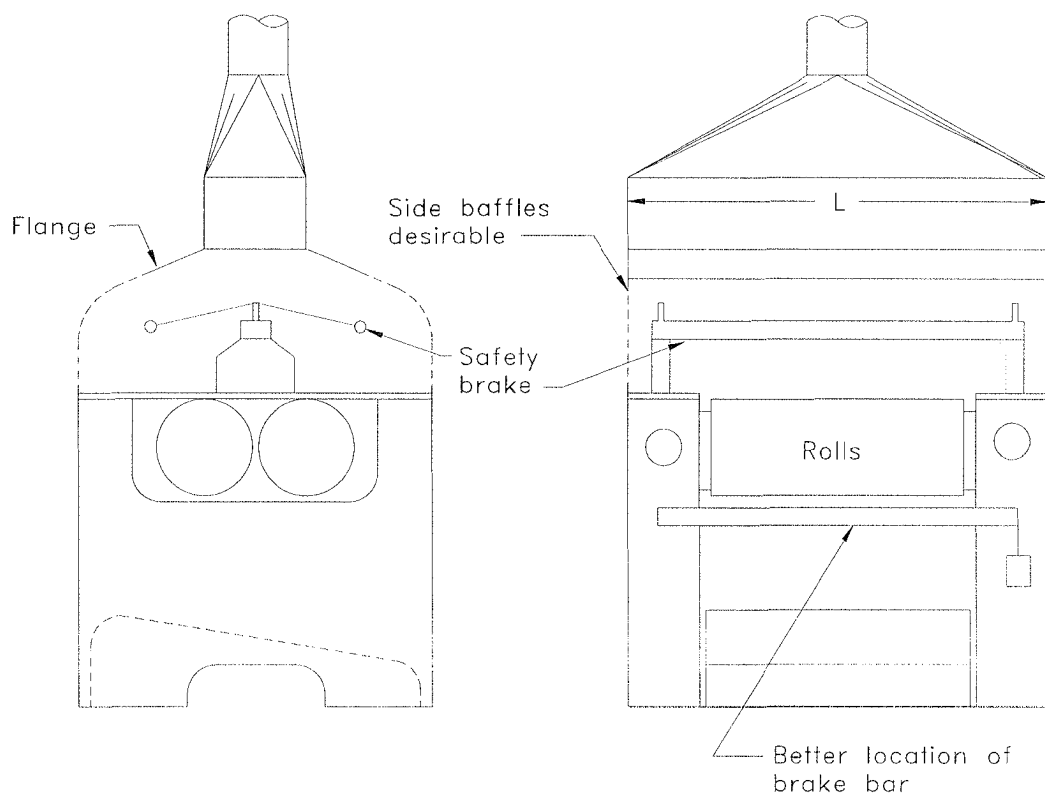
BANBURY MIXER

DATE

11-90

FIGURE

VS-60-10



$Q = 125 \text{ cfm/ft}^2 \text{ open face area}$
 Minimum duct velocity = 2000 fpm
 $h_e = 0.25 V P_d$

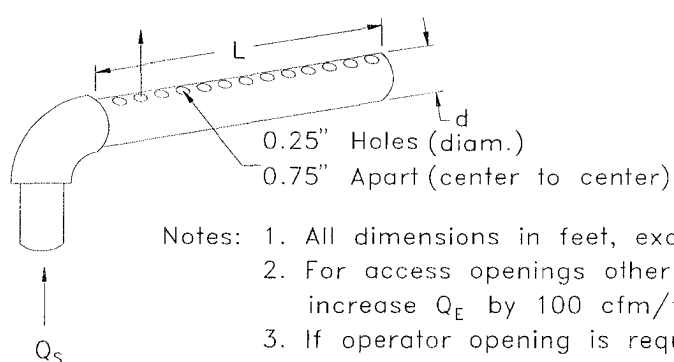
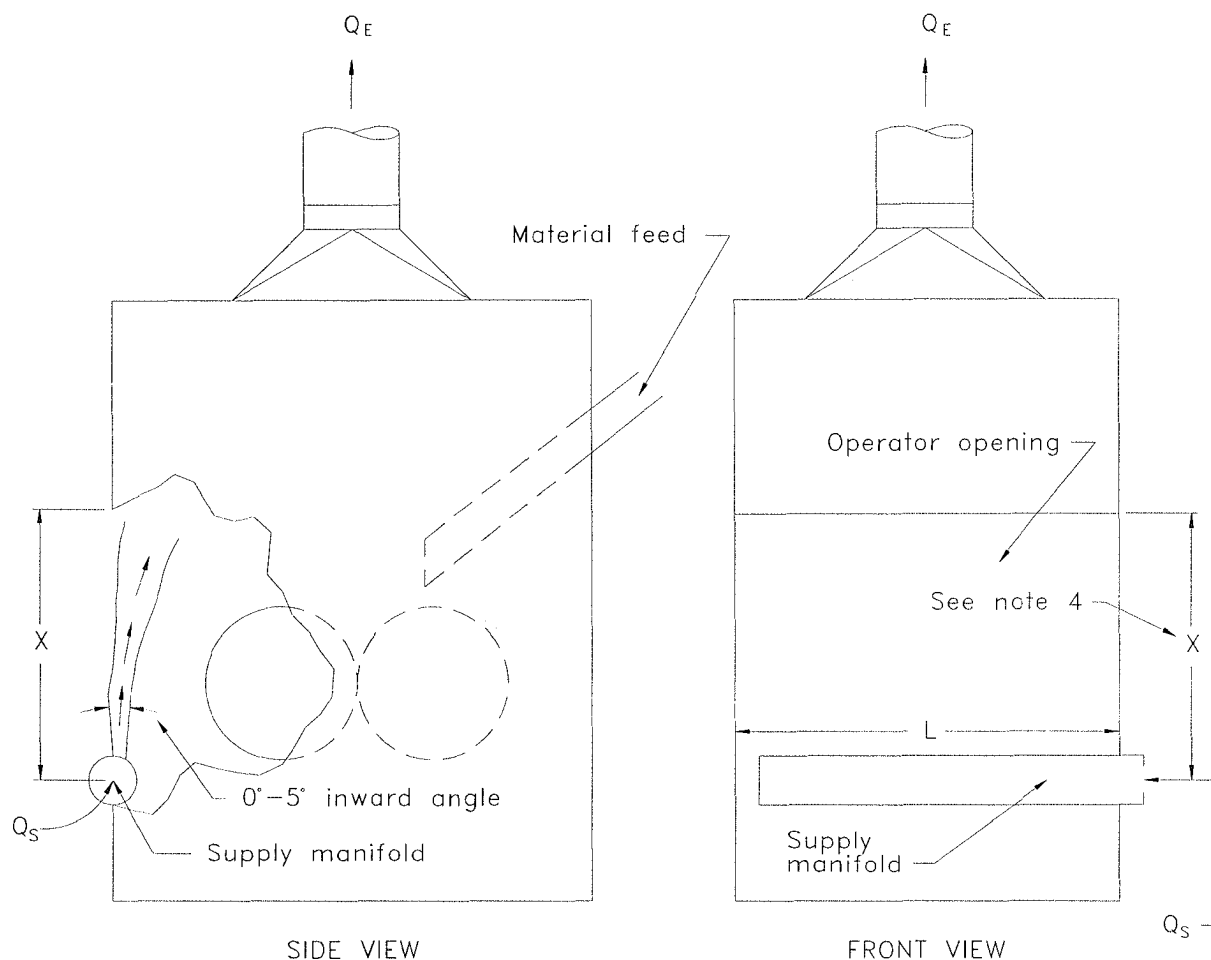
NOTE: Both sides may be open

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RUBBER CALENDER ROLLS

DATE 11-90

FIGURE VS-60-11



$$d \geq 0.145 \sqrt{L} \text{ ft (Manifold diameter)}$$

$$Q_s = 29.6 L \text{ cfm (Supply flow)}$$

$$Q_E = 320 \sqrt{X} L \text{ cfm (Exhaust flow)}$$

$$SP_s = \text{Manifold pressure} = +1.2 \text{ ''wg}$$

- Notes:
1. All dimensions in feet, except as otherwise noted.
 2. For access openings other than operator opening, increase Q_E by 100 cfm/ft^2 of opening area.
 3. If operator opening is required on both sides of mill, total Q_E will be sum of Q_E for both sides.
 4. X not to exceed 6 ft.

Reference 10.60.1

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ROLLER MILL VENTILATION

DATE

11-90

FIGURE

VS-60-12

10.65 MOVABLE EXHAUST HOODS

Movable exhaust hoods provide control for moving contaminant sources. In general, movable hoods are associated with flexible exhaust ducts; traveling exhaust hoods, swivel, slip, or telescoping joints in duct sections; or systems which separate the hood from the duct for access to the process.

Flexible exhaust duct is possibly the most common way of providing a movable exhaust hood. A section of flexible duct connects to a relatively small exhaust hood. The duct section and hood may be supported by a counter-weighted or spring-loaded, hinged arm that allows the positioning of the exhaust hood near the source of contaminant generation. This type of device is known as "snorkel," "elephant trunk," or "flex-arm" exhaust. Illustrations of this type of exhaust in Chapter 10 include Welding Exhaust (VS-65-01) and Granite Cutting and Finishing (VS-65-02). Flexible exhaust duct use is also illustrated in Barrel Filling (VS-15-01), Metal Spraying (VS-90-30) and Service Garage Ventilation (VS-85-01 and VS-85-02) and low volume/high velocity systems (Section 10.40). Frictional resistance can be very high in the flexible duct section as can the negative pressure or suction. Materials used in the construction of the duct may be metal or non-metal and the losses vary over a wide range depending on the type and use. The application data provided by the manufacturer must be included in the design development of the system. When used, flexible duct should be non-collapsible with minimal length to reduce undesirable bends which will result in excessive static pressure losses.

Traveling exhaust hoods may be used for a variety of operations where the contaminant source moves from one point to another. This type is more suited to heavy-duty requirements than the flexible exhaust duct. Examples of

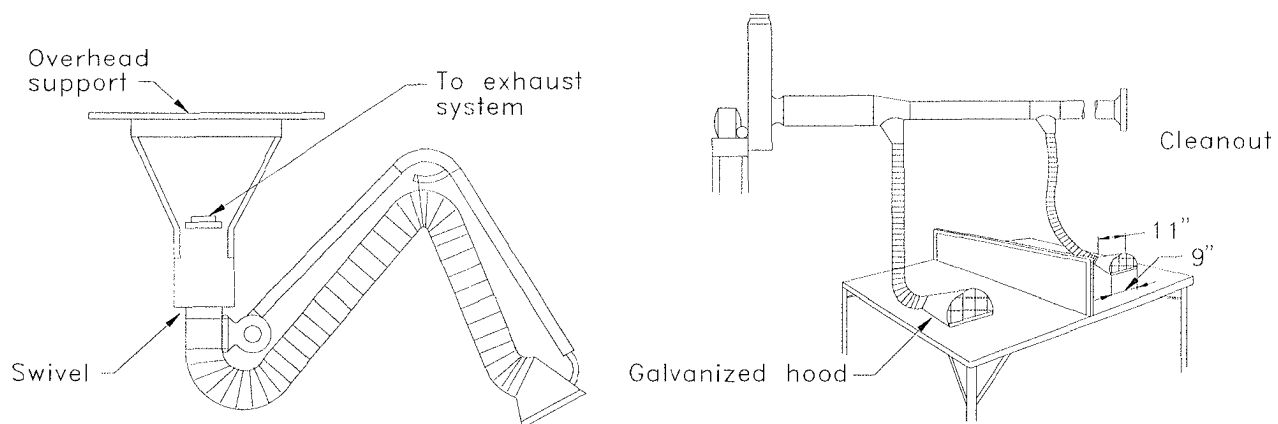
these operations include flame and plasma cutting, foundry pouring, heavy abrasive cutting, and similar operations. An illustration of a traveling exhaust hood is the Hawley Trav-L-Vent (VS-65-03).^(10.65.1)

Telescoping or slip joints are duct sections that overlap, slide, or rotate to allow a section of one exhaust duct to slide into or rotate around another section of duct. This arrangement allows an exhaust hood to be moved from one position to another without disconnecting the duct or hood. Illustrations of the slip joint include the Core Grinder (VS-80-13) and Granite Cutting and Finishing (VS-65-02). This joint or duct section may include a swivel feature to permit rotation of the hood away from the process equipment being exhausted and may be used in the horizontal or vertical plane. Guide rails may be required for the horizontal application while pulleys and counter-weights may be required for vertical applications.

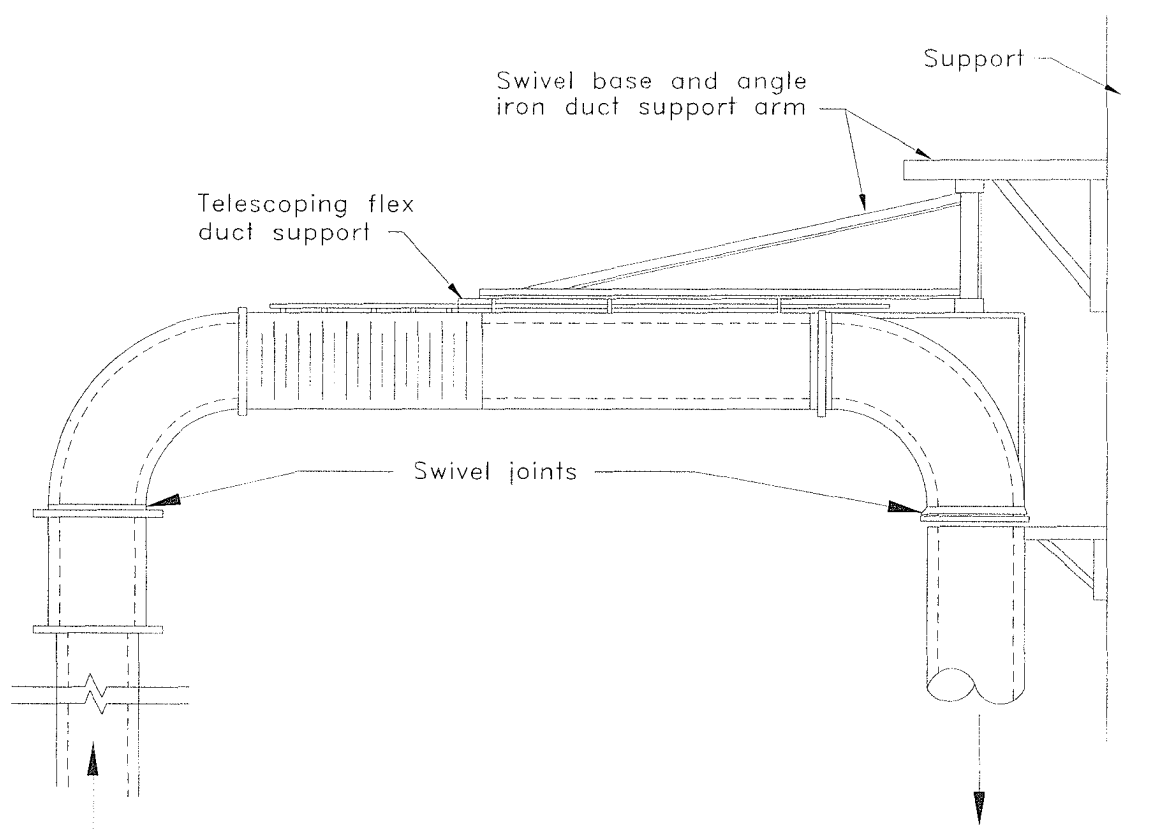
Separating exhaust duct sections is another method of providing a movable exhaust hood. This concept requires that the exhaust duct separate at or near the exhaust hood when the hood is moved for access to the process equipment or the process equipment moves. Illustrations of this method include hoods for Top Electrode Melting Furnace (VS-55-03), Core Making Machine—Small Roll-Over Type (VS-20-11), and Mobile Hood, Die Casting (VS-55-21). Alignment of the exhaust duct when the hood is in place is critical, and any opening at this point should be included in the total exhaust calculations. Also, during the separation period, little or no exhaust control will be available at the contaminant source.

REFERENCES

- 10.65.1 Vulcan Engineering Co.: Hawley Trav-L-Vent Equipment Specifications and Layout. Helena, AL.



FLEXIBLE EXHAUST CONNECTIONS



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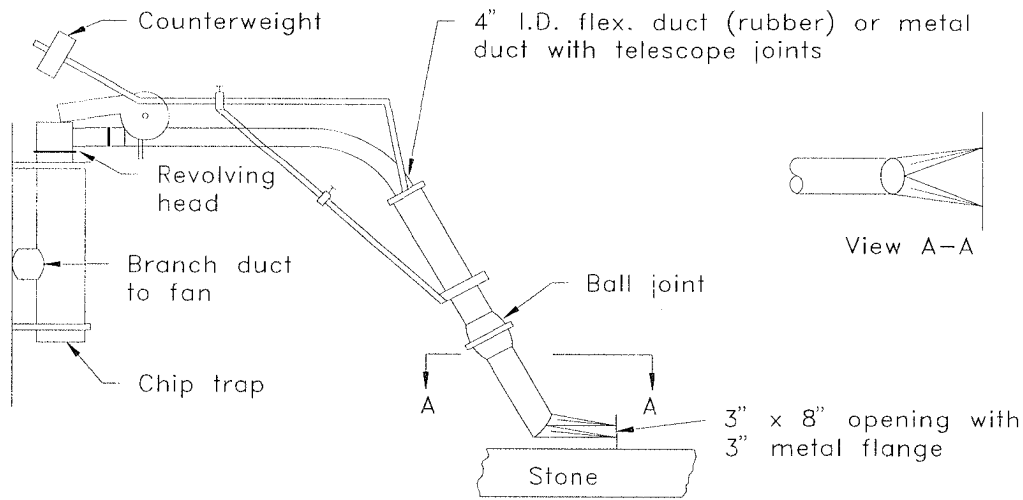
MOVEABLE EXHAUST HOODS

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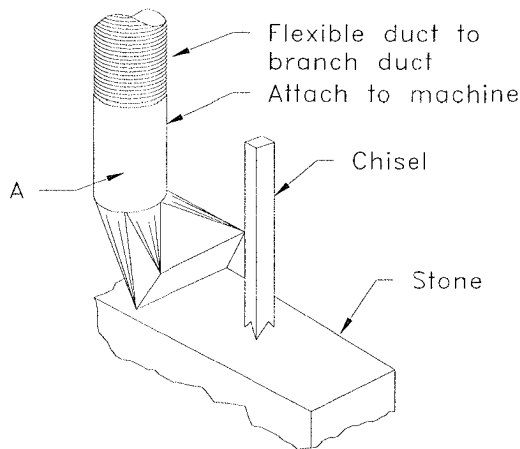
FIGURE

VS-65-01



PNEUMATIC HAND TOOLS

$Q = 400$ cfm minimum, tool 10" max distance from hood
Minimum duct velocity = 4000 fpm



Abrasive blasting to be done in a room or cabinet; 500 fpm at all openings. See "Abrasive Blasting", VS-80-01

SURFACE MACHINE HOODS

Hood	cfm	Branch diam.
Baby surfacer	400	4"
Medium surfacer	600	5"

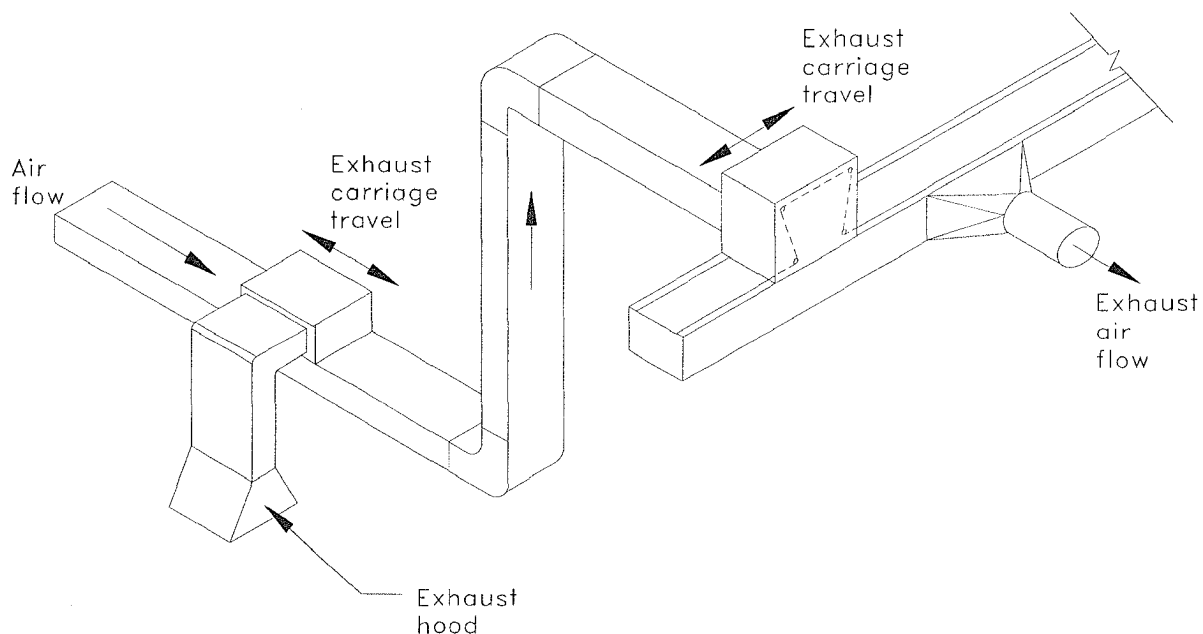
$h_e = 1.0 V_D$ (at point A)

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GRANITE CUTTING
AND FINISHING

DATE 02-91

FIGURE VS-65-02



Typical Design Ranges

$$Q = 2500 - 10,000 \text{ cfm}$$

$$V = 3500 - 4000 \text{ fpm}$$

$$h_e = 6'' \text{ wg approx. total}$$

$$SP_h = \text{calculate separately}$$

Note: This is a patented system with many variations.
Consult manufacturer for applications.

See Reference 10.65.1

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*HAWLEY TRAV-L-VENT
PERSPECTIVE LAYOUT*

DATE

02-91

FIGURE VS-65-03

10.70 OPEN SURFACE TANKS

Ventilation rates for plating, cleaning, and other open surface tank operations will depend on a number of parameters which include materials, tank configuration and location, and type of ventilation system. This section describes four hood/ventilation types: enclosing and canopy hoods, lateral exhaust, and push-pull.

Enclosing hoods usually consist of a lateral hood with one end panel (two sides open) or panels at both tank ends (one side open). This hood configuration can provide increased efficiency by reducing the effects of cross-drafts and by directing more of the hood air flow over the tank open surface.

Canopy hoods may be open on four sides (free standing) or on three sides (such as against a wall). Control is achieved by air flow into the hood. It is, however, difficult in many cases to achieve sufficient control velocity without excessive air flow rates. Canopy hoods should not be used with highly toxic materials, in locations where high cross-drafts are unavoidable, or where the worker must bend over the tank.

Lateral exhaust consists of a slot hood which controls emissions by pulling air across the tank. A single slot may be used on one side of the tank where the tank width is 36 inches or less. For widths greater than 36 inches and where the process configuration will allow, two slot hoods on opposite sides of the tank or a slot hood along the tank centerline may be used. A single slot may be used up to a tank width of 48 inches but only if the material hazard class is low and if cross-drafts are not present. (see Section 10.70.1)

The air flow required will be that necessary to achieve a minimum control velocity determined by the hazard class of the material used for operation and the particular tank/ventilation system configuration. The procedure for determining the class and minimum control velocity for the three preceding hood types is provided in Tables 10.70.1–10.70.7 and the accompanying text. Exhaust flow for a canopy hood is determined from VS-99-03 and for a booth hood from Figure 3-11 where W is the total opening width. The exhaust flow for a lateral hood is determined from Table 10.70.4.

Air and/or mechanical agitation of the tank solution may be used as an aid to the plating or cleaning process. Mechanical agitation creates a rolling motion and usually will not affect tank emissions. However, air agitation creates a boiling-like condition and may significantly increase tank emissions, thus creating need for increased exhaust flow to provide effective control.

Push-pull ventilation consists of a push jet located on one side of a tank with a lateral exhaust hood on the other side. (10.70.1–10.70.3) Tank emissions are controlled by the jet formed over the tank surface. The jet captures the emissions and carries them into the hood. As the jet velocity, at all locations across the tank, is higher than the maximum control velocities specified for canopy, enclosing, or lateral exhaust

hoods (Table 10.70.3), the push-pull exhaust flow is determined on the basis of that necessary to capture the jet flow and is independent of the hazard classification. Push-pull design criteria are provided in VS-70-10, -11 and -12.

10.70.1 Tank Design Considerations:

1. Duct velocity = any desired velocity (see Chapter 3).
2. Entry loss = 1.78 slot VP plus duct entry loss for slot hoods. For canopy or enclosure hoods, entry loss = duct entry loss.
3. Maximum slot hood plenum velocity = 1/2 slot velocity (see Chapter 3).
4. Slot velocity = 2000 fpm unless distribution is provided by well-designed, tapered takeoff.
5. Provide ample area at the small end of the plenum.
6. If $L = 6'$ or greater, multiple takeoffs are desirable. If $L = 10'$ or greater, multiple takeoffs are necessary.
7. Tank width (W) means the effective width over which the hood must pull air to operate (e.g., where the hood face is set back from the edge of the tank, this setback must be added in measuring tank width).

If $W = 20"$, slot on one side is suitable.

If $W = 20 - 36"$, slots on both sides are desirable.

If $W = 36 - 48"$, use slots on both sides or along tank centerline or use push-pull. A single slot along one side should not be used unless all other conditions are optimum.

If $W = 48"$ or greater, local exhaust usually is not practical. Consider using push-pull.

Enclosure can be used for any width tank if process will permit.

It is not practicable to ventilate across the long dimension of a tank whose ratio W/L exceeds 2.0. It is undesirable to do so when W/L exceeds 1.0

8. Liquid level should be 6" to 8" below top of tank with parts immersed.
9. Lateral hood types A, C, D and E (VS-70-01 and -02) are preferred. Plenum acts as baffle to room air currents.
10. Provide removable covers on tank if possible.
11. Provide duct with cleanouts, drains and corrosion-resistant coating if necessary. Use flexible connection at fan inlet.
12. Install baffles to reduce cross-drafts. A baffle is a vertical plate the same length as the tank and with the top of the plate as high as the tank is wide. If the exhaust hood is on the side of the tank against a building wall or close to it, it is perfectly baffled.

13. Replacement air to the tank area must be supplied evenly and directed toward the tank from above or in front of the tank so that cross-drafts do not occur.

Flow Rate Calculation for Good Conditions: (No cross-drafts, adequate and well-distributed replacement air.)

1. Establish process class by determining hazard potential from Tables 10.70.1 and 10.70.2; information from Threshold Limit Values, Solvent Flash Point, Solvent Drying Time Tables in Appendices A and B and Tables 10.70.5–10.70.8.
2. Process class can also be established directly from Tables 10.70.5–10.70.8 if process parameters are known.
3. From Table 10.70.3, choose minimum control velocity according to hazard potential; evolution rate (process class); and hood design (see Table 10.70.5 for typical processes).
4. From Table 10.70.4, select the cfm/ft² for tank dimensions and tank location.
5. Multiply tank area by value obtained from Table 10.70.4 to calculate required air volume.

EXAMPLE

Given: Chrome Plating Tank 6' x 2.5'

High production decorative chrome

Free standing in room

No cross-drafts

- a. Tank Hood. See VS-70-01. Use hood "A" long 6' side. Hood acts as baffle
- b. Component — Chromic Acid (Chromium, metal; water-soluble CrVI)
Hazard potential: A (from Table 10.70.1; from Appendix A: TLV = 0.05 mg/m³; from Appendix A: Flash point = Negligible)
Rate of Evolution: 1 from Table 10.70.2; from Table 10.70.6: Gassing rate = high)
Class: A-1
Control Velocity = 150 fpm (from Table 10.70.3)
Minimum Exhaust Rate = 225 cfm/ft² (from Table 10.70.4; Baffled tank, W/L = 0.42)
Minimum Exhaust Flow Rate = 225 x 15 = 3375 cfm
- c. Hood Design
Design slot velocity = 2000 fpm
Slot area = $Q/V = 3375 \text{ cfm}/2000 \text{ fpm} = 1.69 \text{ ft}^2$
Slot Width = $A/L = 1.69 \text{ ft}^2/6 \text{ ft} = 0.281' = 3.375"$

Example (continued)

$$\text{Plenum depth} = (2)(\text{slot width}) = (2)(3.375) = 6.75"$$

TABLE 10.70.1. Determination of Hazard Potential

Hazard Potential	HYGIENIC STANDARDS		
	Gas and Vapor (see Appendix A)	Mist (see Appendix A)	Flash Point (see Appendix B)
A	0–10 ppm	0–0.1 mg/m ³	—
B	11–100 ppm	0.11–1.0 mg/m ³	Under 100 F
C	101–500 ppm	1.1–10 mg/m ³	100–200 F
D	Over 500 ppm	Over 10 mg/m ³	Over 200 F

TABLE 10.70.2. Determination of Rate of Gas, Vapor, or Mist Evolution

Rate	Liquid Temperature (F)	Degrees Below Boiling Point (F)	Relative Evaporation* (Time for 100% Evaporation)	Gassing**
1	Over 200	0–20	Fast (0–3 hours)	High
2	150–200	21–50	Medium (3–12 hours)	Medium
3	94–149	51–100	Slow (12–50 hours)	Low
4	Under 94	Over 100	Nil (Over 50 hours)	Nil

*Dry Time Relation (see Appendix B). Below 5 — Fast; 5–15 — Medium; 15–75 — Slow; 70–over — Nil.

**Rate of gassing depends on rate of chemical or electrochemical action and therefore depends on the material treated and the solution used in the tank and tends to increase with 1) amount of work in the tank at any one time, 2) strength of the solution in the tank, 3) temperature of the solution in the tank, and 4) current density applied to the work in electrochemical tanks.

$$\text{Duct area} = Q/V = 3375 \text{ cfm}/2500 \text{ fpm} = 1.35 \text{ ft}^2$$

$$\text{Use 16" duct, area} = 1.396 \text{ ft}^2$$

$$\text{Final duct velocity} = Q/A = 3375/1.396 = 2420 \text{ fpm}$$

$$\text{Hood SP} = \text{Entry loss} + \text{Acceleration}$$

$$= 1.78 \text{ VP}_s + 0.25 \text{ VP}_d + 1.0 \text{ VP}_d \text{ (see Chapter 3)}$$

$$= (1.78 \times 0.25") + (0.25 \times 0.37") + 0.37"$$

$$= 0.45 + 0.09 + 0.37$$

$$\text{Hood SP} = 0.91"$$

REFERENCES

- 10.70.1 Huebener, D.J.; Hughes, R.T.: Development of Push-Pull Ventilation. American Industrial Hygiene Assoc. Journal, Vol. 46:262–267 (1985).
- 10.70.2 Hughes, R.T.: Design Criteria for Plating Tank Push-Pull Ventilation. In: Ventilation '85: Proceedings of the First International Symposium on Ventilation for Contaminant Control. Elsevier Press, Amsterdam, the Netherlands (1986).
- 10.70.3 Sciola, V.: Private Communication, Hamilton Standard.

TABLE 10.70.3. Minimum Control Velocity (FPM) for Undisturbed Locations

Class (see Tables 10.70.1 & 10.70.2)	Enclosing Hood		Lateral Exhaust (see VA-70-01 & 70-02) (Note 1)	Canopy Hoods (see Figure 3-8 & VS-99-03)	
	One Open Side	Two Open Sides		Three Open Sides	Four Open Sides
A-1 and A-2 (Note 2)	100	150	150	Do not use	Do not use
A-3 (Note 2), B-1, B-2, and C-1	75	100	100	125	175
B-3, C-2, and D-1 (Note 3)	65	90	75	100	150
A-4 (Note 2) C-3, and D-2 (Note 3)	50	75	50	75	125
B-4, C-4, D-3 (Note 3), and D-4 — Adequate General Room Ventilation Required (see Chapter 2).					

Notes: 1. Use aspect ratio to determine air volume; see Table 10.70.4 for computation.

2. Do not use canopy hood for Hazard Potential A processes.

3. Where complete control of hot water is desired, design as next highest class.

TABLE 10.70.4. Minimum Rate, cfm/ft² of Tank Area for Lateral Exhaust

Required Minimum Control Velocity, fpm (from Table 10.70.3)	cfm/ft ² to maintain required minimum control velocities at following				
	$\frac{\text{tank width } (W)}{\text{tank length } (L)}$ ratios				
	0.0–0.09	0.1–0.24	0.25–0.49	0.5–0.99	1.0–2.0 (Note 2)
Hood against wall or flanged (see Note 1 below and Section 10.70.1, Note 12). See VS-70-01 A and VS-70-02 D and E.					
50	50	60	75	90	100
75	75	90	110	130	150
100	100	125	150	175	200
150	150	190	225	[250] Note 3	[250] Note 3
Hood on free standing tank (see Note 1). See VS-70-01 B and VS-70-02 F.					
50	75	90	100	110	125
75	110	130	150	170	190
100	150	175	200	225	250
150	225	[250] Note 3	[250] Note 3	[250] Note 3	[250] Note 3

Notes: 1. Use W/2 as tank width in computing W/L ratio for hood along centerline or two parallel sides of tank. See VS-70-01 B and C and VS-70-02 F.

2. See Section 10.70.1, Notes 6 and 7.

3. While bracketed values may not produce 150 fpm control velocity at all aspect ratios, the 250 cfm/ft² is considered adequate for control.

TABLE 10.70.5. Typical Processes Minimum Control Velocity (fpm) for Undisturbed Locations

Operation	Contaminant	Hazard	Contaminant Evolution	Lateral Exhaust Control Velocity (See VS-70-01 & VS-70-02)	Collector Recommended
Anodizing Aluminum	Chromic-Sulfuric Acids	A	1	150	X
Aluminum Bright Dip	Nitric + Sulfuric Acids	A	1	150	X
	Nitric + Phosphoric Acids	A	1	150	X
Plating — Chromium	Chromic Acid	A	1	150	X
	Copper Strike	C	2	75	X
Metal Cleaning (Boiling)	Alkaline Mist	C	1	100	X
Hot Water (if vent desired)	Water Vapor	D	2	50*	
		D	1	75*	
Stripping — Copper	Alkaline-Cyanide Mists	C	2	75	X
	Nickel	A	1	150	X
Pickling — Steel	Hydrochloric Acid	A	2	150	X
	Sulfuric Acid	B	1	100	X
Salt Solution					
Bonderizing & Parkerizing)	Water Vapor	D	2	50*	
	Not Boiling	D	2	50*	
	Boiling	D	1	75*	
Salt Baths (Molten)	Alkaline Mist	C	1	100	X

*Where complete control of water vapor is desired, design as next highest class.

TABLE 10.70.6. Airborne Contaminants Released by Metallic Surface Treatment, Etching, Pickling, Acid Dipping and Metal Cleaning Operations

Process	Type	Notes	Component of Bath Which May be Released to Atmosphere (13)	Physical and Chemical Nature of Major Atmospheric Contaminant	Class (12)	Usual Temp. Range F
Surface Treatment	Anodizing Aluminum		Chromic-Sulfuric Acids	Chromic Acid Mist	A-1	95
	Anodizing Aluminum		Sulfuric Acid	Sulfuric Acid Mist	B-1	60-80
	Black Magic		Conc. Sol. Alkaline Oxidizing Agents	Alkaline Mist, Steam	C-1	260-350
	Bonderizing	1	Boiling Water	Steam	D-2,1 (14, 15)	140-212
	Chemical Coloring		None	None	D-4	70-90
	Descaling	2	Nitric-Sulfuric, Hydrofluoric Acids	Acid Mist, Hydrogen Fluoride Gas, Steam	B-2,1 (15)	70-150
	Ebonol		Conc. Sol. Alkaline Oxidizing Agents	Alkaline Mist, Steam	C-1	260-350
	Galvanic-Anodize	3	Ammonium Hydroxide	Ammonia Gas, Steam	B-3	140
	Hard-Coating Aluminum		Chromic-Sulfuric Acids	Chromic Acid Mist	A-1	120-180
	Hard-Coating Aluminum		Sulfuric Acid	Sulfuric Acid Mist	B-1	120-180
	Jetal		Conc. Sol. Alkaline Oxidizing Agents	Alkaline Mist, Steam	C-1	260-350
	Magcote	4	Sodium Hydroxide	Alkaline Mist, Steam	C-3,2 (15)	105-212
	Magnesium Pre-Dye Dip		Ammonium Hydroxide-Ammonium Acetate	Ammonia Gas, Steam	B-3	90-180
	Parkerizing	1	Boiling Water	Steam	D-2,1 (14,15)	140-212
	Zincete Immersion	5	None	None	D-4	70-90
Etching	Aluminum		Sodium Hydroxide-Soda Ash-Trisodium Phosphate	Alkaline Mist, Steam	C-1	160-180
	Copper	6	Hydrochloric Acid	Hydrogen Chloride Gas	A-2	70-90
	Copper	7	None	None	D-4	70
Pickling	Aluminum		Nitric Acid	Nitrogen Oxide Gases	A-2	70-90
	Aluminum		Chromic, Sulfuric Acids	Acid Mists	A-3	140
	Aluminum		Sodium Hydroxide	Alkaline Mist	C-1	140
	Cast Iron		Hydrofluoric-Nitric Acids	Hydrogen Fluoride-Nitrogen Oxide Gases	A-2,1 (15)	70-90
	Copper		Sulfuric Acid	Acid Mist, Steam	B-3,2 (15)	125-175
	Copper	8	None	None	D-4	70-175
	Duralumin		Sodium Fluoride, Sulfuric Acid	Hydrogen Fluoride Gas, Acid Mist	A-3	70
	Inconel		Nitric, Hydrofluoric Acids	Nitrogen Oxide, HF Gases, Steam	A-1	150-165
	Inconel		Sulfuric Acid	Sulfuric Acid Mist, Steam	B-2	160-180
	Iron and Steel		Hydrochloric Acid	Hydrogen Chloride Gas	A-2	70
	Iron and Steel		Sulfuric Acid	Sulfuric Acid Mist, Steam	B-1	70-175
	Magnesium		Chromic-Sulfuric, Nitric Acids	Nitrogen Oxide Gases, Acid Mist, Steam	A-2	70-160
	Monel and Nickel		Hydrochloric Acid	Hydrogen Chloride Gas, Steam	A-2	180
	Monel and Nickel		Sulfuric Acid	Sulfuric Acid Mist, Steam	B-1	160-190
	Nickel Silver		Sulfuric Acid	Acid Mist, Steam	B-3,2 (15)	70-140
	Silver		Sodium Cyanide	Cyanide Mist, Steam	C-3	70-210
	Stainless Steel	9	Nitric, Hydrofluoric Acids	Nitrogen Oxide, Hydrogen Fluoride Gases	A-2	125-180
	Stainless Steel	9,10	Hydrochloric Acid	Hydrogen Chloride Gas	A-2	130-140
	Stainless Steel	9,10	Sulfuric Acid	Sulfuric Acid Mist, Steam	B-1	180
	Stainless Steel Immunization		Nitric Acid	Nitrogen Oxide Gases	A-2	70-120
	Stainless Steel Passivation		Nitric Acid	Nitrogen Oxide Gases	A-2	70-120

TABLE 10.70.6. Airborne Contaminants Released by Metallic Surfaced Treatment, Etching, Pickling, Acid Dipping and Metal Cleaning Operations (con't)

Process	Type	Notes	Component of Bath Which May be Released to Atmosphere (13)	Physical and Chemical Nature of Major Atmospheric Contaminant	Class (12)	Usual Temp. Range F
Acid Dipping	Aluminum Bright Dip		Phosphoric, Nitric Acids	Nitrogen Oxide Gases	A-1	200
	Aluminum Bright Dip		Nitric, Sulfuric Acids	Nitrogen Oxide Gases, Acid Mist	A-2,1 (15)	70–90
	Cadmium Bright Dip		None	None	D-4	70
	Copper Bright Dip		Nitric, Sulfuric Acids	Nitrogen Oxide Gases, Acid Mist	A-2,1 (15)	70–90
	Copper Semi-Bright Dip		Sulfuric Acid	Acid Mist	B-2	70
	Copper Alloys Bright Dip		Nitric, Sulfuric Acids	Nitrogen Oxide Gases, Acid Mist	A-2,1 (15)	70–90
	Copper Matte Dip		Nitric, Sulfuric Acids	Nitrogen Oxide Gases, Acid Mist	A-2,1 (15)	70–90
	Magnesium Dip		Chromic Acid	Acid Mist, Steam	A-2	190–212
	Magnesium Dip		Nitric, Sulfuric Acids	Nitrogen Oxide Gases, Acid Mist	A-2,1 (15)	70–90
	Monel Dip		Nitric, Sulfuric Acids	Nitrogen Oxide Gases, Acid Mist	A-2,1 (15)	70–90
	Nickel and Nickel Alloys Dip		Nitric, Sulfuric Acids	Nitrogen Oxide Gases, Acid Mist	A-2,1 (15)	70–90
	Silver Dip		Nitric Acid	Nitrogen Oxide Gases	A-1	70–90
	Silver Dip		Sulfuric Acid	Sulfuric Acid Mist	B-2	70–90
	Zinc and Zinc Alloys Dip		Chromic, Hydrochloric Acids	Hydrogen Chloride Gas (If HCl attacks Zn)	A-4,3 (15)	70–90
Metal Cleaning	Alkaline Cleaning	11	Alkaline Sodium Salts	Alkaline Mist, Steam	C-2,1 (15)	160–210
	Degreasing		Trichloroethylene-Perchloroethylene	Trichloroethylene-Perchloroethylene Vapors	B (16)	188–250
	Emulsion Cleaning		Petroleum-Coal Tar Solvents	Petroleum-Coal Tar Vapors	B-3,2 (15)	70–140
	Emulsion Cleaning		Chlorinated Hydrocarbons	Chlorinated Hydrocarbon Vapors	(17) (17)	70–140 70–140

Notes: 1 Also Aluminum Seal, Magnesium Seal, Magnesium Dye Set, Dyeing Anodized Magnesium, Magnesium Alkaline Dichromate Soak, Coloring Anodized Aluminum.
 2 Stainless Steel Before Electropolishing.
 3 On Magnesium.
 4 Also Manodysz, Dow-12.
 5 On Aluminum.
 6 Dull Finish.
 7 Ferric Chloride Bath.
 8 Sodium Dichromate, Sulfuric Acid Bath and Ferrous Sulfate, Sulfuric Acid Bath.
 9 Scale Removal.
 10 Scale Loosening.

11 Soak and Electrocleaning.
 12 Class as described in Table 10.70.3 based on hazard potential (Table 10.70.1) and rate of evolution (Table 10.70.2) for usual operating conditions. Higher temperatures, agitation or other conditions may result in a higher rate of evolution.
 13 Hydrogen gas also released by many of these operations.
 14 Rate where essentially complete control of steam is required. Otherwise, adequate dilution ventilation may be sufficient.
 15 The higher rate is associated with the higher value in the temperature range.
 16 For vapor degreasers, rate is determined by operating procedure. See VS-70-20.
 17 Class of operation is determined by nature of the hydrocarbon. Refer to Appendix A.

TABLE 10.70.7. Airborne Contaminants Released by Electropolishing, Electroplating and Electroless Plating Operations

Process	Type	Notes	Component of Bath Which May be Released to Atmosphere (19)	Physical and Chemical Nature of Major Atmospheric Contaminant	Class (18)	Usual Temp. Range F
Electropolishing	Aluminum	1	Sulfuric, Hydrofluoric Acids	Acid Mist, Hydrogen Fluoride Gas, Steam	A-2	140–200
	Brass, Bronze	1	Phosphoric Acid	Acid Mist	B-3	68
	Copper	1	Phosphoric Acid	Acid Mist	B-3	68
	Iron	1	Sulfuric, Hydrochloric, Perchloric Acids	Acid Mist, Hydrogen Chloride Gas, Steam	A-2	68–175
	Monel	1	Sulfuric Acid	Acid Mist, Steam	B-2	86–160
	Nickel	1	Sulfuric Acid	Acid Mist, Steam	B-2	86–160
	Stainless Steel	1	Sulfuric, Hydrofluoric, Chromic Acids	Acid Mist, Hydrogen Fluoride Gas, Steam	A-2,1 (20)	70–300
	Steel	1	Sulfuric, Hydrochloric, Perchloric Acids	Acid Mist, Hydrogen Chloride Gas, Steam	A-2	68–175
Strike Solutions	Copper		Cyanide Salts	Cyanide Mist	C-2	70–90
	Silver		Cyanide Salts	Cyanide Mists	C-2	70–90
	Wood's Nickel		Nickel Chloride, Hydrochloric Acid	Hydrogen Chloride Gas, Chloride Mist	A-2	70–90
Electroless Plating	Copper		Formaldehyde	Formaldehyde Gas	A-1	75
	Nickel	2	Ammonium Hydroxide	Ammonia Gas	B-1	190
Electroplating Alkaline	Platinum		Ammonium Phosphate, Ammonia Gas	Ammonia Gas	B-2	158–203
	Tin		Sodium Stannate	Tin Salt Mist, Steam	C-3	140–170
	Zinc	3	None	None	D-4	170–180
Electroplating Fluoborate	Cadmium		Fluoborate Salts	Fluoborate Mist, Steam	C-3,2 (20)	70–170
	Copper		Copper Fluoborate	Fluoborate Mist, Steam	C-3,2 (20)	70–170
	Indium		Fluoborate Salts	Fluoborate Mist, Steam	C-3,2 (20)	70–170
	Lead		Lead Fluoborate-Fluoboric Acid	Fluoborate Mist, Hydrogen Fluoride Gas	A-3	70–90
	Lead-Tin Alloy		Lead Fluoborate-Fluoboric Acid	Fluoborate Mist	C-3,2 (20)	70–100
	Nickel		Nickel Fluoborate	Fluoborate Mist	C-3,2 (20)	100–170
	Tin		Stannous Fluoborate, Fluoboric Acid	Fluoborate Mist	C-3,2 (20)	70–100
	Zinc		Fluoborate Salts	Fluoborate Mist, Steam	C-3,2 (20)	70–170
Electroplating Cyanide	Brass, Bronze	4,5	Cyanide Salts, Ammonium Hydroxide	Cyanide Mist, Ammonia Gas	B-4,3 (20)	60–100
	Bright Zinc	5	Cyanide Salts, Sodium Hydroxide	Cyanide, Alkaline Mists	C-3	70–120
	Cadmium	5	None	None	D-4	70–100
	Copper	5,6	None	None	D-4	70–160
	Copper	5,7	Cyanide Salts, Sodium Hydroxide	Cyanide, Alkaline Mists, Steam	C-2	110–160
	Indium	5	Cyanide Salts, Sodium Hydroxide	Cyanide, Alkaline Mists	C-3	70–120
	Silver	5	None	None	D-4	72–120
	Tin-Zinc Alloy	5	Cyanide Salts, Potassium Hydroxide	Cyanide, Alkaline Mists, Steam	C-3,2 (20)	120–140
	White Alloy	5,8	Cyanide Salts, Sodium Stannate	Cyanide, Alkaline Mists	C-3	120–150
	Zinc	5,9	Cyanide Salts, Sodium Hydroxide	Cyanide, Alkaline Mists	C-3,2 (7)	70–120

TABLE 10.70.7. Airborne Contaminants Released by Electropolishing, Electroplating and Electroless Plating Operations (con't)

Process	Type	Notes	Component of Bath Which May be Released to Atmosphere (19)	Physical and Chemical Nature of Major Atmospheric Contaminant	Class (18)	Usual Temp. Range F
Electroplating Acid	Chromium		Chromic Acid	Chromic Acid Mists	A-1	90–140
	Copper	10	Copper Sulfate, Sulfuric Acid	Sulfuric Acid Mist	B-4,3 (20,21)	75–120
	Indium	12	None	None	D-4	70–120
	Indium	13,14	Sulfamic Acid, Sulfamate Salts	Sulfamate Mist	C-3	70–90
	Iron		Chloride Salts, Hydrochloric Acid	Hydrochloric Acid Mist, Steam	A-2	190–210
	Iron	12	None	None	D-4	70–120
	Nickel	3	Ammonium Fluoride, Hydrofluoric Acid	Hydrofluoric Acid Mist	A-3	102
	Nickel and Black Nickel	12,15	None	None	C-4 (22)	70–150
	Nickel	9,12	Nickel Sulfate	Nickel Sulfate Mist	B-2	70–90
	Nickel	13,14	Nickel Sulfamate	Sulfamate Mist	C-3	75–160
	Palladium	15	None	None	D-4	70–120
	Rhodium	12,17	None	None	D-4	70–120
	Tin		Tin Halide	Halide Mist	C-2	70–90
	Tin	12	None	None	D-4	70–120
	Zinc		Zinc Chloride	Zinc Chloride Mist	B-3	75–120
	Zinc	12	None	None	D-4	70–120

Notes: 1 Arsine may be produced due to the presence of arsenic in the metal or polishing bath.
 2 Alkaline Bath.
 3 On Magnesium.
 4 Also Copper-Cadmium Bronze.
 5 HCN gas may be evolved due to the acidic action of CO₂ in the air at the surface of the bath.
 6 Conventional Cyanide Bath.
 7 Except Conventional Cyanide Bath.
 8 Albaloy, Spekwhite, Bonwhite (Alloys of Copper, Tin, Zinc).
 9 Using Insoluble Anodes.
 10 Over 90 F.
 11 Mild Organic Acid Bath.
 12 Sulfate Bath.
 13 Sulfamate Bath.

14 Air Agitated.
 15 Chloride Bath.
 16 Nitrite Bath.
 17 Phosphate Bath.
 18 Class as described in Table 10.70.3 based on hazard potential (Table 10.70.1) and rate of evolution (Table 10.70.2) for usual operating conditions. Higher temperatures, agitation, high current density or other conditions may result in a higher rate of evolution.
 19 Hydrogen gas also released by many of these operations.
 20 The higher rate is associated with the higher value in the temperature range.
 21 Baths operated at a temperature of over 140 F with a current density of over 45 amps/ft² and with air agitation will have a higher rate of evolution.
 22 Local exhaust ventilation may be desired to control steam and water vapor.

TABLE 10.70.8. Airborne Contaminants Released by Stripping Operations

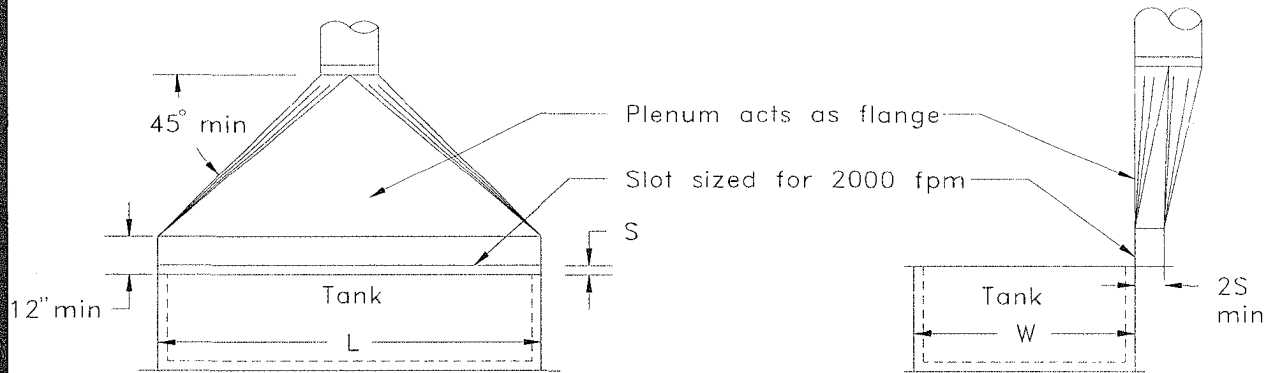
Coating to be Stripped	Base Metal (Footnote)		Component of Batch Which May be Released to Atmosphere (f)	Physical and Chemical Nature of Major Atmospheric Contaminant	Class (e)	Usual Temp. Range F
Anodized Coatings	1,7		Chromic Acid	Acid Mist, Steam	A-2	120–200
Black Oxide Coatings	14		Hydrochloric Acid	Hydrogen Chloride Gas	A-3,2 (g)	70–125
Brass and Bronze	8,14	(a)	Sodium Hydroxide, Sodium Cyanide	Alkaline, Cyanide Mists	C-3,2 (g)	70–90
Cadmium	8,14	(a)	Sodium Hydroxide, Sodium Cyanide	Alkaline, Cyanide Mists	C-3,2 (g)	70–90
	2,4,14		Hydrochloric Acid	Acid Mist, Hydrogen Chloride Gas	A-3,2 (g)	70–90
Chromium	7,8,14	(a)	Sodium Hydroxide	Alkaline Mist, Steam	C-3	70–150
	2,4,8,14		Hydrochloric Acid	Hydrogen Chloride Gas	A-2	70–125
	2,4,8,18	(a)	Sulfuric Acid	Acid Mist	B-2	70–90
Copper	8,14		Sodium Hydroxide, Sodium Cyanide	Alkaline, Cyanide Mists	C-3,2 (g)	70–90
	7,12,14	(b)	None	None	D-4	70–90
	14	(a)	Alkaline Cyanide	Cyanide Mist	C-3,2 (g)	70–160
	1		Nitric Acid	Nitrogen Oxide Gases	A-1	70–120
	18	(a)	Sodium Hydroxide-Sodium Sulfide	Alkaline Mist, Steam	C-2	185–195
Gold	4,5,6,8,9,14	(a)	Sodium Hydroxide, Sodium Cyanide	Alkaline, Cyanide Mists	C-3,2 (g)	70–90
	4,5,18	(a)	Sulfuric Acid	Acid Mist	B-3,2 (g)	70–100
Lead	13	(c)	Acetic Acid, Hydrogen Peroxide	Oxygen Mist	D-3	70–90
	14	(a),(c)	Sodium Hydroxide	Alkaline Mist, Steam	C-3,2 (g)	70–140
Nickel	2,4		Sulfuric, Nitric Acids	Nitrogen Oxide Gases	A-2,1 (g)	70–90
	2,4	(a)	Hydrochloric Acid	Hydrogen Chloride Gas	A-3	70–90
	2,4,14	(a)	Sulfuric Acid	Acid Mist	B-3	70–90
	7		Hydrofluoric Acid	Hydrogen Fluoride Gas	A-3,2 (g)	70–90
	14		Fuming Nitric Acid	Nitrogen Oxide Gases	A-1	70–90
		(a),(d)	Hot Water	Steam	D-2 (h)	200
	1,18,19	(a)	Sulfuric Acid	Acid Mist, Steam	B-3,2 (g)	70–150
Phosphate Coatings	15		Chromic Acid	Acid Mist, Steam	A-3	165
	16		Ammonium Hydroxide	Ammonia Gas	B-3,2 (g)	70–90
Rhodium	10		Sulfuric, Hydrochloric Acids	Acid Mist, Hydrogen Chloride Gas	A-3,2 (g)	70–100
Silver	1		Nitric Acid	Nitrogen Oxide Gases	A-1	70–90
	2,11		Sulfuric, Nitric Acids	Nitrogen Oxide Gases, Steam	A-1	180
	8,14	(a)	Sodium Hydroxide, Sodium Cyanide	Alkaline, Cyanide Mists	C-3	70–90
	17	(a)	Sodium Cyanide	Cyanide Mist	C-3	70–90
Tin	2,3,4		Ferric Chloride, Copper Sulfate Acetic Acid	Acid Mist	B-4,3 (g)	70–90
		(a)	Sodium Hydroxide	Alkaline Mist	C-3	70–90
	2,14		Hydrochloric Acid	Hydrogen Chloride Gas	A-3,2 (g)	70–90
	14	(a)	Sodium Hydroxide	Alkaline Mist, Steam	C-2	70–200

TABLE 10.70.8. Airborne Contaminants Released by Stripping Operations (con't)

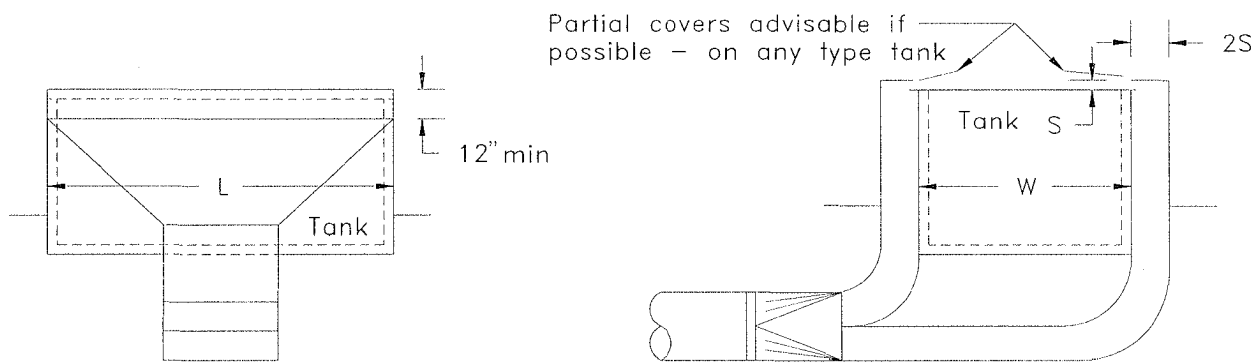
Coating to be Stripped	Base Metal (Footnote)	Component of Batch Which May be Released to Atmosphere (f)	Physical and Chemical Nature of Major Atmospheric Contaminant	Class (e)	Usual Temp. Range F
Zinc	1 8,14	Nitric Acid Sodium Hydroxide, Sodium	Nitrogen Oxide Gases Alkaline, Cyanide Mists	A-1 C-3	70-90 70-90
Base Metal:	1. Aluminum 2. Brass 3. Bronze 4. Copper 5. Copper Alloys 6. Ferrous Metals 7. Magnesium	8. Nickel 9. Nickel Alloys 10. Nickel Plated Brass 11. Nickel Silver 12. Non-Ferrous Metals 13. Silver	14. Steel 15. Steel (Manganese Type Coatings) 16. Steel (Zinc Type Coatings) 17. White Metal 18. Zinc 19. Zinc Base Die Castings		

Notes: (a) Electrolytic Process.
 (b) Refers only to steel (14) when Chromic, Sulfuric Acids Bath is used.
 (c) Also Lead Alloys.
 (d) Sodium Nitrate Bath.
 (e) Class as described in Table 10.70. 3 based on hazard potential (Table 10.70.1) and rate of evolution (Table 10.70.2) for usual operating conditions. Higher temperatures, agitation or other conditions may result in a higher rate of evolution.

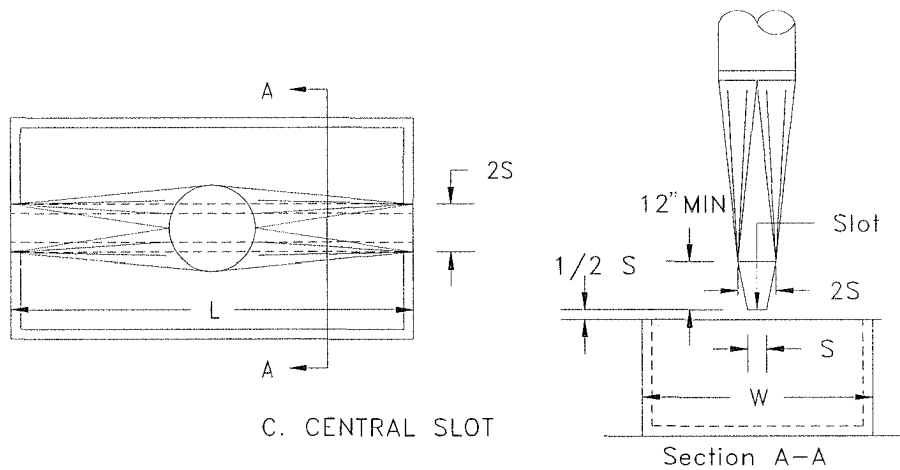
(f) Hydrogen gas also released by some of these operations.
 (g) The higher rate is associated with the higher value in the temperature range.
 (h) Rate where essentially complete control of steam is required. Otherwise, adequate dilution ventilation may be sufficient.



A. UPWARD PLENUM



B. DOWNWARD PLENUM



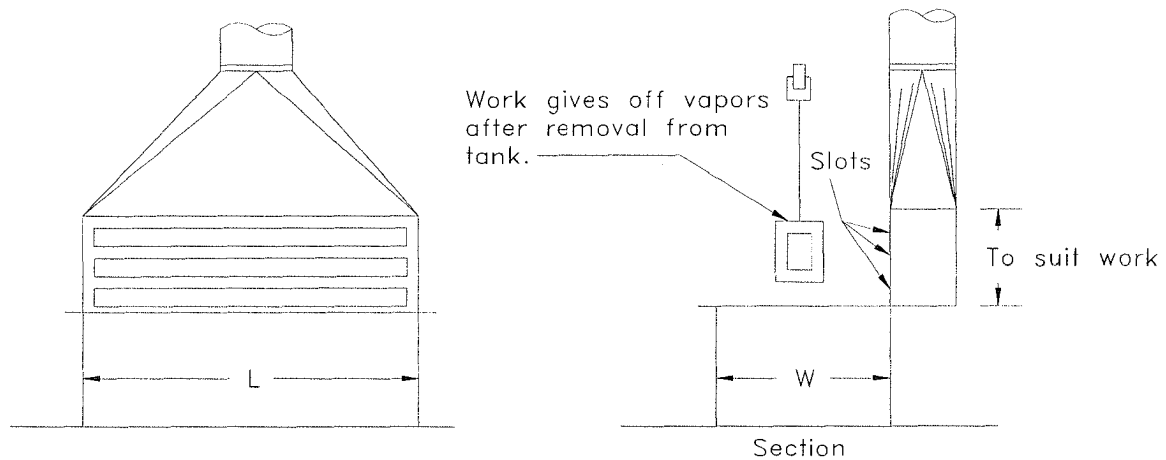
C. CENTRAL SLOT

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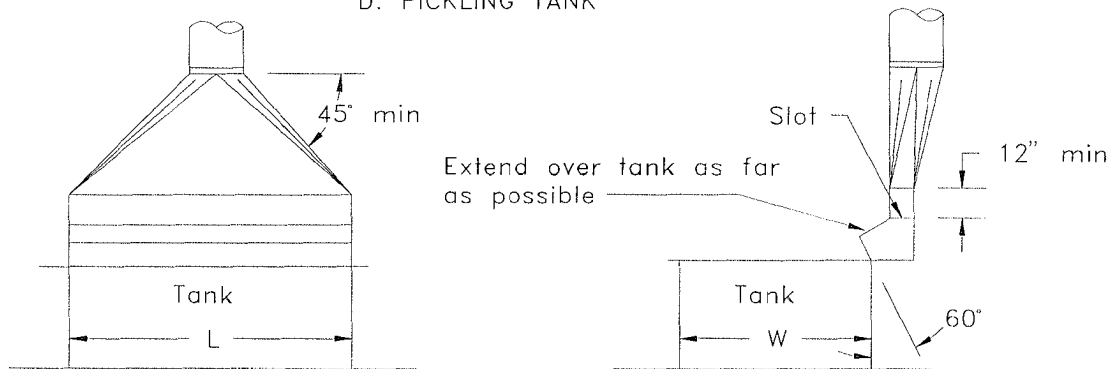
OPEN SURFACE TANKS

DATE 12-90

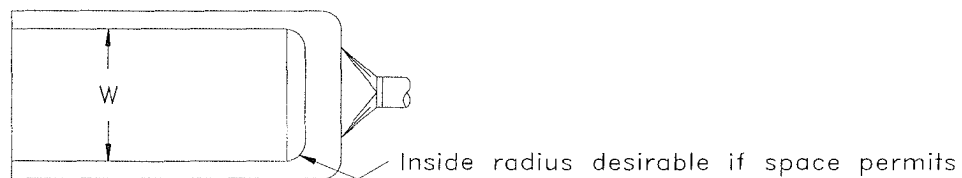
FIGURE VS-70-01



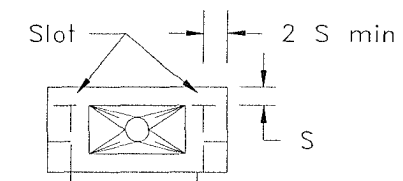
D. PICKLING TANK



E. LATERAL



F. END TAKE-OFF

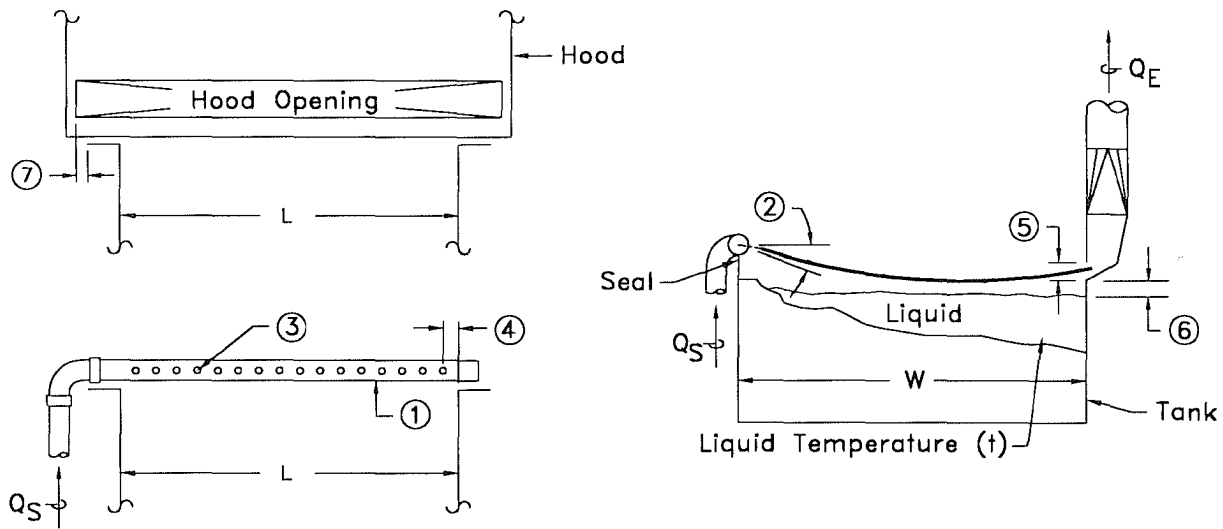


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OPEN SURFACE TANKS

DATE 12-90

FIGURE VS-70-02



Push nozzle manifold (1) - Circular, rectangular or square. Manifold cross-sectional area should be at least 2.5 times the total nozzle flow area.

Push nozzle angle (2) - 0° to 20° down.

Nozzle openings (3) - 1/8" to 1/4" slot or 5/32" to 1/4" dia. holes with 3 to 8 dia. spacing. Outer holes or slot ends (4) must be 1/2" to 1" inside tank inner edges.

Exhaust opening (5) - Size to achieve 2000 fpm slot velocity. Outer edges of opening (7) must extend to edge of tank including flanges.

Liquid surface (6) - Tank freeboard must not exceed 8" with parts removed.

Push nozzle supply $Q_j = 243 \sqrt{A_j}$
 where Q_j = push nozzle supply, cfm/ft manifold length
 A_j = total nozzle opening per foot of manifold length

Total push supply $Q_s = Q_j \times L$ cfm

Exhaust flow $Q_E = 75 \text{ cfm/ft}^2$ tank surface area for $t \leq 150 \text{ F}$
 $Q_E = (0.4 T + 15) \text{ cfm/ft}^2$ tank surface area for $t > 150 \text{ F}$.

Tank surface area = L (length of tank) x W (width of tank)

Design Procedure: Select nozzle opening within above limits and calculate push supply and exhaust air flow. See VS-70-11 and VS-70-12.

Reference 10.70.1, 10.70.2, & 10.70.3

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PUSH-PULL HOOD DESIGN
 DATA FOR WIDTHS UP TO 10'

DATE 4-94

FIGURE VS-70-10

In push-pull ventilation, a nozzle pushes a jet of air across the vessel surface into an exhaust hood. Effectiveness of a push jet is a function of its momentum which can be related to the product of the nozzle supply air flow (Q_j) and the nozzle exit velocity (V_j). For a jet used for plating tanks or other open surface vessels, a push supply flow can be determined from:

$$Q_j = 243\sqrt{A_j}$$

where: Q_j = push nozzle supply, cfm per foot of push nozzle plenum length

A_j = nozzle exit area, ft²/per foot of push nozzle plenum length

Using this approach, a push nozzle design is first selected and the nozzle exit area (A_j) determined.

The push nozzle manifold may be round, rectangular or square in cross-section. The push nozzle may be a 1/8" to 1/4" horizontal slot or 5/32" to 1/4" diameter drilled holes on 3 to 8 diameter spacing.

It is important that the air flow from the nozzle be evenly distributed along the length of the supply plenum. To achieve this, the total nozzle exit area should not exceed 40% of the plenum cross-sectional area. Multiple supply plenum inlets should be used where practical.

The push nozzle manifold should be located as near the vessel edge as possible to minimize the height above the liquid surface. The manifold should be adjustable to optimize the push jet angle. The manifold axis can be angled down a maximum of 20° to permit the jet to clear obstructions and to maintain the jet at the vessel surface. It is essential any opening between the manifold and tank be sealed.

An exhaust flow of 75 cfm/ft² of vessel surface area should be used for tank liquid temperatures (t) of 150 F or lower. For tank liquid temperatures greater than 150 F use an exhaust flow of $(0.4t + 15)$ cfm/ft². These flow rates are independent of the "class" used in determining exhaust flow for side draft hoods. "Control velocity" is achieved by the push jet blowing over the tank and will be considerably higher than that which can be achieved by a side draft hood. The purpose of the exhaust hood is to capture and remove the jet—not to provide capture velocity. A flanged hood design is to be used wherever practical. The exhaust hood should be located at the vessel edge so as not to leave a gap between the hood and the vessel.

Design and location of an open surface vessel encompasses a number of variables. In some cases vessel shape, room location, cross-drafts, etc., may create conditions requiring adjustment of the push and/or pull flow rates in order to achieve effective control. Cross-draft velocities over 75 fpm, very wide vessels (eight feet or more), or very large or flat surface parts may require increased push and/or pull flows. To account for the effects of these variables, a flow adjustment of $\pm 20\%$ should be designed into the push and $+20\%$ into the pull flow system. Wherever practical, construction and evaluation of a pilot system is recommended. Once designed and installed, push-pull systems can be initially evaluated by use of a visual tracer technique and appropriate flow adjustments can be made as required.

The exhaust hood opening should be sized to assure even flow distribution across the opening. This can be achieved by sizing the slot for 2000 fpm slot velocity.

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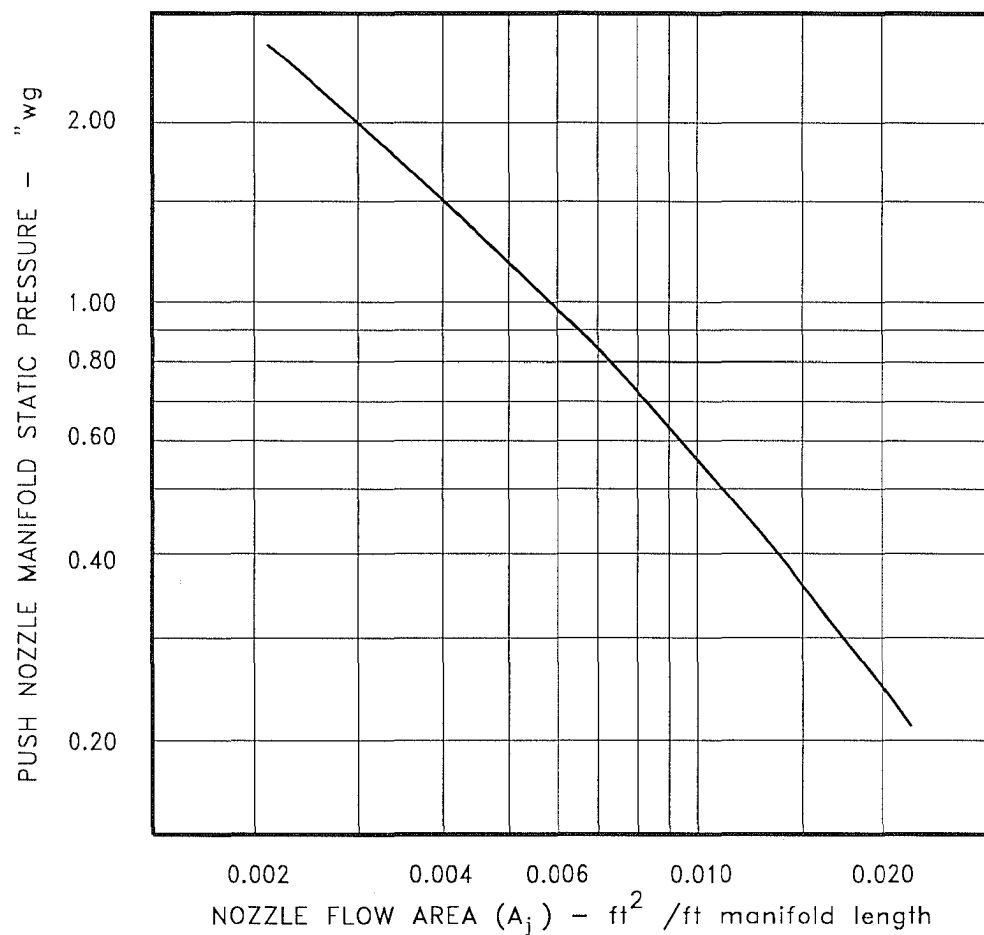
DESIGN DATA
PUSH-PULL HOOD

DATE

4-94

FIGURE

VS-70-11



PUSH NOZZLE SUPPLY = $243\sqrt{A_j}$ cfm/ft of length for nozzles with 1/8 to 1/4 inch wide slots or 5/32 to 1/4 inch diameter holes on 3 to 8 diameter spacing.

For holes $A_j (\text{ft}^2/\text{ft}) = 0.065 \times \text{hole diameter (in)}/\text{hole spacing (no. of diameters (in).)}$

For slot $A_j (\text{ft}^2/\text{ft}) = \frac{\text{slot width (in)}}{12}$ (See VS-70-10)

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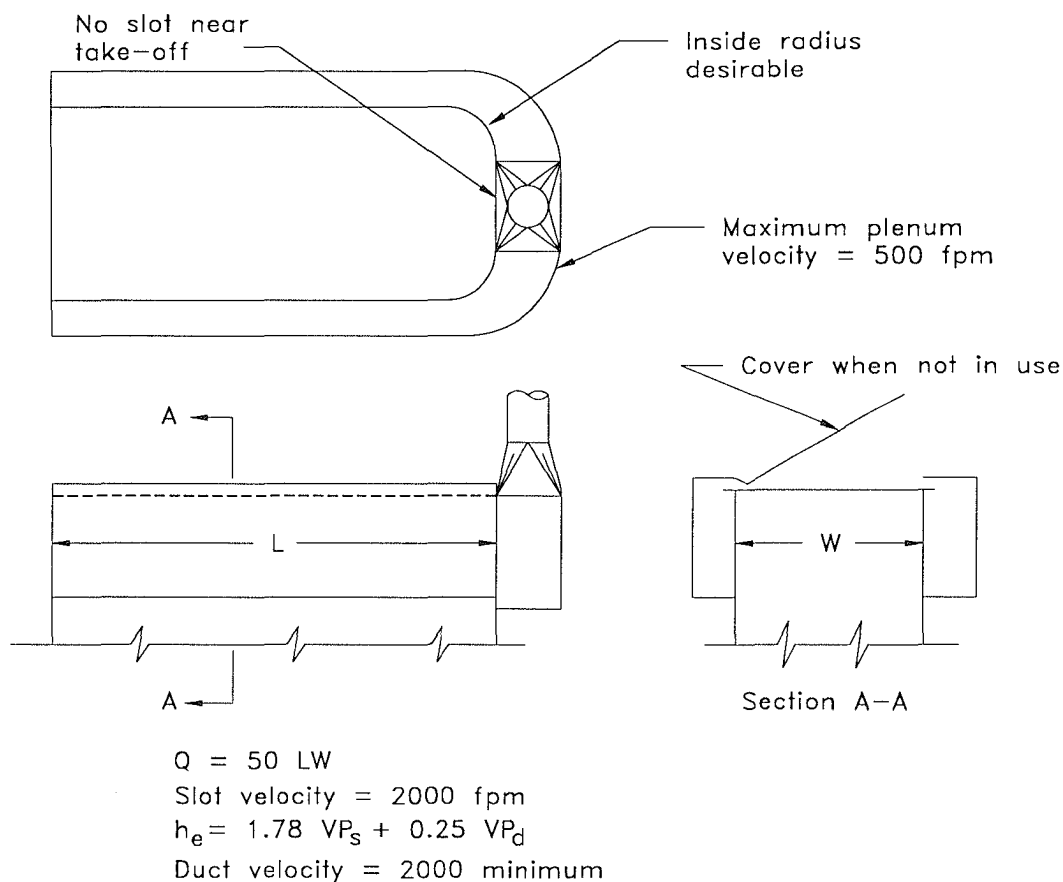
*PUSH NOZZLE
MANIFOLD PRESSURE*

DATE

4-94

FIGURE

VS-70-12



- Also provide:
1. Separate flue for combustion products
 2. For cleaning operation, appropriate respiratory protection is necessary.
 3. For pit units, the pit should be mechanically ventilated.
 4. For further safeguards, see VS-501.1

NOTE: Provide downdraft grille for parts that cannot be removed dry; $Q = 50 \text{ cfm/ft}^2$ grille area.

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SOLVENT DEGREASING TANKS

DATE 12-90

FIGURE VS-70-20

Solvent vapor degreasing refers to boiling liquid cleaning systems utilizing trichloroethylene, perchlorethylene, methylene chloride, freons® or other halogenated hydrocarbons. Cleaning action is accomplished by the condensation of the solvent vapors in contact with the work surface producing a continuous liquid rinsing action. Cleaning ceases when the temperature of the work reaches the temperature of the surrounding solvent vapors. Since halogenated hydrocarbons are somewhat similar in their physical, chemical and toxic characteristics, the following safeguards should be provided to prevent the creation of a health or life hazard:

1. Vapor degreasing tanks should be equipped with a condenser or vapor level thermostat to keep the vapor level below the top edge of the tank by a distance equal to one-half the tank width or 36 inches, whichever is shorter.
2. Where water type condensers are used, inlet water temperature should not exceed 80 F and the outlet temperature should not exceed 110 F. For some solvents, lower water temperatures may be required.
3. Degreasers should be equipped with a boiling liquid thermostat to regulate the rate of vapor generation, and with a safety control at an appropriate height above the vapor line to prevent the escape of solvent in case of a malfunction.
4. Tanks or machines of more than 4 square feet of vapor area should be equipped with suitable gasketed cleanout or sludge doors, located near the bottom, to facilitate cleaning.
5. Work should be placed in and removed slowly from the degreaser, at a rate no greater than 11 fpm, to prevent sudden disturbances of the vapor level.
6. CARE MUST BE TAKEN TO PREVENT DIRECT SOLVENT CARRYOUT DUE TO THE SHAPE OF THE PART.
7. Maximum rated workloads as determined by the rate of heat transfer (surface area and specific heat) should not be exceeded.
8. Special precautions should be taken where natural gas or other open flames are used to heat the solvent to prevent vapors* from entering the combustion air supply.
9. Heating elements should be designed and maintained so that their surface temperature will not cause the solvent or mixture to breakdown* or produce excessive vapors.
10. Degreasers should be located in such a manner that vapors* will not reach or be drawn into atmospheres used for gas or electric arc welding, high temperature heat treating, combustion air or open electric motors.
11. Whenever spray or other mechanical means are used to disperse solvent liquids, sufficient enclosure or baffling should be provided to prevent direct release of airborne vapor above the top of the tank.
12. An emergency quick-drenching facility should be located in near proximity to the degreaser for use in the event of accidental eye contact with the degreasing liquid.

*Electric arcs, open flames and hot surfaces will thermally decompose halogenated hydrocarbons to toxic and corrosive substances (such as hydrochloric and/or hydrofluoric acid). Under some circumstances, phosgene may be formed.

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SOLVENT VAPOR DEGREASING

DATE

1-91

FIGURE

VS-70-21

10.75 PAINTING OPERATIONS

Application of industrial paints and coatings usually is accomplished by one of three techniques: air-atomization, electrostatics, or airless methods. Potential health hazards exist from exposure to solid and liquid aerosols as well as to solvent vapors. In addition to the airborne exposures, hazards include the use of flammable and combustible liquids and the accumulation of flammable paint residues. Fire safety and proper electrical wiring are important concerns in most paint applications.^(10.75.1, 10.75.2)

Control of airborne pollutants by ventilation may be accomplished through the use of spray booths such as shown in VS-75-01 and VS-75-04. The typical booth is a partial enclosure of sheet metal construction with openings for conveying the work piece into and out of the booth. Several factors are important in the performance of these booths. Booth depth is critical; spray rebound may escape from shallow booths and increase exposures. The size of the booth is governed principally by the size of the object being coated. Sufficient space must be provided to permit air flow on all sides of the object, to provide room to work, and to enable the air to enter the booth in a smooth, controlled manner without excessive wrap-around. In some cases, downdraft booths may be employed when large objects are painted.

In air-atomization applications, the most common spray technique, it is important to use the minimum air pressure needed to accomplish the task. Excess air pressure results in increased dispersion of the paint and overspray as well as poor work quality.

Airless application results in aerosols with fewer particles in the respirable range. One study^(10.75.3) suggests that approximately 20% of the particles in air-atomization applications are less than 12 microns while airless methods produce aerosols with only 2% less than this value. The larger aerosols produced by the airless technique will deposit more efficiently on the work piece, due to impaction, than the smaller particles produced by the compressed air method.

Electrostatic applications result in more efficient deposition of paint aerosols due to electrostatic forces. As a result, ventilation air flow requirements for control of electrostatic applications tend to be lower than for compressed air methods.

Many spray booths are equipped with disposable particu-

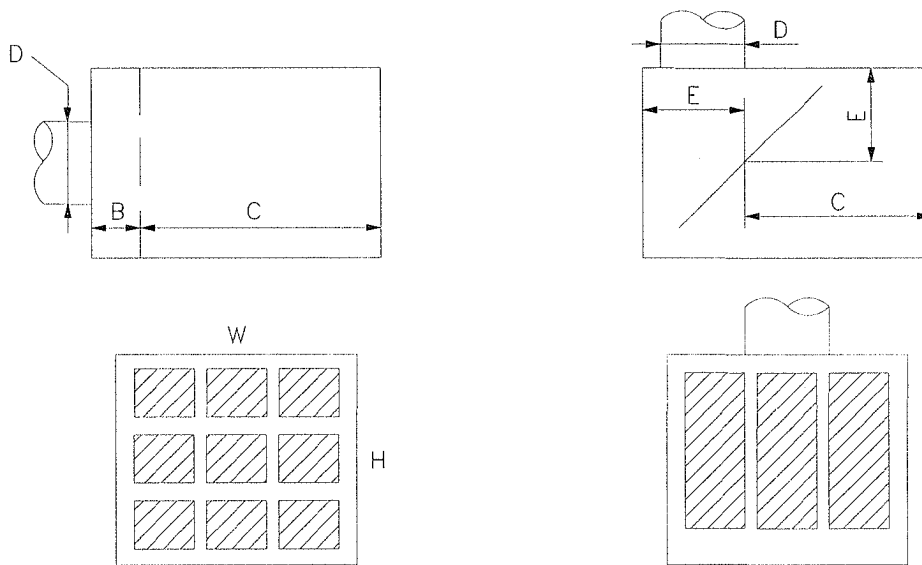
late filters which become loaded over time and result in increased pressure loss. This loss eventually can reduce air flow to unacceptable levels and, hence, system performance must be monitored. Water wash systems are available for cleaning particulate matter from the exhausted air but do little for solvent vapors. Fan selection is an important component of a spray booth installation. Often the fan is an integral part of the system when purchased and may be installed in a different configuration than originally designed. This can result in reduced air flow, particularly if additional system resistance is encountered in the actual installation.^(10.75.4)

Work practices remain an important aspect of controlling exposure to paint aerosols and solvent vapors. The worker should not stand downstream of the object being sprayed. A turntable can help to facilitate easy access to all sides of the object without the worker having to move. Extension arms on spray guns should be employed for hard-to-reach cavities. Proper location of the booth with respect to replacement air and obstructions is essential. Locating booths in corners or near disruptive air currents can defeat the protection of these hoods. Poor location of the booth may result in turbulent air flow which may reduce the protection provided by the booth.

Respiratory protection may be required in applications using toxic materials. This includes heavy metal pigments and organics such as isocyanates in urethane paints and amines in epoxy paints.

REFERENCES

- 10.75.1 National Fire Protection Association: Flammable and Combustible Liquids Code. No. 30. NFPA. Boston, MA (1990).
- 10.75.2 National Fire Protection Association: National Electric Code. No. 70. NFPA. Boston, MA, (1990).
- 10.75.3 National Institute for Occupational Safety and Health: An Evaluation of Engineering Control Technology for Spray Painting. DHHS (NIOSH) Pub. No. 81-121; NTIS Pub. No. PB-82-162-264. National Technical Information Service, Springfield, VA (1981).
- 10.75.4 Burgess, W.A.; Ellenbecker, M.J.; Treitman, R.D.: Ventilation for Control of the Work Environment. John Wiley and Sons, NY (1989).



1. Split baffle or filters

$$B = 0.75 D$$

$$\text{Baffle area} = 0.75 WH$$

For filter area, see note 2

2. Angular baffle

$$E = D + 6''$$

$$\text{Baffle area} = 0.40 WH$$

For filter area, see note 2

Air spray paint design data.

Any combination of duct connections and baffles may be used. Large, deep booths do not require baffles. Consult manufacturers for water-curtain designs. Use explosion proof fixtures and a non-sparking fan. Electrostatic spray booth requires automatic high-voltage disconnects for conveyor failure, fan failure or grounding.

Walk-in booth

$$W = \text{work size} + 6'$$

$$H = \text{work size} + 3' \text{ (minimum} = 7')$$

$$C = \text{work size} + 6'$$

$$Q = 100 \text{ cfm/ft}^2 \text{ booth cross section}$$

May be 75 cfm/ft² for very large, deep booth. Operator may require a NIOSH certified respirator.

$$h_e = 1.78 VP_s + 0.50 VP_d \text{ (baffles)}$$

$$h_e = \text{Dirty filter resistance} + 0.50 VP_d \text{ (filters)}$$

$$\text{Duct velocity} = 2000 \text{ fpm}$$

Operator outside booth

$$W = \text{work size} + 2'$$

$$H = \text{work size} + 2'$$

$$C = 0.75 \times \text{larger front dimension}$$

$$Q = 100 - 150 \text{ cfm/ft}^2 \text{ of open area, including conveyor openings.}$$

Airless spray paint design

$$Q = 60 \text{ cfm/ft}^2 \text{ booth cross section, walk-in booth}$$

$$Q = 60 - 100 \text{ cfm/ft}^2 \text{ of total open area, operator outside of booth}$$

Notes : 1. Baffle arrangements shown are for air distribution only.

2. Paint arresting filters usually selected for 100 - 500 fpm, consult manufacturer for specific details.

3. For construction and safety, consult NFPA - See Reference 10.75.1.

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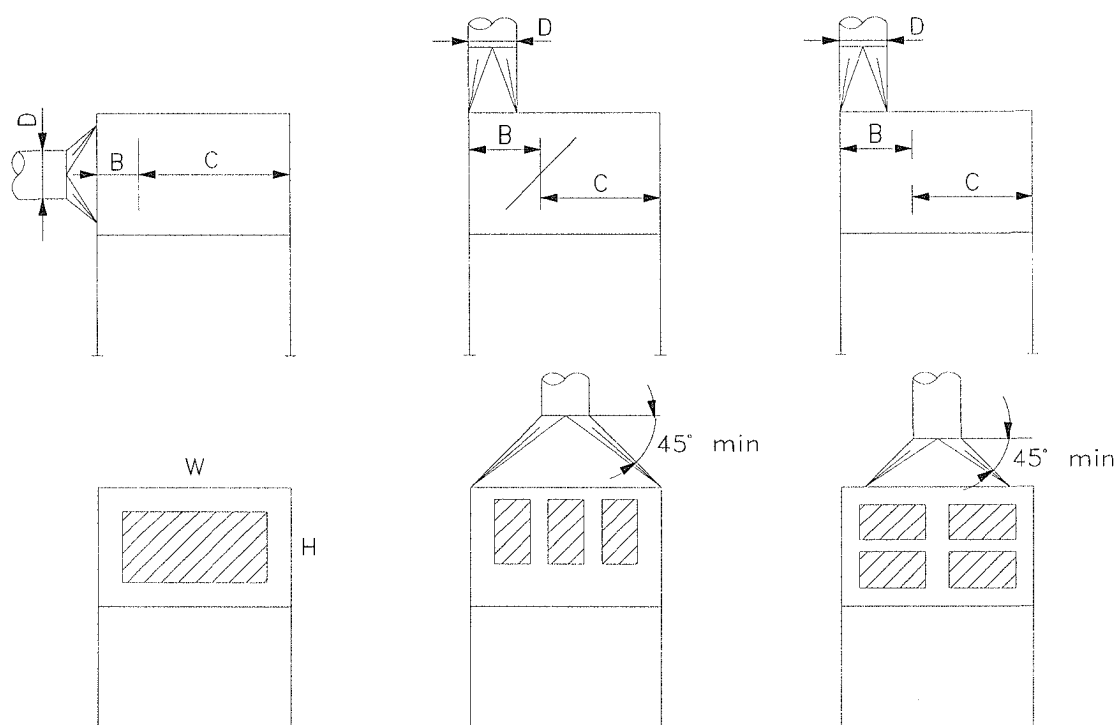
LARGE PAINT BOOTH

DATE

1-91

FIGURE

VS-75-01



1. Solid baffle
 $B = 0.75 D$
 Baffle area = $0.60 WH$
2. Angular baffle
 $B = D + 6"$
 Baffle area = $0.60 WH$
3. Split baffle or filters
 $B = D + 6"$
 Baffle area = $0.75 WH$

Air spray design data

Any combination of branch ducts and baffles may be used.

W = work size + 12"

H = work size + 12"

C = $0.75 W$ or H , whichever is larger

Q = 200 cfm/ft^2 ($200 WH$) – for face area up to 4 ft^2

= 150 cfm/ft^2 – for face area over 4 ft^2

$h_e = 1.78 VP_s + 0.25 VP_d$ (baffles)

= dirty filter resistance + $0.25 VP_d$ (filters)

Duct velocity = 2000 fpm

Airless spray paint design data

Q = 125 cfm/ft^2 ($125 WH$) – for face area up to 4 ft^2

= 100 cfm/ft^2 – for face area over 4 ft^2

- Notes:
1. Baffle arrangements shown are for air distribution only.
 2. Paint arresting filters usually selected for $100 - 500 \text{ fpm}$, consult manufacturer for specific details.
 3. For construction and safety, consult NFPA (Reference 10.75.1).

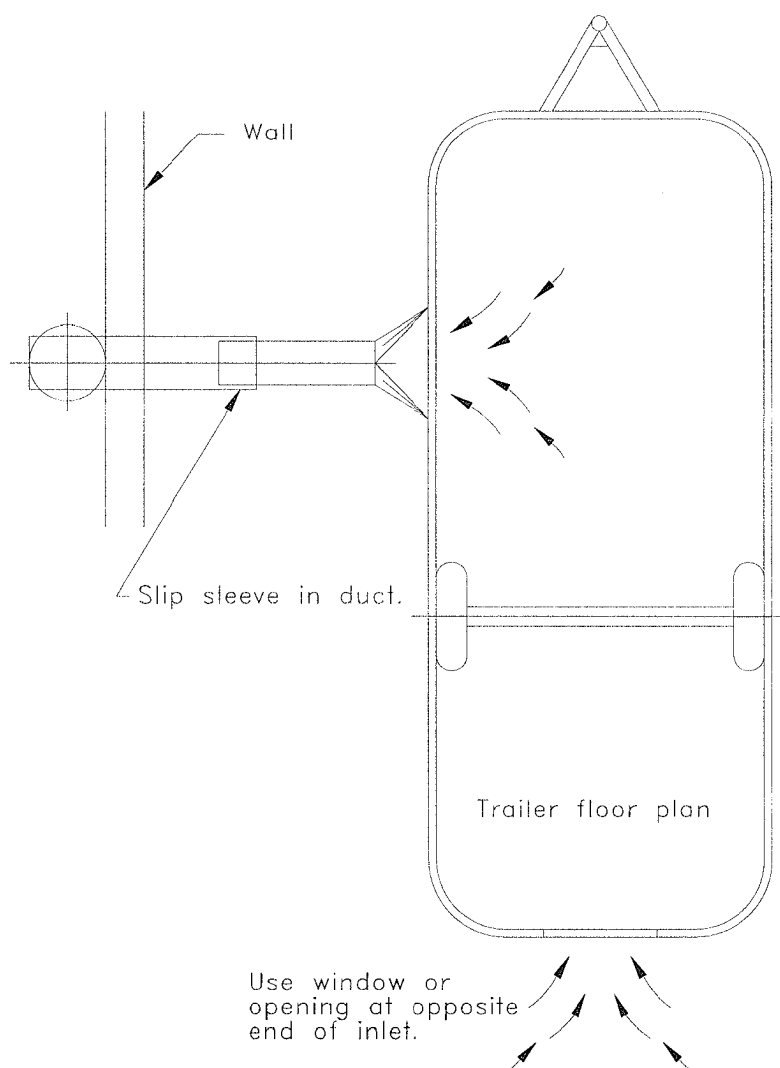
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SMALL PAINT BOOTH

DATE

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FIGURE VS-75-02



$Q = 50 \text{ cfm/ft}^2$ of cross-sectional trailer area
 $h_e = 0.25 \text{ VR}_d$
 Minimum duct velocity = 2000 fpm

Notes:

1. Paint arresting filters usually selected for 100–500 fpm, consult manufacturer for specific details.
2. For construction and safety, consult NFPA, Reference 10.75.1.
3. Operator must wear an appropriate, NIOSH certified respirator.

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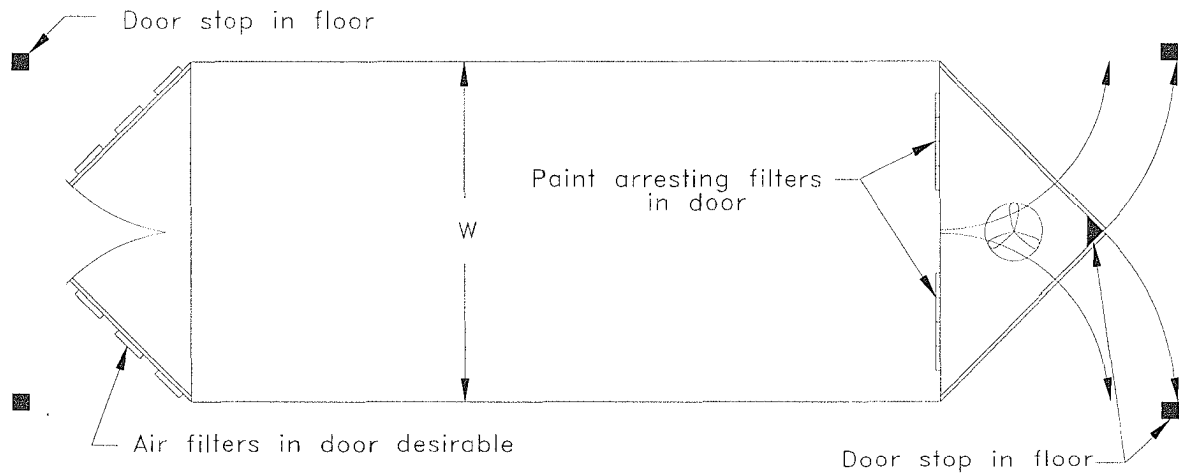
*TRAILER INTERIOR
SPRAY PAINTING*

DATE

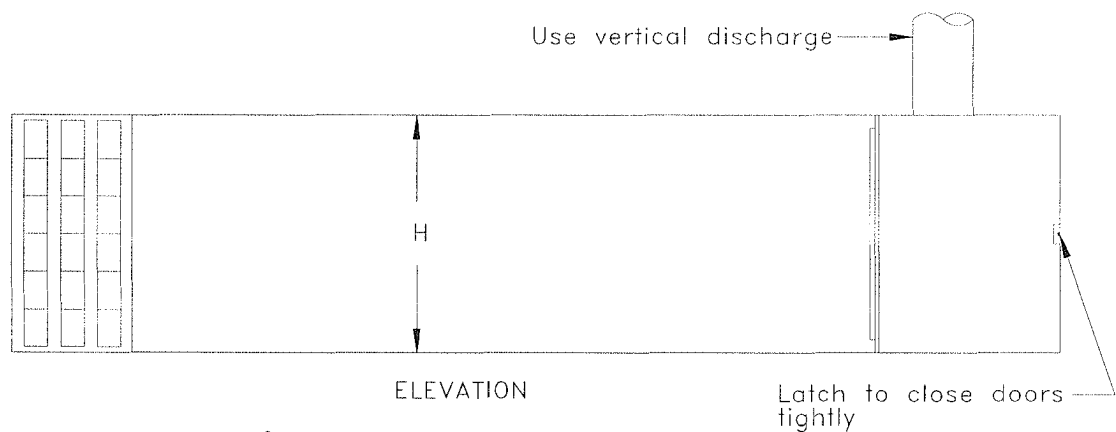
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FIGURE

VS-75-03



PLAN VIEW



ELEVATION

$Q = 100 \text{ cfm/ft}^2$ of cross sectional area
 (When $W \times H$ is greater than 150 ft^2 , $Q = 50 \text{ cfm/ft}^2$)
 $h_e = 0.50 \text{ VP}_d$ plus resistance of each filter bank when dirty
 Minimum duct velocity = 2000 fpm

Notes:

1. Exhaust fan interlock with make-up air supply and compressed air to spray gun is desirable.
2. Paint arresting filters usually selected for 100-500 fpm. Consult manufacturer for specific details.
3. For construction and safety, consult NFPA, Reference 10.75.1.
4. For airless spray painting use
 $Q = 60 \text{ cfm/ft}^2$ of cross section area.

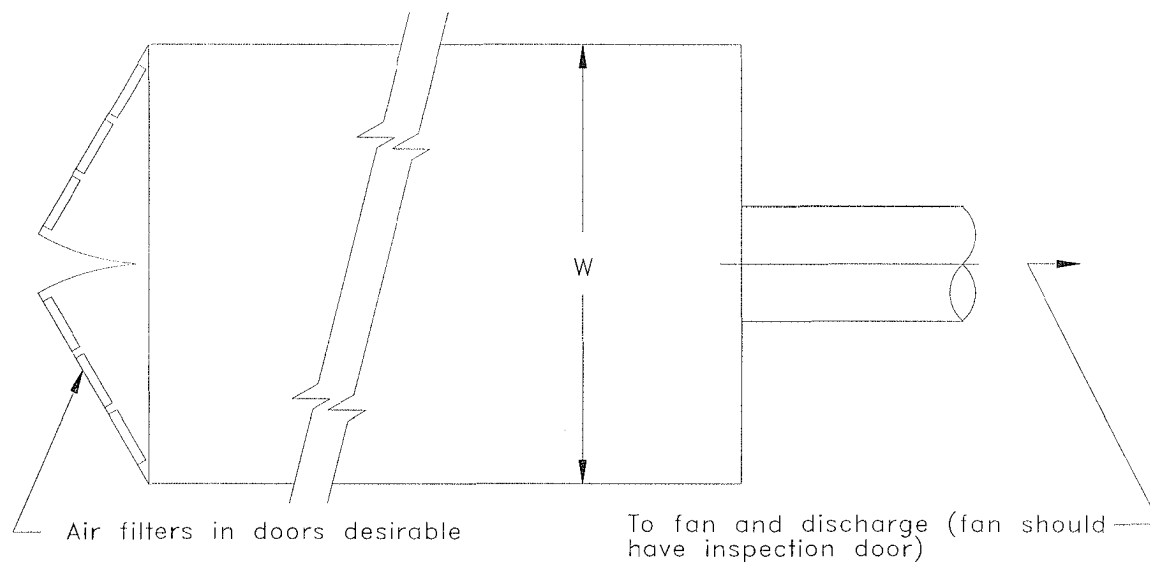
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LARGE DRIVE-THROUGH
 SPRAY PAINT BOOTH

DATE

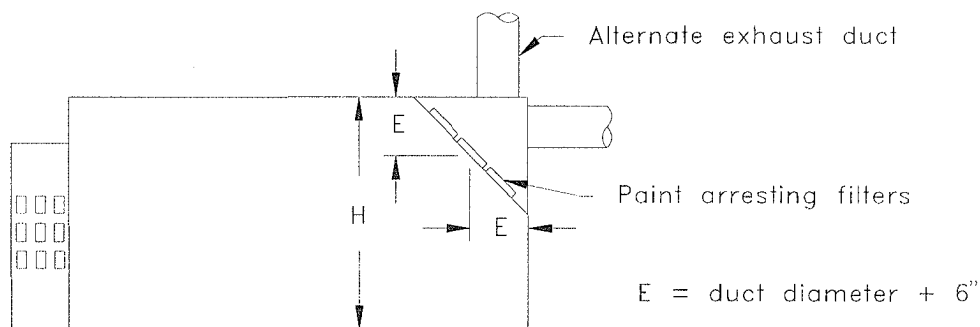
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FIGURE VS-75-04



PLAN VIEW

$Q = 100 \text{ cfm/ft}^2$ of cross sectional area
 (when $W \times H$ is greater than 150 ft^2 , $Q = 50 \text{ cfm/ft}^2$)
 $h_e = 0.50 VP_d$ plus resistance of each filter bank when dirty
 Minimum duct velocity = 2000 fpm
 Paint arresting filters to be sized for 100–500 cfm/ft²
 of filter. Consult manufacturer for specific details.



ELEVATION

Typical filter installation

Note: For airless spray painting use
 $Q = 60 \text{ cfm/ft}^2$ of cross-sectional area

For construction and safety
 consult NFPA code (Reference 10.75.1).

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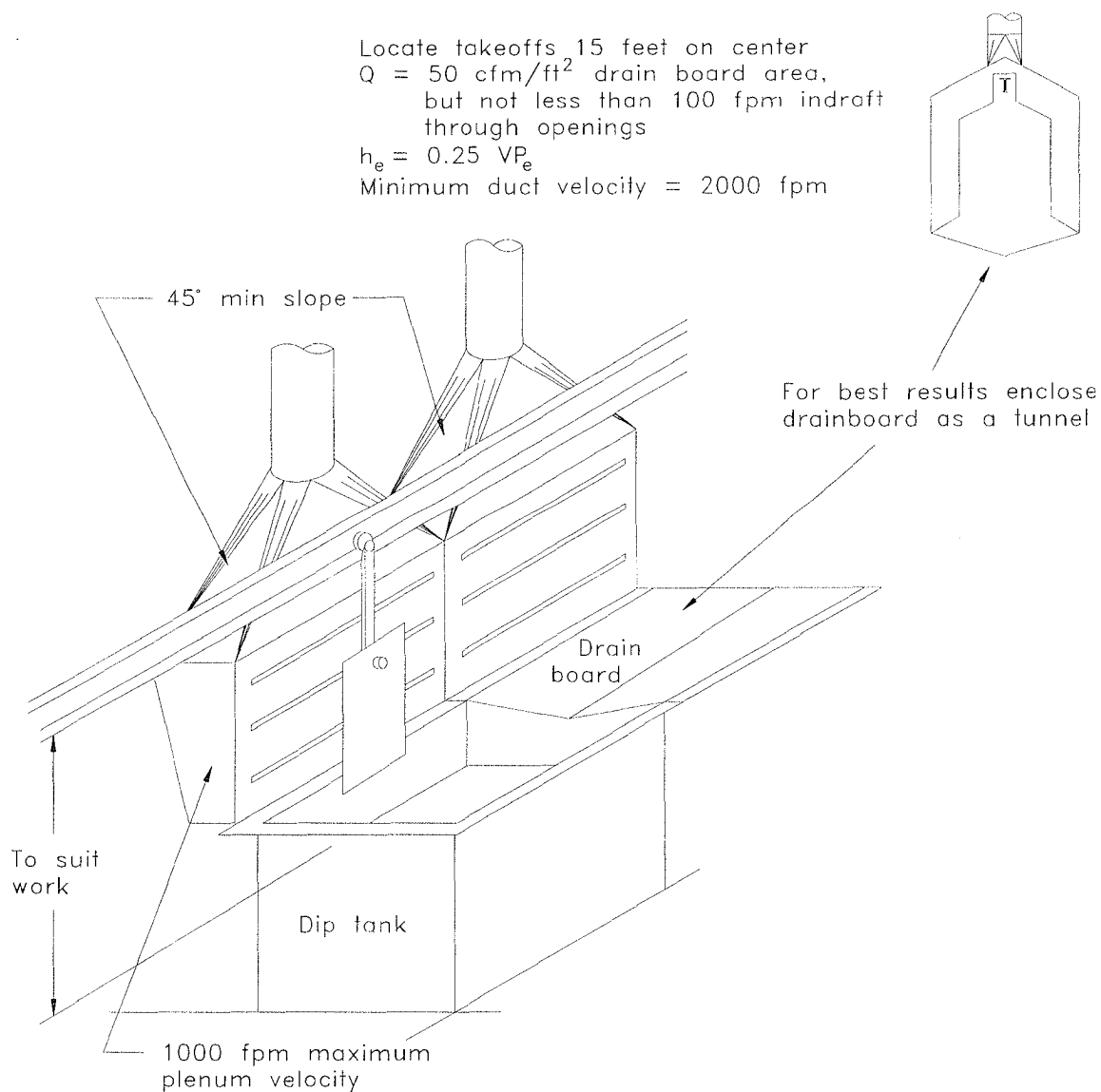
PAINT BOOTH
 VEHICLE SPRAY

DATE

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FIGURE

VS-75-05



$Q = 125 \text{ cfm/ft}^2$ of tank and drainboard area
 Slot velocity = 2000 fpm
 $h_e = 1.78 VP_s + 0.25 VP_d$
 Minimum duct velocity = 2000 fpm

For air drying in a room or enclosure, see Chapter 2 for dilution ventilation required.

For construction and safety, consult NFPA codes, Reference 10.75.1

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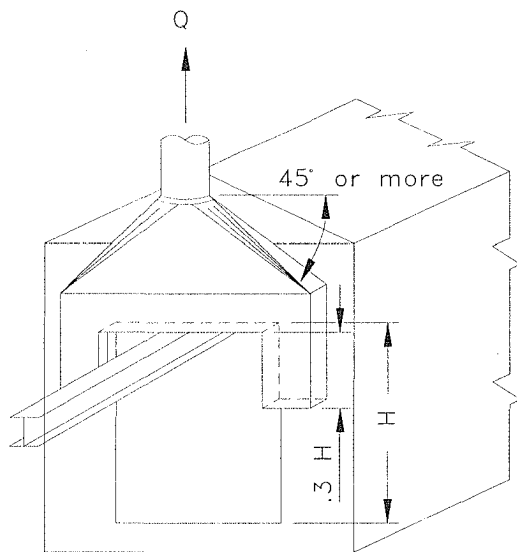
DIP TANK

DATE

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FIGURE

VS-75-06



SLOT TYPE

$Q = 100 \text{ cfm/ft}^2 \text{ door plus } 1/2 \text{ Products of combustion}$

Minimum duct velocity = 2000 fpm

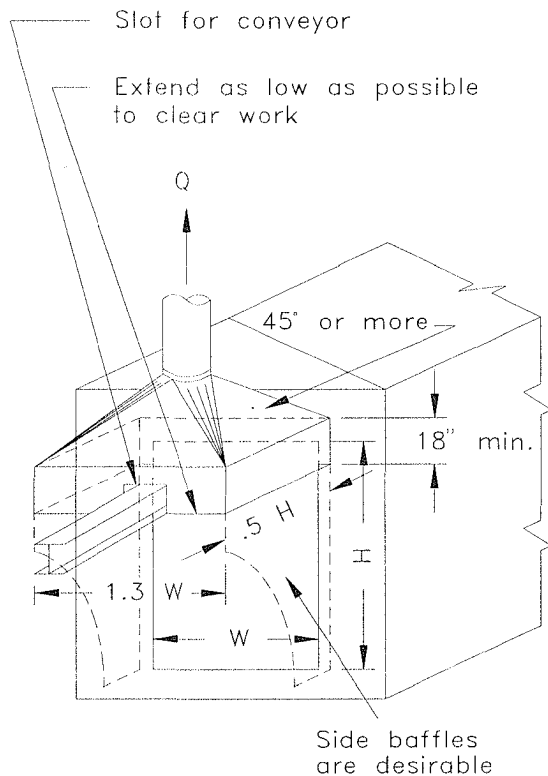
$h_e = 1.0 \text{ VP}_s + 0.25 \text{ VP}_d$

Size plenum for
1000 fpm maximum

Slot on three sides with

$V_s = 2000 \text{ fpm.}$

Locate on inside or outside of door.



CANOPY TYPE

$Q = 200 \text{ cfm/ft}^2 \text{ of hood face plus } 1/2 \text{ products of combustion}$

$h_e = 0.25 \text{ VP}_d$

Duct velocity = 2000 fpm

Notes:

1. For dryers, include rate of water vapor liberated.
2. For flammable solvent drying refer to Chapter 2, "General Industrial Ventilation".
3. Hoods at each end of oven. Reduce size of doors as much as possible. Separate vent must be added for products of combustion.
4. For construction and safety, consult NFPA code (Reference 10.75.1).

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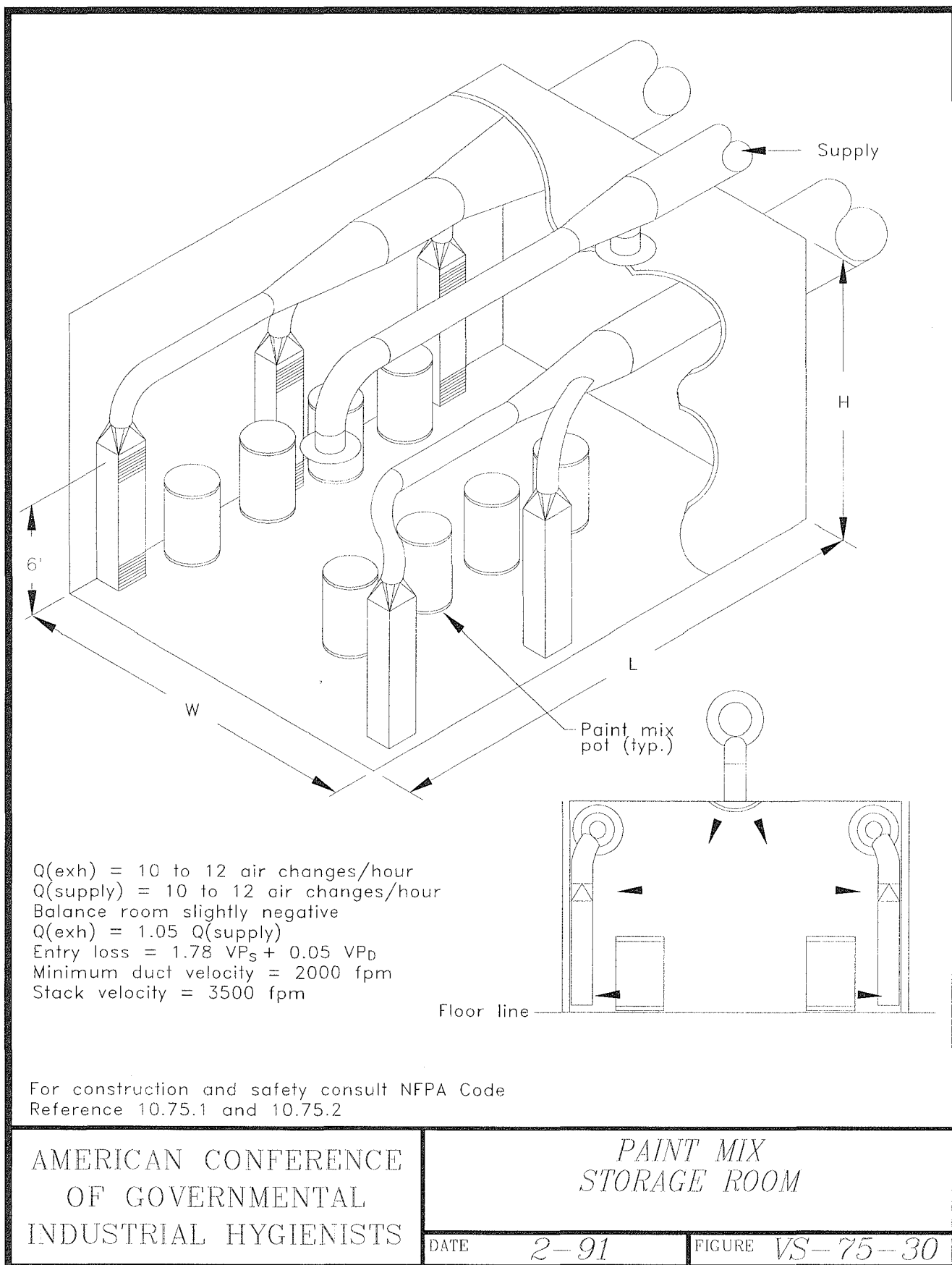
DRYING OVEN VENTILATION

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FIGURE

VS-75-20



10.80 MECHANICAL SURFACE CLEANING AND FINISHING

Mechanical surface cleaning is generally used to clean a surface in preparation for painting, welding, or other operations. The surfaces may be coated with paint, rust, or oxidation; plated with other metals; or covered with molding sand, inorganic, organic, or biological matter. Mechanical cleaning may be accomplished by abrasive blasting, wire wheels, sand paper/sanding belts, grinding wheels, or use of abrasive chips in tumbling mills. The capture velocity needed to entrain large particles is often very high and the collection hood must be positioned so the materials are directed toward the hood. A minimum duct transport velocity of 3500 fpm is needed but 4000 to 5000 fpm is recommended. A hood that encloses as much of the operation as practical is desired. The toxicity of the material removed must be considered when cleaning mechanically. Complete enclosures may be used or the worker may need to wear a respirator in addition to using local exhaust ventilation.

For many grinding, buffing and polishing operations, regulations from the Occupational Safety and Health Administration (OSHA)^(10.80.1) and National Fire Protection Association (NFPA)^(10.80.2) may apply.

10.80.1 Abrasive Cleaning: VS-80-01, -02, and -03 show suggested designs for abrasive blasting and tumbling mills. A supplied air respirator must be used in abrasive blasting rooms.

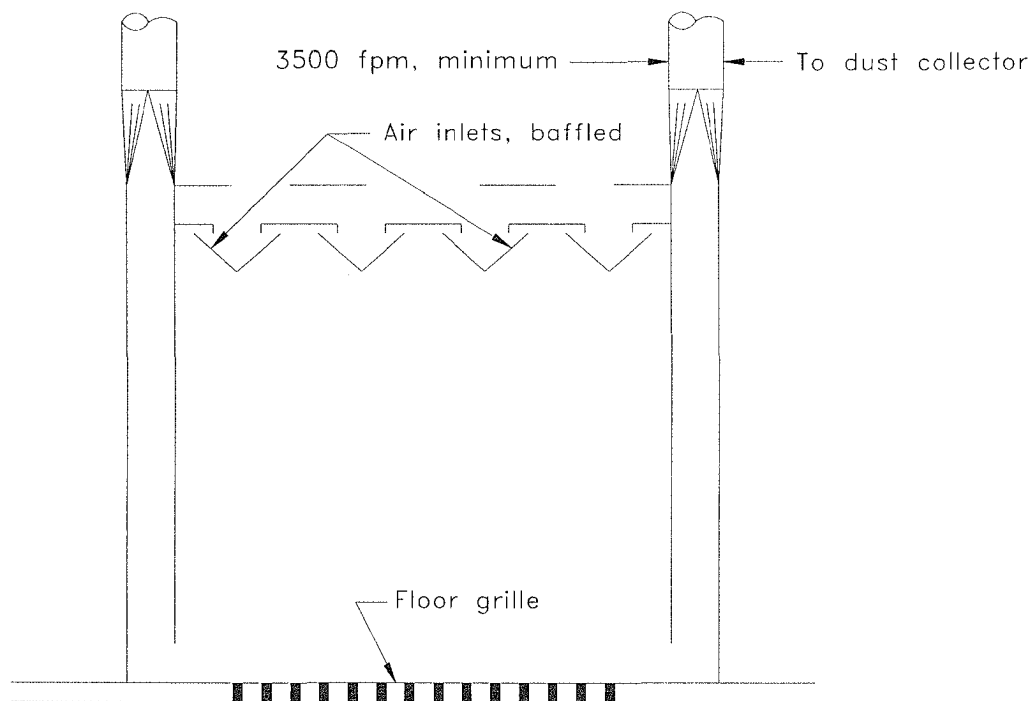
10.80.2 Grinding: Mechanical surface finishing uses organic bonded wheels, cones, saws or other shapes rotating at a high rate of speed to smooth a surface; reduce an object or part in size; or perform other operations. As the object is being surfaced or finished, metallic particles are removed and leave the object at a high speed. In addition, the abrasive wheel is reduced in size and generates particles that must be controlled. Frequently, grinding is accomplished using fluids to keep the parts cool. This cooling fluid will be emitted as an aerosol or mist and needs to be controlled and provisions must be made in the duct to drain off the liquids that accumulate.

The hood used to capture the particles should enclose the operation as much as possible and be positioned to take advantage of the velocity and direction of the particles as they are generated. Design specifications for grinding and surfacing operations are shown in VS-80-10 through VS-80-19.

10.80.3 Buffing and Polishing: The same principles apply for buffing and polishing as for grinding and surfacing. The buffing wheel or belt should be enclosed as much as practical and positioned to take advantage of the centrifugal force of the particles as they leave the wheel or belt. The minimum duct velocity for the generated particles is 3500 fpm and 4500 fpm if the material is wet or sticky. Since many varieties of metals and alloys are buffed and polished, it is extremely important not to mix ferrous and non-ferrous metals in the same exhaust systems (see NFPA codes).^(10.80.2) VS-80-30 through VS-80-35 show suggested designs for buffing and polishing.

REFERENCES

- 10.80.1 U. S. Department of Labor, Occupational Safety and Health Administration: 29 CFR.1910. US-DOL/OSHA, Washington, DC (1970).
- 10.80.2 National Fire Protection Association: National Fire Codes—in particular NFPA-65 (Processing and Finishing of Aluminum); NFPA-68 (Guide for Explosion Prevention Systems); NFPA-77 (Practice on Static Electricity); NFPA-91 (Installation of Blowers and Exhaust Systems for Dust, Stack and Vapor Removal or Conveying); NFPA-480 (Storage, Handling and Processing of Magnesium); NFPA-481 (Production, Processing, Handling and Storage of Titanium); NFPA-482 (Production, Processing, Handling and Storage of Zirconium); and NFPA-561 (Manufacture of Aluminum and Magnesium Powder). NFPA, Quincy, MA.
- 10.80.3 National Institute for Occupational Safety and Health: Recommended Industrial Ventilation Guidelines. DHEW (NIOSH) Pub. No. 76-162; NTIS Pub. No. PB-266-227. National Technical Information Service, Springfield, VA (1975).
- 10.80.4 American Foundrymen's Society, Inc.: Foundry Ventilation Manual. AFS, Des Plaines, IL (1985).
- 10.80.5 National Institute for Occupational Safety and Health: Ventilation Requirements for Grinding, Buffing and Polishing Operations. DHEW (NIOSH) Pub. No. 75-107; NTIS Pub. No. PB-277-332. National Technical Information Service, Springfield, VA (1975).



SECTION THROUGH TYPICAL ROOM

$Q = 60-100 \text{ cfm/ft}^2$ of floor for downdraft with typical choice 80 cfm/ft^2 .
 $Q = 100 \text{ cfm/ft}^2$ of wall for crossdraft.

Lower control velocities may be used depending on toxicity of the contaminant, object and blasting media and the size of the blasting room.

- Notes:
1. The above ventilation is for operator visibility and to control escape of contaminants into adjacent work areas.
 2. Operator in an abrasive blasting room is required to wear appropriate NIOSH certified respiratory protection.
 3. For rotary tables use 200 cfm/ft^2 of total opening (taken without curtains).
 4. For blasting cabinets see VS-80-02.

Reference 10.80.3 and 10.80.4

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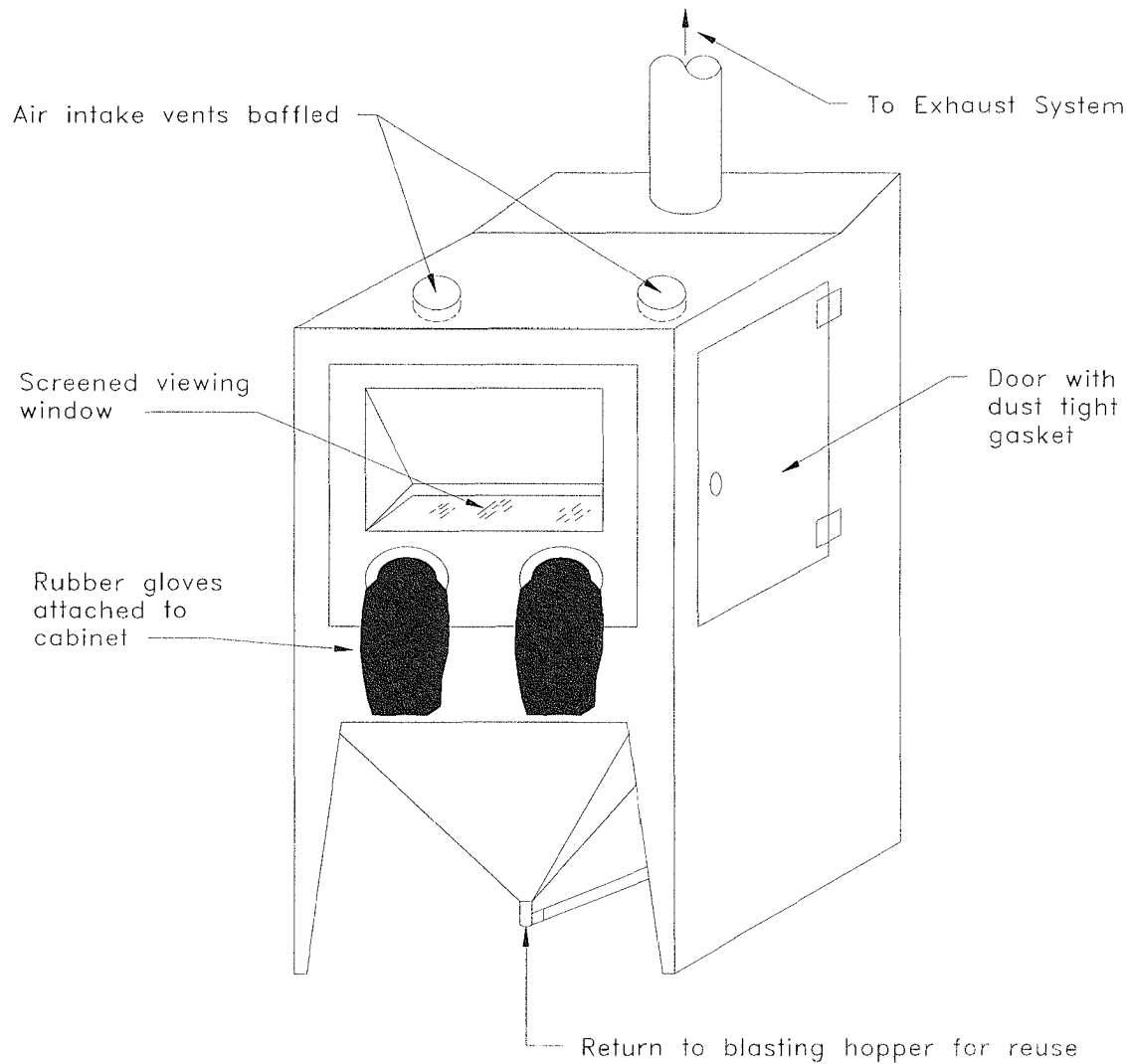
*ABRASIVE BLASTING
ROOM*

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FIGURE

VS-80-01



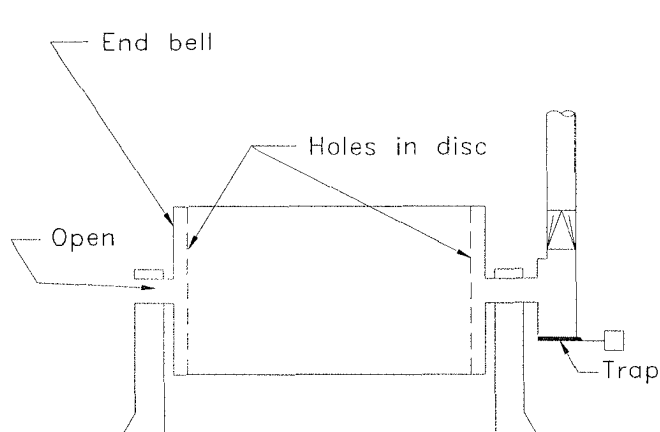
Notes: 20 air changes per minute
 At least 500 fpm inward velocity at all openings
 Minimum duct velocity = 4000 fpm
 If cabinet has self-contained dust collector,
 consult manufacturer for losses.
 $h_e = 1.0 VP_d$

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ABRASIVE BLASTING CABINET

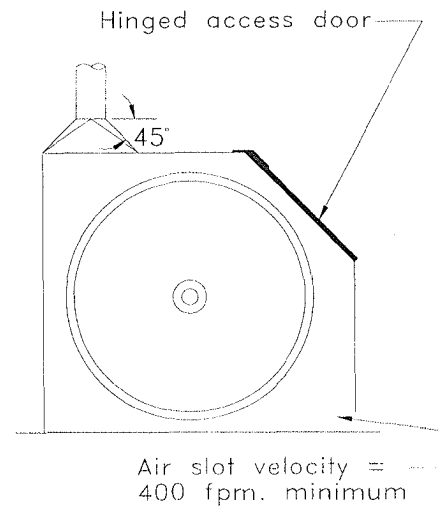
DATE 02-91

FIGURE VS-80-02



SECTION THROUGH HOLLOW TRUNNION TUMBLER

Minimum duct velocity = 5000 fpm
Entry loss (h_e) depends on design and typically ranges from 3 to 9 "wg.



STAVE MILL (END SECTION)

Minimum duct velocity = 3500 fpm
 $h_e = 0.25 V_p^2$

EXHAUST RATES

Square mill side diam in.	Round mill I.D. in inches	Exhaust cfm**	
		Trunnion	Stave
	Up to 24 incl.	430	800
Up to 24 incl.	24 - 30	680	900
25 - 30	31 - 36	980	980
31 - 36	37 - 42	1330	1330
37 - 42	43 - 48	1750	1750
43 - 48	49 - 54	2200	2200
49 - 54	55 - 60	2730	2730
55 - 60	61 - 66	3300	3300
61 - 66	67 - 72	3920	3920
67 - 72		4600	4600

* Low-loss designs have large air inlet openings in end bell.
Holes in end discs are sized for velocities of 1250 - 1800 fpm.

** For lengths over 72", increase exhaust rate proportionately

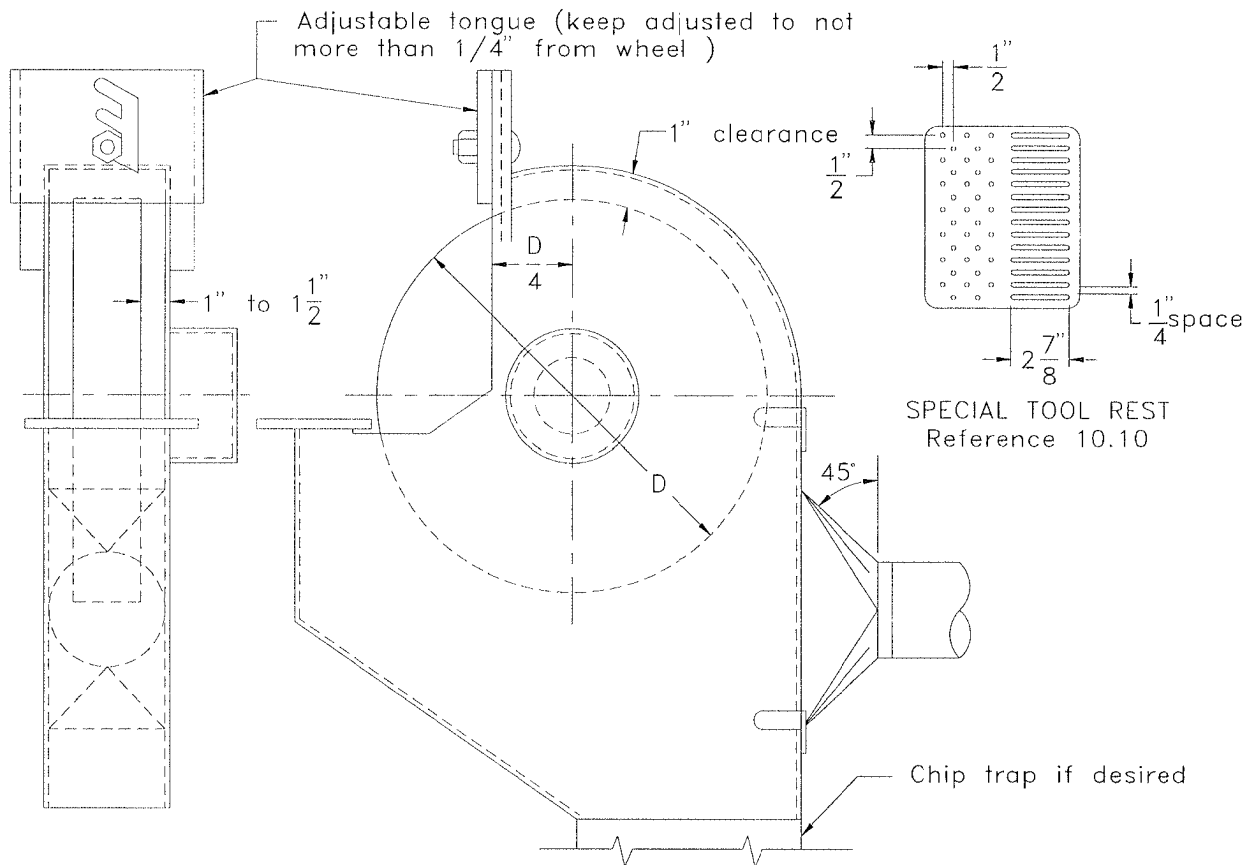
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TUMBLING MILLS

DATE

02-91

FIGURE VS-80-03



EXHAUST FLOW RATES, cfm

Wheel diam. inches	Wheel width inches	Good enclosure*	Poor enclosure
Up to 5	1	220	390
5 to 10	1.5	390	610
10 to 14	2	500	740
14 to 16	2	610	880
16 to 20	3	740	1000
20 to 24	4	880	1200
24 to 30	5	1200	1600
30 to 36	6	1600	2000

*Special hood and tool rest as shown, no more than 25% of the wheel exposed.
Minimum duct velocity = 4000 fpm

$$h_e = 0.65 VP_d \text{ for straight take-off}$$

$$= 0.40 VP_d \text{ for tapered take-off}$$

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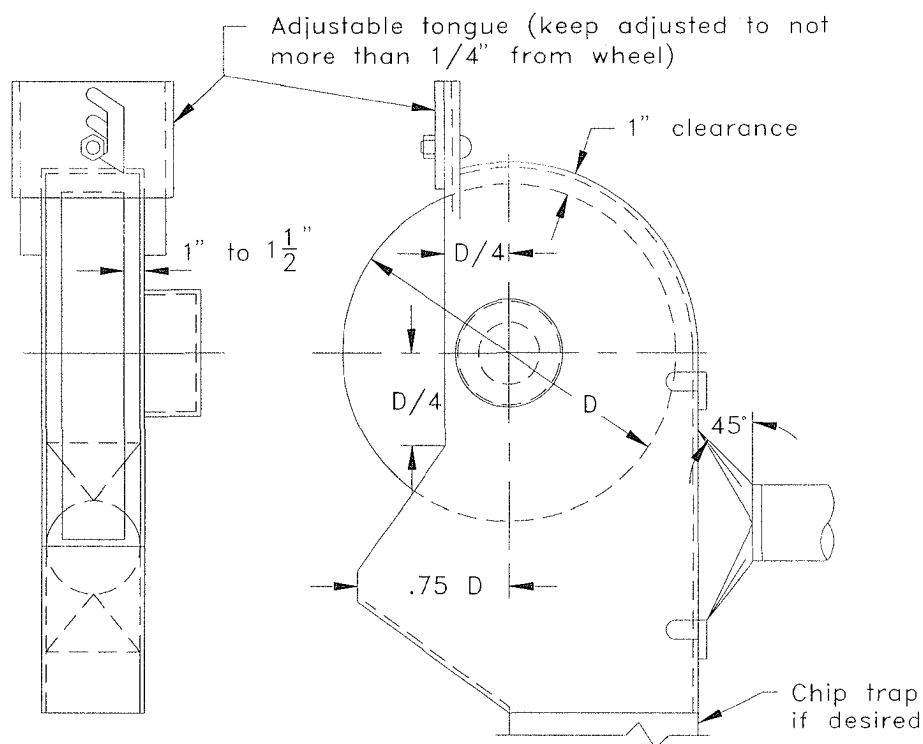
GRINDING WHEEL HOOD
SURFACE SPEEDS
ABOVE 6500 sfpm

DATE

02-91

FIGURE

VS-80-10



EXHAUST FLOW RATES, cfm

Wheel diam inches	Wheel width inches	Good enclosure*	Poor enclosure
Up to 5	1	220	220
5 to 10	1.5	220	300
10 to 14	2	300	500
14 to 16	2	390	610
16 to 20	3	500	740
20 to 24	4	610	880
24 to 30	5	880	1200
30 to 36	6	1200	1600

* No more than 25% of wheel exposed.

Minimum duct velocity = 4000 fpm

$h_e = 0.65 VP_d$ for straight takeoff

$0.40 VP_d$ for tapered takeoff

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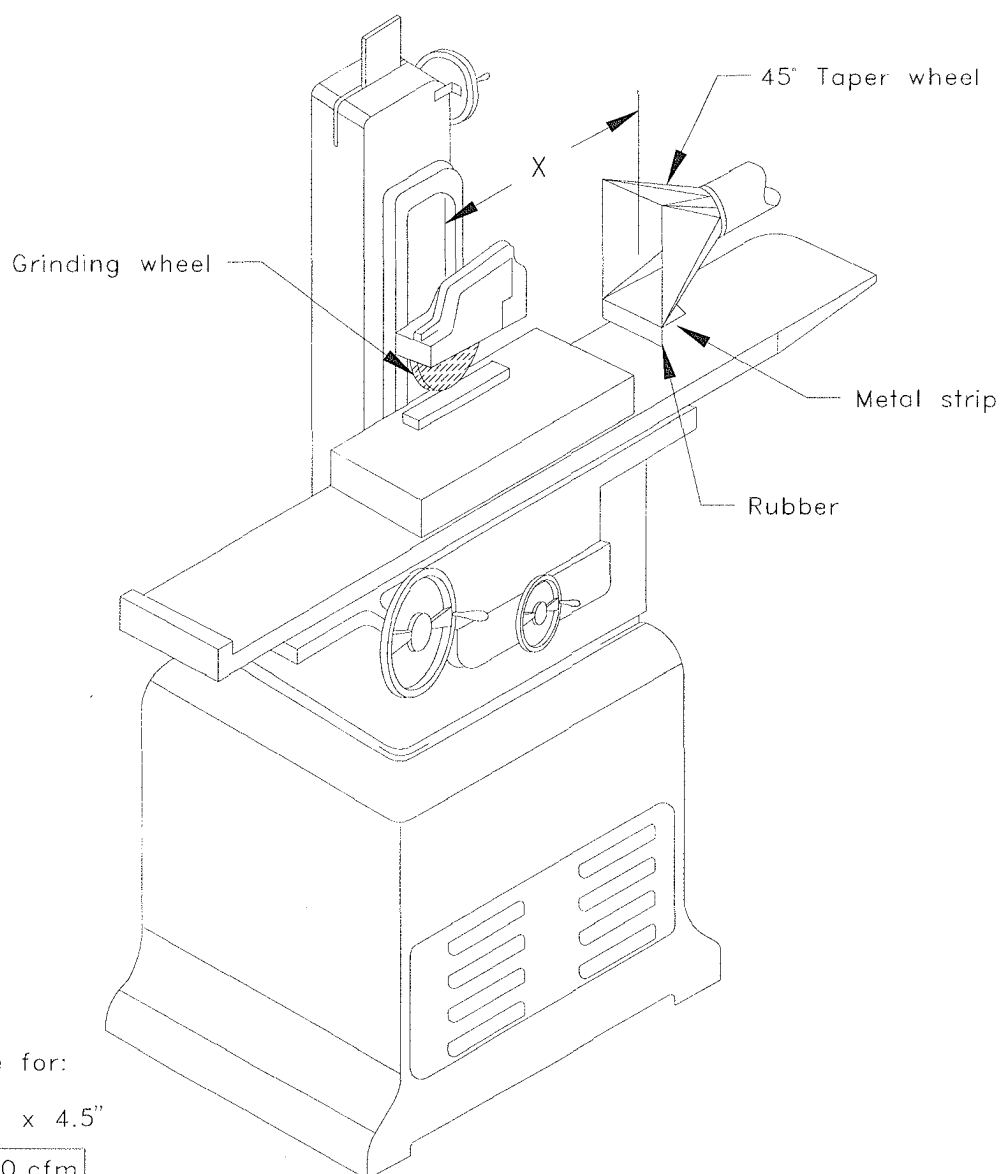
GRINDING WHEEL HOOD
SURFACE SPEEDS
BELOW 6500 sfpm

DATE

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FIGURE

VS-80-11



Example for:

$x = 4''$

$A = 3'' \times 4.5''$

V_s	Q, cfm
1000	52
2000	100
3000	160
4000	210
5000	260
6000	310
7000	360
8000	420
9000	470
10000	520

$$Q = 0.043 V_s (10X^2 + A)$$

Minimum duct velocity = 3500 fpm

$$h_e = 0.25 VP_d$$

X = distance from hood face to center of wheel, ft

A = hood face area, ft^2

V_s = Wheel Speed, surface feet per min. (SFM)

$$= \pi (D/12) R$$

D = diameter in inches

R = rpm of grinding wheel

Reference 10.80.5

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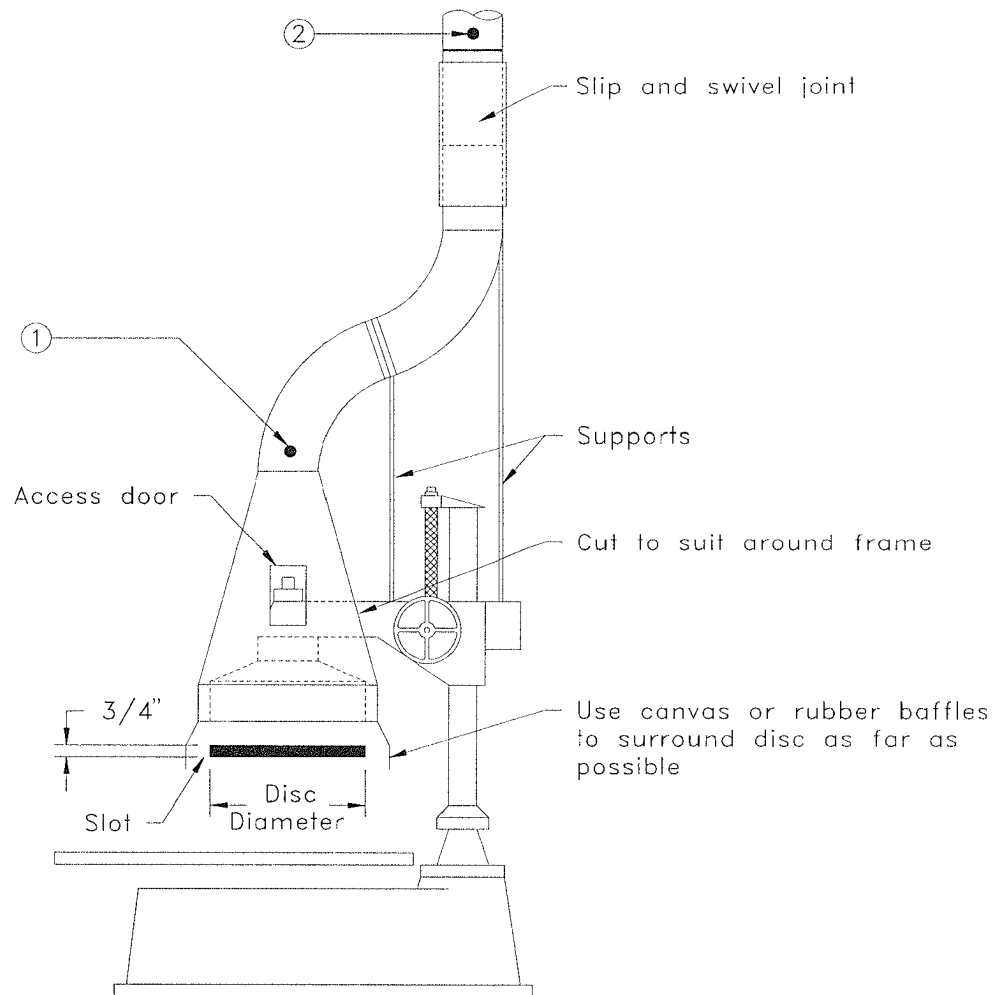
SURFACE GRINDER

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02-91

FIGURE

VS-80-12



Disc diameter	Duct diameter	cfm
Up to 20"	6"	900
20" to 30"	8"	1600
30" to 53"	12"	3500
53" to 72"	16"	6300

Minimum duct velocity = 4000 fpm
 Minimum slot velocity = 2000 fpm

① $h_e = 1.0 VP_s + 0.40 VP_d$

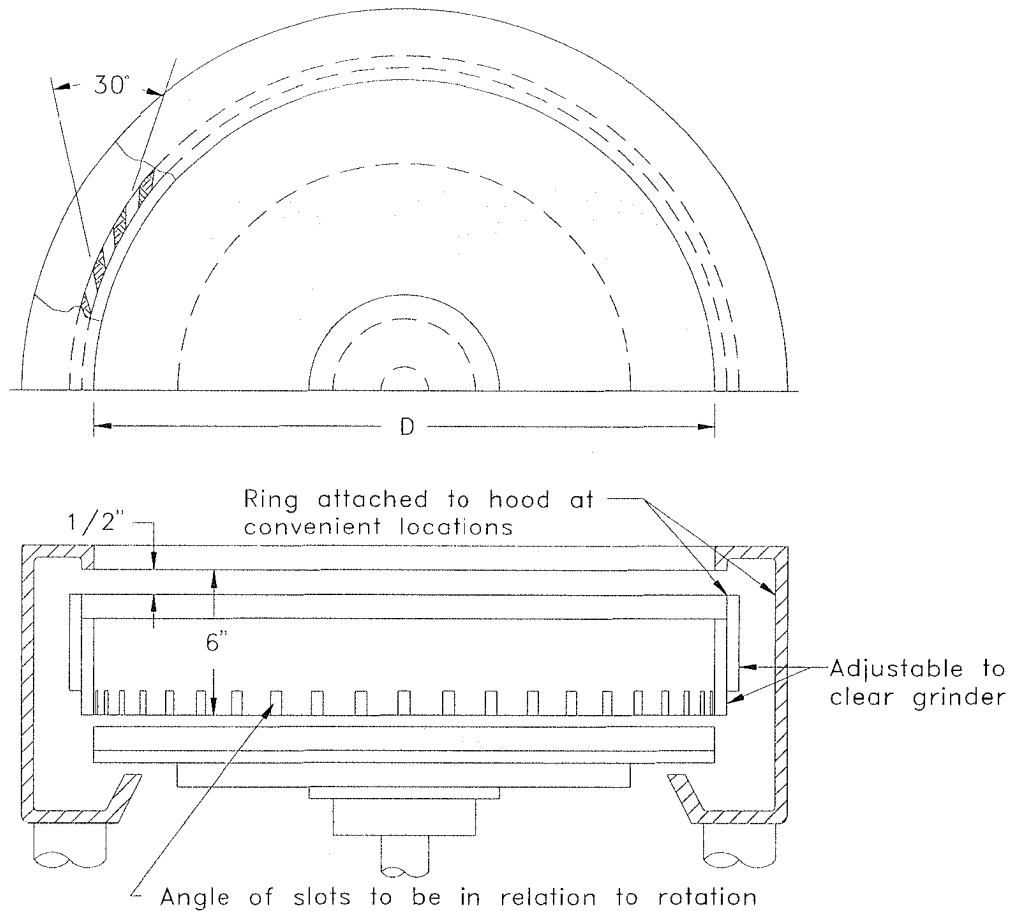
② $h_e + \text{elbow losses} + \text{joint losses}$

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CORE GRINDER

DATE 02-91

FIGURE VS-80-13



EXHAUST FLOW RATE, cfm

Disc diameter inches	1/2 or more of disc covered		Disc not covered	
	No.*	Exhaust flow rate, cfm	No.*	Exhaust flow rate, cfm
Up to 20	1	500	2	780
20 to 30	2	780	2	1500
30 to 53	2	1800	4	3500
53 to 72	2	3100	5	6000

* Number of exhaust outlets around periphery of hood or equal distribution provided by other means.

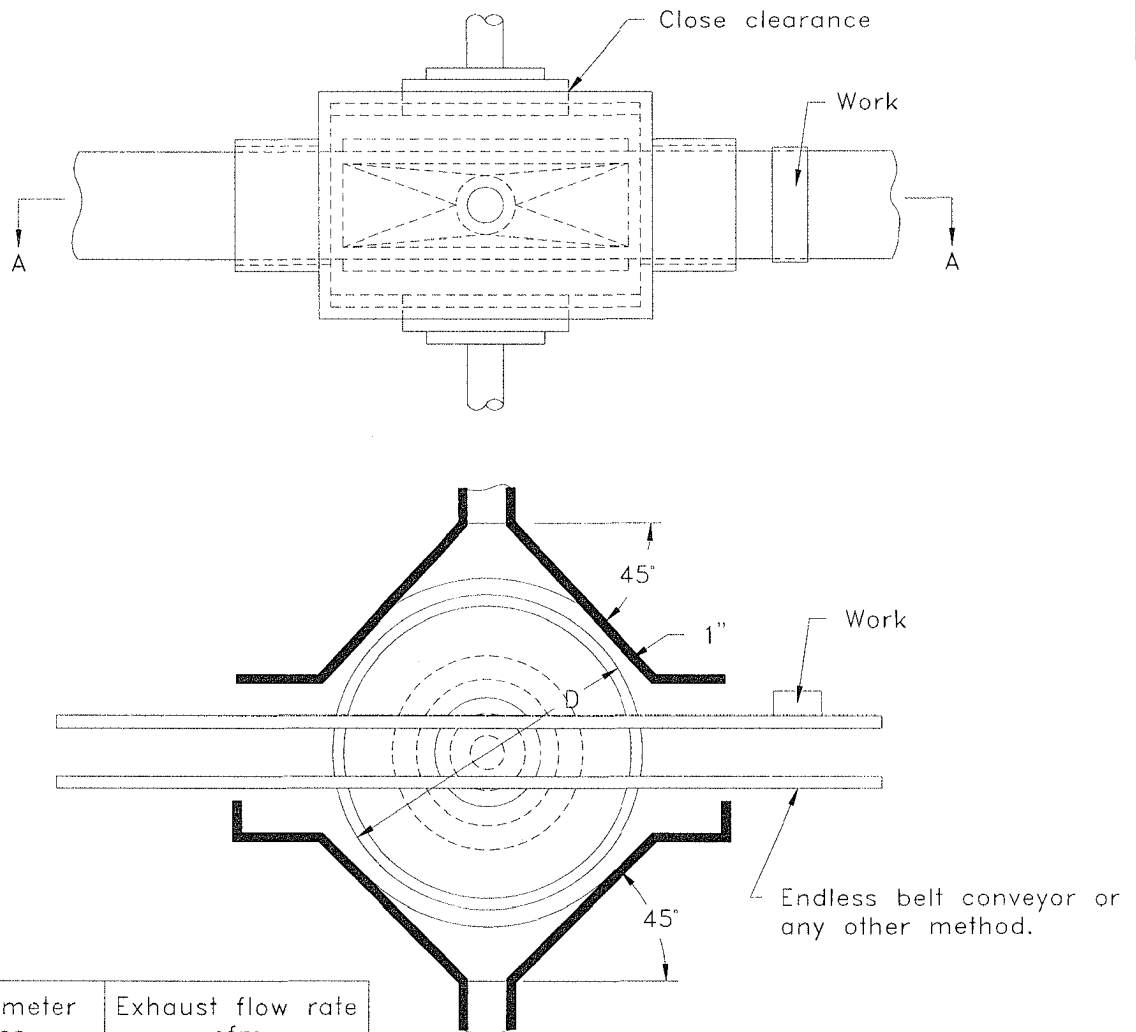
Minimum slot velocity = 2000 fpm
 Minimum duct velocity = 4000 fpm
 $h_e = 1.0 VP_s + 0.5 VP_d$

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VERTICAL SPINDLE DISC
 GRINDER

DATE 02-91

FIGURE VS-80-14



Disc diameter inches	Exhaust flow rate cfm
Up to 19	610
19 to 25	880
25 to 30	1200
30 to 53	2000
53 to 72	6300

Note: If the disc is tightly enclosed by machine housing, then exhaust from the housing is acceptable.

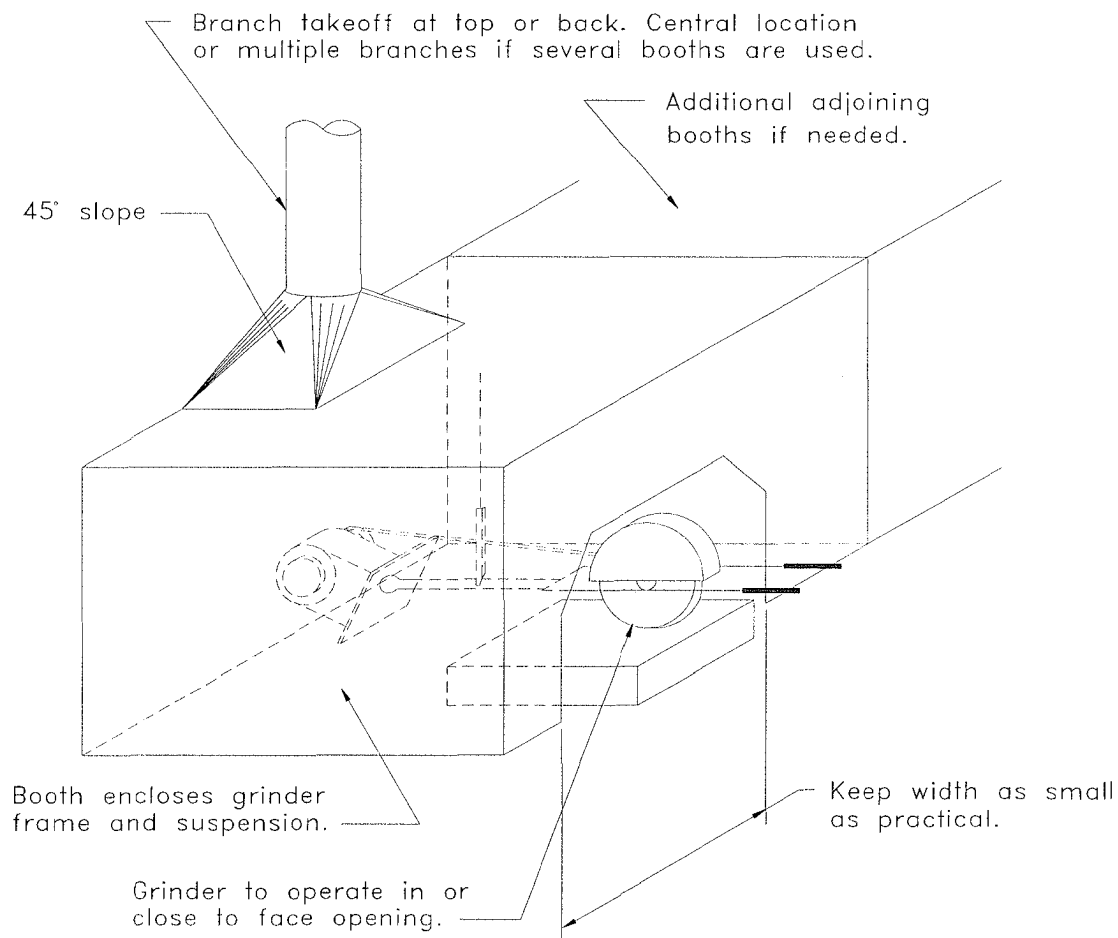
Minimum duct velocity = 4000 fpm
 $h_e = 0.65 VP_d$ straight take-off
 $= 0.45 VP_d$ tapered take-off

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*HORIZONTAL DOUBLE-SPINDLE
DISC GRINDER*

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FIGURE VS-80-15



For a large opening, 4' to 6' wide
 $Q = 150 \text{ cfm/ft}^2$ of opening

For a small opening, 2' to 2'-6" with
 grinder in front
 $Q = 200 \text{ cfm/ft}^2$ of opening

Minimum duct velocity = 3500 fpm
 $h_e = 0.25 \text{ VP}_d$

NOTE: Small local exhaust hoods mounted behind grinder wheel may trap the stream of sparks, but are usually not effective in control of airborne dust.

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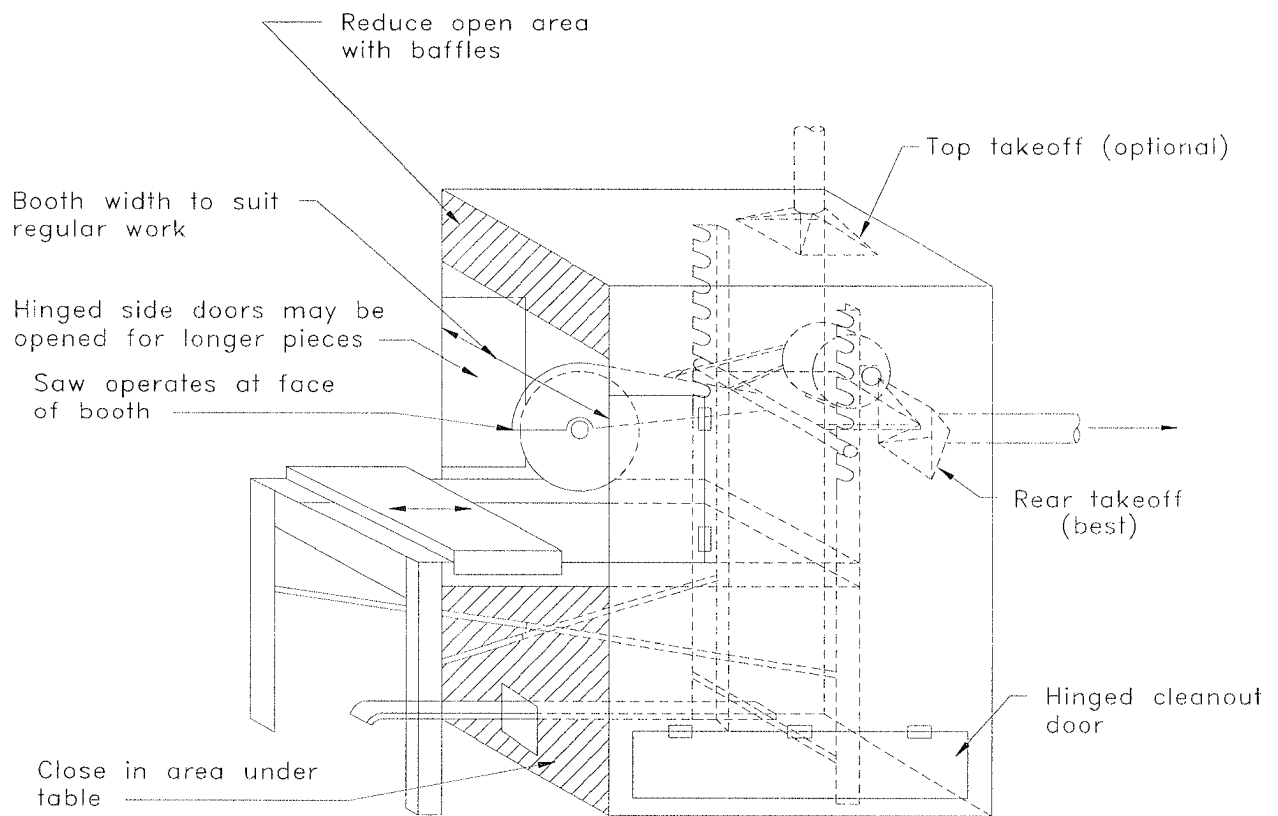
SWING GRINDER

DATE

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FIGURE

VS-80-16



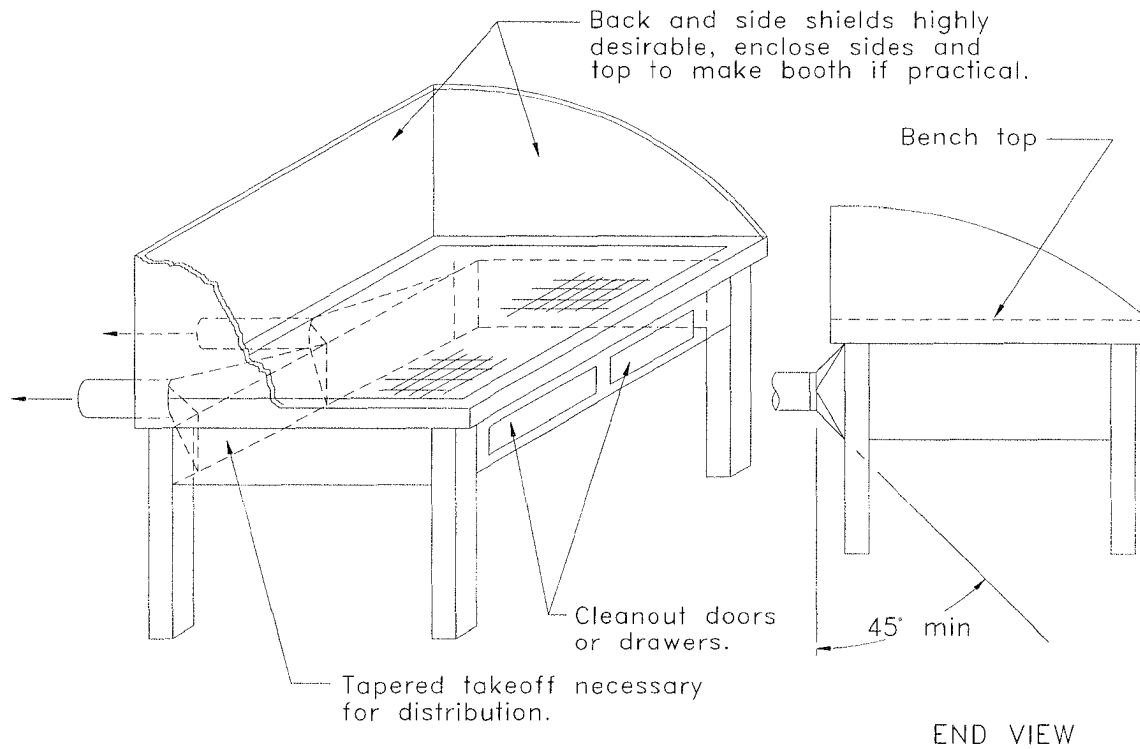
$Q = 250 \text{ cfm/ft}^2$ of open face area
 Minimum duct velocity = 4000 fpm
 $h_e = 0.50 VP_d$ (no taper)
 $0.25 VP_d$ (with 45° taper)

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ABRASIVE CUT-OFF
SAW

DATE 02-91

FIGURE VS-80-17



$Q = 150-250 \text{ cfm/ft}^2$ of bench area.

Minimum duct velocity = 3500 fpm

$h_e = 0.25 VP_d$

If slots are used for distribution

$h_e = 1.78 VP_s + 0.25 VP_d$

- Notes: 1. If grinding in a booth, use 100 fpm face velocity.
 2. For downdraft grilles in floor: $Q=100 \text{ cfm/ft}^2$ of working area.
 3. Provide equal distribution.
 4. Provide for cleanout.

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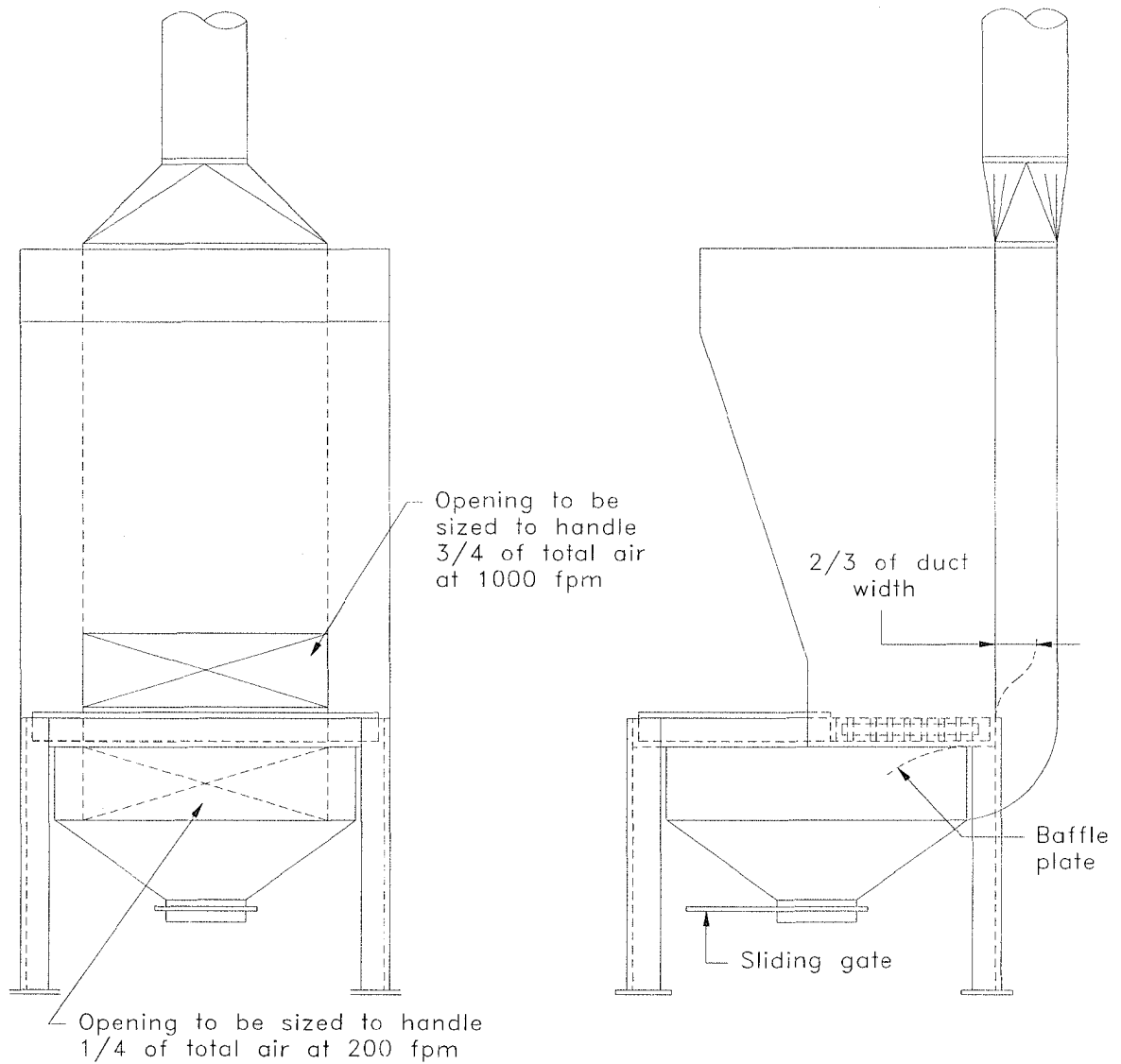
HAND GRINDING BENCH

DATE

02-91

FIGURE

VS-80-18



$Q = 150 \text{ cfm/ft}^2 \text{ opening}$
 Minimum duct velocity = 3500 fpm
 $h_e = 0.25 VP_d$

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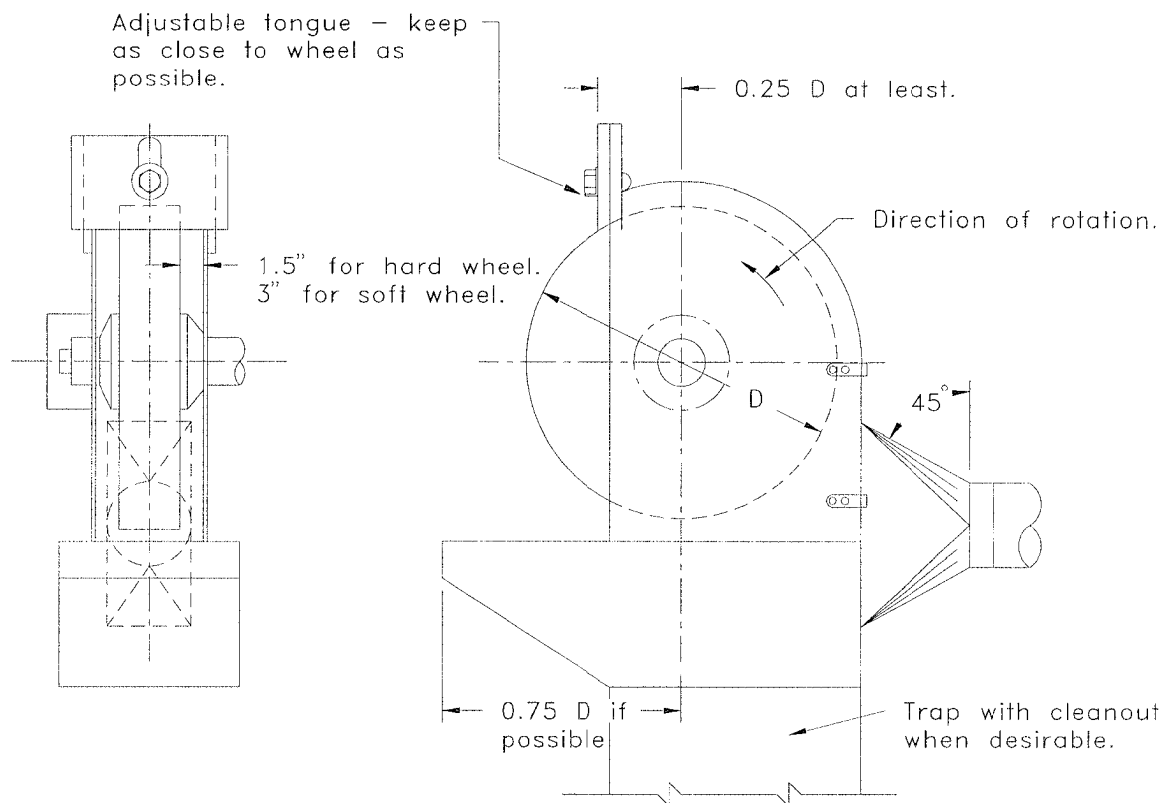
PORTABLE CHIPPING AND
GRINDING TABLE

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FIGURE

VS-80-19



Minimum duct velocity : 3500 fpm,
4500 fpm if material is wet or sticky
 $h_e = 0.65 VP_D$ for straight take-off.
 $h_e = 0.40 VP_D$ for tapered take-off.

Wheel diam. inches	Wheel width inches	Exhaust flow rate cfm	Exhaust flow rate cfm
		Good enclosure *	Poor enclosure
Up to 9	2	300	400
over 9 to 16	3	500	610
over 16 to 19	4	610	740
over 19 to 24	5	740	1200
over 24 to 30	6	1040	1500
over 30 to 36	6	1200	2000

* not more than 25% of the wheel is exposed

Note : Consult applicable NFPA codes Reference 10.80.2

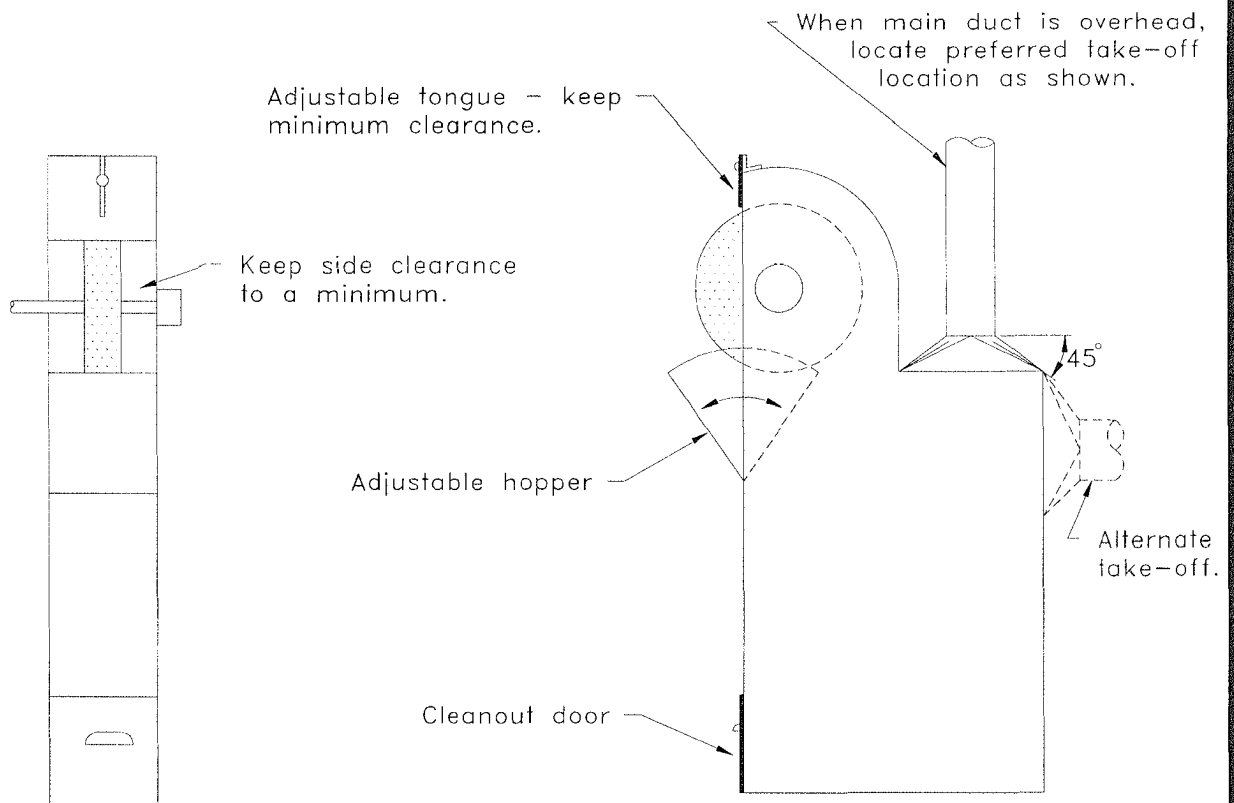
Caution : Do not mix ferrous and non-ferrous metals
in same exhaust system.

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MANUAL
BUFFING AND POLISHING

DATE 02-91

FIGURE VS-80-30



Wheel diam. inches	Wheel width inches	Exhaust flow rate cfm
Up to 9	2	400
9 to 16	3	610
16 to 19	4	740
19 to 24	5	1200
24 to 30	6	1500
30 to 36	6	1900

Note: For wider wheels than listed, increase cfm with width

Minimum duct velocity = 3500 fpm

4500 fpm if material is wet or sticky

$$h_e = 0.40 VP_d$$

Notes: 1. Consult applicable NFPA codes. See Reference 10.80.2

2. For titanium, aluminum, and magnesium, eliminate hopper use 5000 fpm through hood cross-section.

3. Caution : Do not mix ferrous and non-ferrous metals in same exhaust system.

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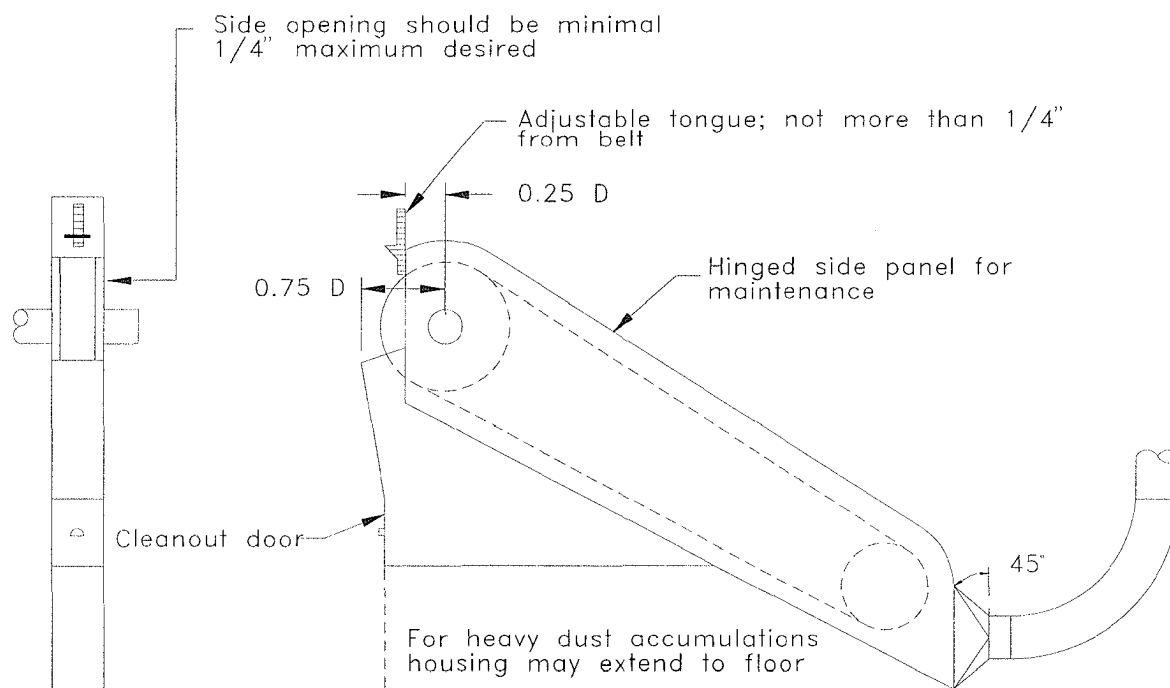
BUFFING LATHE

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02-91

FIGURE

VS-80-31



Belt width inches	Exhaust flow rate cfm	Exhaust flow rate cfm
	Good enclosure *	Enclosure
1 1/2	220	300
2	390	610
3	500	740
4	610	880
5	880	1200
6	1200	1570

* Hood as shown; no more than 25% of wheel exposed.

$$h_e = 0.40 VP_d$$

Minimum duct velocity = 3500 fpm, 4500 fpm if wet or sticky.

Notes:

1. Consult applicable NFPA codes, 10.80.2
2. For titanium, aluminum and magnesium eliminate hopper and use 5000 fpm through hood cross section.
3. Caution: do not mix ferrous and non-ferrous metals in same exhaust system.

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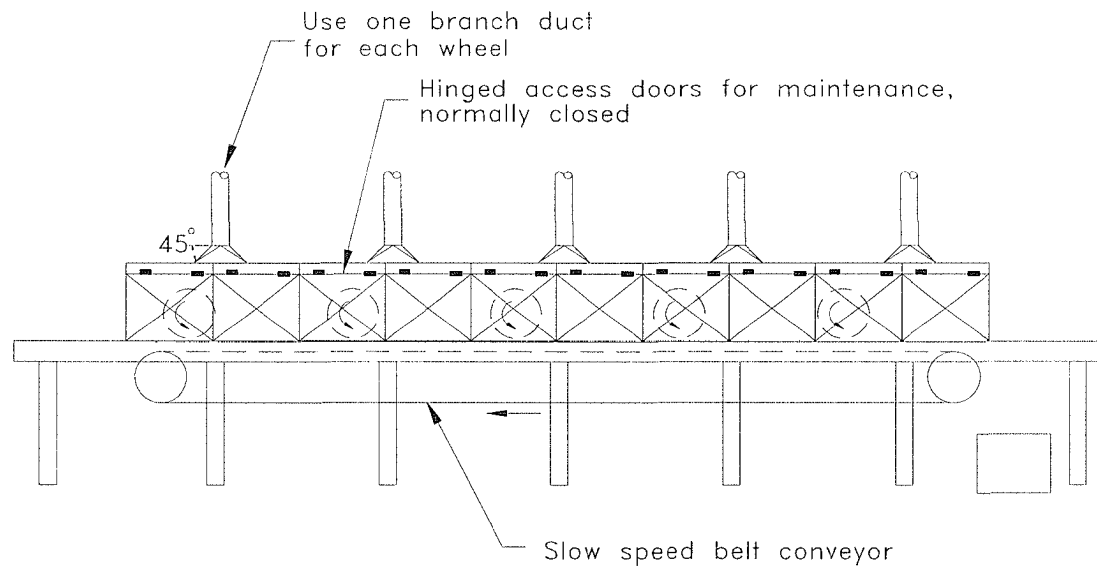
BACKSTAND IDLER
POLISHING MACHINE

DATE

02-91

FIGURE

VS-80-32



$Q = 500 \text{ cfm/wheel, minimum}_2$

Not less than 250 cfm/ft^2 total open area

Minimum duct velocity = 3500 fpm, 4500 fpm if
material is wet or sticky

$$h_e = 1.78 VP_s + 0.25 VP_d$$

Note:

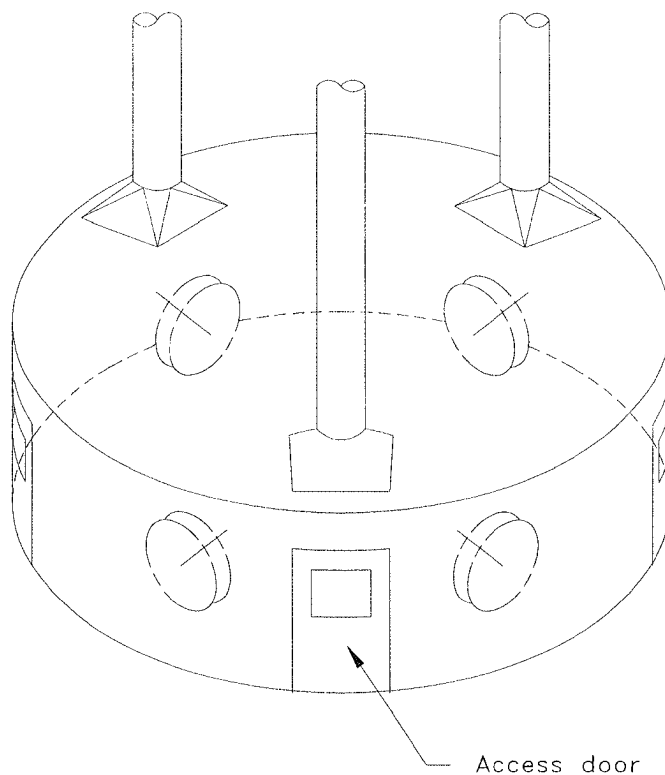
1. Consult applicable NFPA standards. Reference 10.80.2
2. Caution: Do not mix ferrous and non-ferrous metals in same exhaust system.
3. Wheel adjustments on outside of enclosure.
4. For highly toxic material, enclose the return strand of the belt conveyor.

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*STRAIGHT LINE AUTOMATIC
BUFFING*

DATE 02-91

FIGURE VS-80-33



$Q = 500 \text{ cfm/wheel, minimum}_2$

Not less than 250 cfm/ft^2 total open area

Minimum duct velocity = 3500 fpm, 4500 fpm if
material is wet or sticky

$$h_e = 1.78 VP_s + 0.25 VP_d$$

On small, 2 or 3 spindle machines, one take-off may be used.
Multiple take-offs desirable.

Note:

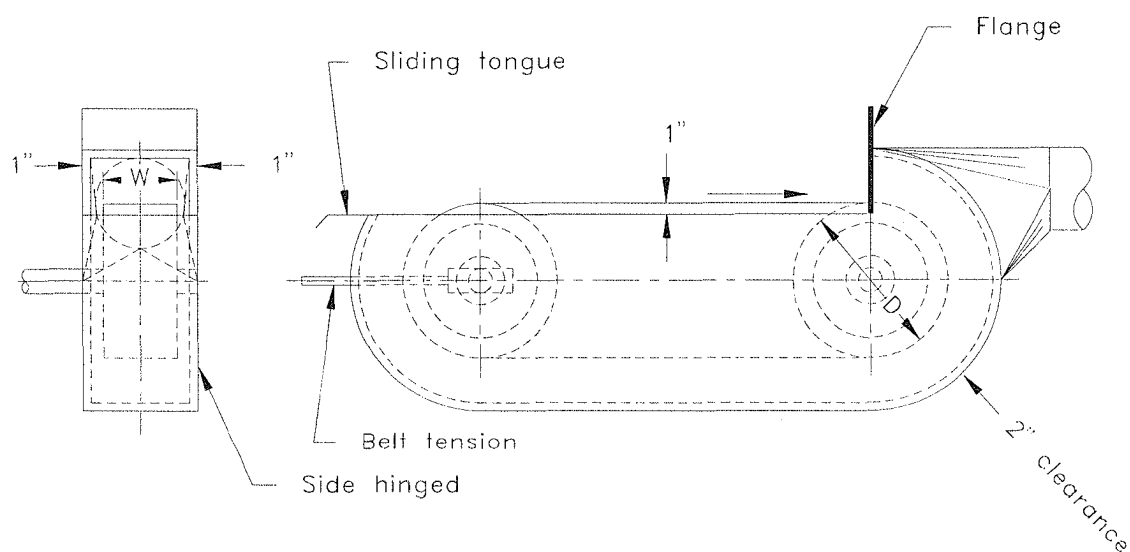
1. Consult applicable NFPA standards, Reference 10.80.2
2. Caution: Do not mix ferrous and non-ferrous metals in same exhaust system.

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CIRCULAR AUTOMATIC BUFFING

DATE 02-91

FIGURE VS-80-34



Belt width, inches	Exhaust flow rate, cfm
Up to 3	220
3 to 5	300
5 to 7	390
7 to 9	500
9 to 11	610
11 to 13	740

Minimum duct velocity = 3500 fpm, 4500 fpm if material is wet or sticky

$h_e = 0.65 VP_d$ for straight take-off
 $0.45 VP_d$ for tapered take-off

Notes:

1. Consult applicable NFPA codes Reference 10.80.2
2. Caution : Do not mix ferrous and non-ferrous metals in same exhaust system

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METAL POLISHING BELT

DATE

02-91

FIGURE VS-80-35

10.85 VEHICLE VENTILATION

The objective of providing ventilation for vehicles in an environment is to keep a worker's exposure to toxic exhaust fumes and gases below the TLV, both the TWA and STEL, or other appropriate standards. This can be achieved either by dilution or local exhaust ventilation.

It is difficult to establish dilution ventilation requirements accurately for the operating vehicles in a plant. For an existing facility, the designer has the opportunity to measure the emission in the field. Standard techniques can be used to measure gas flow rates, composition and temperatures, and contaminant levels. Using the equations in Chapter 2 and the measurements, the dilution rates can be calculated. However, it is not always possible to accurately determine the contaminant generation rate because generation is not uniform. Moreover, no such data are available to the designer for new vehicles.

The use of dilution ventilation is usually considered only after rejection of the source capture concept. Common reasons for rejecting source capture (local exhaust) are operating interference problems or layout constraints. For lift trucks or cars in motion or idling outside of stalls, local exhaust is not feasible. Hence the only method for control of health hazards is dilution ventilation.

Over the years, some empirical rates have been developed which have been applied successfully to achieve contaminant control. The recommended dilution rates based on average operating conditions are:

- 5,000 cfm/propane fueled lift truck
- 8,000 cfm/gasoline fueled lift truck
- 5,000 cfm/operating automobile
- 10,000 cfm (or more)/operating truck
- 100 cfm/horsepower for diesel fueled vehicle

The above dilution rates for lift trucks apply under the following conditions:^(10.85.1)

1. A regular maintenance program incorporating final engine tuning through carbon monoxide analysis of exhaust gas must be provided. CO concentration of gases should be limited to 1% for propane fueled trucks; 2% for gasoline fueled trucks.
2. The periods of lift truck engine operation do not exceed 50% of the working day (total engine operation of lift truck equal to or less than 4 hours in an 8-hour shift).
3. A reasonably good distribution of air flow must be provided.

4. The volume of space must amount to 150,000 ft³/lift truck or more.
5. The lift truck is powered by an engine of less than 60 HP.

Where actual operating conditions vary from the above, the ventilation rate should be increased. On the other hand, mechanical ventilation may not be required in large buildings where lift truck operation is intermittent and where natural infiltration based on a maximum of one air change/hour for the net building volume exceeds the recommended dilution ventilation rate.

The alternative to dilution ventilation is to capture the contaminant at the source by installing local exhaust ventilation. For stationary vehicles in service garages, effective systems are shown in VS-85-01 (overhead) and VS-85-02 (under floor). The systems should be connected directly to the vehicle exhaust and should terminate outdoors above the roof. The design procedure outlined in Chapter 5 must be followed. For friction loss data of flexible ducts, manufacturers should be contacted. As with all flexible systems, the length of flexible duct must be minimized, and non-collapsible duct should be used. Unnecessary and/or sharp bends should be avoided. Exhaust requirements for automobiles are shown in VS-85-02 and for diesel engines in VS-85-03.

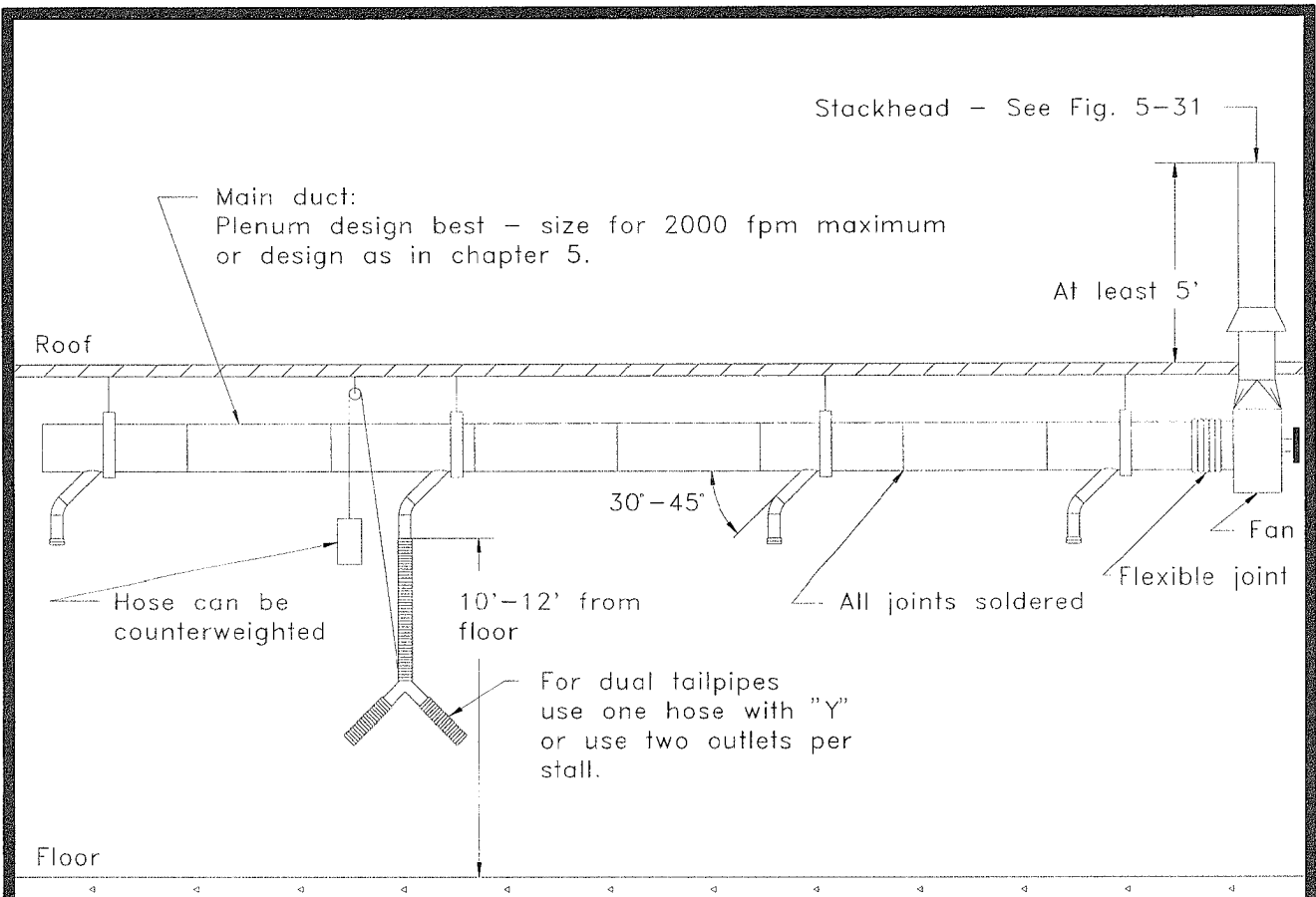
The requirements for parking or storage garages should be based on short-term exposure of drivers to exhaust emissions when entering or departing. A continuous supply of 500 cfm fresh air/parking space should be adequate. Additional ventilation may be required if there are long periods of engine idling (winter warm-ups, loading, etc.) or if the general traffic pattern is such that clusters of vehicles arrive or depart.

Attendant booths of parking garages should be pressurized with a supply of fresh air from uncontaminated sources.

For indoor loading docks, continuous supply of 2 cfm/ft² of dock area should be adequate where truck motors are shut off except when entering or leaving the dock.

REFERENCES

- 10.85.1 Hama, G.M.; Butler, Jr., K.E.: Ventilation Requirements for Lift Truck Operation. Heating, Piping and Air Conditioning (January, 1970).
- 10.85.2 Goldfield, J.; Sheehy, J.W.; Gunter, B.J.; Daniels, W.J.: An Affordable Ventilation Control for Radiator Repair Shops. Ventilation '91: 3rd International Symposium on Ventilation for Contaminant Control. American Conference of Governmental Industrial Hygienists, Cincinnati, OH (1993).



Vehicle horsepower (hp)	cfm/vehicle	Flexible duct diam.	Branch connection
Up to 200	100	3"	4"
Over 200	200	4"	4"
Diesel trucks	See VS-85-03		

On dynamometer test rolls

Automobiles and light duty trucks = 350 cfm

Heavy duty trucks = 1200 cfm minimum

For friction loss of flexible duct; consult manufacturers' data.

See VS-85-02

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*SERVICE GARAGE VENTILATION
OVERHEAD*

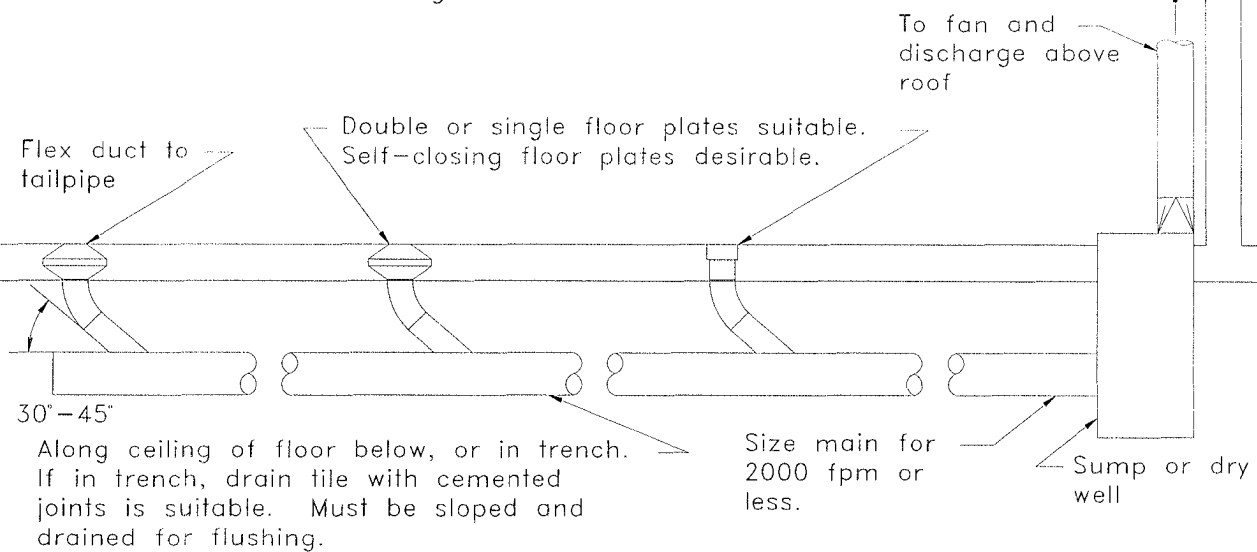
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FIGURE

VS-85-01

Note : In ventilating a garage use either the overhead or under floor system.
Exhaust to be discharged above roof.



UNDER FLOOR SYSTEM
EXHAUST REQUIREMENTS*

Type	cfm per vehicle	Flex duct ID (min)
Automobiles and trucks up to 200 hp	100	3"
Automobiles and trucks over 200 hp	200	4" **
Diesel	See VS-85-03	

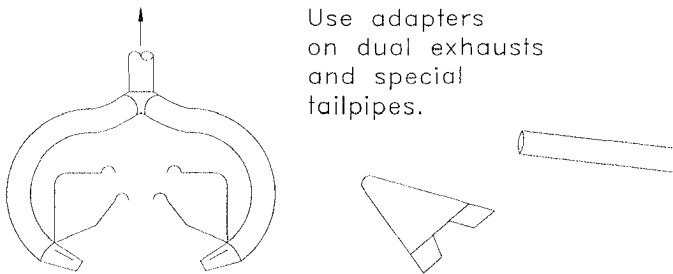
* On dynamometer test rolls

Automobiles and light duty trucks = 350 cfm

Heavy duty trucks = 1200 cfm minimum

** 3" diam. permissible for short runs with proper fan.

For friction loss of flexible duct; consult manufacturers' data.

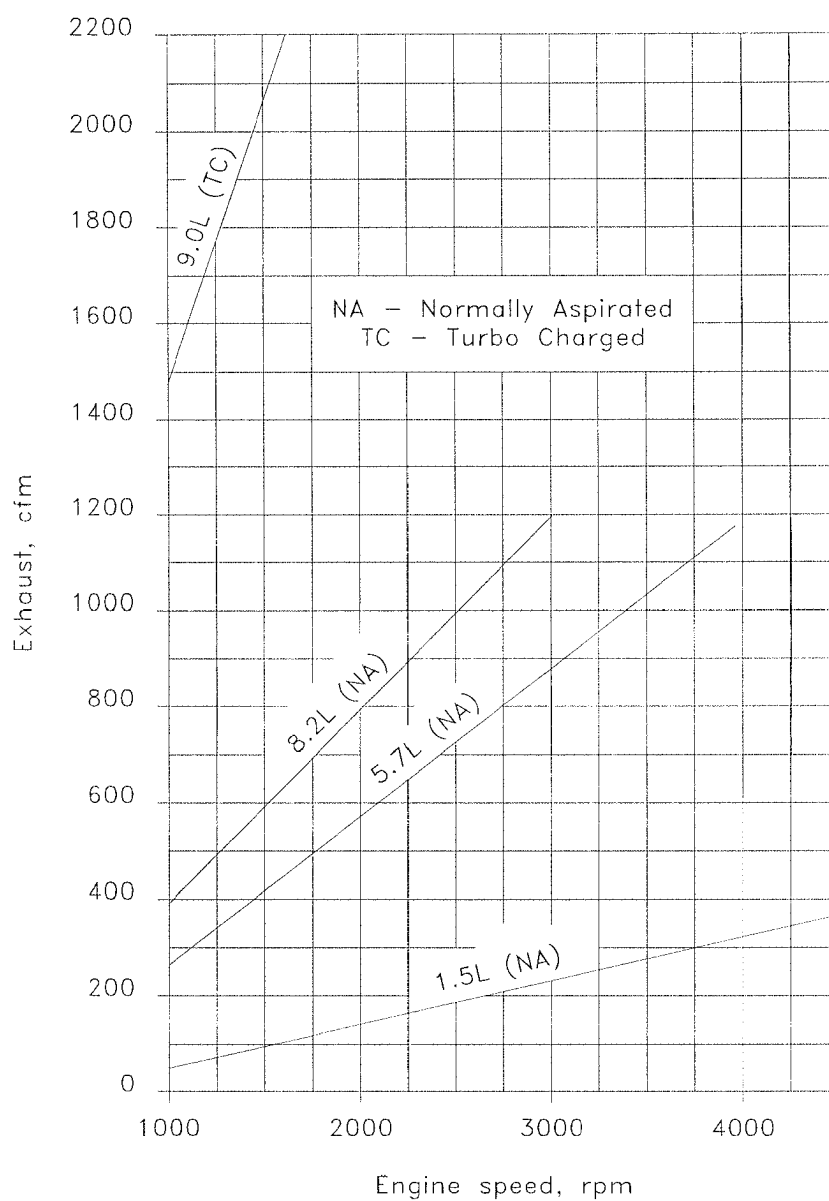


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SERVICE GARAGE
VENTILATION UNDERFLOOR

DATE 12-90

FIGURE VS-85-02



Exhaust, cfm = acfm + 20% excess

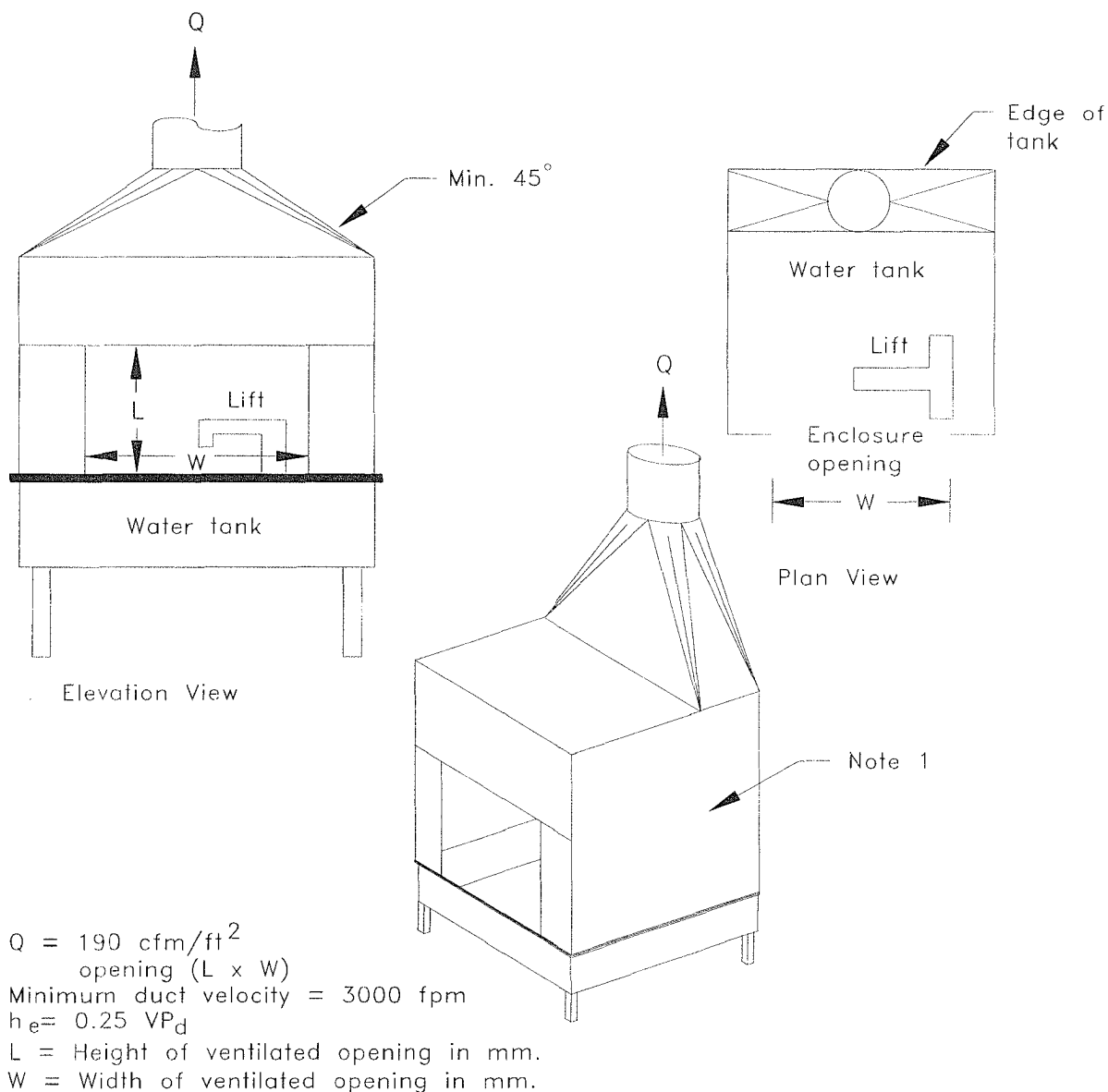
For specific design information request manufacturers 13 mode
EPA engine bench test

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*EXHAUST SYSTEM REQUIREMENTS
FOR TYPICAL DIESEL ENGINES
UNDER LOAD*

DATE 12-90

FIGURE VS-85-03



Notes: 1. Enclosure can be made of metal or fire resistant curtain material

Reference: 10.85.3

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INDUSTRIAL HYGIENISTS

*VENTILATED BOOTH FOR
RADIATOR REPAIR SOLDERING*

DATE

4-94

FIGURE

VS-85-10

10.90 WELDING AND CUTTING

The purpose of welding and cutting ventilation is to control gases, fumes, and particulate generated during the welding and cutting operations.

10.90.1 Hazards: The generation rate of fumes and gases varies with the composition of the base metal, fluxes, and fillers, and with the rate and depth of welding. Exposure to the welder varies with the generation rate, duration and frequency of operations, work practices (particularly distance of the plume from the breathing zone), and the effectiveness of ventilation.

Contaminants from welding may include:

1. Fume from the base metals and filler or electrode metals.
2. Fume from coatings (e.g., zinc oxide from galvanized surfaces, thoria from T.I.G. welding, and fluorides and NO₂ from electrode coatings).
3. Ozone due to ionization of oxygen by the ultraviolet light from arc welding.
4. Carbon monoxide from ultraviolet effects on carbon dioxide in shield gas.
5. Shield gases such as carbon dioxide, helium and argon.
6. Fluoride gases and other thermal decomposition products of fluxes and electrode coatings.
7. Flammable gases such as acetylene.

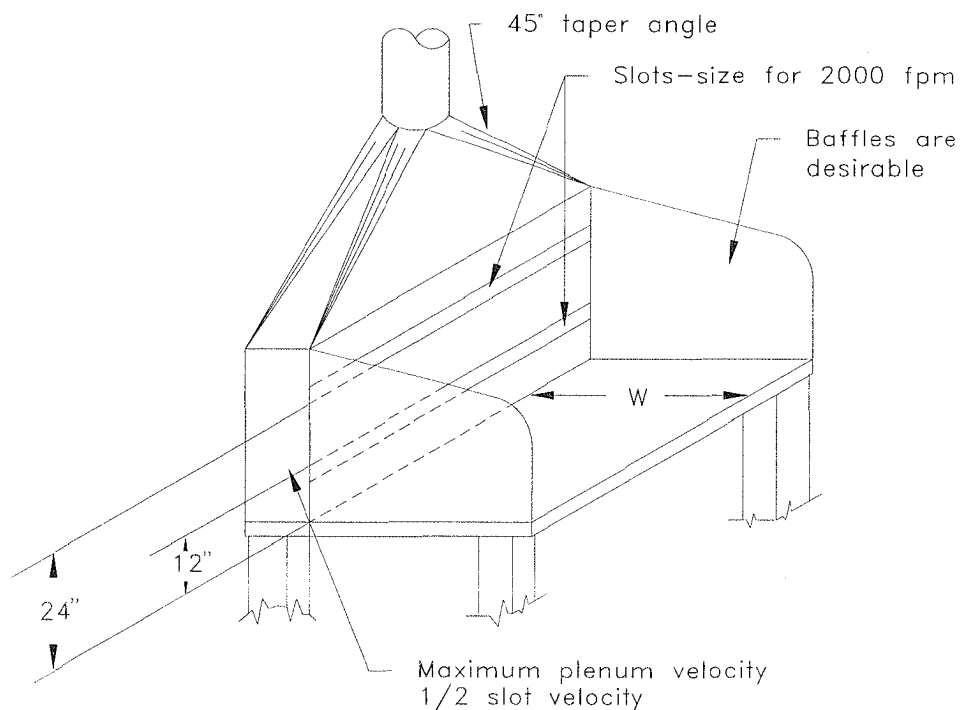
There are welding tasks that present enhanced hazards such as welding on materials containing or contaminated with heavy metals or welding in the presence of flammable vapors or halogenated hydrocarbons. If such welding is required, extraordinary precautions must be taken on a case-by-case basis. Even in the absence of such hazard materials, any

welding operation in a confined space is potentially lethal and requires continuous and copious dilution ventilation.

10.90.2 General Recommendations:

1. Choose hood designs in the following descending order of effectiveness: enclosing hoods; vacuum nozzles; fixed slot/plenum hood on a worktable or rectangular hood fixed above a worktable; moveable hood above a worktable; moveable hood hanging freely or overhead canopy; dilution ventilation.
2. Integrate planning for ventilation systems with planning for materials handling.
3. Place welding curtains or other barriers to block cross-drafts.
4. Install turntables, work rests, and other aids to improve utilization of the hoods.
5. Avoid recirculating filtered air from welding hoods back into occupied spaces unless the welding is low hazard and produces low quantities of gaseous contaminants.
6. Face velocity for enclosing hoods should be 100–130 fpm with the higher values used for poor conditions such as high cross-draft velocities.
7. Capture velocity for non-enclosing hoods should be 100–170 fpm with the higher values used for poor conditions such as high cross-draft velocities and with higher hazard levels.

Enclosing hoods are by far the most effective in controlling welding contaminants; however, they restrict access and force reconsideration of material and product handling. Capturing hoods are less effective than enclosures but for low hazard conditions can be adequate if properly used.



$Q = 350 \text{ cfm/ft of hood length}$
 Hood length = required working space
 $W = 24''$ maximum, if $W > 24''$ see chapter 3
 Minimum duct velocity = 2000 fpm
 $h_e = 1.78 VP_s + 0.25 VP_d$

General ventilation, where local exhaust can not be used:

Rod, diam.	Cfm/welder
5/32	1000
3/16	1500
1/4	3500
3/8	4500

or

- A. For open areas, where welding fume can rise away from the breathing zone:
 cfm required = $800 \times \text{lb/hour rod used}$
- B. For enclosed areas or positions where fume does not readily escape breathing zone:
 cfm required = $1600 \times \text{lb/hour rod used}$

For toxic materials higher airflows are necessary and operator may require respiratory protection equipment.

Other types of hoods

Local exhaust: See VS-90-02

Booth: For design see VS-90-30

$Q = 100 \text{ cfm/ft}^2$ of face opening

MIG welding may require precise air flow control

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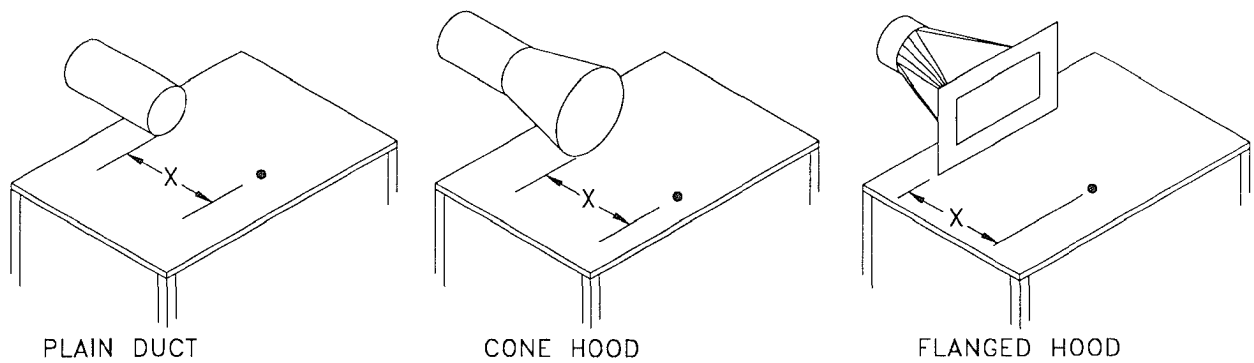
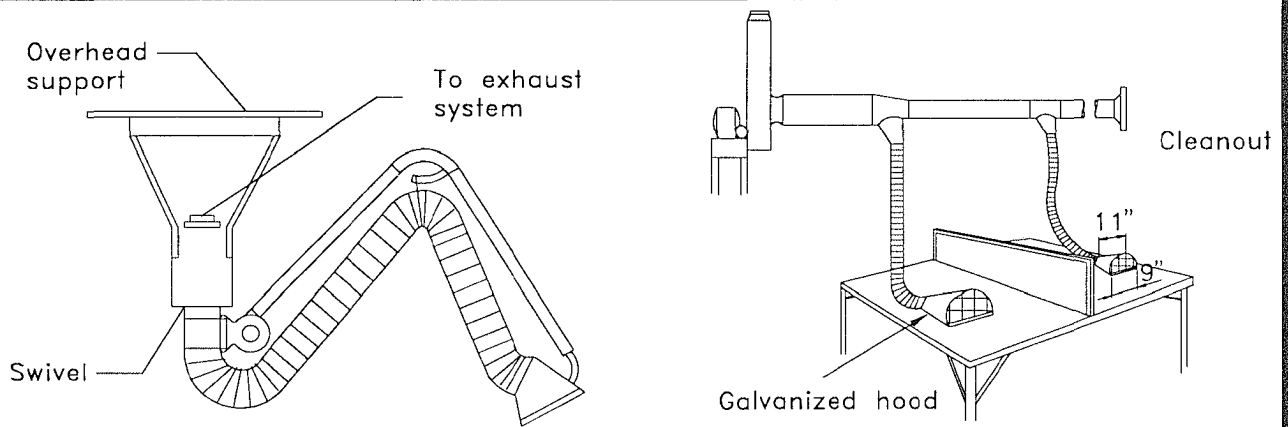
WELDING VENTILATION BENCH HOOD

DATE

1-91

FIGURE

VS-90-01



RATE OF EXHAUST

X, inches	Plain duct cfm	Flange or cone, cfm
Up to 6	335	250
6-9	755	560
9-12	1335	1000

Face velocity = 1500 fpm
 Minimum duct velocity = 3000 fpm
 Plain duct entry loss = $0.93 VP_d$
 Flange or cone entry loss = $0.25 VP_d$

Notes:

1. Locate work as close as possible to hood.
2. Hoods perform best when located to the side of the work.
3. Ventilation rates may be inadequate for toxic materials.
4. Velocities above 100-200 fpm may disturb shield gas.

GENERAL VENTILATION, where local exhaust cannot be used :

Rod, diam.	cfm/welder
5/32	1000
3/16	1500
1/4	3500
3/8	4500

OR

- A. For open areas, where welding fume can rise away from the breathing zone:
cfm required = $800 \times \text{lb/hour rod used}$
- B. For enclosed areas or positions where fume does not readily escape breathing zone:
cfm required = $1600 \times \text{lb/hour rod used}$

For toxic materials higher airflows are necessary and operator should use respiratory protection equipment.

Other types of hoods

Bench, see VS-90-01

Booth, for design see VS-90-30

$Q = 100 \text{ cfm/ft}^2$ of face opening

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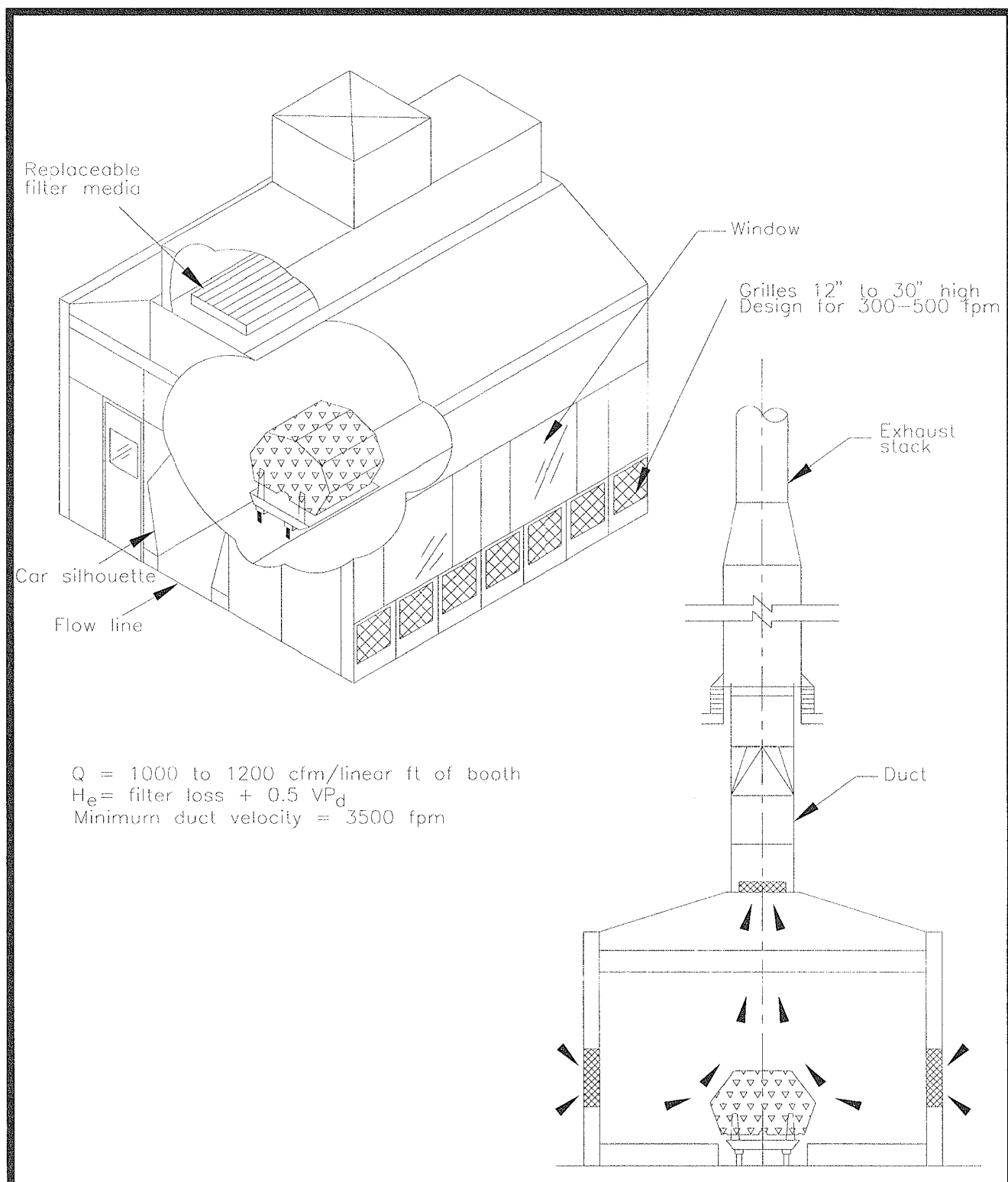
WELDING VENTILATION
MOVABLE EXHAUST HOODS

DATE

1-91

FIGURE

VS-90-02

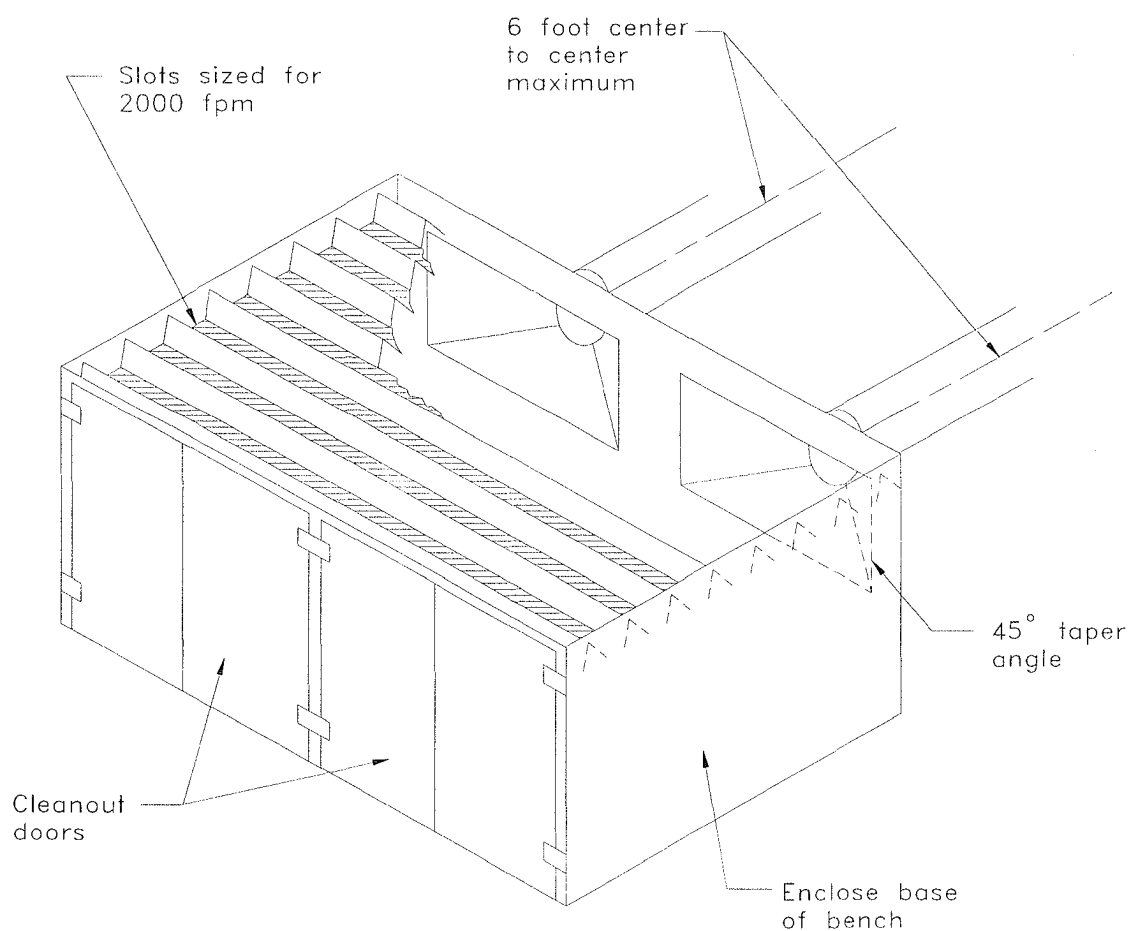


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PRODUCTION LINE
WELDING BOOTH

DATE 2-91

FIGURE VS-90-03



$$Q = 150 \text{ cfm/ft}^2 \text{ of gross bench area}$$

$$\text{Minimum duct velocity} = 4000 \text{ fpm}$$

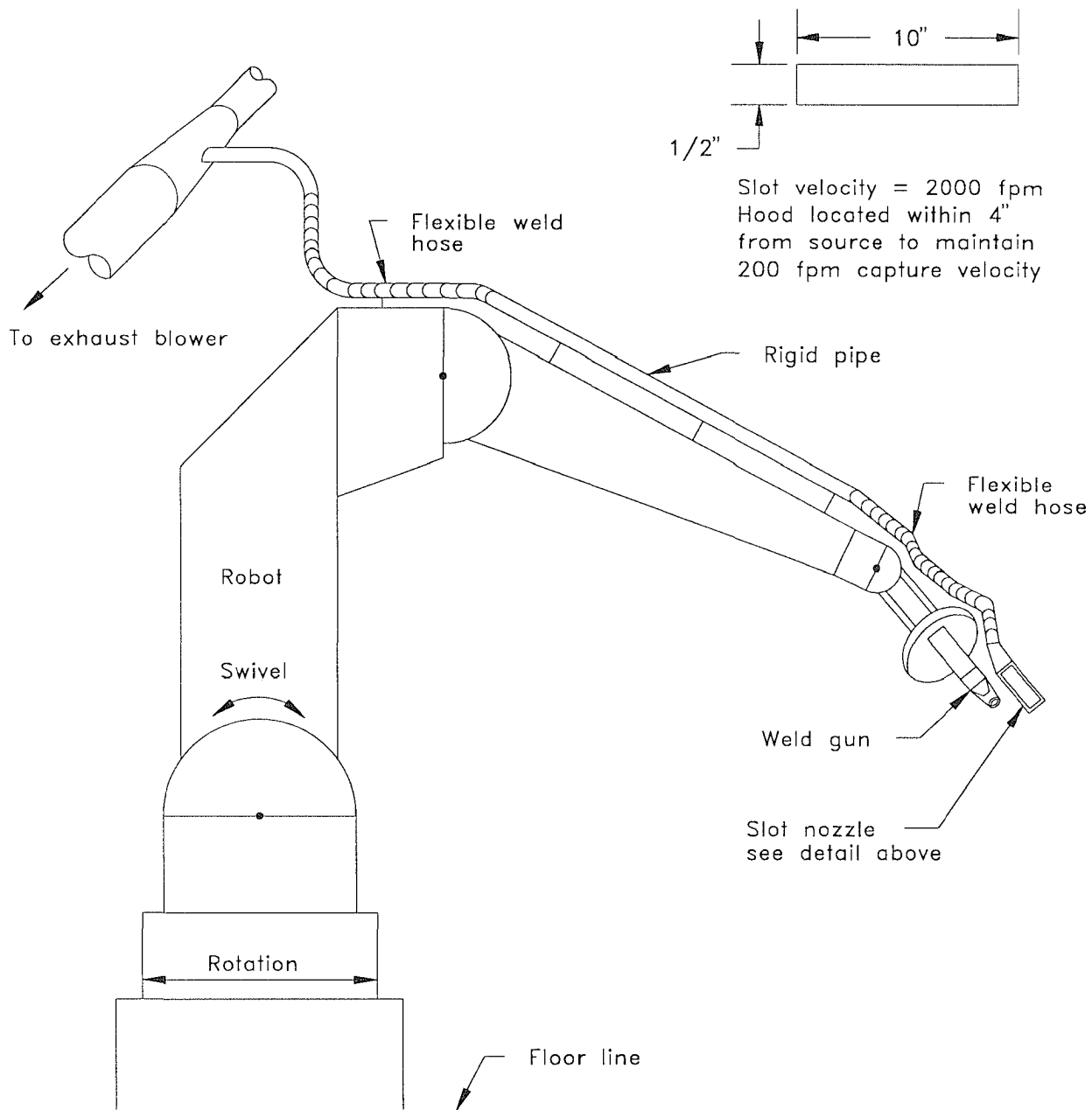
$$h_e = 1.78 VP_s + 0.25 VP_d$$

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TORCH CUTTING VENTILATION

DATE 1-91

FIGURE VS-90-10



$$h_e = 1.78 V_p^2 + 0.25 V_{p_d}^2$$

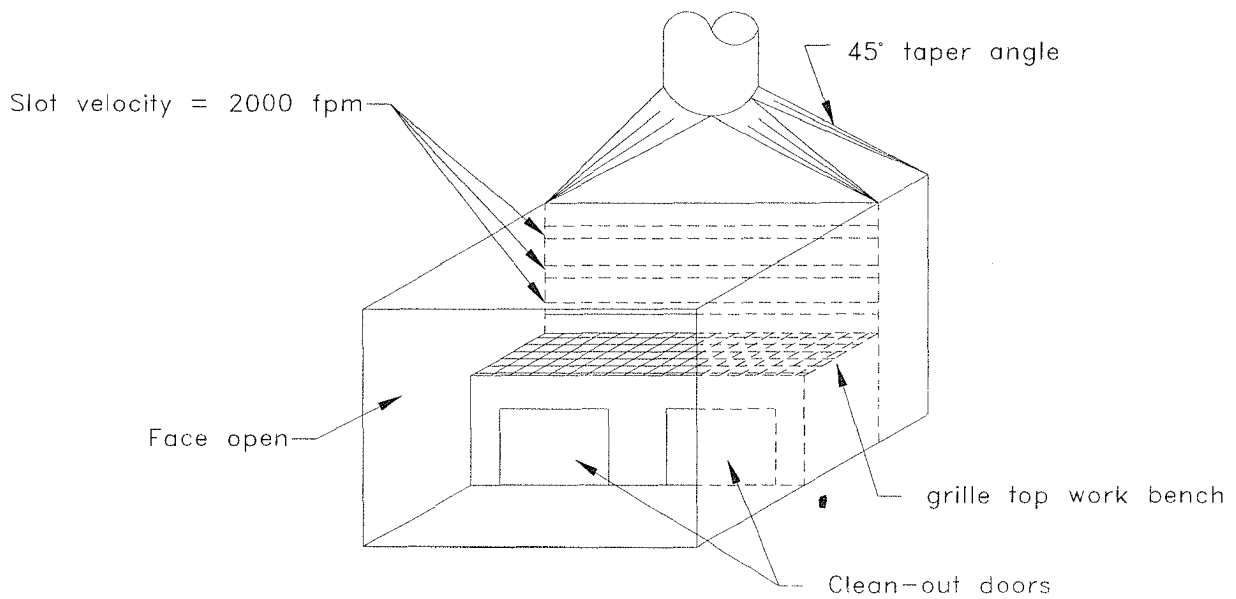
Minimum duct velocity = 3500 fpm

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ROBOTIC APPLICATION

DATE 02-91

FIGURE VS-90-20



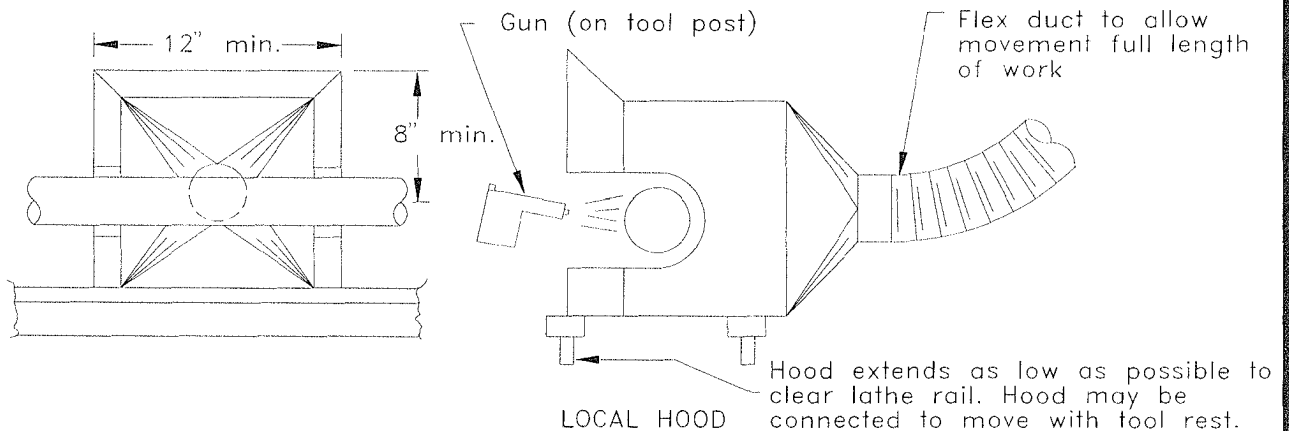
METALLIZING BOOTH

Non-toxic: $Q = 150 \text{ cfm/ft}^2 \text{ face area}$ Toxic: Provide appropriate NIOSH certified respirator
 $Q = 200 \text{ cfm/ft}^2 \text{ face area}$

Minimum duct velocity = 3500 fpm

$$h_e = 1.78 VP_s + 0.25 VP_d$$

Small lathe, etc., may be mounted in booth



Note: Local hood may not be satisfactory for spraying toxic metals.

$$Q = 200 \text{ cfm/ft}^2 \text{ face openings}$$

Minimum duct velocity = 3500 fpm

$$h_e = 0.25 VP_d$$

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METAL SPRAYING

DATE

1-91

FIGURE VS-90-30

10.95 WOODWORKING

Woodworking equipment generates large amounts of wood dust by abrasive or cutting action. It is important to provide good ventilation for all equipment as the broad particle size distribution of wood dust creates the potential for health and housekeeping problems as well as fire hazards. Excessive amounts of dust, if allowed to accumulate inside equipment and in shop areas, can create fire or explosion hazards. An additional consideration should be the toxicity of the wood species used.

In many instances, woodworking equipment, such as saws and sanders, generates air flow patterns which make dust control difficult. Exhaust hoods should enclose the operation as much as possible. Where the equipment tends to eject wood dust (e.g., at sanding belt pulleys) the exhaust hood should be placed in the ejection path.

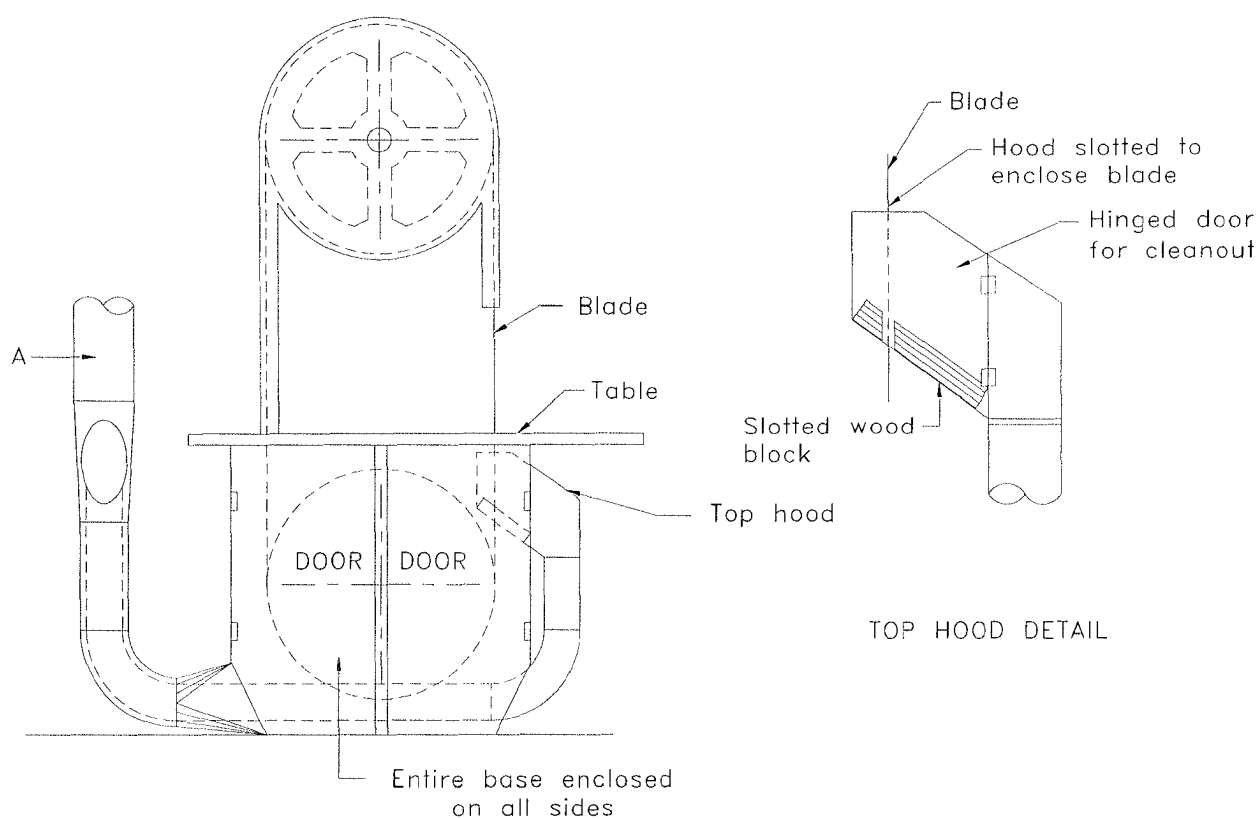
Enclosures must incorporate cleanout doors to prevent dust build-up. Duct velocities should be maintained at a minimum of 3500 fpm to prevent settling and subsequent clogging of the duct.

Exhaust flow rates will vary with equipment type and size. Design data are provided for a number of operations in

VS-10-95-1 through 10-95-20 and in Table 10.95.1. Additional information for hand-held sanders using Low Volume-High Velocity (LV-HV) can be found in sub-section 10.40. Where information for a specific operation is not provided, data for similar listed operations can be used.

REFERENCES

- 10.95.1 Hampl, V.; Johnston, O.: Control of Wood Dust from Horizontal Belt Sanding. *American Industrial Hygiene Association Journal* 46(10):567-577 (1985).
- 10.95.2 Hampl, V.; Johnston, O.: Control of Wood Dust from Disc Sanders. *Applied Occupational Hygiene* 6(11):938-944 (November 1991).
- 10.95.3 Topmiller, J.L.; Watkins, D.A.; Schulman, S.A.; Murdock, D.J.: Controlling Wood Dust from Orbital Hand Sanders. *Applied Occupational Hygiene* 11(9):1131-1138 (September 1996).
- 10.95.4 Hampl, V.; Topmiller, J.L.; Watkins, D.S.; Murdock, D.J.: Control of Wood Dust from Rotational Hand-Held Sanders. *Applied Occupational Environmental Hygiene* 7(4):263-270 (April 1992).



Blade width, inches	Exhaust flow rate, cfm		
	Bottom	Top	Total
Up to 2	350	350	700
2 to 3	350	550	900
3 to 4	550	800	1350
4 to 6	550	1100	1650
6 to 8	550	1400	1950

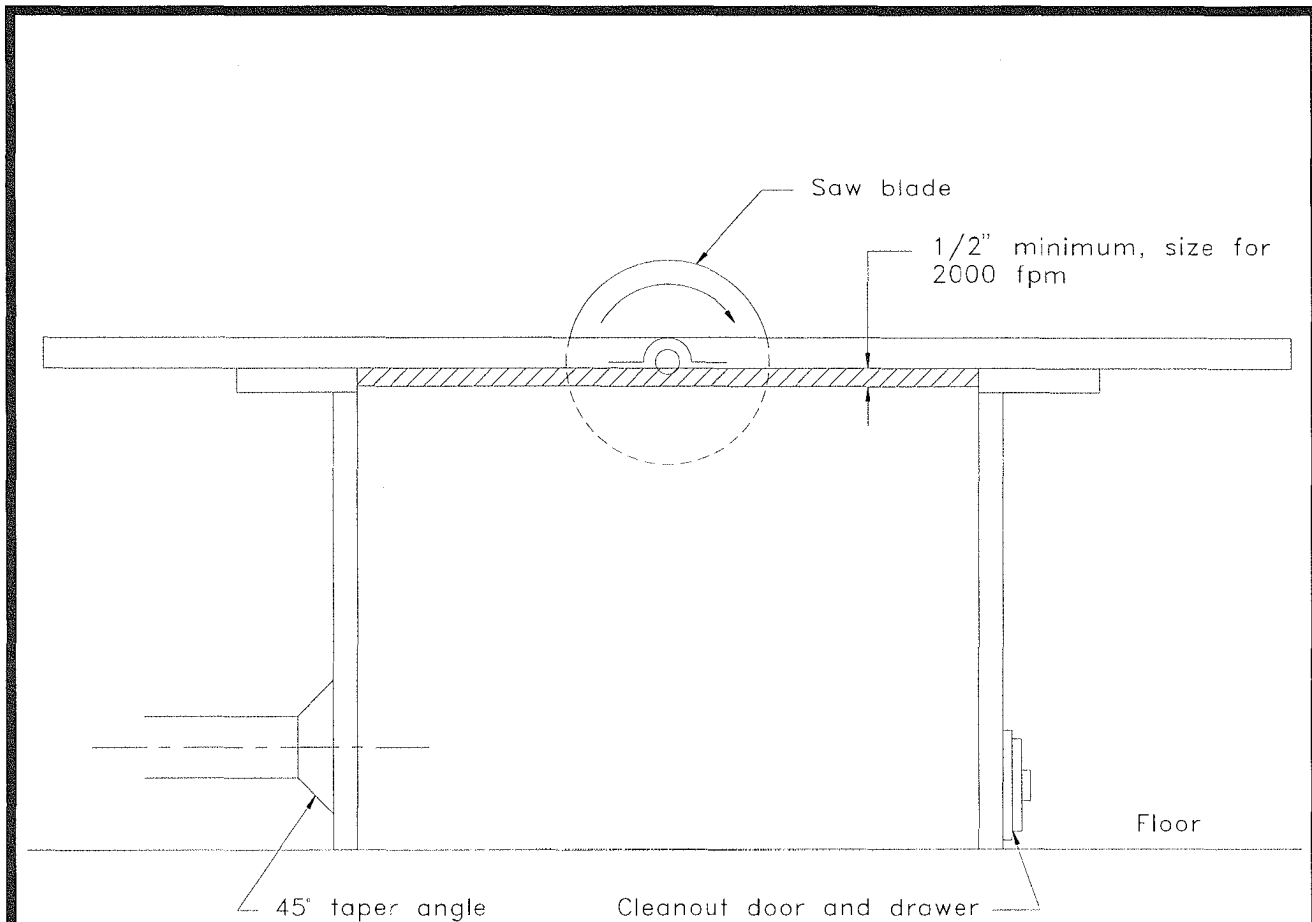
Minimum duct velocity = 3500 fpm
 $h_e = 1.75 VP_d$ (Point A in duct riser)

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BAND SAW

DATE 12-90

FIGURE VS-95-01



Saw blade diameter, inches	Exhaust flow rate, cfm
Up to 16	350
16 to 24	440
over 24	550
Saw with dado blade	550

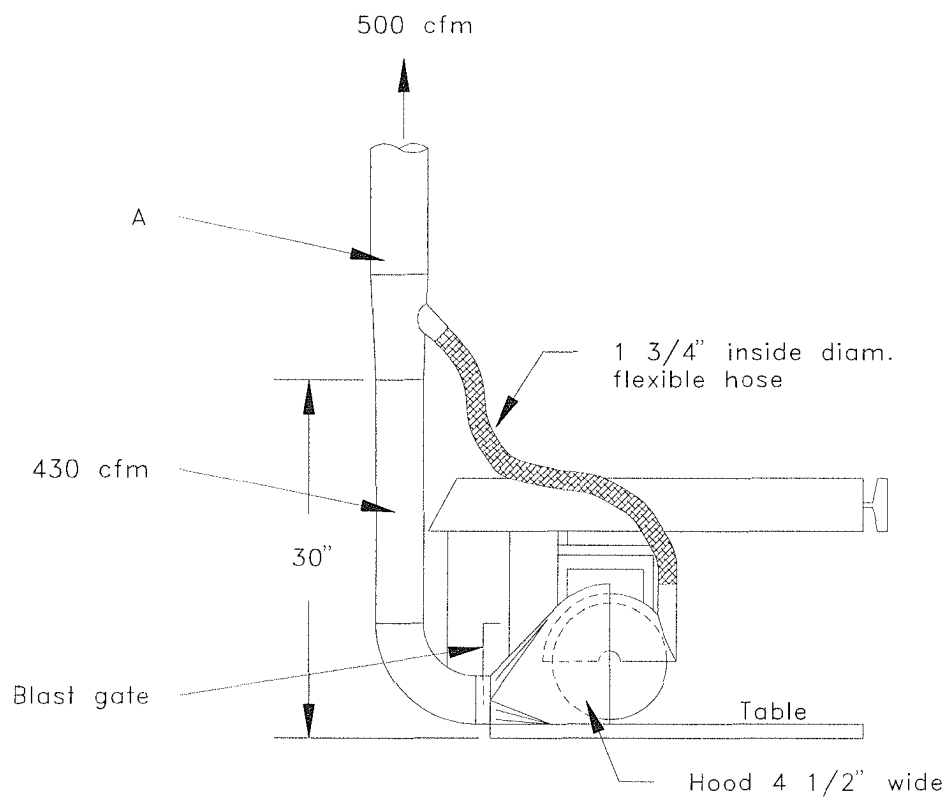
Minimum duct velocity = 4000 fpm
 $h_e = 1.78 VP_s + 0.25 VP_d$

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FLOOR TABLE SAW

DATE *12-90*

FIGURE *VS-95-02*



Minimum duct velocity = 4000 fpm
 $h_e = 3.5 VP_d$ (point A in duct riser)

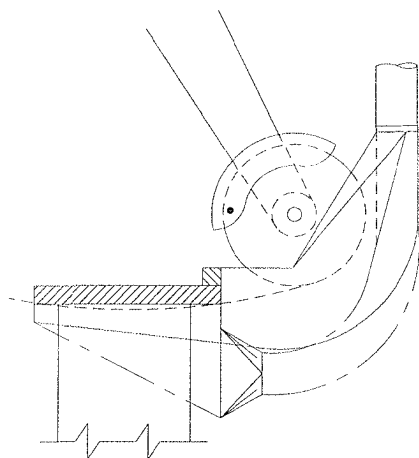
For booth enclosure, see VS-80-17

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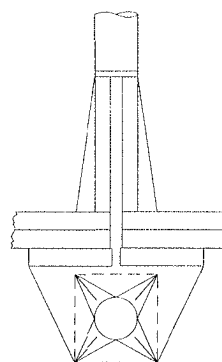
RADIAL ARM SAW

DATE 12-90

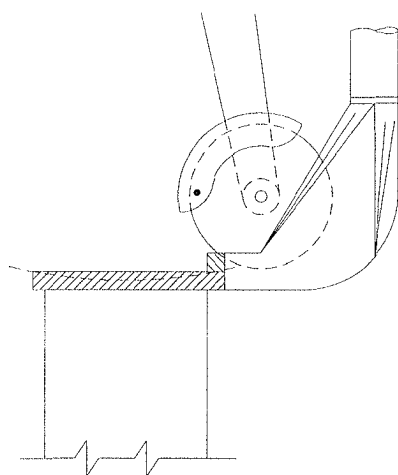
FIGURE VS-95-03



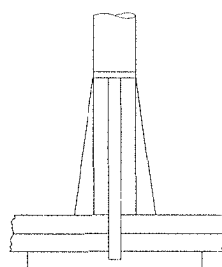
Type hood where table
is cut through



Front view
of hood



Type hood where table
is not cut through



Front view
of hood

Saw diameter, inches	Exh. flow rate cfm
Up to 20 incl.	350
over 20	440

Minimum duct velocity = 4000 fpm

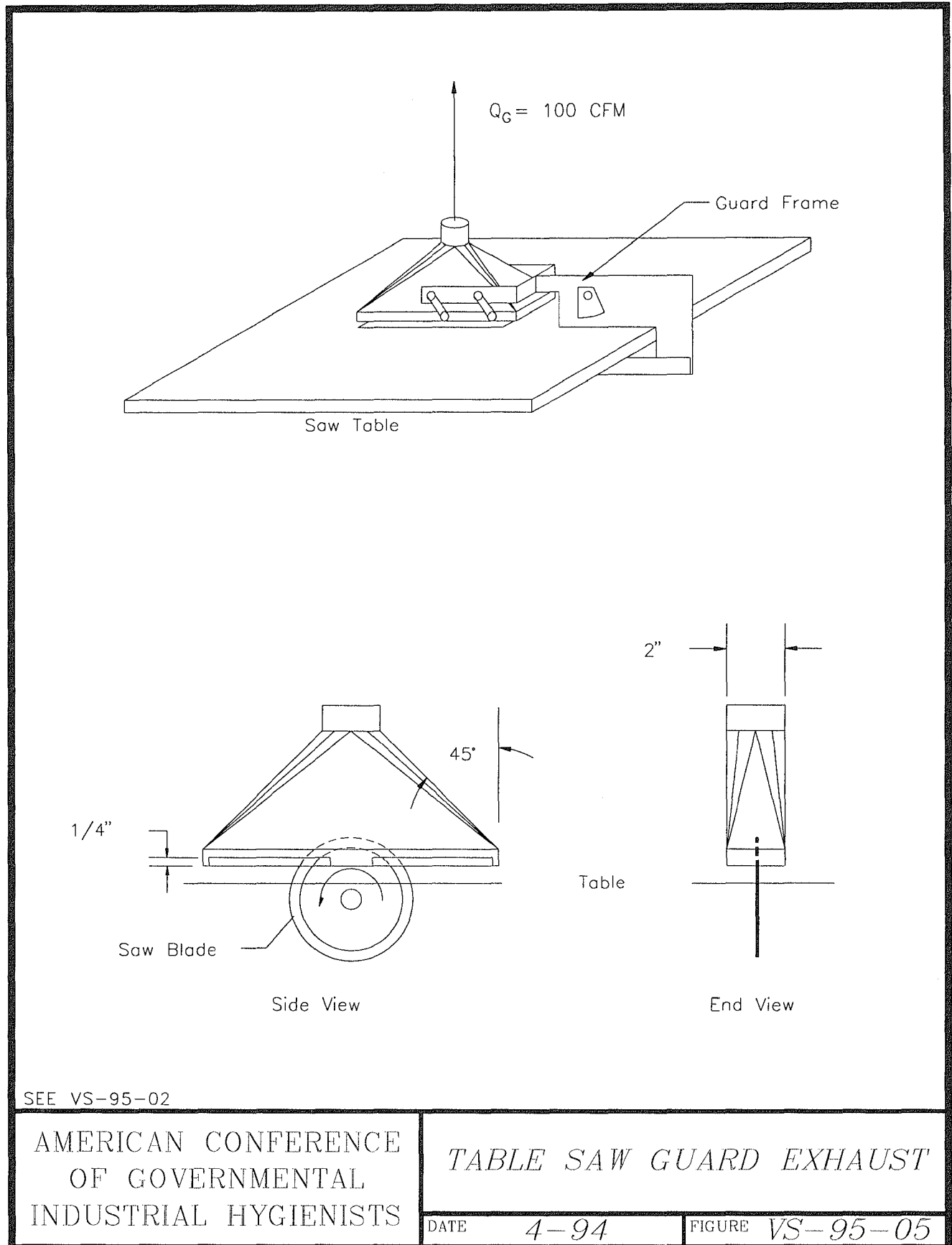
$$h_e = 1.78 VP_s + 0.25 VP_d$$

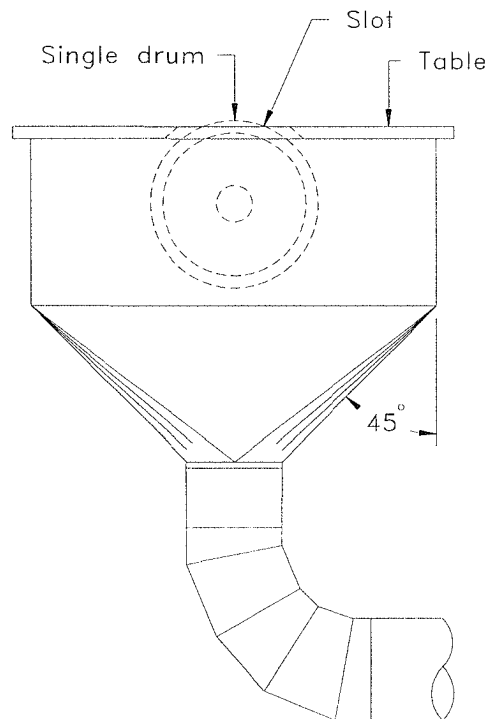
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SWING SAW

DATE 12-90

FIGURE VS-95-04





Drum surface square inches	Exhaust flow rate cfm
Up to 200 (and less than 10" diam.)	350
200 to 400	550
400 to 700	790
700 to 1400	1100
1400 to 2400	1400

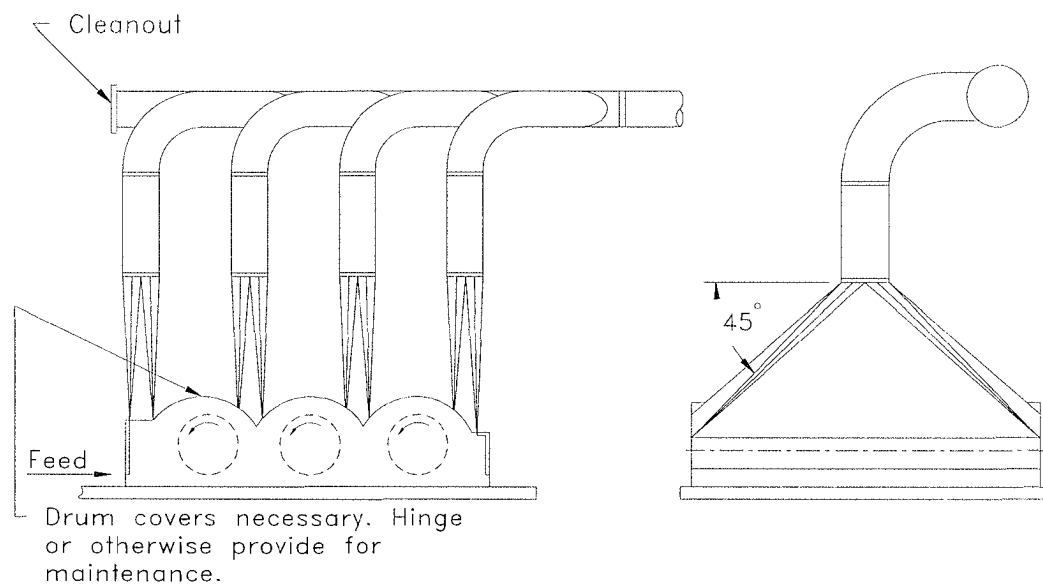
Minimum duct velocity = 3500 fpm
 Entry loss depends on hood design.
 $h_e = 1.78 VP_s + 0.25 VP_d$ as illustrated

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SINGLE DRUM SANDER

DATE 12-90

FIGURE VS-95-10



Exhaust flow rates

Drum length	Total exhaust for machine cfm/drum *
Up to 31"	550
31" to 49"	790
49" to 67"	1100
over 67"	1400
Brush rolls	350 cfm at brush

Note: Provide one more take off than the number of drums.

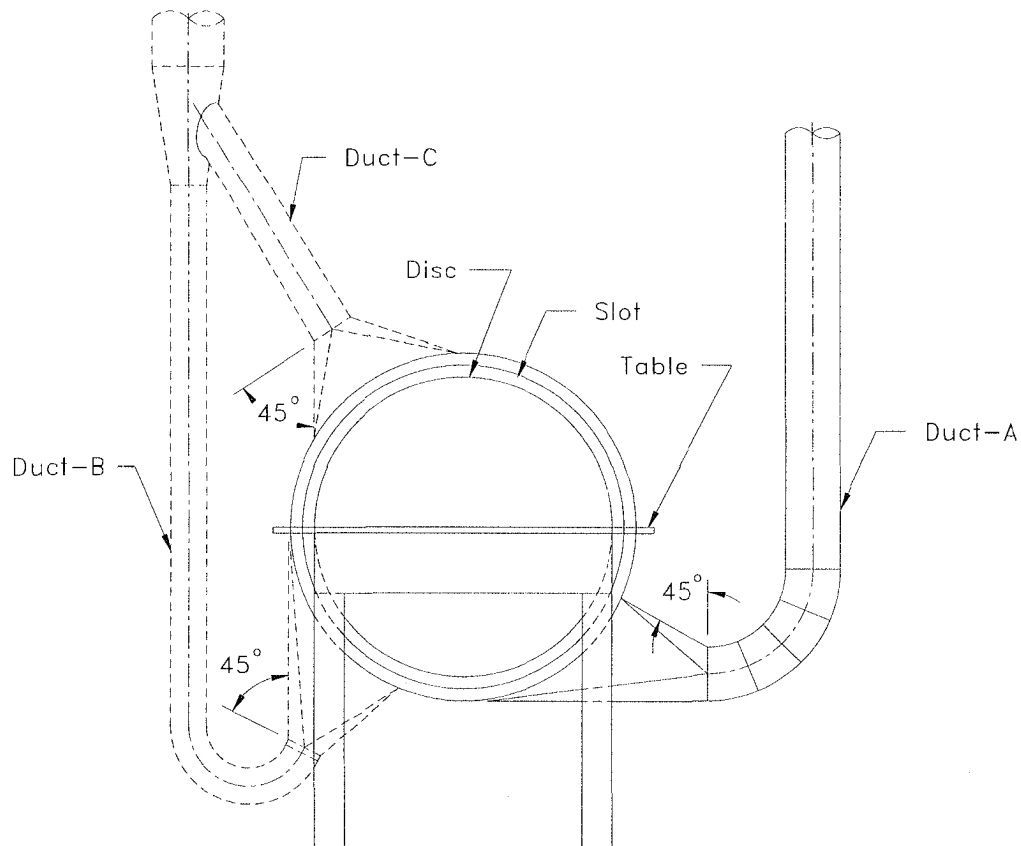
Minimum duct velocity = 3500 fpm
 $h_e = 0.25 VP_d$

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MULTIPLE DRUM SANDER

DATE 12-90

FIGURE VS-95-11



Disc diameter, inches	Total exhaust flow rate cfm	Applies to duct
Up to 12	350	A
12 to 18	440	A
18 to 26	550	A
26 to 32	700 *	A-B
32 to 38	900 *	A-B
38 to 48	1300 **	A-B-C

* Two bottom branches.

** One top and two bottom branches.

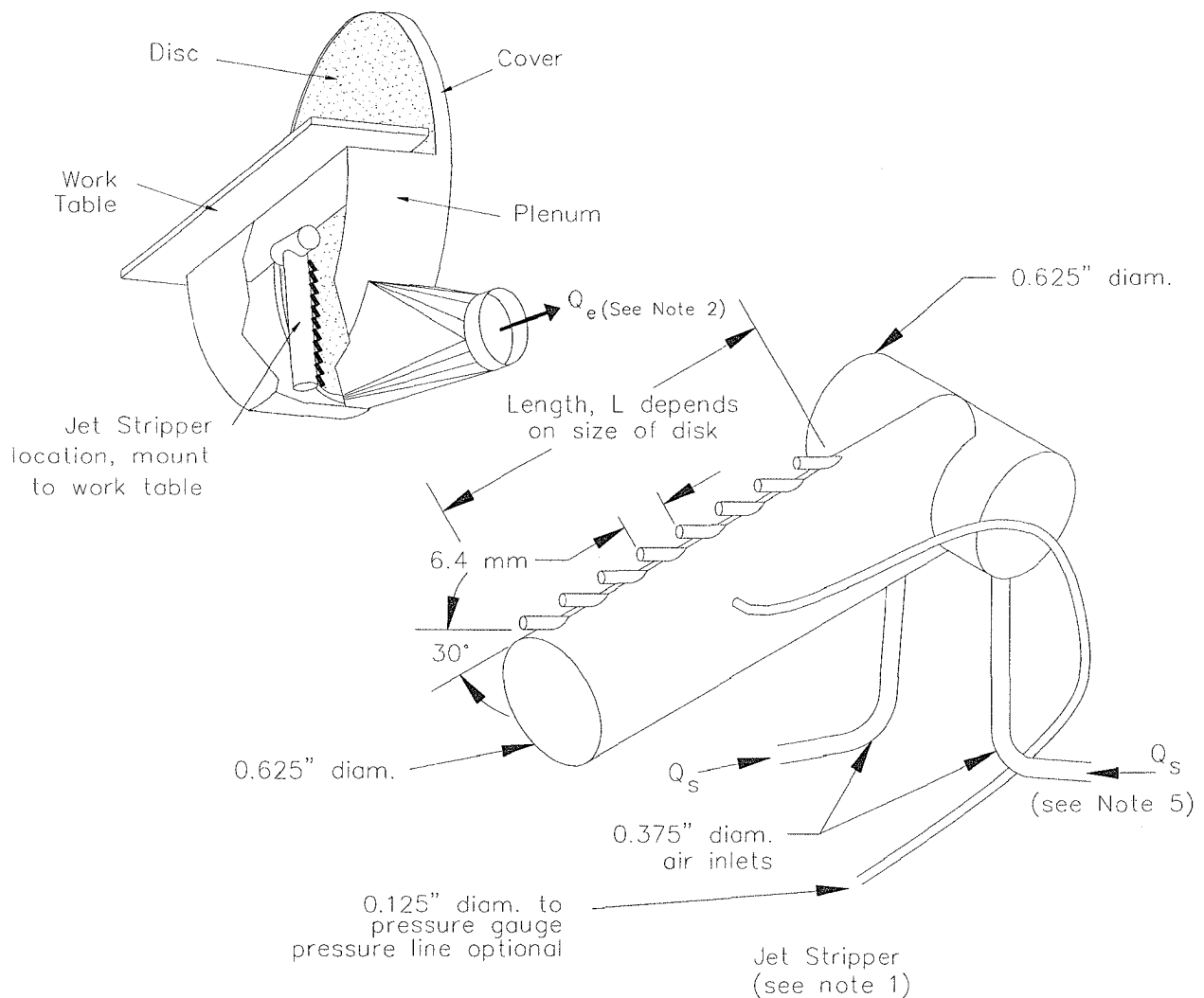
Minimum duct velocity = 3500 fpm
 Entry loss depends on hood design.
 $h_e = 1.0 VP_s + 0.25 VP_d$ as illustrated

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DISC SANDERS

DATE 12-90

FIGURE VS-95-12



NOTES:

1. Jet stripper may be added as shown. It is used in addition to the ventilation specified in VS-95-12.
2. Q_e is same as shown in VS-95-12.
3. Clearance between stripper nozzle and disk is 0.25".
4. Nozzle opening diameter is 0.035".
5. Total air flow (Q_s) to stripper = 1.8 L. (L in inches)
6. Stripper inlet pressure = 15 psi.

The Jet Stripper is patented (#5,099,616). Use of the device (not for sale) is permissible. To obtain information regarding license for commercial production of the device contact NIOSH, 1-800-35NIOASH.

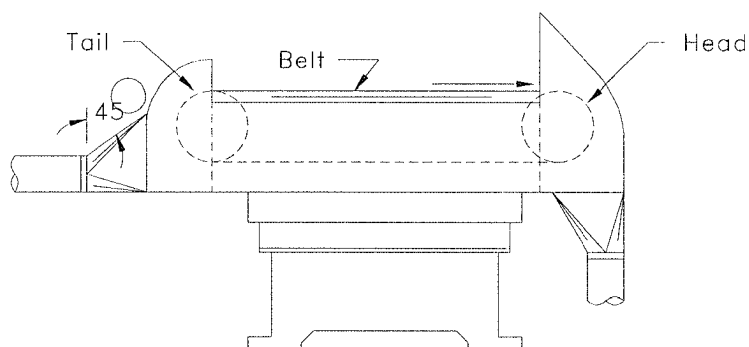
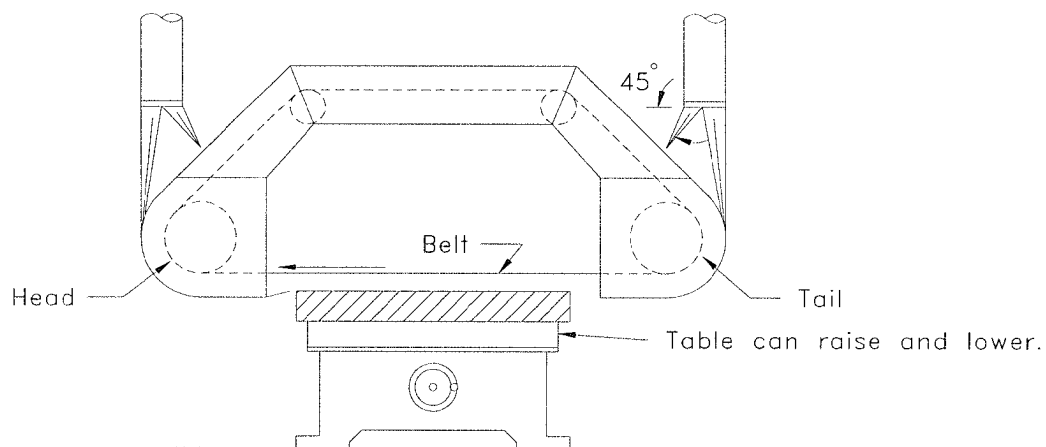
Reference 10.95.2

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OPTIONAL JET STRIPPER
FOR DISK SANDER

DATE 11/96

FIGURE VS-95-12a



HORIZONTAL BELT SANDERS

Belt width, inches	Exhaust flowrate CFM		
	Head end	Tail end	Total
up to 6	440	350	790
6 to 9	550	350	900
9 to 14	800	440	1200
over 14	1100	550	1700

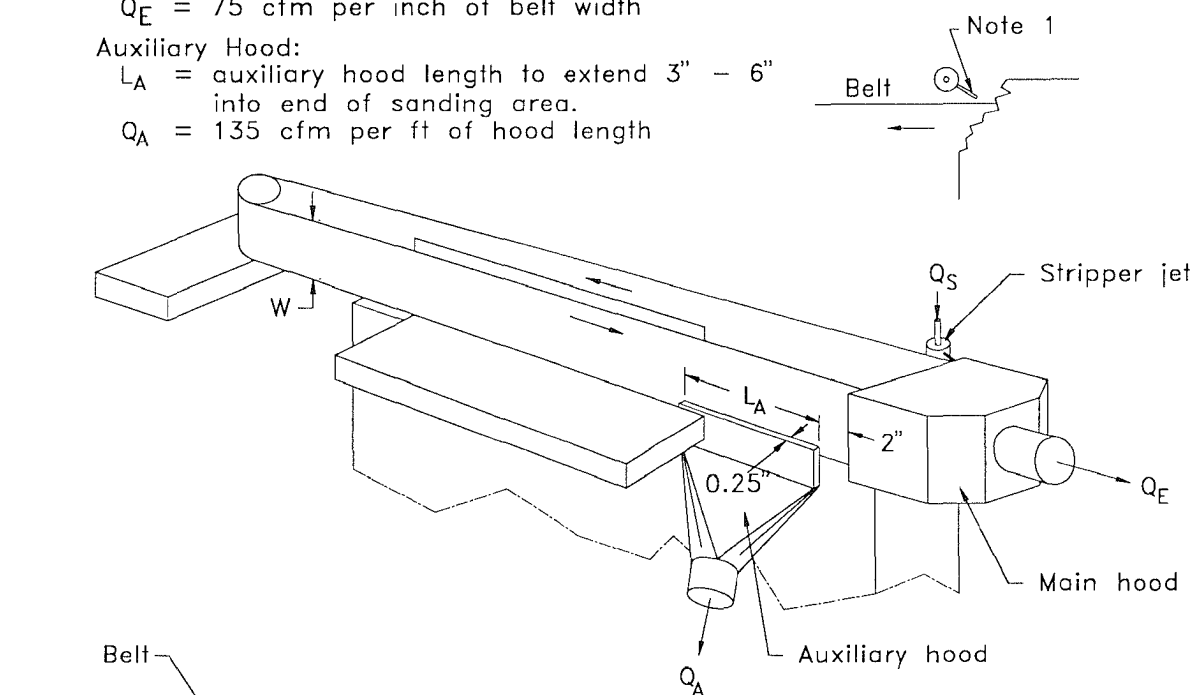
Minimum duct velocity = 3500 fpm
 $h_e = 0.40 VP_d$

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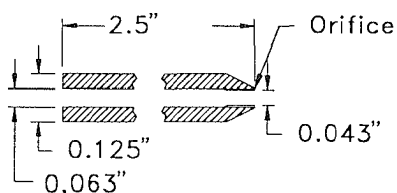
HORIZONTAL BELT SANDERS

DATE 12-90

FIGURE VS-95-13

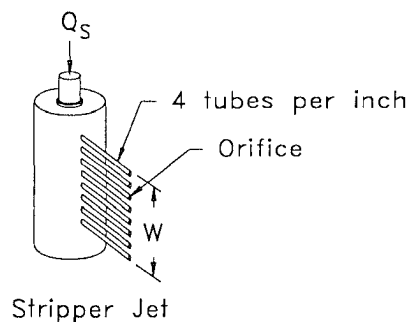
Main Hood: W = belt width, in. Q_E = 75 cfm per inch of belt width**Auxiliary Hood:** L_A = auxiliary hood length to extend 3" - 6" into end of sanding area. Q_A = 135 cfm per ft of hood length**Notes:**

1. Tips of stripper jets must be positioned 0.25" from belt and inside hood face.
2. Keep clearances between hood, belt and workrest to a minimum.



Tube and Orifice Detail

Reference 10.95.1

 W = belt width, in. Q_S = 0.5 cfm per tube,
inlet pressure 10 - 12 psig

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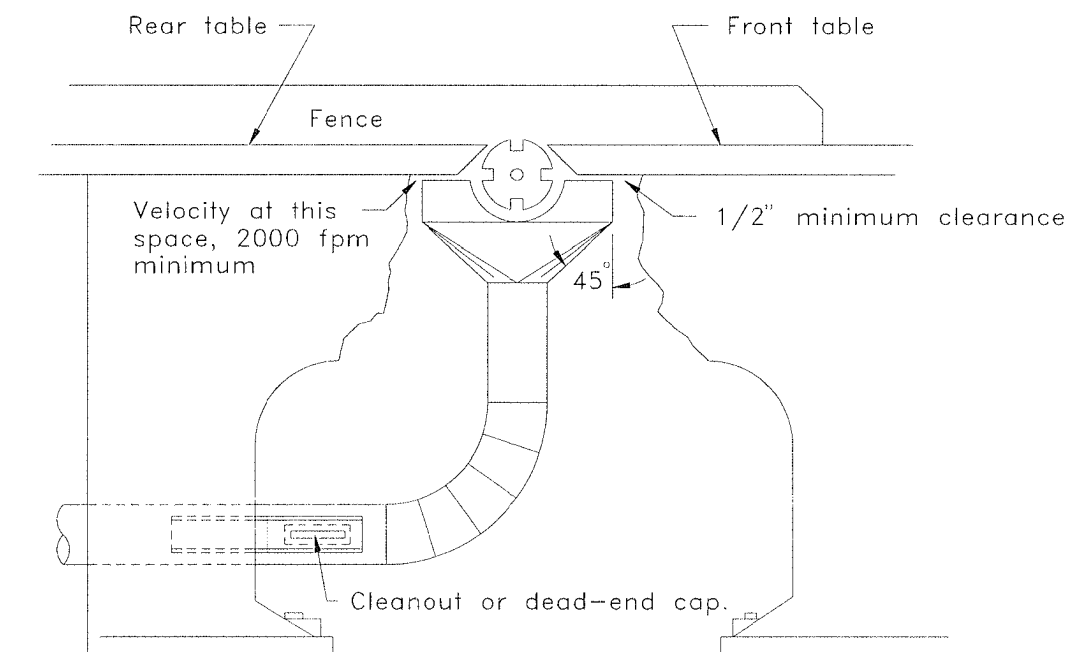
*HORIZONTAL BELT SANDER
PUSH-PULL SYSTEM*

DATE

12-90

FIGURE

VS-95-14



Knife length, inches	Exhaust flowrate cfm
Up to 6	350
6 to 12	440
12 to 20	550
over 20	800

Minimum duct velocity = 4000 fpm
 $h_e = 1.0 VP_s + 0.25 VP_d$

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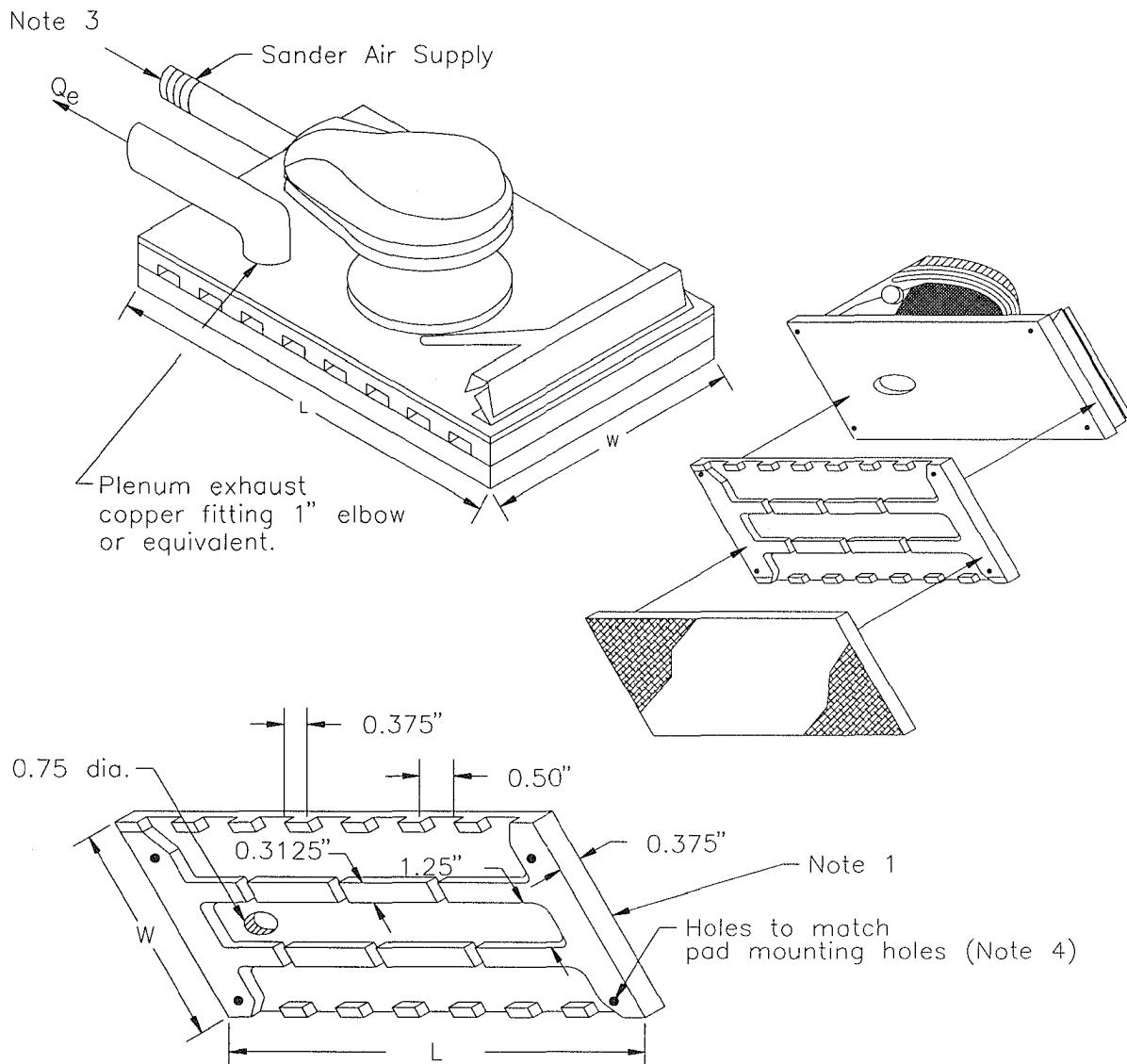
JOINTERS

DATE

12-90

FIGURE

VS-95-20



1. Plenum machined from metal or plastic.
2. Plenum can be used with air or electrically powered sander.
3. $Q_e = 50$ cfm at approximately 25" wg.
4. Mounting holes located to match sander pad mounting holes.

The Exhaust Plenum Retrofit has a patent pending. Use of the device (not for sale) is permissible. To obtain information regarding the license for commercial production of the device contact NIOSH, 1-800-35NIOASH

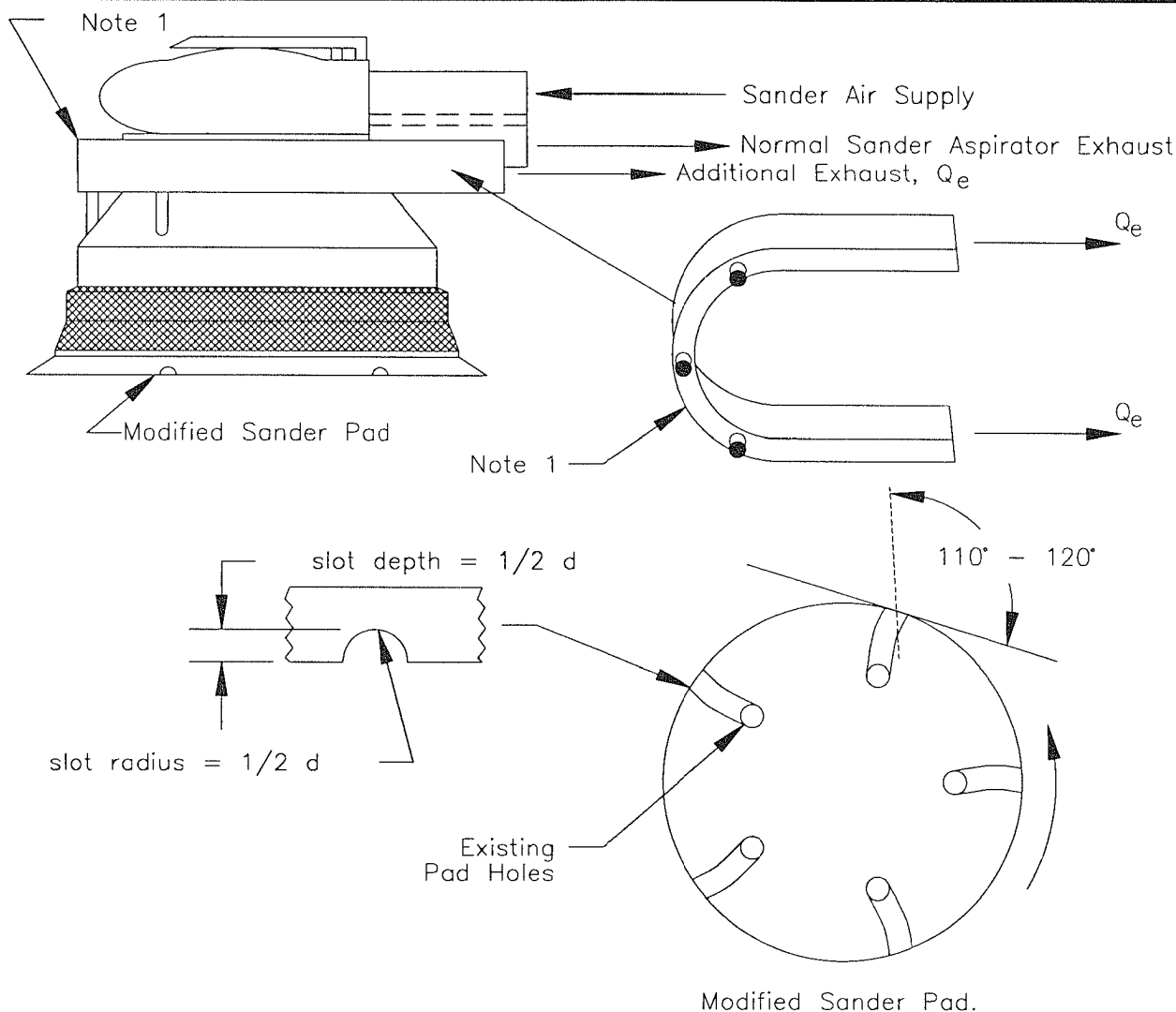
Reference: 10.95.3

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EXHAUST PLENUM RETROFIT FOR ORBITAL HAND SANDER

DATE 12-96

FIGURE VS-95-30



NOTES:

1. Additional exhaust plenum to fit sander.
2. $Q_e = 50$ cfm at 25" wg
3. Q_e is in addition to normal sander aspirator exhaust.
4. Additional exhaust may be supplied by standard shop vacuum cleaner.

The Auxiliary Exhaust Modification Configuration is patented (#5,105,858). Use of this device (not for sale) is permissible. To obtain information regarding license for commercial production of the device contact NIOSH, 1-800-35NOSH.

Reference: 10.95.4

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AUXILIARY EXHAUST RETROFIT
FOR AIR POWERED RANDOM
ORBITAL HAND SANDER

DATE

12/97

FIGURE

VS-95-31

TABLE 10.95.1. Miscellaneous Woodworking Machinery not Given in VS Prints

The following list of recommended exhaust volumes is for average-sized woodworking machines and is based on many years of experience. It must be noted that some modern, high-speed, or extra-large machines will produce such a large volume of waste that greater exhaust volumes must be used. Similarly, some small machines of the home workshop or bench type may use less exhaust air than listed.

Self-feed Table Rip Saw			
Exhaust Flow Rate, cfm			
Saw Diameter, inches	Bottom	Top	Total
Up to 16 inclusive	440	350	790
Over 16	550	350	900
Self-feed, not on table	800	550	1350
Gang Rip Saws			
Exhaust Flow Rate, cfm			
Saw Diameter, inches	Bottom	Top	Total
Up to 24, inclusive	550	350	900
Over 24 to 36, incl.	800	440	1240
Over 36 to 48, incl.	1100	550	1650
Over 48	1400	550	2060
Vertical Belt Sanders			
(rear belt and both pulleys enclosed)			
and			
Top Run Horizontal Belt Sanders			
Belt Width, inches	Exhaust Flow Rate, cfm		
Up to 6, incl.	440		
Over 6 to 9, incl.	550		
Over 9 to 14, incl.	800		
Over 14	1100		
Swing Arm Sander: 440 cfm			
Single Planers or Surfacers			
Exhaust Flow Rate, cfm			
Up to 20" knives	785		
Over 20" to 26" knives	1100		
Over 26" to 32" knives	1400		
Over 32" to 38" knives	1765		
Over 38" knives	2200		
Double Planers or Surfacers			
Exhaust Flow Rate, cfm			
	Bottom	Top	Total
Up to 20" knives	550	785	1355
Over 20" to 26" knives	785	1100	1885
Over 26" to 32" knives	1100	1400	2500
Over 32" to 38" knives	1400	1800	3200
Over 38" knives	1400	2200	3600

Molders, Matchers, & Sizers				
Exhaust Flow Rate, cfm				
Size, inches	Bottom	Top	Right	Left
Up to 7, incl.	440	550	350	350
Over 7 to 12, incl.	550	800	440	440
Over 12 to 18, incl.	800	1100	550	550
Over 18 to 24, incl.	1100	1400	800	800
Over 24	1400	1770	1100	1100
Exhaust Flow Rate, cfm				
Sash stickers	550			
Woodshapers	440 to 1400			
Tenoner	Same as moulder			
Automatic lathe	800 to 5000			
Forming lathe	350 to 1400			
Chain mortise	350			
Dowel machine	350 to 800			
Panel raiser	550			
Dove-tail and lock corner	550 to 800			
Pulley pockets	550			
Pulley stile	550			
Glue jointer	800			
Gainer	350 to 1400			
Router	350 to 800			
Hogs				
Up to 12" wide	1400			
Over 12" wide	3100			
Floorsweep				
6" to 8" diameter	800 to 1400			

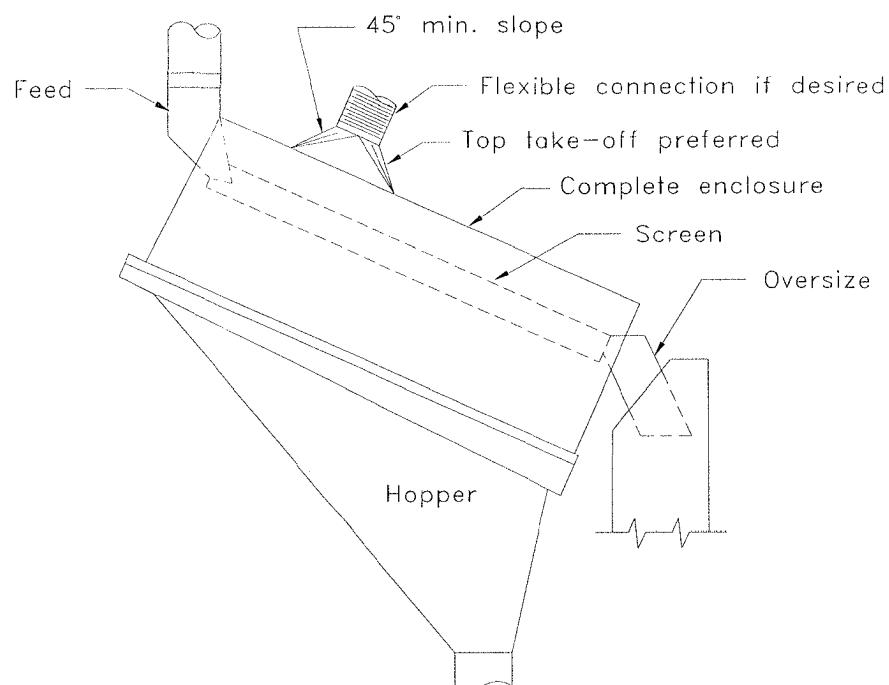
10.99 MISCELLANEOUS OPERATIONS

In the previous sections of the chapter, hood ventilation sketches were grouped together because they provided ventilation concepts for similar operations, used the same ventilation approach, or were applicable within the same industry. However, not all hood ventilation sketches are so easily categorized.

This section provides a location for those hood ventilation sketches that do not fit in other sections. Some have a unique application such as VS-99-04 for the Pistol Range. Others have such broad application that they could appear in many of sections (e.g., the canopy hood in VS-99-03). In other cases, this section will be used for new ventilation sketches for a particular application or industry. Such sketches will reside in this section until other hood ventilation sketches are developed and a new section formed. Finally, this section will be used for tabular presentation of specific design parameters for a variety of operations which could not be adequately described in previous sections.

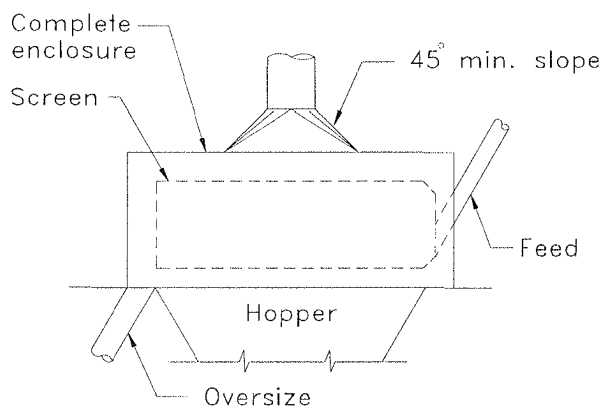
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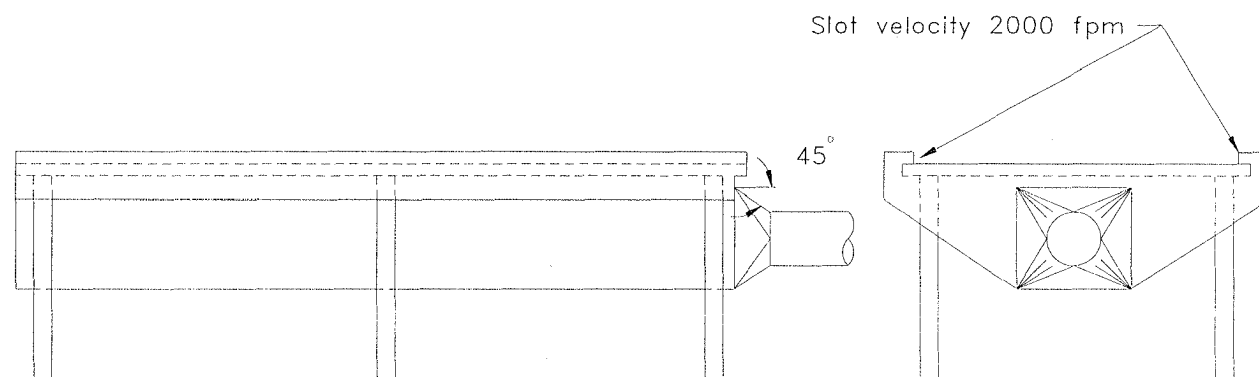
FLAT DECK SCREEN

$Q = 200 \text{ cfm/ft}^2$ through hood openings, but not less than
 50 cfm/ft^2 screen area. No increase for multiple decks
 Minimum duct velocity = 3500 fpm
 $h_e = 0.50 VP_d$



CYLINDRICAL SCREEN

$Q = 100 \text{ cfm/ft}^2$ circular cross section of
 screen; at least 400 cfm/ft^2 of
 enclosure opening
 Minimum duct velocity = 3500 fpm
 $h_e = 0.50 VP_d$



$Q = 50-100 \text{ cfm/ft}^2$ of table top.
 Minimum duct velocity = 2000 fpm

$h_e = 1.78 V_{P_s} + 0.25 V_{P_d}$

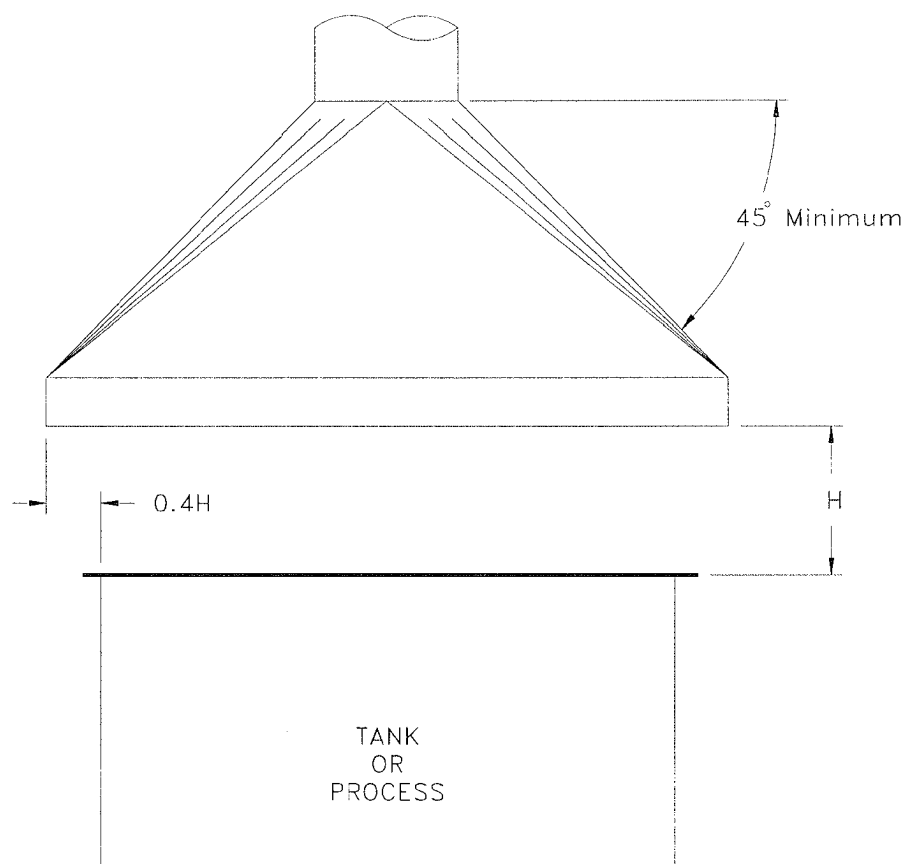
Note: See "Open Surface Tanks", VS-70-01 and VS-70-02
 for other suitable slot types. Air flow rate may
 be calculated on dilution basis if data is available.
 Maximum plenum velocity = $1/2$ slot velocity.
 Large plenum essential for good distribution.

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TABLE SLOT

DATE 12-90

FIGURE VS-99-02



Not to be used where material is toxic and worker must bend over tank or process.

Side curtains are necessary when cross-drafts are present.

$Q = 1.4PHV$	for open type canopy. P = perimeter of tank, feet. V = 50–500 fpm. See Chapter 3
$Q = (W + L)HV$	for two sides adjacent enclosed. W & L are open sides of hood. V = 50–500 fpm. See Chapter 3
$Q = WHV$ or LHV	for three sides enclosed. (booth) V = 50–500 fpm. See Chapter 3

$$h_e = 0.25 VP_d$$

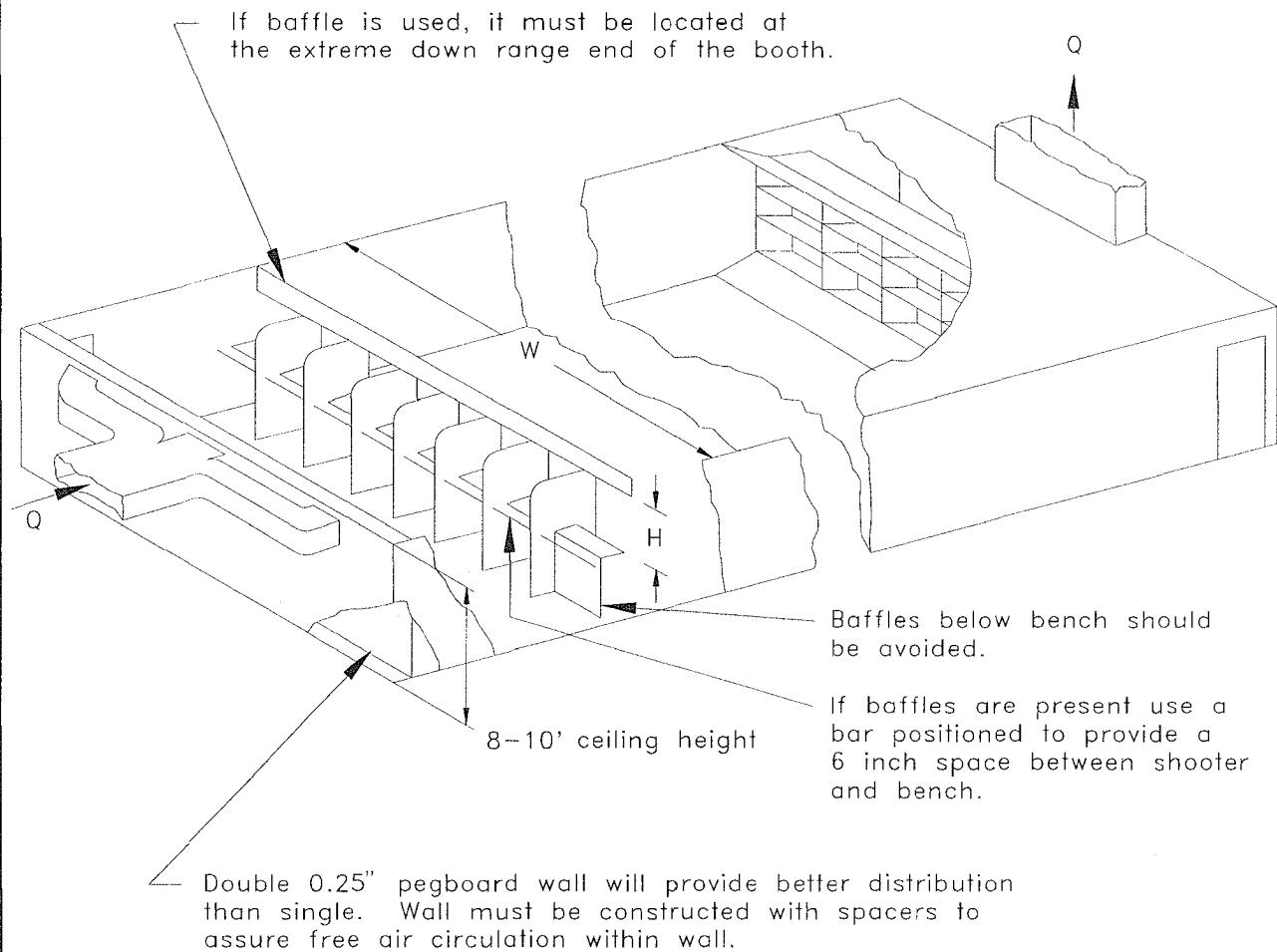
Duct velocity = 1000–3000 fpm

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CANOPY HOOD

DATE 12-90

FIGURE VS-99-03



$Q(\text{minimum}) = 50 \text{ HW}$, but not less than 20 cfm/ft^2 of room cross-sectional area

Notes:

Replacement air distribution:

Uniform air distribution necessary.
Perforated rear wall or ceiling plenum system preferred. Grilles and diffusers are not recommended.

NIOSH certified dust respirator for lead is necessary during clean-up and lead removal from bullet trap.

Acoustical material on walls, ceiling and thick fabric on bench top are recommended

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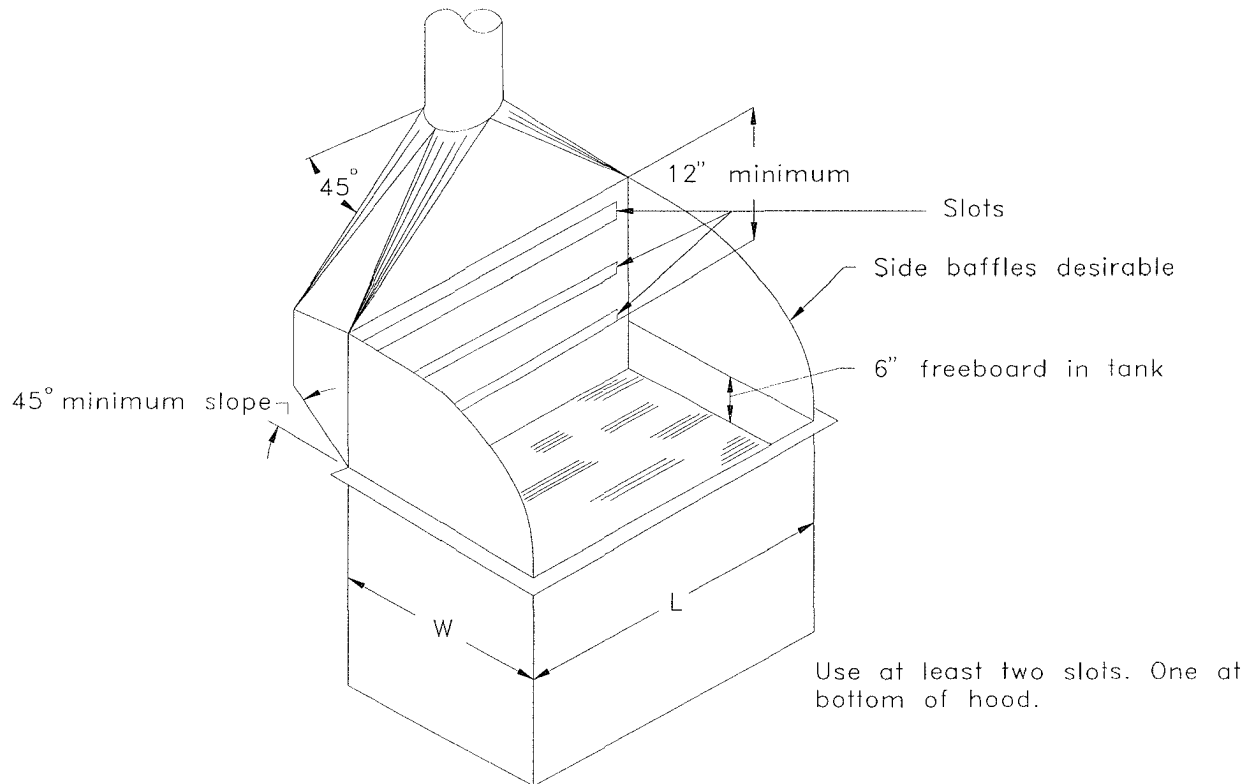
INDOOR PISTOL AND
SMALL BORE RIFLE
RANGE VENTILATION

DATE

12-90

FIGURE

VS-99-04



$Q = 150 \text{ cfm/ft}^2 \text{ of bed (150LW)}$
 Slot velocity = 2000 fpm
 $h_e = 1.78 VP_s + 0.25 VP_d$
 Minimum duct velocity = 3500 fpm
 W not to exceed 36"

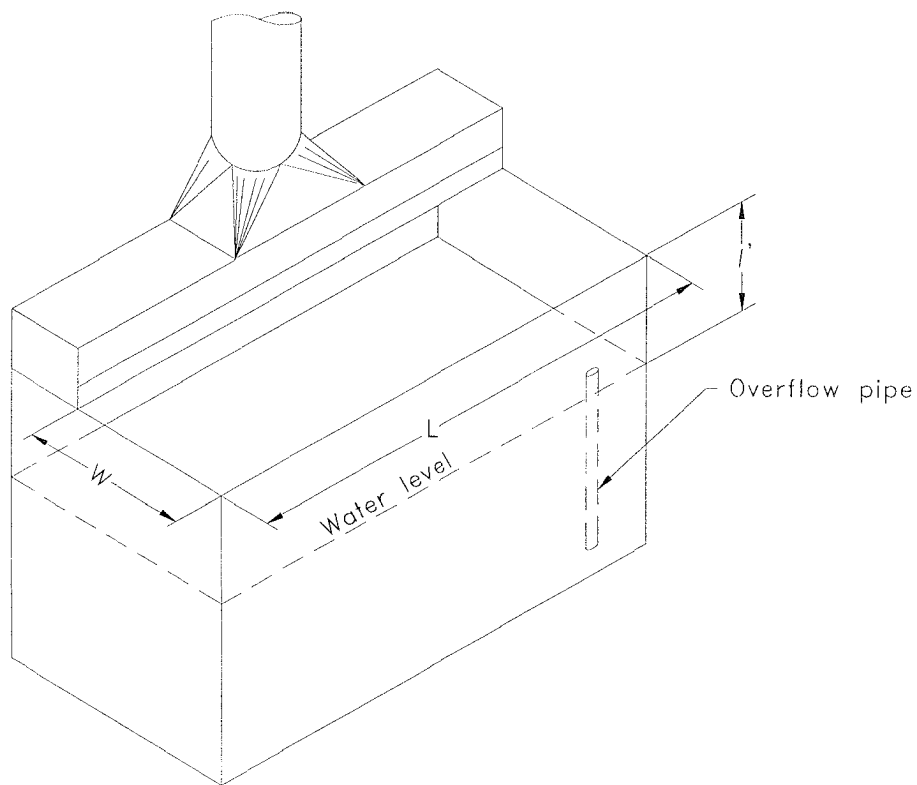
Free board must be maintained to prevent material carryout.

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FLUIDIZED BEDS

DATE 12-90

FIGURE VS-99-05



$$Q = 200 L \times W$$

$$h_e = 1.78 \text{ slot VP}_s + 0.25 \text{ duct VP}_d$$

Duct velocity = 2500 - 3000 fpm

Slot size for 1500 - 2000 fpm

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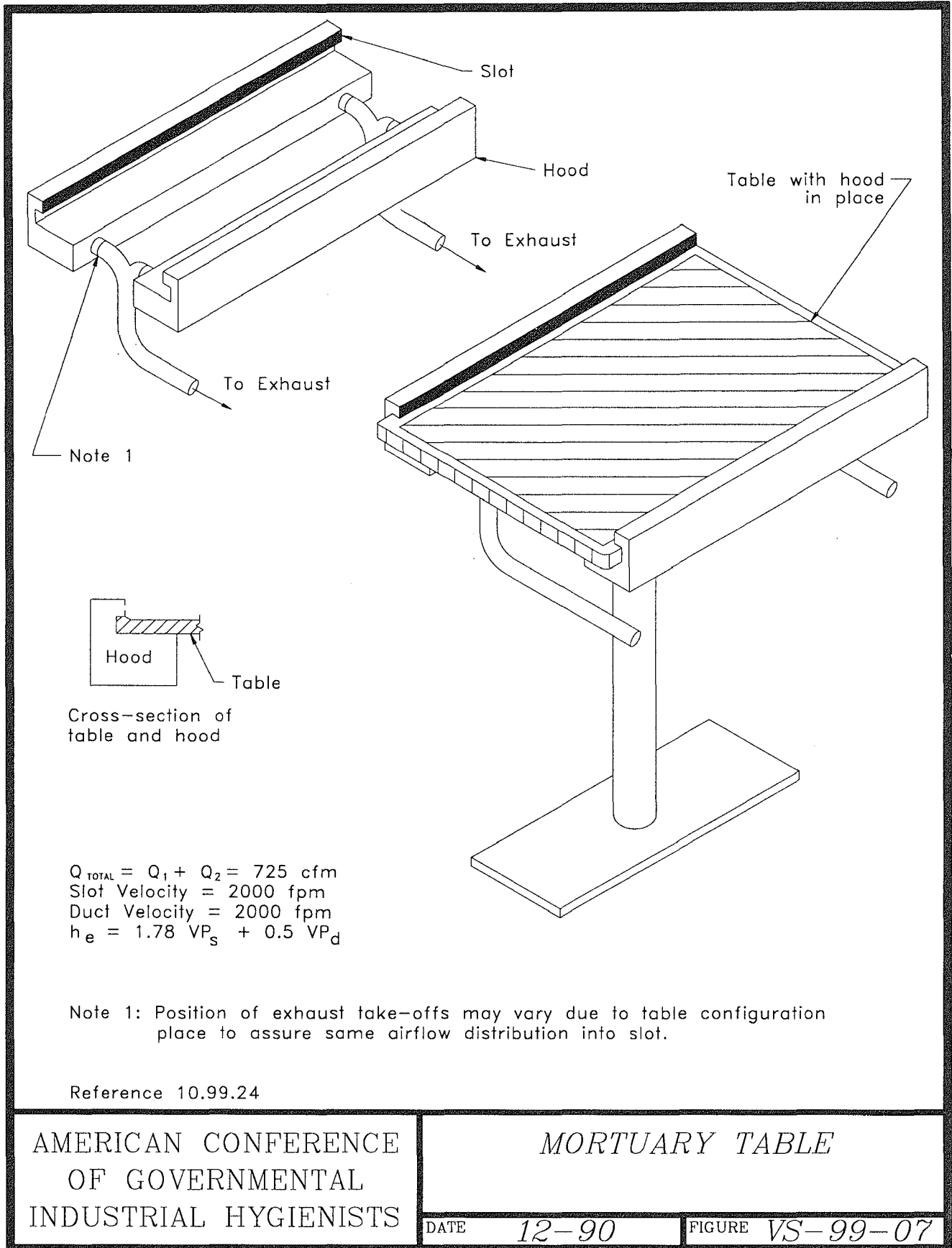
OUTBOARD MOTOR TEST

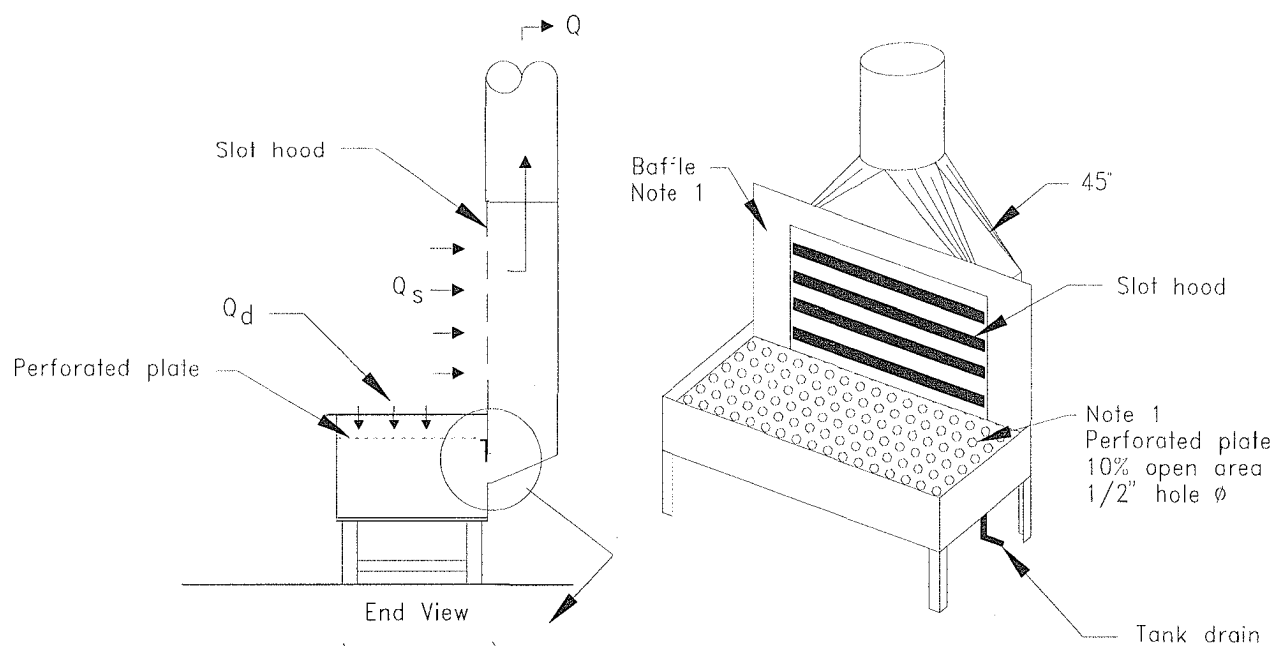
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FIGURE

VS-99-06





$$Q_s = 45 \text{ cfm/ft}^2 \text{ of tank area}$$

$$Q_d = 65 \text{ cfm/ft}^2 \text{ of tank area}$$

$$\text{Plenum velocity} < 1/2 \text{ slot velocity}$$

$$\text{Slot velocity} = 2000 \text{ fpm}$$

$$h_e = 1.78 VP_d + .25 VP_s$$

Note 1: Build perforated plate in sections for easy cleaning
Allow freeboard height at least 9"

Note 2: Baffle to extend to ends of tank
6" (minimum) above top of workpiece

Note 3: Adjust slot to achieve Q_s/Q_d flow distribution

Reference 10.99.25

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FURNITURE STRIPPING
TANK

DATE

4-94

FIGURE

VS-99-08

TABLE 10.99.1. Grain Elevators, Feed Mills, Flour Mills (10.99.23)

The following data are offered as guides. Air flow rates can vary considerably depending on degree of enclosure, flow rate of material, and dustiness of the grain. Minimum duct velocity = 3500 fpm. Ventilation control is desirable for these operations to minimize the explosive characteristics of grain dusts and to preserve plant housekeeping standards.

Operation	Hood Design	Air Volume					
Bag Loading	VS-15-02, VS-15-03 Booth VS-15-01	As shown 100 cfm/ft ² open face area					
Belt Discharge	To belt — VS-50-20 To bin — VS-50-10 To elevator — VS-50-1, VS-50-20	350 cfm/ft of belt width up to 200 fpm belt speed 500 cfm/ft of belt width over 200 fpm belt speed Increase 1/3 if material drop is over 10 ft					
Bins	Direct exhaust. Use taper.	500 cfm/bin					
Bucket Elevator	VS-50-01	100 cfm/ft ² cross-section					
Cleaning Machines	Consult manufacturer						
Distributors	Enclose discharge 200 fpm in-draft through enclosure openings.	No. of Spouts	Diameter of Spouts				
			Exhaust cfm	6"	7"	8"	9"
		0 – 6		550	675	950	1250
		6 – 12		950	1250	1500	1900
		12 – 24	1500	1900	2250	2750	
Feed Grinders	Consult manufacturer						
Floor Dump	Booth	200 cfm/ft ² open face area					
Floor Sweep		950 cfm in 4" × 8" opening					
Garner Bin	Direct exhaust. Use taper.	cfm = 1.25 × bushels/min					
Mixers	Ventilated cover	Mixer Capacity		Exhaust, cfm			
		Up to 0.5 ton		300			
		0.5 to 1.5 tons		675			
		Over 1.5 tons		950			
Percentage Feeders	Enclosed conveyor	200 cfm at each feeder					
Purifiers	Enclosure	30 – 40 cfm/ft ² screen area					
Roll Stands	Enclosure	60 cfm/lineal ft					
Scales	Enclosure	Scale Capacity, Bushels		Exhaust, cfm			
		Up to 5		250			
		6 to 10		400			
		Over 11		600			
Scale Hopper	Direct exhaust. Use taper.	cfm = 1.25 × bushels/min					
Screw Conveyor	Direct exhaust. Use taper.	200 cfm — ducts on 30 ft centers					
Sifters	Enclosure	200 cfm/compartment					
Track Sink	Direct exhaust from hopper. Use taper.	100 cfm/ft ² grate area					
Tripper Car	Belt discharge. VS-50-01, VS-50-10, VS-50-20. Spout ends — tapered connection. Spillage — exhaust under head pulley.	See "Belt Discharge" above. 200 cfm/ft ² spout cross-section 90 cfm/ft belt width					

TABLE 10.99.2. Miscellaneous Specific Operation Standards

Operation or Industry	Ventilation		Minimum Design Duct Velocity (fpm)	Reference No. and Remarks
	Type of Hood	Air Flow or Capture Velocity		
Abrasive Wheel Mfg. Grading screen Barrels	Enclosure — booth Close canopy	50 fpm at face 400 fpm at face	4000 4000	10.99.1 Bbls. receive dust from cyclone
Grinding wheel dressing	Enclosure — booth	400 fpm at face	3000	
Aluminum Furnaces	Enclosure	150-200 fpm through opening	2000	10.99.2
Asbestos				
Bagging	Enclosure — booth	250 fpm through all openings	3500	10.99.3, 10.99.4
Carding	Enclosure	1600 cfm/card	3500	
Crushing	Enclosure	150 fpm through all openings	3500	10.99.5
Drilling of panels containing asbestos	Moveable hood	400 fpm capture velocity	4500	10.99.6
Dumping	Booth	250 fpm face velocity	3500	10.99.6
Grinding of brake shoes	Enclosure	400 fpm minimum capture at the tool rest	3500	10.99.6
Hot press for brake shoes	Enclosure	250 fpm through all openings	3500	10.99.6
Mixing	Booth	250 fpm face velocity	3500	10.99.6
Preform Press	Enclosure	250 fpm through all openings	3500	10.99.6
Screening	Enclosure	200 fpm through all openings but not less than 25 cfm/ft ² screen areas	3000	10.99.3
Spool winding	Local Hoods	50 cfm/spool	3500	
Spinning and twisting	Partial	50 cfm/spool	3500	Hinged front panels and skirt, wet twisting preferred
Weaving	Canopy with baffles	50 fpm through openings	3500	Wet weaving preferred
Auto Parking Garage	2-Level	500 cfm/parking space		10.99.7
Ceramic	Enclosure	200 fpm through all openings	3500	10.99.8, 10.99.9, 10.99.10
Dry pan	Local at die	500 cfm	3500	Automatic feed
Dry Press	Local at die	500 cfm	3500	Manual feed
	At supply bin	500 cfm	3500	Manual feed
Aerographing	Booth	100 fpm (face)	—	
Spraying (lead glaze)	Booth	400 fpm (face)	2000	
Coating Pans (pharmaceutical)	Air flow into opening of pan	100–150 fpm through opening	3000	10.99.8, 10.99.11 If heated air supplied to pan, add volume of heated air to exhaust
Cooling Tunnels (foundry)	Enclosure	75-100 cfm per running foot of enclosure	—	10.99.12, 10.99.13
Core Knockout (manual)	Large side-draft or semi-booth—exhaust near floor	200–250 cfm/ft ² dust producing working area	3500	10.99.12, 10.99.13, 10.99.14
Core Sanding (on lathe)	Downdraft under work	100 fpm at source	3500	10.99.15
Crushers and Grinders	Enclosure	200 fpm through openings	3500	10.99.16
Drilling (rocks)	Special trap (see references)	60 cfm — vertical (downward) work 200 cfm—horizontal work		10.99.17, 10.99.18, 10.99.19 May vary with size and speed of drill
Forge (hand)	Booth	200 fpm at face	1500	10.99.2
Outboard Motor Test Tank	Side draft	200 cfm/ft ² of tank opening	—	
Packaging Machines	Booth Downdraft Complete enclosure	50 – 100 fpm at face 95 – 150 fpm down 100 – 400 fpm opening	3000 to 4000	
Paper Machine	Canopy	200 – 300 fpm at face	1500	10.99.20, 10.99.21

TABLE 10.99.2. Miscellaneous Specific Operation Standards (con't)

Operation or Industry	Ventilation		Minimum Design Duct Velocity (fpm)	Reference No. and Remarks
	Type of Hood	Air Flow or Capture Velocity		
Quartz Fusing	Booth on bench	150–200 fpm at face	—	10.99.20, 10.99.21
Rotary Blasting Table	Enclosure	500 fpm through all openings when in operation	3500	
Silver Soldering	Free hanging	100 fpm at source	2000	
Steam Kettles	Canopy	150 fpm at face	2000	
Varnish Kettles	Canopy	200–250 fpm at face	1500	10.99.2, 10.99.10
Wire Impregnating	Covered tanks	200 cfm/ft ² of opening	—	10.99.22 Chlorinated naphthalenes & diphenyls

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APPENDICES

Appendix A Threshold Limit Values for Chemical Substances in the Work Environment with Intended Changes for 1997	12-3
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APPENDIX A

1997 Threshold Limit Values for Chemical Substances in the Work Environment

Adopted by ACGIH
with Intended Changes

INTRODUCTION TO THE CHEMICAL SUBSTANCES

Threshold Limit Values (TLVs) refer to airborne concentrations of substances and represent conditions under which it is believed that nearly all workers may be repeatedly exposed day after day without adverse health effects. Because of wide variation in individual susceptibility, however, a small percentage of workers may experience discomfort from some substances at concentrations at or below the threshold limit; a smaller percentage may be affected more seriously by aggravation of a pre-existing condition or by development of an occupational illness. Smoking of tobacco is harmful for several reasons. Smoking may act to enhance the biological effects of chemicals encountered in the workplace and may reduce the body's defense mechanisms against toxic substances.

Individuals may also be hypersusceptible or otherwise unusually responsive to some industrial chemicals because of genetic factors, age, personal habits (e.g., smoking, alcohol, or other drugs), medication, or previous exposures. Such workers may not be adequately protected from adverse health effects from certain chemicals at concentrations at or below the threshold limits. An occupational physician should evaluate the extent to which such workers require additional protection.

TLVs are based on available information from industrial experience; from experimental human and animal studies; and, when possible, from a combination of the three. The basis on which the values are established may differ from substance to substance; protection against impairment of health may be a guiding factor for some, whereas reasonable freedom from irritation, narcosis, nuisance, or other forms of stress may form the basis for others. Health impairments considered include those that shorten life expectancy, compromise physiological function, impair the capability for resisting other toxic substances or disease processes, or adversely affect reproductive function or developmental processes.

The amount and nature of the information available for establishing a TLV varies from substance to substance; consequently, the precision of the estimated TLV is also subject to variation and the latest TLV *Documentation* should be consulted in order to assess the extent of the data available for a given substance.

These limits are intended for use in the practice of industrial hygiene as guidelines or recommendations in the control of potential workplace health hazards and for no other use, e.g., in the evaluation or control of community air pollution nuisances; in estimating the toxic potential of continuous, uninterrupted exposures or other extended work periods; as proof or disproof of an existing disease or physical condition; or adoption or use by countries whose working conditions or cultures differ from those in the United States of America and where substances and processes differ. These limits are *not* fine lines between safe and dangerous concentrations nor are they a relative index of toxicity. They *should not* be used by anyone untrained in the discipline of industrial hygiene.

The TLVs, as issued by ACGIH, are recommendations and should be used as guidelines for good practices. In spite of the fact that serious adverse health effects are not believed likely as a result of exposure to the threshold limit concentrations, the best practice is to maintain concentrations of all atmospheric contaminants as low as is practical.

ACGIH disclaims liability with respect to the use of TLVs.

Notice of Intended Changes. Each year, proposed actions of the Chemical Substances TLV Committee for the forthcoming year are issued in the form of a "Notice of Intended Changes." This Notice provides an opportunity for comment and *solicits suggestions of substances to be added to the list. The suggestions should be accompanied by substantiating evidence.* The "Notice of Intended Changes" is presented after the Adopted Values in this section. Values listed in parentheses in the "Adopted" list are to be used during the period in which a proposed change for that Value is listed in the Notice of Intended Changes.

Definitions. Three categories of Threshold Limit Values (TLVs) are specified herein, as follows:

a) Threshold Limit Value—Time-Weighted Average (TLV—TWA)—the time-weighted average concentration for a conventional 8-hour workday and a 40-hour workweek, to which it is believed that nearly all workers may be repeatedly exposed, day after day, without adverse effect.

b) Threshold Limit Value—Short-Term Exposure Limit (TLV—STEL)—the concentration to which it is believed that workers can be exposed continuously for a short period of time without suffering from 1) irritation, 2) chronic or irreversible tissue damage, or 3) narcosis of sufficient degree to increase the likelihood of accidental injury, impair self-rescue or materially reduce work efficiency, and provided that the daily TLV—TWA is not exceeded. It is not a separate independent exposure limit; rather, it supplements the time-weighted average (TWA) limit where there are recognized acute effects from a substance whose toxic effects are primarily of a chronic nature. STELs are recommended only where toxic effects have been reported from high short-term exposures in either humans or animals.

A STEL is defined as a 15-minute TWA exposure which should not be exceeded at any time during a workday even if the 8-hour TWA is within the TLV—TWA. Exposures above the TLV—TWA up to the STEL should not be longer than 15 minutes and should not occur more than four times per day. There should be at least 60 minutes between successive exposures in this range. An averaging period other than 15 minutes may be recommended when this is warranted by observed biological effects.

c) Threshold Limit Value—Ceiling (TLV—C)—the concentration that should not be exceeded during any part of the working exposure.

In conventional industrial hygiene practice if instantaneous monitoring is not feasible, then the TLV—C can be assessed by sampling over a period that should not exceed 15 minutes, except for those substances that may cause immediate irritation when exposures are short.

For some substances, e.g., irritant gases, only one category, the TLV—Ceiling, may be relevant. For other substances, one or two categories may be relevant, depending upon their physiologic action. It is important to observe that if any one of these types of TLVs is exceeded, a potential hazard from that substance is presumed to exist.

The Chemical Substances TLV Committee holds to the opinion that TLVs based on physical irritation should be considered no less binding than those based on physical impairment. There is increasing evidence that

physical irritation may initiate, promote, or accelerate physical impairment through interaction with other chemical or biologic agents.

Time-Weighted Average (TWA) vs Ceiling (C) Limits. TWAs permit excursions above the TLV provided they are compensated by equivalent excursions below the TLV-TWA during the workday. In some instances, it may be permissible to calculate the average concentration for a workweek rather than for a workday. The relationship between the TLV and permissible excursion is a rule of thumb and in certain cases may not apply. The amount by which the TLVs may be exceeded for short periods without injury to health depends upon a number of factors such as the nature of the contaminant, whether very high concentrations—even for short periods—produce acute poisoning, whether the effects are cumulative, the frequency with which high concentrations occur, and the duration of such periods. All factors must be taken into consideration in arriving at a decision as to whether a hazardous condition exists.

Although the TWA concentration provides the most satisfactory, practical way of monitoring airborne agents for compliance with the TLVs, there are certain substances for which it is inappropriate. In the latter group are substances which are predominantly fast acting and whose TLV is more appropriately based on this particular response. Substances with this type of response are best controlled by a ceiling limit that should not be exceeded. It is implicit in these definitions that the manner of sampling to determine noncompliance with the limits for each group must differ; a single, brief sample, that is applicable to a ceiling limit, is not appropriate to the TWA; here, a sufficient number of samples are needed to permit determination of a TWA concentration throughout a complete cycle of operations or throughout the workshift.

Whereas the ceiling limit places a definite boundary that concentrations should not be permitted to exceed, the TWA requires an explicit limit to the excursions that are permissible above the listed TLVs. It should be noted that the same factors are used by the Chemical Substances TLV Committee in determining the magnitude of the value of the STEL or whether to include or exclude a substance for a ceiling listing.

Excursion Limits. For the vast majority of substances with a TLV-TWA, there is not enough toxicological data available to warrant a STEL. Nevertheless, excursions above the TLV-TWA should be controlled even where the 8-hour TLV-TWA is within recommended limits. Earlier editions of the TLV list included such limits whose values depended on the TLV-TWAs of the substance in question.

While no rigorous rationale was provided for these particular values, the basic concept was intuitive: in a well-controlled process exposure, excursions should be held within some reasonable limits. Unfortunately, neither toxicology nor collective industrial hygiene experience provide a solid basis for quantifying what those limits should be. The approach here is that the maximum recommended excursion should be related to variability generally observed in actual industrial processes. In reviewing large numbers of industrial hygiene surveys conducted by the National Institute for Occupational Safety and Health, Leidel, Busch, and Crouse⁽¹⁾ found that short-term exposure measurements were generally lognormally distributed with geometric standard deviations mostly in the range of 1.5 to 2.0.

While a complete discussion of the theory and properties of the lognormal distribution is beyond the scope of this section, a brief description of some important terms is presented. The measure of central tendency in a lognormal distribution is the antilog of the mean logarithm of the sample values. The distribution is skewed, and the geometric mean is always smaller than the arithmetic mean by an amount which depends on the geometric standard deviation. In the lognormal distribution, the geometric standard deviation (sd_g) is the antilog of the standard deviation of the sample value logarithms and 68.26% of all values lie between m_g/sd_g and $m_g \times sd_g$.

If the short-term exposure values in a given situation have a geometric standard deviation of 2.0, 5% of all values will exceed 3.13 times the geometric mean. If a process displays a variability greater than this, it is not under good control and efforts should be made to restore control. This concept is the basis for the following excursion limit recommendations which apply to those TLV-TWAs that do not have STELs:

Excursions in worker exposure levels may exceed 3 times the TLV-TWA for no more than a total of 30 minutes during a workday, and under no circumstances should they exceed 5 times the TLV-TWA, provided that the TLV-TWA is not exceeded.

The approach is a considerable simplification of the idea of the lognormal concentration distribution but is considered more convenient to use by the practicing industrial hygienist. If exposure excursions are maintained within the recommended limits, the geometric standard deviation of the concentration measurements will be near 2.0 and the goal of the recommendations will be accomplished.

When the toxicological data for a specific substance are available to establish a STEL, this value takes precedence over the excursion limit regardless of whether it is more or less stringent.

"Skin" Notation. Listed substances followed by the designation "Skin" refer to the potential significant contribution to the overall exposure by the cutaneous route, including mucous membranes and the eyes, either by contact with vapors or, of probable greater significance, by direct skin contact with the substance. Vehicles present in solutions or mixtures can also significantly enhance potential skin absorption. It should be noted that while some materials are capable of causing irritation, dermatitis, and sensitization in workers, these properties are *not considered relevant* when assigning a skin notation. It should be noted, however, that the development of a dermatological condition can significantly affect the potential for dermal absorption.

While relatively limited quantitative data currently exist with regard to skin absorption of gases, vapors, and liquids by workers, the Chemical Substances TLV Committee recommends that the integration of data from acute dermal studies and repeated dose dermal studies in animals and/or humans, along with the ability of the chemical to be absorbed, be used in deciding on the appropriateness of the skin notation. In general, available data which suggest that the potential for absorption via the hands/forearms during the workday could be significant, especially for chemicals with lower TLVs, could justify a skin notation. From acute animal toxicity data, materials having a relatively low dermal LD_{50} (1000 mg/kg of body weight or less) would be given a skin notation. Where repeated dermal application studies have shown significant systemic effects following treatment, a skin notation would be considered. When chemicals penetrate the skin easily (higher octanol-water partition coefficients) and where extrapolations of systemic effects from other routes of exposure suggest dermal absorption may be important in the expressed toxicity, a skin notation would be considered.

Substances having a skin notation and a low TLV may present special problems for operations involving high airborne concentrations of the material, particularly under conditions where significant areas of the skin are exposed for a long period of time. Under these conditions, special precautions to significantly reduce or preclude skin contact may be required.

Biological monitoring should be considered to determine the relative contribution of exposure via the dermal route to the total dose. The TLV/BEI Booklet contains a number of adopted Biological Exposure Indices, which provide an additional tool when assessing the worker's total exposure to selected materials. For additional information, refer to "Dermal Absorption" in the "Introduction to the Biological Exposure Indices," 6th edition of the *Documentation of Threshold Limit Values and Biological Exposure Indices*, and to Leung and Paustenbach.⁽²⁾

Use of the skin designation is intended to alert the reader that air sampling alone is insufficient to accurately quantitate exposure and that measures to prevent significant cutaneous absorption may be required.

"Sensitizer" Notation. Listed substances followed by the designation "SEN" refer to the confirmed potential for worker sensitization as a result of dermal contact and/or inhalation exposure, based on the weight of scientific evidence. Lack of the sensitizer notation does not necessarily mean that the substance is not a sensitizer. The *Documentation of the Threshold Limit Values and Biological Exposure Indices* should be consulted for detailed information on the specific substance, the relative sensitizing potency, and whether its sensitization potential is related to dermal contact, inhalation exposure, or both.

Mixtures. Special consideration should be given also to the application of the TLVs in assessing the health hazards that may be associated with

(1) Leidel, N.A.; Busch, K.A.; Crouse, W.E.: Exposure Measurement Action Level and Occupational Environmental Variability. DHEW (NIOSH) Pub. No. 76-131; NTIS Pub. No. PB-267-509. National Technical Information Service, Springfield, VA (December 1975).

(2) Leung, H.; Paustenbach, D.J.: Techniques for Estimating the Percutaneous Absorption of Chemicals Due to Occupational and Environmental Exposure. *Appl. Occup. Environ. Hyg.* 9(3):187-197 (March 1994).

exposure to mixtures of two or more substances. A brief discussion of basic considerations involved in developing TLVs for mixtures and methods for their development, amplified by specific examples, are given in Appendix C.

Particulate Matter. For solid and liquid particulate matter, TLVs are expressed in terms of total particulate, except where the terms inhalable, thoracic, or respirable particulate are used. Refer to Endnotes. See Appendix D, Particle Size-Selective Sampling Criteria for Airborne Particulate Matter, for the definitions of inhalable, thoracic, and respirable particulate matter. The term total particulate refers to airborne material sampled with the 37mm closed face cassette traditionally used in the United States for aerosol sampling.

The intent of the Chemical Substances TLV Committee is to replace all total particulate TLVs with inhalable, thoracic, and/or respirable particulate matter TLVs. All proposed changes will be included on the Notice of Intended Changes and comments invited. Publication of the results of side-by-side sampling studies using older total and newer inhalable, thoracic, or respirable sampling techniques is encouraged to aid in the appropriate replacement of current total particulate TLVs.

Particulates Not Otherwise Classified (PNOC). There are many substances on the TLV list, and many more that are not on the list, for which there is no evidence of specific toxic effects. Those that are particulates have frequently been called "nuisance dusts." Although these materials may not cause fibrosis or systemic effects, they are not biologically inert. At high concentrations, otherwise nontoxic particulates have been associated with the occasionally fatal condition known as alveolar proteinosis. At lower concentrations, they can inhibit the clearance of toxic particulates from the lung by decreasing the mobility of the alveolar macrophages. Accordingly, the Chemical Substances TLV Committee recommends the use of the term "Particulates Not Otherwise Classified (PNOC)" to emphasize that all materials are potentially toxic and to avoid the implication that these materials are harmless at all exposure concentrations. Particulates identified under the PNOC heading are those containing no asbestos and <1% crystalline silica. To recognize the adverse effects of exposure to otherwise nontoxic particulate matter, a TLV-TWA of 10 mg/m³ for inhalable particulate and a TLV-TWA of 3 mg/m³ for respirable particulate have been established and are included in the adopted TLV list. Refer to the Documentation for Particulates Not Otherwise Classified (PNOC) for a complete discussion of this subject.

Simple Asphyxiants—"Inert" Gases or Vapors. A number of gases and vapors, when present in high concentrations in air, act primarily as simple asphyxiants without other significant physiologic effects. A TLV may not be recommended for each simple asphyxiant because the limiting factor is the available oxygen. The minimal oxygen content should be 18% by volume under normal atmospheric pressure (equivalent to a partial pressure, pO₂ of 135 torr). Atmospheres deficient in O₂ do not provide adequate warning and most simple asphyxiants are odorless. Several simple asphyxiants present an explosion hazard. Account should be taken of this factor in limiting the concentration of the asphyxiant.

Biological Exposure Indices (BEI). A cross reference (↔) is indicated for those substances for which there are also Biological Exposure Indices. For such substances, biological monitoring should be instituted to evaluate the total exposure, e.g., dermal, ingestion, or nonoccupational. See the BEI section in this Booklet.

Physical Factors. It is recognized that such physical factors as heat, ultraviolet and ionizing radiation, humidity, abnormal pressure (altitude), and the like may place added stress on the body so that the effects from exposure at a TLV may be altered. Most of these stresses act adversely to increase the toxic response of a substance. *Although most TLVs have built-in safety factors to guard against adverse effects to moderate deviations from normal environments, the safety factors of most substances are not of such a magnitude as to take care of gross deviations.* For example, continuous, heavy work at temperatures above 25°C WBGT, or overtime extending the workweek more than 25%, might be considered gross deviations. In such instances, judgment must be exercised in the proper adjustments of the TLVs.

Unlisted Substances. The list of TLVs is by no means a complete list of all hazardous substances or of all hazardous substances used in industry. For a large number of materials of recognized toxicity, little or no data

are available that could be used to establish a TLV. Substances that do not appear on the TLV list should not be considered to be harmless or nontoxic. When unlisted substances are introduced into a workplace, the medical and scientific literature should be reviewed to identify potentially dangerous toxic effects. It may also be advisable to conduct preliminary toxicity studies. In any case, it is necessary to remain alert to adverse health effects in workers which may be associated with the use of new materials. The TLV Committee strongly encourages industrial hygienists and other occupational health professionals to bring to the Committee's attention any information which would suggest that a TLV should be established. Such information should include exposure concentrations and correlated health effects data (dose-response) that would support a recommended TLV.

Unusual Work Schedules. Application of TLVs to workers on work schedules markedly different from the conventional 8-hour day, 40-hour week requires particular judgement in order to provide, for such workers, protection equal to that provided to workers on conventional workshifts.

As tentative guidance, field hygienists are referred to the "Brief and Scala model" which is described and explained at length in Patty.⁽³⁾

The Brief and Scala model reduces the TLV proportionately for both increased exposure time and reduced recovery (nonexposure) time. The model is generally intended to apply to work schedules longer than 8 hours/day or 40 hours/week. The model should not be used to justify very high exposures as "allowable" where the exposure periods are short (e.g., exposure to 8 times the TLV-TWA for one hour and zero exposure during the remainder of the shift). In this respect, the general limitations on TLV excursions and STELs should be applied to avoid inappropriate use of the model with very short exposure periods or shifts.

Since adjusted TLVs do not have the benefit of historical use and long-time observation, medical supervision during initial use of adjusted TLVs is advised. In addition, the hygienist should avoid unnecessary exposure of workers even if a model shows such exposures to be "allowable" and should not use models to justify higher-than-necessary exposures.

The Brief and Scala model is easier to use than some of the more complex models based on pharmacokinetic actions. However, hygienists thoroughly familiar with such models may find them more appropriate in specific instances. Use of such models usually requires knowledge of the biological half-life of each substance, and some models require additional data.

Short workweeks can allow workers to have two full-time jobs, perhaps with similar exposures, and may result in overexposure even if neither job by itself entails overexposure. Hygienists should be alert to such situations.

Conversion of TLVs in ppm to mg/m³. TLVs for gases and vapors are usually established in terms of parts per million of substances in air by volume (ppm). For convenience to the user, these TLVs are also listed in terms of milligrams of substance per cubic meter of air (mg/m³), where 24.45 = molar volume of air in liters at normal temperature and pressure (NTP) conditions (25°C and 760 torr), giving a conversion equation of:

$$\text{TLV in mg/m}^3 = \frac{(\text{TLV in ppm}) (\text{gram molecular weight of substance})}{24.45}$$

Conversely, the equation for converting TLVs in mg/m³ to ppm is:

$$\text{TLV in ppm} = \frac{(\text{TLV in mg/m}^3) (24.45)}{\text{gram molecular weight of substance}}$$

Resulting values are rounded to two significant figures below 100 and to three significant figures above 100. This is not done to give any converted value a greater precision than that of the original TLV but to avoid increasing or decreasing the TLV significantly merely by the conversion of units.

The above equation may be used to convert TLVs to any degree of precision desired. When converting TLVs to mg/m³ for other temperatures and pressures, the reference TLVs should be used as a starting point. When converting values expressed as an element (e.g., as Fe, as Ni), the molecular value of the element should be used, not that of the entire compound.

In making conversions for substances with variable molecular weights, appropriate molecular weights have been estimated or assumed (see the TLV Documentation).

(3) Paustenbach, D.J.: Occupational Exposure Limits, Pharmacokinetics, and Unusual Work Schedules. In: Patty's Industrial Hygiene and Toxicology, 3rd ed., Vol. 3A, The Work Environment, Chap. 7, pp. 222-348. R.L. Harris, L.J. Cralley and L.V. Cralley, Eds. John Wiley and Sons, Inc., New York (1994).

Biologically Derived Airborne Contaminants. TLVs exist for certain substances of biological origin, including cellulose; some wood, cotton, and grain dusts; nicotine; pyrethrum; starch; subtilisins (proteolytic enzymes); sucrose; and vegetable oil mist. However, for the reasons identified below, there are no TLVs against which to compare environmental air concentrations of most materials of biological origin. The ACGIH Bioaerosols Committee has developed and separately published guidelines to assess, control, remediate, and prevent biologically derived contamination in indoor environments.⁽⁴⁾ For the purposes of the guidelines, indoor biological contamination is defined as the presence of a) biologically derived aerosols of a kind and concentration likely to cause disease or predispose people to disease; b) inappropriate concentrations of outdoor bioaerosols, especially in buildings designed to prevent their entry; or c) indoor microbial growth and remnants of biological growth that may become aerosolized and to which people may be exposed. The guidelines define an approach to assessing and controlling bioaerosol exposures. This approach relies on visually inspecting buildings, assessing occupant symptoms, evaluating building performance, monitoring potential environmental sources, and applying professional judgement.

Biologically derived airborne contaminants include bioaerosols (airborne particles composed of or derived from living organisms) and volatile organic compounds that organisms release. Bioaerosols include microorganisms (i.e., culturable, nonculturable, and dead microorganisms) and fragments, toxins, and particulate waste products from all varieties of living things. Biologically derived contaminants are ubiquitous in nature and may be modified by human activity. All persons are repeatedly exposed, day after day, to a wide variety of such materials.

The guidelines provide background information on the major groups of bioaerosols including their sources and health effects. The guidelines also describe methods to collect, analyze, and interpret bioaerosol samples from potential environmental sources. Occasionally, environmental monitoring detects a single or predominating biological contaminant. More commonly, monitoring reveals a mixture of many biologically derived materials, reflecting the diverse and interactive nature of indoor microenvironments. Environmental sampling for bioaerosols should be conducted only following careful formulation of testable hypotheses about potential bioaerosol sources and mechanisms by which workers may be exposed to bioaerosols from these sources. Even when investigators work from testable hypotheses and well-formulated sampling plans, results from environmental bioaerosol monitoring may be inconclusive and possibly misleading.

For the reasons identified below, there are no TLVs for interpreting environmental measurements of a) total culturable or countable bioaerosols (e.g., total bacteria or fungi); b) specific culturable or countable bioaerosols (e.g., *Aspergillus fumigatus*); c) infectious agents (e.g., *Legionella pneumophila*, *Mycobacterium tuberculosis*); or d) assayable biological contaminants (e.g., endotoxin, mycotoxin, antigens, or microbial volatile organic compounds).

A. Total culturable or countable bioaerosols. Culturable bioaerosols are those bacteria and fungi that can be grown in laboratory culture. Such results are reported as the number of colony-forming units. Countable bioaerosols are those pollen grains, fungal spores, bacterial cells, and other material that can be identified and counted by microscope. A general TLV for culturable or countable bioaerosol concentrations is not scientifically supportable because of the following.

1. Culturable microorganisms and countable biological particles do not comprise a single entity, i.e., bioaerosols in occupational settings are generally complex mixtures of many different microbial, animal, and plant particles.
2. Human responses to bioaerosols range from innocuous effects to serious, even fatal, diseases, depending on the specific material involved and workers' susceptibility to it. Therefore, an appropriate exposure limit for one bioaerosol may be entirely inappropriate for another.
3. It is not possible to collect and evaluate all bioaerosol components using a single sampling method. Many reliable methods are available

to collect and analyze bioaerosol materials. However, different methods of sample collection and analysis may result in different estimates of the concentrations of culturable and countable bioaerosols.

4. At present, information relating culturable or countable bioaerosol concentrations to health effects is generally insufficient to describe exposure-response relationships.

B. Specific culturable or countable bioaerosols other than infectious agents. Specific TLVs for individual culturable or countable bioaerosols have not been established to prevent hypersensitivity, irritant, or toxic responses. At present, information relating culturable or countable bioaerosol concentrations to health effects consists largely of case reports and qualitative exposure assessments. The data available are generally insufficient to describe exposure-response relationships. Reasons for the absence of good epidemiologic data on such relationships include the following.

1. Most data on concentrations of specific bioaerosols are derived from indicator measurements rather than from measurements of actual effector agents. For example, investigators use the air concentration of culturable fungi to represent exposure to airborne fungal antigens. In addition, most measurements are from either area or source samples. These monitoring approaches are less likely to reflect human exposure accurately than would personal sampling for actual effector agents.
2. Bioaerosol components and concentrations vary widely within and among different occupational and environmental settings. Unfortunately, replicate sampling is uncommon in bioaerosol assessments. Further, the most commonly used air sampling devices for indoor monitoring are designed to collect "grab" samples over relatively short time intervals. Measurements from single, short-term grab samples may be orders of magnitude higher or lower than long-term average concentrations and are unlikely to represent workplace exposures accurately. Some organisms and sources release aerosols as "concentration bursts," which may only rarely be detected by limited grab sampling. Nevertheless, such episodic bioaerosol releases may produce significant health effects.

C. Infectious agents. Human dose-response data are available for only a few infectious bioaerosols. At present, air sampling protocols for infectious agents are limited and suitable primarily for research endeavors. In most routine exposure settings, public health measures, such as immunization, active case finding, and medical treatment, remain the primary defenses against infectious bioaerosols. Facilities associated with increased risks for transmission of airborne infectious diseases (e.g., microbiology laboratories, animal handling facilities, and health care settings) should employ engineering controls to minimize air concentrations of infectious agents. Further, such facilities should consider the need for administrative controls and personal protective equipment to prevent the exposure of workers to these bioaerosols.

D. Assayable biological contaminants. Assayable, biologically derived contaminants (e.g., endotoxin, mycotoxins, antigens, and volatile organic compounds) are microbial, animal, or plant substances that can be detected using chemical, immunological, or biological assays. Evidence does not yet support TLVs for any of these substances. However, assay methods for certain common airborne antigens and endotoxin are steadily improving, and field validation of these assays is also progressing. Dose-response relationships for some assayable bioaerosols have been observed in experimental studies and occasionally in epidemiologic surveys. Therefore, TLVs for some of these substances may be appropriate in the future. Also, innovative molecular techniques are becoming available for specific bioaerosols currently detectable only by culture or counting.

ACGIH actively solicits information, comments, and data that will help the Bioaerosols Committee evaluate the potential for health effects associated with bioaerosol exposures in occupational and related environments. Such information should be sent to ACGIH Technical Affairs office.

(4) American Conference of Governmental Industrial Hygienists: Guidelines for the Assessment of Bioaerosols in the Indoor Environment. ACGIH, Cincinnati, OH (1989).

Note: In the interest of keeping this section as compact as possible, the footnotes for the adopted values are as follows:

FOOTNOTES

- * 1997 Adoption.
- ‡ See Notice of Intended Changes.
- () Adopted values enclosed are those for which changes are proposed. Consult the Notice of Intended Changes for current proposal.
- † 1997 Revision or Addition to the Notice of Intended Changes.
- ◀ Identifies substances for which there are also BEIs (see BEI section). Substances identified in the BEI documentations for methemoglobin inducers (for which methemoglobin is the principal cause of toxicity) and organophosphorus cholinesterase inhibitors are part of this notation.
- Substance for which the TLV is higher than the OSHA Permissible Exposure Limit (PEL) and/or the NIOSH Recommended Exposure Limit (REL). See Fed. Reg. 58(124):35338–35351, June 30, 1993, and subsequent amendments for revised OSHA PELs.
- Substance identified by other sources as a suspected or confirmed human carcinogen.
- A Refers to Appendix A — Carcinogens.
- B Refers to Appendix B — Substances of Variable Composition.
- C Denotes Ceiling limit.
- (a) Parts of vapor or gas per million parts of contaminated air by volume at NTP conditions (25°C; 760 torr).
- (b) Milligrams of substance per cubic meter of air.
- (c) Simple asphyxiant; see definition in the "Introduction to the Chemical Substances."
- (d) NOC = not otherwise classified.
- (e) The value is for particulate matter containing no asbestos and < 1% crystalline silica.
- (f) Fibers longer than 5 μm and with an aspect ratio equal to or greater than 3:1 as determined by the membrane filter method at 400–450X magnification (4-mm objective) phase contrast illumination.
- (g) The value is for particulate matter containing < 5% crystalline silica. For particulates containing more than this percentage of crystalline silica, the environment should be evaluated against the TLV–TWA of 0.1 mg/m^3 for respirable quartz. The concentration of respirable particulates for the application of this limit is to be determined from the fraction passing a size-selector with the characteristics defined in the "C" paragraph of Appendix D.
- (h) Fibers longer than 5 μm ; diameter less than 3 μm ; aspect ratio greater than 5:1 as determined by the membrane filter method at 400–450X magnification (4-mm objective) phase contrast illumination.
- (i) Inhalable fraction. The concentration of inhalable particulate for the application of this TLV is to be determined from the fraction passing a size-selector with the characteristics defined in the "A" paragraph of Appendix D.
- (j) Respirable fraction. The concentration of respirable dust for the application of this limit is to be determined from the fraction passing a size-selector with the characteristics defined in the "C" paragraph of Appendix D.
- (k) As sampled by method that does not collect vapor.
- (l) Does not include stearates of toxic metals.
- (m) Based on "high-volume" sampling.
- (n) However, should not exceed 2 mg/m^3 respirable particulates.
- (o) For greater assurance of worker protection, biological monitoring is recommended.
- (p) Except castor, cashew nut, or similar irritant oils.
- (q) Lint-free particulate matter as measured by the vertical elutriator cotton-dust sampler described in the Transactions of the National Conference on Cotton Dust, p. 33, by J.R. Lynch (May 2, 1970).

ADOPTED VALUES				
Substance	[CAS #]	TWA ppm ^a	mg/m ^{3b}	STEL/CEILING (C) ppm ^a mg/m ^{3b}
• Acetaldehyde [75-07-0] (1993) ..	—	—	—	C 25,A3 C 45,A3
Acetic acid [64-19-7] (1976)	10	25	15	37
• Acetic anhydride [108-24-7] (1993)	5	21	—	—
• Acetone [67-64-1] (1997)	500,A4	1188,A4	750,A4	1782,A4
• Acetone cyanohydrin [75-86-5], as CN—Skin (1994)	—	—	C 4.7	C 5
• Acetonitrile [75-05-8] (1996)	40,A4	67,A4	60,A4	101,A4
Acetophenone [98-86-2] (1993)	10	49	—	—
Acetylene [74-86-2] (1981)	—(c)	—	—	—
Acetylene dichloride, <i>see</i> 1,2-Dichloroethylene				
Acetylene tetrabromide [79-27-6] (1986)	1	14	—	—
Acetylsalicylic acid (Aspirin) [50-78-2] (1980)	—	5	—	—
‡ Acrolein [107-02-8] (1976)	(0.1)	(0.23)	(0.3)	(0.69)
■ Acrylamide [79-06-1]—Skin (1996)	—	0.03,A3	—	—
Acrylic acid [79-10-7]—Skin (1996)	2,A4	5.9,A4	—	—
• Acrylonitrile [107-13-1]—				
Skin (1984)	2,A2	4.3,A2	—	—
Adipic acid [124-04-9] (1993)	—	5	—	—
Adiponitrile [111-69-3]—				
Skin (1994)	2	8.8	—	—
■ Aldrin [309-00-2]—Skin (1996) ..	—	0.25,A3	—	—
Allyl alcohol [107-18-6]—				
Skin (1976)	2	4.8	4	9.5
■ Allyl chloride [107-05-1] (1996) ..	1,A3	3,A3	2,A3	6,A3
‡ Allyl glycidyl ether (AGE) [106-92-3] (1976)	(5)	(23)	(10)	(47)
Allyl propyl disulfide [2179-59-1] (1976)	2	12	3	18
α-Alumina, <i>see</i> Aluminum oxide				
Aluminum [7429-90-5]				
Metal dust (1986)	—	10	—	—
Pyro powders, as Al (1979) ..	—	5	—	—
Welding fumes, as Al (1979) ..	—	5	—	—
Soluble salts, as Al (1979) ..	—	2	—	—
Alkyls (NOC ^(d)), as Al (1979)	—	—	2	—
Aluminum oxide [1344-28-1] (1996)	—	10,(e)A4	—	—
■ 4-Aminodiphenyl [92-67-1]—				
Skin (1972)	—	A1	—	—
2-Aminoethanol, <i>see</i> Ethanolamine				
2-Aminopyridine [504-29-0] (1986)	0.5	1.9	—	—
3-Amino-1,2,4-triazole, <i>see</i> Amitrole				
■ Amitrole [61-82-5] (1996)	—	0.2,A3	—	—
Ammonia [7664-41-7] (1976)	25	17	35	24
Ammonium chloride fume [12125-02-9] (1976)	—	10	—	20
Ammonium perfluorooctanoate [3825-26-1]—Skin (1994)	—	0.01,A3	—	—
Ammonium sulfamate [7773-06-0] (1986)	—	10	—	—
Amosite, <i>see</i> Asbestos				
n-Amyl acetate [628-63-7] (1987)	100	532	—	—
sec-Amyl acetate [626-38-0] (1987)	125	665	—	—
• Aniline [62-53-3] and homologues—Skin (1996) ..	2,A3	7.6,A3	—	—
■ o-Anisidine [90-04-0]—				
Skin (1996)	0.1,A3	0.5,A3	—	—
■ p-Anisidine [104-94-9]—				
Skin (1996)	0.1,A4	0.5,A4	—	—
Antimony [7440-36-0] and compounds, as Sb (1980) ..	—	0.5	—	—
■ Antimony trioxide [1309-64-4] production (1980)	—	A2	—	—
ANTU [86-88-4] (1996)	—	0.3,A4	—	—
Argon [7440-37-1] (1981)	—(c)	—	—	—
• Arsenic, elemental [7440-38-2] and inorganic compounds (except Arsine), as As (1993) ..	—	0.01,A1	—	—

ADOPTED VALUES				
Substance	[CAS #]	TWA ppm ^a	mg/m ^{3b}	STEL/CEILING (C) ppm ^a mg/m ^{3b}
• Arsine [7784-42-1] (1977)	0.05	0.16	—	—
‡ Asbestos ^(f)				
‡• Amosite [12172-73-5] (1980) ..	—	(0.5 fiber/cc, A1)	—	—
‡• Chrysotile [12001-29-5] (1980) ..	—	(2 fibers/cc, A1)	—	—
‡• Crocidolite [12001-28-4] (1980) ..	—	(0.2 fiber/cc, A1)	—	—
‡• Other forms (1980)	—	(2 fibers/cc, A1)	—	—
‡ Asphalt (petroleum) fumes [8052-42-4] (1996)	—	(5,A4)	—	—
Atrazine [1912-24-9] (1983)	—	5,A4	—	—
◄ Azinphos-methyl [86-50-0]—				
Skin (1996)	—	0.2,A4	—	—
Barium [7440-39-3] and soluble compounds, as Ba (1996)	—	0.5,A4	—	—
Barium sulfate [7727-43-7] (1986) ..	—	10 ^(e)	—	—
Benomyl [17804-35-2] (1996)	0.84,A4	10,A4	—	—
■ Benz[a]anthracene [56-55-3] (1993)	A2	A2	—	—
• Benzene [71-43-2]—Skin (1997)	0.5,A1	1.6,A1	2.5,A1	8,A1
■ Benzidine [92-87-5]—Skin (1982)	—	A1	—	—
■ Benzo[b]fluoranthene [205-99-2] (1992)	A2	A2	—	—
p-Benzoquinone, <i>see</i> Quinone				
Benzotrifluoride [98-07-7]—Skin (1997)	—	—	C 0.1,A2	C 0.8,A2
Benzoyl chloride [98-88-4] (1995) ..	—	—	C 0.5,A4	C 2.8,A4
Benzoyl peroxide [94-36-0] (1996) ..	—	5,A4	—	—
■ Benzo[a]pyrene [50-32-8] (1976)	A2	A2	—	—
Benzyl acetate [140-11-4] (1995)	10,A4	61,A4	—	—
■ Benzyl chloride [100-44-7] (1996)	1,A3	5.2,A3	—	—
• Beryllium [7440-41-7] and compounds, as Be (1997)	—	0.002,A1	—	0.01, A1
Biphenyl [92-52-4] (1987)	0.2	1.3	—	—
Bismuth telluride, as Bi ₂ Te ₃ Undoped [1304-82-1] (1996) ..	—	10,A4	—	—
Se-doped (1996)	—	5,A4	—	—
Borates, tetra, sodium salts [1303-96-4]				
Anhydrous (1977)	—	1	—	—
Decahydrate (1977)	—	5	—	—
Pentahydrate (1977)	—	1	—	—
Boron oxide [1303-86-2] (1986) ..	—	10	—	—
Boron tribromide [10294-33-4] (1986)	—	—	C 1	C 10
Boron trifluoride [7637-07-2] (1977)	—	—	C 1	C 2.8
Bromacil [314-40-9] (1996)	—	10,A3	—	—
Bromine [7726-95-6] (1994)	0.1	0.66	0.2	1.3
Bromine pentafluoride [7789-30-2] (1986)	0.1	0.72	—	—
Bromochloromethane, <i>see</i> Chlorobromomethane				
Bromoform [75-25-2]—				
Skin (1996)	0.5,A3	5.2,A3	—	—
• 1,3-Butadiene [106-99-0] (1994) ..	2,A2	4.4,A2	—	—
Butane [106-97-8] (1981)	800	1900	—	—
Butanethiol, <i>see</i> Butyl mercaptan				
‡ n-Butanol [71-36-3]—				
(Skin) (1977)	—	—	(C 50)	(C 152)
sec-Butanol [78-92-2] (1990)	100	303	—	—
tert-Butanol [75-65-0] (1995)	100,A4	303,A4	—	—
2-Butanone, <i>see</i> Methyl ethyl ketone (MEK)				
• 2-Butoxyethanol (EBGE) [111-76-2]—Skin (1987)	25	121	—	—
‡ n-Butyl acetate [123-86-4] (1996)	150,(A4)	713,(A4)	200,(A4)	950,(A4)
sec-Butyl acetate [105-46-4] (1987)	200	950	—	—
tert-Butyl acetate [540-88-5] (1987)	200	950	—	—
n-Butyl acrylate [141-32-2] (1996)	10,A4	52,A4	—	—
n-Butylamine [109-73-9]—				
Skin (1976)	—	—	C 5	C 15
• tert-Butyl chromate, as CrO ₃ [1189-85-1]—Skin (1977)	—	—	—	C 0.1
• n-Butyl glycidyl ether (BGE) [2426-08-6] (1981)	25	133	—	—
n-Butyl lactate [138-22-7] (1977)	5	30	—	—

ADOPTED VALUES					
Substance	[CAS #]	TWA ppm ^{a)}	TWA mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)}	STEL/CEILING (C) mg/m ^{3b)}
■ Butyl mercaptan [109-79-5] (1977)		0.5	1.8	—	—
o-sec-Butylphenol					
[89-72-5]—Skin (1980)		5	31	—	—
p-tert-Butyl toluene [98-51-1] (1993)		1	6.1	—	—
◄■ Cadmium, elemental [7440-43-9] and compounds, as Cd (1993)		—	0.01, ⁽ⁱ⁾ A2	—	—
		—	0.002, ⁽ⁱ⁾ A2	—	—
Calcium carbonate					
[1317-65-3] (1986)		—	10 ^(e)	—	—
■ Calcium chromate					
[13765-19-0], as Cr (1991)		—	0.001,A2	—	—
Calcium cyanamide					
[156-62-7] (1996)		—	0.5,A4	—	—
Calcium hydroxide					
[1305-62-0] (1978)		—	5	—	—
Calcium oxide [1305-78-8] (1978)		—	2	—	—
Calcium silicate (synthetic)					
[1344-95-2] (1996)		—	10, ^(e) A4	—	—
Calcium sulfate					
[7778-18-9] (1986)		—	10 ^(e)	—	—
Camphor, synthetic					
[76-22-2] (1996)		2,A4	12,A4	3,A4	19,A4
Caprolactam [105-60-2]					
Dust (1996)		—	1,A4	—	3,A4
• Vapor (1996)		5,A4	23,A4	10,A4	46,A4
■ Captafol [2425-06-1]—					
Skin (1996)		—	0.1,A4	—	—
■ Captan [133-06-2] (1996)		—	5,A3	—	—
Carbaryl [63-25-2] (1996)		—	5,A4	—	—
Carbofuran [1563-66-2] (1996)		—	0.1,A4	—	—
■ Carbon black [1333-86-4] (1996)		—	3.5,A4	—	—
Carbon dioxide [124-38-9] (1986)		5000	9000	30,000	54,000
◄■ Carbon disulfide [75-15-0]—					
Skin (1980)		10	31	—	—
◄ Carbon monoxide					
[630-08-0] (1992)		25	29	—	—
Carbon tetrabromide					
[558-13-4] (1976)		0.1	1.4	0.3	4.1
■ Carbon tetrachloride (Tetrachloromethane)					
[56-23-5]—Skin (1996)		5,A2	31,A2	10,A2	63,A2
Carbonyl chloride, <i>see</i> Phosgene					
Carbonyl fluoride					
[353-50-4] (1986)		2	5.4	5	13
Catechol [120-80-9]—					
Skin (1996)		5,A3	23,A3	—	—
Cellulose [9004-34-6] (1986)		—	10	—	—
Cesium hydroxide					
[21351-79-1] (1977)		—	2	—	—
■ Chlordane [57-74-9]—					
Skin (1996)		—	0.5,A3	—	—
■ Chlorinated camphene (Toxaphene)					
[8001-35-2]—Skin (1996)		—	0.5,A3	—	1,A3
Chlorinated diphenyl oxide					
[31242-93-0] (1990)		—	0.5	—	—
• Chlorine [7782-50-5] (1996)		0.5,A4	1.5,A4	1,A4	2.9,A4
Chlorine dioxide					
[10049-04-4] (1976)		0.1	0.28	0.3	0.83
Chlorine trifluoride					
[7790-91-2] (1977)		—	—	C 0.1	C 0.38
Chloroacetaldehyde					
[107-20-0] (1977)		—	—	C 1	C 3.2
Chloroacetone [78-95-5]—					
Skin (1989)		—	—	C 1	C 3.8
α-Chloroacetophenone					
[532-27-4] (1996)		0.05,A4	0.32,A4	—	—
Chloroacetyl chloride					
[79-04-9]—Skin (1991)		0.05	0.23	0.15	0.69

ADOPTED VALUES					
Substance	[CAS #]	TWA ppm ^{a)}	TWA mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)}	STEL/CEILING (C) mg/m ^{3b)}
◄ Chlorobenzene [108-90-7] (1996)	10,A3	46,A3	—	—	—
o-Chlorobenzylidene malononitrile					
[2698-41-1]—Skin (1996)	—	—	C 0.05,A4	C 0.39,A4	—
Chlorobromomethane					
[74-97-5] (1990)	200	1060	—	—	—
2-Chloro-1,3-butadiene, <i>see</i> β-Chloroprene					
Chlorodifluoromethane					
[75-45-6] (1996)	1000,A4	3540,A4	—	—	—
■ Chlorodiphenyl (42% chlorine)					
[53469-21-9]—Skin (1990)	—	1	—	—	—
■ Chlorodiphenyl (54% chlorine)					
[11097-69-1]—Skin (1996)	—	0.5,A3	—	—	—
1-Chloro-2,3-epoxy propane, <i>see</i> Epichlorohydrin					
2-Chloroethanol, <i>see</i> Ethylene chlorohydrin					
Chloroethylene, <i>see</i> Vinyl chloride					
■ Chloroform [67-66-3] (1996)	10,A3	49,A3	—	—	—
■ bis(Chloromethyl) ether					
[542-88-1] (1981)	0.001,A1	0.0047,A1	—	—	—
■ Chloromethyl methyl ether					
[107-30-2] (1983)	A2	A2	—	—	—
1-Chloro-1-nitropropane					
[600-25-9] (1981)	2	10	—	—	—
Chloropentafluoroethane					
[76-15-3] (1981)	1000	6320	—	—	—
Chloropicrin [76-06-2] (1996)	0.1,A4	0.67,A4	—	—	—
■ β-Chloroprene					
[126-99-8]—Skin (1980)	10	36	—	—	—
2-Chloropropionic acid					
[598-78-7]—Skin (1991)	0.1	0.44	—	—	—
o-Chlorostyrene					
[2039-87-4] (1976)	50	283	75	425	—
o-Chlorotoluene [95-49-8] (1990)	50	259	—	—	—
2-Chloro-6-(trichloromethyl) pyridine, <i>see</i> Nitrapyrin					
Chlorpyrifos [2921-88-2]—					
Skin (1996)	—	0.2,A4	—	—	—
• Chromite ore processing (Chromate), as Cr (1978)	—	0.05,A1	—	—	—
Chromium, metal [7440-47-3], and inorganic compounds, as Cr					
■ Metal and Cr III compounds (1994)	—	0.5,A4	—	—	—
◄■ Water-soluble Cr VI compounds, NOC ^(d) (1994)	—	0.05,A1	—	—	—
■ Insoluble Cr VI compounds, NOC ^(d) (1994)	—	0.01,A1	—	—	—
■ Chromyl chloride					
[14977-61-8] (1982)	0.025	0.16	—	—	—
■ Chrysene [218-01-9] (1996)	—	A3	—	—	—
Chrysotile, <i>see</i> Asbestos					
Clopidol [2971-90-6] (1996)	—	10,A4	—	—	—
‡ Coal dust (1987)	—	(2, ^(g,l))	—	—	—
■ Coal tar pitch volatiles					
[65996-93-2], as benzene solubles (1981)	—	0.2,A1	—	—	—
◄■ Cobalt, elemental [7440-48-4], and inorganic compounds, as Co (1994)	—	0.02,A3	—	—	—
Cobalt carbonyl					
[10210-68-1], as Co (1983)	—	0.1	—	—	—
Cobalt hydrocarbonyl					
[16842-03-8], as Co (1983)	—	0.1	—	—	—
‡ Copper [7440-50-8]					
‡ Fume (1977)	—	(0.2)	—	—	—
‡ Dusts & mists, as Cu (1986)	—	(1)	—	—	—
Cotton dust, raw (1986)	—	0.2 ^(q)	—	—	—
• Cresol [1319-77-3], all isomers—					
Skin (1977)	5	22	—	—	—
Cristobalite, <i>see</i> Silica—Crystalline					
Crocidolite, <i>see</i> Asbestos					

ADOPTED VALUES					
Substance	[CAS #]	TWA ppm ^{a)}	TWA mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)}	STEL/CEILING (C) mg/m ^{3b)}
‡ Crotonaldehyde [4170-30-3] (1996)	(2,A3)	(5.7,A3)	—	—	—
Crufomate [299-86-5] (1996)	—	5,A4	—	—	—
Cumene [98-82-8]—Skin (1987) ..	50	246	—	—	—
Cyanamide [420-04-2] (1977)	—	2	—	—	—
Cyanogen [460-19-5] (1977)	10	21	—	—	—
Cyanogen chloride [506-77-4] (1980)	—	—	C 0.3	C 0.75	—
Cyclohexane [110-82-7] (1987) ..	300	1030	—	—	—
Cyclohexanol [108-93-0]— Skin (1977)	50	206	—	—	—
Cyclohexanone [108-94-1]— Skin (1996)	25,A4	100,A4	—	—	—
Cyclohexene [110-83-8] (1977) ..	300	1010	—	—	—
Cyclohexylamine [108-91-8] (1996)	10,A4	41,A4	—	—	—
* Cyclonite [121-82-4]—Skin (1997) ..	—	0.5, A4	—	—	—
Cyclopentadiene [542-92-7] (1987) ..	75	203	—	—	—
Cyclopentane [287-92-3] (1987) ..	600	1720	—	—	—
Cyhexatin [13121-70-5] (1996) ...	—	5,A4	—	—	—
2,4-D [94-75-7] (1996)	—	10,A4	—	—	—
■ DDT (Dichlorodiphenyltrichloroethane) [50-29-3] (1996)	—	1,A3	—	—	—
Decaborane [17702-41-9]— Skin (1976)	0.05	0.25	0.15	0.75	—
◀ Demeton [8065-48-3]— Skin (1986)	0.01	0.11	—	—	—
Diacetone alcohol [123-42-2] (1987)	50	238	—	—	—
1,2-Diaminoethane, <i>see</i> Ethylenediamine					
Diatomaceous earth, <i>see</i> Silica—Amorphous					
◀ Diazinon [333-41-5]— Skin (1996)	—	0.1,A4	—	—	—
■ Diazomethane [334-88-3] (1996) ..	0.2,A2	0.34,A2	—	—	—
Diborane [19287-45-7] (1977)	0.1	0.11	—	—	—
1,2-Dibromoethane, <i>see</i> Ethylene dibromide					
2-N-Dibutylaminoethanol [102-81-8]—Skin (1994)	0.5	3.5	—	—	—
Dibutyl phenyl phosphate [2528-36-1]—Skin (1990)	0.3	3.5	—	—	—
Dibutyl phosphate [107-66-4] (1976)	1	8.6	2	17	—
Dibutyl phthalate [84-74-2] (1987) ..	—	5	—	—	—
■ Dichloroacetylene [7572-29-4] (1995)	—	—	C 0.1,A3	C 0.39,A3	—
o-Dichlorobenzene [95-50-1] (1996)	25,A4	150,A4	50,A4	301,A4	—
■ p-Dichlorobenzene [106-46-7] (1993)	10,A3	60,A3	—	—	—
■ 3,3'-Dichlorobenzidine [91-94-1]—Skin (1996)	—	A3	—	—	—
1,4-Dichloro-2-butene [764-41-0]—Skin (1993)	0.005,A2	0.025,A2	—	—	—
Dichlorodifluoromethane [75-71-8] (1996)	1000,A4	4950,A4	—	—	—
1,3-Dichloro-5,5-dimethyl hydantoin [118-52-5] (1976) ...	—	0.2	—	0.4	—
1,1-Dichloroethane [75-34-3] (1996)	100,A4	405,A4	—	—	—
1,2-Dichloroethane, <i>see</i> Ethylene dichloride					
1,1-Dichloroethylene, <i>see</i> Vinylidene chloride					
1,2-Dichloroethylene, sym [540-59-0], cis & trans (1987)	200	793	—	—	—
■ Dichloroethyl ether [111-44-4]—Skin (1996)	5,A4	29,A4	10,A4	58,A4	—
Dichlorodifluoromethane [75-43-4] (1980)	10	42	—	—	—

ADOPTED VALUES					
Substance	[CAS #]	TWA ppm ^{a)}	TWA mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)}	STEL/CEILING (C) mg/m ^{3b)}
Dichloromethane, <i>see</i> Methylene chloride					
1,1-Dichloro-1-nitroethane [594-72-9] (1986)	2	12	—	—	—
1,2-Dichloropropane, <i>see</i> Propylene dichloride					
■ 1,3-Dichloropropene [542-75-6]—Skin (1996)	1,A4	4.5,A4	—	—	—
2,2-Dichloropropionic acid [75-99-0] (1980)	1	5.8	—	—	—
Dichlorotetrafluoroethane [76-14-2] (1996)	1000,A4	6990,A4	—	—	—
◀ Dichlorvos [62-73-7]— Skin (1996)	0.1,A4	0.90,A4	—	—	—
Dicrotophos [141-66-2]— Skin (1996)	—	0.25,A4	—	—	—
Dicyclopentadiene [77-73-6] (1977)	5	27	—	—	—
Dicyclopentadienyl iron [102-54-5] (1986)	—	10	—	—	—
■ Dieldrin [60-57-1]—Skin (1996) ..	—	0.25,A4	—	—	—
Diethanolamine [111-42-2]—Skin (1994)	0.46	2	—	—	—
Diethylamine [109-89-7]— Skin (1994)	5,A4	15,A4	15,A4	45,A4	—
2-Diethylaminoethanol [100-37-8]—Skin (1994)	2	9.6	—	—	—
Diethylene triamine [111-40-0]—Skin (1977)	1	4.2	—	—	—
Diethyl ether, <i>see</i> Ethyl ether					
Di(2-ethylhexyl)phthalate, <i>see</i> Di-sec-octyl phthalate					
‡ Diethyl ketone [96-22-0] (1981) ..	200	705	(—)	(—)	—
Diethyl phthalate [84-66-2] (1987)	—	5	—	—	—
Difluorodibromomethane [75-61-6] (1986)	100	858	—	—	—
■ Diglycidyl ether (DGE) [2238-07-5] (1996)	0.1,A4	0.53,A4	—	—	—
Dihydroxybenzene, <i>see</i> Hydroquinone					
Diisobutyl ketone [108-83-8] (1977)	25	145	—	—	—
Diisopropylamine [108-18-9]— Skin (1977)	5	21	—	—	—
Dimethoxymethane, <i>see</i> Methylal					
◀ N,N-Dimethylacetamide [127-19-5]—Skin (1996)	10,A4	36,A4	—	—	—
Dimethylamine [124-40-3] (1996) ..	5,A4	9.2,A4	15,A4	27.6,A4	—
Dimethylaminobenzene, <i>see</i> Xylidene					
◀ Dimethylaniline [121-69-7] (N,N-Dimethylaniline)— Skin (1996)	5,A4	25,A4	10,A4	50,A4	—
Dimethylbenzene, <i>see</i> Xylene					
■ Dimethyl carbamoyl chloride [79-44-7] (1978)	—	A2	—	—	—
Dimethylethoxysilane [14857-34-2] (1996)	0.5	2.1	1.5	6.4	—
Dimethyl-1,2-dibromo-2,2-dichloroethyl phosphate, <i>see</i> Naled					
◀ Dimethylformamide [68-12-2]— Skin (1996)	10,A4	30,A4	—	—	—
2,6-Dimethyl-4-heptanone, <i>see</i> Diisobutyl ketone					
■ 1,1-Dimethylhydrazine [57-14-7]—Skin (1995)	0.01,A3	0.025,A3	—	—	—
Dimethylnitrosoamine, <i>see</i> N-Nitrosodimethylamine					
Dimethylphthalate [131-11-3] (1986)	—	5	—	—	—
■ Dimethyl sulfate [77-78-1]—Skin (1996)	0.1,A3	0.52,A3	—	—	—
Dinitolmide [148-01-6] (1996)	—	5,A4	—	—	—
◀ Dinitrobenzene [528-29-0; 99-65-0; 100-25-4] (all isomers)—Skin (1986) ..	0.15	1.0	—	—	—

ADOPTED VALUES					
Substance	[CAS #]	TWA ppm ^{a)}	mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)}	mg/m ^{3b)}
Dinitro-o-cresol [534-52-1]—Skin (1986)	—	—	0.2	—	—
3,5-Dinitro-o-toluamide, <i>see</i> Dinitolmide					
*◄Dinitrotoluene [25321-14-6]—Skin (1997)	—	—	0.2,A3	—	—
**■ Dioxane [123-91-1]— Skin (1997)	25,A3	—	90, A3	—	—
◄Dioxathion [78-34-2]— Skin (1996)	—	—	0.2,A4	—	—
Diphenyl, <i>see</i> Biphenyl					
Diphenylamine [122-39-4] (1996)	—	—	10,A4	—	—
Diphenylmethane diisocyanate, <i>see</i> Methylene bisphenyl isocyanate					
Dipropylene glycol methyl ether [34590-94-8]—Skin (1976)	100	—	606	150	909
Dipropyl ketone [123-19-3] (1981)	—	50	233	—	—
Diquat [2764-72-9]—Skin (1996)	—	—	0.5, ⁽ⁱ⁾ A4	—	—
	—	—	0.1, ⁽ⁱ⁾ A4	—	—
■ Di-sec-octyl phthalate [117-81-7] (1996)	—	—	5,A3	—	10,A3
Disulfiram [97-77-8] (1996)	—	—	2,A4	—	—
Disulfoton [298-04-4]— Skin (1986)	—	—	0.1	—	—
2,6-Di-tert-butyl-p-cresol [128-37-0] (1996)	—	—	10,A4	—	—
Diuron [330-54-1] (1996)	—	—	10,A4	—	—
Divinyl benzene [1321-74-0] (1980)	10	—	53	—	—
Emery [1302-74-5] (1986)	—	—	10 ^(e)	—	—
Endosulfan [115-29-7]— Skin (1996)	—	—	0.1,A4	—	—
Endrin [72-20-8]—Skin (1996) ..	—	—	0.1,A4	—	—
• Enflurane [13838-16-9] (1996) ...	75,A4	—	566,A4	—	—
Enzymes, <i>see</i> Subtilisins					
**■ Epichlorohydrin [106-89-8]—Skin (1997)	0.5, A3	—	1.9, A3	—	—
◄EPN [2104-64-5]—Skin (1996) ..	—	—	0.1,A4	—	—
1,2-Epoxypropane, <i>see</i> Propylene oxide					
2,3-Epoxy-1-propanol, <i>see</i> Glycidol					
Ethane [74-84-0] (1981)	— ^(c)	—	—	—	—
Ethanethiol, <i>see</i> Ethyl mercaptan					
Ethanol [64-17-5] (1996)	1000,A4	—	1880,A4	—	—
Ethanolamine [141-43-5] (1978) ..	3	—	7.5	6	15
◄Ethion [563-12-2]—Skin (1977) ..	—	—	0.4	—	—
◄• 2-Ethoxyethanol (EGEE) [110-80-5]—Skin (1984)	5	—	18	—	—
◄• 2-Ethoxyethyl acetate (EGEEA) [111-15-9]—Skin (1984)	5	—	27	—	—
Ethyl acetate [141-78-6] (1996) ..	400,A4	—	1440,A4	—	—
•■ Ethyl acrylate [140-88-5] (1990) ..	5,A4	—	20,A4	15,A4	61,A4
Ethyl alcohol, <i>see</i> Ethanol					
Ethylamine [75-04-7]— Skin (1994)	5	—	9.2	15	27.6
Ethyl amyl ketone [541-85-5] (1977)	25	—	131	—	—
◄Ethyl benzene [100-41-4] (1976) ..	100	—	434	125	543
Ethyl bromide [74-96-4]— Skin (1996)	5,A3	—	22,A3	—	—
‡ Ethyl butyl ketone [106-35-4] (1987)	50	—	234	(—)	(—)
Ethyl chloride [75-00-3]— Skin (1995)	100, A3	—	264, A3	—	—
Ethylene [74-85-1] (1996)	A4 ^(c)	—	—	—	—
Ethylene chlorohydrin [107-07-3]—Skin (1996)	—	—	—	C 1,A4	C 3.3,A4
Ethylenediamine [107-15-3]—Skin (1996)	10,A4	—	25,A4	—	—
■ Ethylene dibromide [106-93-4]—Skin (1996)	A3	—	A3	—	—

ADOPTED VALUES					
Substance	[CAS #]	TWA ppm ^{a)}	mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)}	mg/m ^{3b)}
•■ Ethylene dichloride [107-06-2] (1996)	10,A4	—	40,A4	—	—
Ethylene glycol [107-21-1], aerosol (1995)	—	—	—	—	C 100,A4
Ethylene glycol dinitrate [628-96-6]—Skin (1985)	0.05	—	0.31	—	—
Ethylene glycol methyl ether acetate, <i>see</i> 2-Methoxyethyl acetate					
•■ Ethylene oxide [75-21-8] (1984) ..	1,A2	—	1.8,A2	—	—
•■ Ethylenimine [151-56-4]— Skin (1996)	0.5,A3	—	0.88,A3	—	—
Ethyl ether [60-29-7] (1976)	400	—	1210	500	1520
Ethyl formate [109-94-4] (1987) ..	100	—	303	—	—
Ethylidene chloride, <i>see</i> 1,1-Dichloroethane					
Ethylidene norbornene [16219-75-3] (1977)	—	—	—	C 5	C 25
• Ethyl mercaptan [75-08-1] (1986) ..	0.5	—	1.3	—	—
N-Ethylmorpholine [100-74-3]—Skin (1986)	5	—	24	—	—
Ethyl silicate [78-10-4] (1986)	10	—	85	—	—
◄Fenamiphos [22224-92-6]— Skin (1996)	—	—	0.1,A4	—	—
◄Fensulfthion [115-90-2] (1996) ..	—	—	0.1,A4	—	—
◄Fenthion [55-38-9]— Skin (1996)	—	—	0.2,A4	—	—
Ferbam [14484-64-1] (1996)	—	—	10,A4	—	—
Ferrovandium dust [12604-58-9] (1983)	—	—	1	—	3
Fibrous glass dust, <i>see</i> Synthetic Vitreous Fibers— Continuous filament glass fibers					
◄Fluorides, as F (1996)	—	—	2.5,A4	—	—
• Fluorine [7782-41-4] (1976)	1	—	1.6	2	3.1
Fluorotrichloromethane, <i>see</i> Trichlorofluoromethane					
◄Fonofos [944-22-9]—Skin (1996) ..	—	—	0.1,A4	—	—
•■ Formaldehyde [50-00-0] (1992) ...	—	—	—	C 0.3,A2	C 0.37,A2
Formamide [75-12-7]— Skin (1988)	10	—	18	—	—
Formic acid [64-18-6] (1991)	5	—	9.4	10	19
◄Furfural [98-01-1]—Skin (1996) ..	2,A3	—	7.9,A3	—	—
Furfuryl alcohol [98-00-0]— Skin (1982)	10	—	40	15	60
•■ Gasoline [8006-61-9] (1996)	300,A3	—	890,A3	500,A3	1480,A3
Germanium tetrahydride [7782-65-2] (1986)	0.2	—	0.63	—	—
Glass, fibrous or dust, <i>see</i> Synthetic Vitreous Fibers					
‡ Glutaraldehyde [111-30-8], activated & inactivated (1997) ..	—	—	—	(C 0.05)	(C 0.2)
Glycerin mist [56-81-5] (1981) ..	—	—	10	—	—
Glycidol [556-52-5] (1996)	2,A3	—	6.1,A3	—	—
Glycol monoethyl ether, <i>see</i> 2-Ethoxyethanol					
Grain dust (oat, wheat, barley) (1986)	—	—	4 ^(e)	—	—
Graphite (all forms except graphite fibers) [7782-42-5] (1991)	—	—	2 ⁽ⁱ⁾	—	—
Gypsum, <i>see</i> Calcium sulfate					
Hafnium [7440-58-6] (1986)	—	—	0.5	—	—
• Halothane [151-67-7] (1996)	50,A4	—	404,A4	—	—
Helium [7440-59-7] (1981)	— ^(c)	—	—	—	—
■ Heptachlor [76-44-8] and Heptachlor epoxide [1024-57-3]—Skin (1994)	—	—	0.05,A3	—	—
• Heptane [142-82-5] (n-Heptane) (1976)	400	—	1640	500	2050
2-Heptanone, <i>see</i> Methyl n-amyl ketone					
3-Heptanone, <i>see</i> Ethyl butyl ketone					

ADOPTED VALUES					
Substance	[CAS #]	TWA ppm ^{a)}	mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)}	mg/m ^{3b)}
▪ Hexachlorobenzene [118-74-1]— Skin (1997).....	—	—	0.002,A3	—	—
▪ Hexachlorobutadiene [87-68-3]—Skin (1996).....	0.02,A3	0.21,A3	—	—	—
Hexachlorocyclopentadiene [77-47-4] (1996).....	0.01,A4	0.11,A4	—	—	—
▪ Hexachloroethane [67-72-1]—Skin (1996).....	1,A3	9.7,A3	—	—	—
Hexachloronaphthalene [1335-87-1]—Skin (1986).....	—	0.2	—	—	—
Hexafluoroacetone [684-16-2]—Skin (1986).....	0.1	0.68	—	—	—
Hexamethylene diisocyanate [822-06-0] (1988).....	0.005	0.034	—	—	—
▪ Hexamethyl phosphoramide [680-31-9]—Skin (1996).....	A3	A3	—	—	—
‡ Hexane (n-Hexane) [110-54-3] (1982).....	(50)	(176)	—	—	—
• Other isomers (1982).....	500	1760	1000	3500	—
1,6-Hexanediamine [124-09-4] (1992).....	0.5	2.3	—	—	—
2-Hexanone, <i>see</i> Methyl n-butyl ketone					
Hexone, <i>see</i> Methyl isobutyl ketone					
sec-Hexyl acetate [108-84-9] (1977)	50	295	—	—	—
Hexylene glycol [107-41-5] (1977).....	—	—	C 25	C 121	—
▪ Hydrazine [302-01-2]— Skin (1995).....	0.01,A3	0.013,A3	—	—	—
Hydrogen [1333-74-0] (1981).....	— ^(c)	—	—	—	—
Hydrogenated terphenyls [61788-32-7] (1977).....	0.5	4.9	—	—	—
Hydrogen bromide [10035-10-6] (1986).....	—	—	C 3	C 9.9	—
Hydrogen chloride [7647-01-0] (1977).....	—	—	C 5	C 7.5	—
Hydrogen cyanide and Cyanide salts as CN					
Hydrogen cyanide [74-90-8]— Skin (1994).....	—	—	C 4.7	C 5	—
Calcium cyanide [592-01-8]— Skin (1994).....	—	—	—	C 5	—
Potassium cyanide [151-50-8]— Skin (1994).....	—	—	—	C 5	—
Sodium cyanide [143-33-9]— Skin (1994).....	—	—	—	C 5	—
Hydrogen fluoride [7664-39-3], as F (1986).....	—	—	C 3	C 2.3	—
Hydrogen peroxide [7722-84-1] (1996).....	1,A3	1.4,A3	—	—	—
Hydrogen selenide [7783-07-5], as Se (1977).....	0.05	0.16	—	—	—
• Hydrogen sulfide [7783-06-4] (1976).....	10	14	15	21	—
• Hydroquinone [123-31-9] (1996).....	—	2,A3	—	—	—
4-Hydroxy-4-methyl-2-pentanone, <i>see</i> Diacetone alcohol					
2-Hydroxypropyl acrylate [999-61-1]—Skin (1980).....	0.5	2.8	—	—	—
Indene [95-13-6] (1987).....	10	48	—	—	—
Indium [7440-74-6] & compounds, as In (1986).....	—	0.1	—	—	—
Iodine [7553-56-2] (1977).....	—	—	C 0.1	C 1.0	—
Iodoform [75-47-8] (1986).....	0.6	10	—	—	—
Iron oxide dust & fume (Fe ₂ O ₃) [1309-37-1], as Fe (1996).....	—	5, ^(e) A4	—	—	—
Iron pentacarbonyl [13463-40-6], as Fe (1982).....	0.1	0.23	0.2	0.45	—
Iron salts, soluble, as Fe (1986)	—	1	—	—	—
Isoamyl acetate [123-92-2] (1987)	100	532	—	—	—

ADOPTED VALUES					
Substance	[CAS #]	TWA ppm ^{a)}	mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)}	mg/m ^{3b)}
Isoamyl alcohol [123-51-3] (1976)	100	361	125	452	—
Isobutyl acetate [110-19-0] (1990)	150	713	—	—	—
Isobutyl alcohol [78-83-1] (1987)	50	152	—	—	—
Isooctyl alcohol [26952-21-6]— Skin (1982).....	50	266	—	—	—
Isophorone [78-59-1] (1995).....	—	—	C 5,A3	C 28,A3	—
Isophorone diisocyanate [4098-71-9] (1988).....	0.005	0.045	—	—	—
Isopropoxyethanol [109-59-1]—Skin (1987).....	25	106	—	—	—
Isopropyl acetate [108-21-4] (1976).....	250	1040	310	1290	—
Isopropyl alcohol [67-63-0] (1976).....	400	983	500	1230	—
Isopropylamine [75-31-0] (1976).....	5	12	10	24	—
N-Isopropylaniline [768-52-5]—Skin (1986).....	2	11	—	—	—
Isopropyl ether [108-20-3] (1976).....	250	1040	310	1300	—
• Isopropyl glycidyl ether (IGE) [4016-14-2] (1976).....	50	238	75	356	—
Kaolin [1332-58-7] (1996).....	—	2, ^(e),i) A4	—	—	—
Ketene [463-51-4] (1976).....	0.5	0.86	1.5	2.6	—
▪ Lead, elemental [7439-92-1], and inorganic compounds, as Pb (1995).....	—	0.05,A3	—	—	—
▪ Lead arsenate [7784-40-9], as Pb ₃ (AsO ₄) ₂ (1985).....	—	0.15	—	—	—
▪ Lead chromate [7758-97-6], as Pb (1991).....	—	0.05,A2	—	—	—
as Cr (1991).....	—	0.012,A2	—	—	—
Limestone, <i>see</i> Calcium carbonate					
▪ Lindane [58-89-9]—Skin (1996)	—	0.5,A3	—	—	—
Lithium hydride [7580-67-8] (1977).....	—	0.025	—	—	—
L.P.G. (Liquified petroleum gas) [68476-85-7] (1987).....	1000	1800	—	—	—
Magnesite [546-93-0] (1986).....	—	10 ^(e)	—	—	—
Magnesium oxide fume [1309-48-4] (1977).....	—	10	—	—	—
◀ Malathion [121-75-5]— Skin (1996).....	—	10,A4	—	—	—
Maleic anhydride [108-31-6] (1977).....	0.25	1.0	—	—	—
Manganese, elemental [7439-96-5], and inorganic compounds, as Mn (1995).....	—	0.2	—	—	—
Manganese cyclopentadienyl tricarbonyl [12079-65-1], as Mn—Skin (1986).....	—	0.1	—	—	—
Marble, <i>see</i> Calcium carbonate					
◀ Mercury [7439-97-6], as Hg—Skin					
Alkyl compounds (1980).....	—	0.01	—	0.03	—
• Aryl compounds (1982).....	—	0.1	—	—	—
Inorganic forms including metallic mercury (1994).....	—	0.025,A4	—	—	—
• Mesityl oxide [141-79-7] (1981).....	15	60	25	100	—
Methacrylic acid [79-41-4] (1981)	20	70	—	—	—
Methane [74-82-8] (1981).....	— ^(c)	—	—	—	—
Methanethiol, <i>see</i> Methyl mercaptan					
◀ Methanol [67-56-1]— Skin (1976).....	200	262	250	328	—
◀ Methomyl [16752-77-5] (1996).....	—	2.5,A4	—	—	—
• Methoxychlor [72-43-5] (1996).....	—	10,A4	—	—	—
◀ 2-Methoxyethanol (EGME) [109-86-4]—Skin (1984).....	5	16	—	—	—
◀ 2-Methoxyethyl acetate (EGMEA) [110-49-6]—Skin (1984).....	5	24	—	—	—

ADOPTED VALUES					
Substance	[CAS #]	TWA ppm ^{a)}	mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)}	mg/m ^{3b)}
4-Methoxyphenol [150-76-5] (1982)	—	—	5	—	—
Methyl acetate [79-20-9] (1976)	200	606	250	757	—
Methyl acetylene [74-99-7] (1990)	1000	1640	—	—	—
Methyl acetylene-propadiene mixture (MAPP) (1976)	1000	1640	1250	2050	—
• Methyl acrylate [96-33-3]— Skin (1997)	2,A4	7,A4	—	—	—
Methylacrylonitrile [126-98-7]—Skin (1986)	1	2.7	—	—	—
Methylal [109-87-5] (1987)	1000	3110	—	—	—
Methyl alcohol, <i>see</i> Methanol	—	—	—	—	—
Methylamine [74-89-5] (1992)	5	6.4	15	19	—
Methyl amyl alcohol, <i>see</i> Methyl isobutyl carbinol	—	—	—	—	—
Methyl n-amyl ketone [110-43-0] (1987)	50	233	—	—	—
◀ N-Methyl aniline [100-61-8]— Skin (1986)	0.5	2.2	—	—	—
•• Methyl bromide [74-83-9]— Skin (1997)	1,A4	3.9,A4	—	—	—
▪ Methyl-tert butyl ether [1634-04-4] (1995)	40,A3	144,A3	—	—	—
‡ Methyl n-butyl ketone [591-78-6]—Skin (1981)	5	20	(—)	(—)	—
• Methyl chloride [74-87-3]— Skin (1996)	50,A4	103,A4	100,A4	207,A4	—
•◀ Methyl chloroform [71-55-6] (1996)	350,A4	1910,A4	450,A4	2460,A4	—
‡ Methyl 2-cyanoacrylate [137-05-3] (1976)	(2)	(9.1)	(4)	(18)	—
Methylcyclohexane [108-87-2] (1987)	400	1610	—	—	—
Methylcyclohexanol [25639-42-3] (1987)	50	234	—	—	—
o-Methylcyclohexanone [583-60-8]—Skin (1976)	50	229	75	344	—
2-Methylcyclopentadienyl manganese tricarbonyl [12108-13-3], as Mn —Skin (1986)	—	0.2	—	—	—
◀ Methyl demeton [8022-00-2]—Skin (1986)	—	0.5	—	—	—
Methylene bisphenyl isocyanate (MDI) [101-68-8] (1988)	0.005	0.051	—	—	—
• Methylene chloride (Dichloromethane) [75-09-2] (1996)	50,A3	174,A3	—	—	—
•• 4,4'-Methylene bis(2-chloroaniline) [MOCA; MBOCA] [101-14-4]—Skin (1993)	0.01,A2	0.11,A2	—	—	—
Methylene bis(4-cyclohexylisocyanate) [5124-30-1] (1988)	0.005	0.054	—	—	—
•• 4,4'-Methylene dianiline [101-77-9]—Skin (1996)	0.1,A3	0.81,A3	—	—	—
◀ Methyl ethyl ketone (MEK) [78-93-3] (1976)	200	590	300	885	—
Methyl ethyl ketone peroxide [1338-23-4] (1977)	—	—	C 0.2	C 1.5	—
Methyl formate [107-31-3] (1976)	100	246	150	368	—
5-Methyl-3-heptanone, <i>see</i> Ethyl amyl ketone	—	—	—	—	—
▪ Methyl hydrazine [60-34-4]—Skin (1995)	0.01,A3	0.019,A3	—	—	—
▪ Methyl iodide [74-88-4]— Skin (1996)	2	12	—	—	—
Methyl isoamyl ketone [110-12-3] (1982)	50	234	—	—	—
Methyl isobutyl carbinol [108-11-2]—Skin (1976)	25	104	40	167	—
◀ Methyl isobutyl ketone [108-10-1] (1981)	50	205	75	307	—

ADOPTED VALUES					
Substance	[CAS #]	TWA ppm ^{a)}	mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)}	mg/m ^{3b)}
Methyl isocyanate [624-83-9]—Skin (1977)	0.02	0.047	—	—	—
Methyl isopropyl ketone [563-80-4] (1981)	200	705	—	—	—
• Methyl mercaptan [74-93-1] (1977)	0.5	0.98	—	—	—
Methyl methacrylate [80-62-6] (1996)	100,A4	410,A4	—	—	—
◀ Methyl parathion [298-00-0]—Skin (1996)	—	0.2,A4	—	—	—
• Methyl propyl ketone [107-87-9] (1976)	200	705	250	881	—
Methyl silicate [681-84-5] (1986)	1	6	—	—	—
α-Methyl styrene [98-83-9] (1981)	50	242	100	483	—
Metribuzin [21087-64-9] (1996)	—	5,A4	—	—	—
◀ Mevinphos [7786-34-7]— Skin (1976)	0.01	0.092	0.03	0.27	—
Mica [12001-26-2] (1986)	—	3(e,i)	—	—	—
Mineral wool fiber, <i>see</i> Synthetic Vitreous Fibers— Glass, Rock, or Slag wool fibers	—	—	—	—	—
Molybdenum [7439-98-7], as Mo	—	5	—	—	—
Soluble compounds (1986)	—	10	—	—	—
Insoluble compounds (1986)	—	—	—	—	—
Monochlorobenzene, <i>see</i> Chlorobenzene	—	—	—	—	—
Monocrotophos [6923-22-4]—Skin (1996)	—	0.25,A4	—	—	—
Morpholine [110-91-8]— Skin (1996)	20,A4	71,A4	—	—	—
◀ Naled [300-76-5]—Skin (1996)	—	3,A4	—	—	—
Naphthalene [91-20-3] (1996)	10,A4	52,A4	15,A4	79,A4	—
▪ β-Naphthylamine [91-59-8] (1972)	—	A1	—	—	—
Neon [7440-01-9] (1981)	—(c)	—	—	—	—
‡• Nickel [7440-02-0]	—	(1)	—	—	—
‡• Metal (1966)	—	(1)	—	—	—
‡• Insoluble compounds, as Ni (1974)	—	(1)	—	—	—
‡• Soluble compounds, as Ni (1976)	—	(0.1)	—	—	—
‡• Nickel carbonyl [13463-39-3], as Ni (1977)	(0.05)	(0.12)	—	—	—
‡• Nickel sulfide roasting, fume & dust, as Ni (1978)	—	(1,A1)	—	—	—
Nicotine [54-11-5]— Skin (1986)	—	0.5	—	—	—
Nitrapyrin [1929-82-4] (1996)	—	10,A4	—	20,A4	—
Nitric acid [7697-37-2] (1976)	2	5.2	4	10	—
◀ Nitric oxide [10102-43-9] (1986)	25	31	—	—	—
◀ p-Nitroaniline [100-01-6]— Skin (1996)	—	3,A4	—	—	—
◀ Nitrobenzene [98-95-3]— Skin (1996)	1,A3	5,A3	—	—	—
•◀ p-Nitrochlorobenzene [100-00-5]—Skin (1996)	0.1,A3	0.64,A3	—	—	—
▪ 4-Nitrodiphenyl [92-93-3]— Skin (1996)	—	A2	—	—	—
Nitroethane [79-24-3] (1986)	100	307	—	—	—
Nitrogen [7727-37-9] (1989)	—(c)	—	—	—	—
• Nitrogen dioxide [10102-44-0] (1996)	3,A4	5.6,A4	5,A4	9.4,A4	—
◀ Nitrogen trifluoride [7783-54-2] (1986)	10	29	—	—	—
• Nitroglycerin (NG) [55-63-0]—Skin (1985)	0.05	0.46	—	—	—
Nitromethane [75-52-5] (1994)	20	50	—	—	—
1-Nitropropane [108-03-2] (1996)	25,A4	91,A4	—	—	—
• 2-Nitropropane [79-46-9] (1996)	10,A3	36,A3	—	—	—

ADOPTED VALUES					
Substance	[CAS #]	TWA ppm ^{a)}	mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)}	mg/m ^{3b)}
■ N-Nitrosodimethylamine [62-75-9]—Skin (1996)	—	—	A3	—	—
◀ Nitrotoluene [88-72-2; 99-08-1; 99-99-0] —Skin (1982)	2	11	—	—	—
Nitrotrichloromethane, <i>see</i> Chloropicrin					
Nitrous oxide [10024-97-2] (1996)	50,A4	90,A4	—	—	—
Nonane [111-84-2], all isomers (1976)	200	1050	—	—	—
Nuisance particulates, <i>see</i> Particulates Not Otherwise Classified (PNOC)					
Octachloronaphthalene [2234-13-1]—Skin (1976)	—	0.1	—	0.3	—
Octane [111-65-9] (1976)	300	1400	375	1750	—
‡ Oil mist, mineral (1976)	—	5(k)	—	(10)	—
Osmium tetroxide [20816-12-0], as Os (1976)	0.0002	0.0016	0.0006	0.0047	—
Oxalic acid [144-62-7] (1976)	—	1	—	2	—
Oxygen difluoride [7783-41-7] (1986)	—	—	C 0.05	C 0.11	—
‡* Ozone [10028-15-6] ‡* Heavy work (1997)	(0.05)	(0.1)	—	—	—
‡* Moderate work (1997)	(0.08)	(0.16)	—	—	—
‡* Light work (1997)	(0.1)	(0.2)	—	—	—
Paraffin wax fume [8002-74-2] (1987)	—	2	—	—	—
Paraquat [4685-14-7], Total particulate (1978)	—	0.5	—	—	—
Respirable fraction (1978)	—	0.1(i)	—	—	—
◀ Parathion [56-38-2]— Skin (1996)	—	0.1,A4	—	—	—
Particulate polycyclic aromatic hydrocarbons (PPAH), <i>see</i> Coal tar pitch volatiles					
Particulates Not Otherwise Classified (PNOC)					
Inhalable (1989)	—	10(e,i)	—	—	—
Respirable (1995)	—	3(e,i)	—	—	—
Pentaborane [19624-22-7] (1976)	0.005	0.013	0.015	0.039	—
Pentachloronaphthalene [1321-64-8]—Skin (1986)	—	0.5	—	—	—
Pentachloronitrobenzene [82-68-8] (1996)	—	0.5,A4	—	—	—
◀ Pentachlorophenol [87-86-5]—Skin (1996)	—	0.5,A3	—	—	—
Pentaerythritol [115-77-5] (1986)	—	10	—	—	—
‡ Pentane (all isomers) (1976) ...	600	1770	(750)	(2210)	—
2-Pentanone, <i>see</i> Methyl propyl ketone					
◀ Perchloroethylene (Tetrachloroethylene) [127-18-4] (1993)	25,A3	170,A3	100,A3	685,A3	—
Perchloromethyl mercaptan [594-42-3] (1977)	0.1	0.76	—	—	—
Perchloryl fluoride [7616-94-6] (1976)	3	13	6	25	—
Perfluoroisobutylene [382-21-8] (1992)	—	—	C 0.01	C 0.082	—
Precipitated silica, <i>see</i> Silica—Amorphous					
Perlite [93763-70-3] (1996)	—	10,(e)A4	—	—	—
Persulfates					
Ammonium [7727-54-0] (1996)	—	0.1	—	—	—
Potassium [7727-21-1] (1996)	—	0.1	—	—	—
Sodium [7775-27-1] (1996)	—	0.1	—	—	—

ADOPTED VALUES					
Substance	[CAS #]	TWA ppm ^{a)}	mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)}	mg/m ^{3b)}
Petroleum distillates, <i>see</i> Gasoline; Stoddard solvent; VM&P naphtha					
Phenacyl chloride, <i>see</i> α-Chloroacetophenone					
◀ Phenol [108-95-2]—Skin (1996)	5,A4	19,A4	—	—	—
Phenothiazine [92-84-2]— Skin (1986)	—	5	—	—	—
■ N-Phenyl-beta-naphthylamine [135-88-6] (1996)	—	A4	—	—	—
o-Phenylenediamine [95-54-5] (1996)	—	0.1,A3	—	—	—
m-Phenylenediamine [108-45-2] (1996)	—	0.1,A4	—	—	—
p-Phenylenediamine [106-50-3] (1996)	—	0.1,A4	—	—	—
Phenyl ether [101-84-8], vapor (1976)	1	7	2	14	—
Phenylethylene, <i>see</i> Styrene, monomer					
■ Phenyl glycidyl ether (PGE) [122-60-1]—Skin (1994)	0.1,A3	0.6,A3	—	—	—
• Phenylhydrazine [100-63-0]—Skin (1996)	0.1,A3	0.44,A3	—	—	—
• Phenyl mercaptan [108-98-5] (1978)	0.5	2.3	—	—	—
Phenylphosphine [638-21-1] (1977)	—	—	C 0.05	C 0.23	—
Phorate [298-02-2]—Skin (1976)	—	0.05	—	0.2	—
Phosdrin, <i>see</i> Mevinphos					
Phosgene [75-44-5] (1978)	0.1	0.40	—	—	—
Phosphine [7803-51-2] (1976) ...	0.3	0.42	1	1.4	—
Phosphoric acid [7664-38-2] (1976)	—	1	—	3	—
Phosphorus (yellow) [7723-14-0] (1986)	0.02	0.1	—	—	—
Phosphorus oxychloride [10025-87-3] (1990)	0.1	0.63	—	—	—
Phosphorus pentachloride [10026-13-8] (1980)	0.1	0.85	—	—	—
Phosphorus pentasulfide [1314-80-3] (1976)	—	1	—	3	—
Phosphorus trichloride [7719-12-2] (1982)	0.2	1.1	0.5	2.8	—
Phthalic anhydride [85-44-9] (1996)	1,A4	6.1,A4	—	—	—
m-Phthalodinitrile [626-17-5] (1977)	—	5	—	—	—
Picloram [1918-02-1] (1996)	—	10,A4	—	—	—
Picric acid [88-89-1] (1990)	—	0.1	—	—	—
Pindone [83-26-1] (1987)	—	0.1	—	—	—
Piperazine dihydrochloride [142-64-3] (1982)	—	5	—	—	—
2-Pivalyl-1,3-indandione, <i>see</i> Pindone					
Plaster of Paris, <i>see</i> Calcium sulfate					
Platinum [7440-06-4] Metal (1981)	—	1	—	—	—
Soluble salts, as Pt (1970) ...	—	0.002	—	—	—
Polychlorobiphenyls, <i>see</i> Chlorodiphenyls					
Polytetrafluoroethylene decomposition products (1972)	—	B1	—	—	—
Portland cement [65997-15-1] (1986)	—	10(e)	—	—	—
Potassium hydroxide [1310-58-3] (1977)	—	—	—	C 2	—
‡ Propane [74-98-6] (1981)	(—) ^(c)	(—)	—	—	—
■ Propane sulfone [1120-71-4] (1996)	—	A3	—	—	—
Propargyl alcohol [107-19-7]—Skin (1987)	1	2.3	—	—	—
■ β-Propiolactone [57-57-8] (1996)	0.5,A3	1.5,A3	—	—	—
Propionic acid [79-09-4] (1990)	10	30	—	—	—
Propoxur [114-26-1] (1996)	—	0.5,A3	—	—	—

ADOPTED VALUES					
Substance	[CAS #]	TWA ppm ^{a)}	TWA mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)}	STEL/CEILING (C) mg/m ^{3b)}
n-Propyl acetate [109-60-4] (1976)	200	835	250	1040	
n-Propyl alcohol [71-23-8]—					
Skin (1976).....	200	492	250	614	
Propylene [115-07-1] (1996).....	A4(C)	—	—	—	—
• Propylene dichloride					
[78-87-5] (1996).....	75,A4	347,A4	110,A4	508,A4	
◀ Propylene glycol dinitrate					
[6423-43-4]—Skin (1985).....	0.05	0.34	—	—	—
Propylene glycol monomethyl ether					
[107-98-2] (1976).....	100	369	150	553	
■ Propylene imine [75-55-8]—					
Skin (1996).....	2,A3	4.7,A3	—	—	—
• Propylene oxide [75-56-9] (1996)	20,A3	48,A3	—	—	—
◀ n-Propyl nitrate [627-13-4] (1978)	25	107	40	172	
Propyne, <i>see</i> Methyl acetylene					
Pyrethrum [8003-34-7] (1996) ...	—	5,A4	—	—	—
Pyridine [110-86-1] (1987).....	5	16	—	—	—
Pyrocatechol, <i>see</i> Catechol					
Quartz, <i>see</i> Silica—Crystalline					
Quinone [106-51-4] (1987).....	0.1	0.44	—	—	—
RDX, <i>see</i> Cyclonite					
Resorcinol [108-46-3] (1996).....	10,A4	45,A4	20,A4	90,A4	
• Rhodium [7440-16-6]					
• Metal (1996).....	—	1,A4	—	—	—
• Insoluble compounds,					
as Rh (1996).....	—	1,A4	—	—	—
• Soluble compounds,					
as Rh (1996).....	—	0.01,A4	—	—	—
Ronnel [299-84-3] (1996).....	—	10,A4	—	—	—
Rosin core solder thermal decomposition					
products, as resin acids—colophony					
[8050-09-7] (1993).....		Sensitizer; reduce exposure	to as low as possible		
Rotenone (commercial)					
[83-79-4] (1996).....	—	5,A4	—	—	—
Rouge (1996).....	—	10,(e)A4	—	—	—
• Rubber solvent (Naphtha)					
[8030-30-6] (1977).....	400	1590	—	—	—
Selenium [7782-49-2] and compounds,					
as Se (1977).....	—	0.2	—	—	—
Selenium hexafluoride					
[7783-79-1], as Se (1979).....	0.05	0.16	—	—	—
Sesone [136-78-7] (1996).....	—	10,A4	—	—	—
Silane, <i>see</i> Silicon tetrahydride					
Silica—Amorphous					
• Diatomaceous earth (uncalcined)					
[61790-53-2]					
Inhalable (1986).....	—	10,(e,i)	—	—	—
Respirable (1995).....	—	3(e,i)	—	—	—
• Precipitated silica					
[112926-00-8] (1987).....	—	10	—	—	—
Silica, fume [69012-64-2] (1992)	—	2(i)	—	—	—
• Silica, fused [60676-86-0] (1992)	—	0.1(i)	—	—	—
• Silica gel [112926-00-8] (1987)	—	—	10	—	—
■ Silica—Crystalline					
Cristobalite [14464-46-1] (1986)	—	0.05(i)	—	—	—
Quartz [14808-60-7] (1986).....	—	0.1(i)	—	—	—
Tridymite [15468-32-3] (1986).....	—	0.05(i)	—	—	—
Tripoli [1317-95-9] (1985).....	—	0.1,(i)	of contained	respirable quartz	—
Silicon [7440-21-3] (1986).....	—	10	—	—	—
Silicon carbide [409-21-2] (1996)	—	10,(e)A4	—	—	—
Silicon tetrahydride					
[7803-62-5] (1983).....	5	6.6	—	—	—
Silver [7440-22-4]					
• Metal (1981).....	—	0.1	—	—	—
Soluble compounds,					
as Ag (1981).....	—	0.01	—	—	—

ADOPTED VALUES					
Substance	[CAS #]	TWA ppm ^{a)}	TWA mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)}	STEL/CEILING (C) mg/m ^{3b)}
Soapstone					
Inhalable (1985).....	—	6(e,i)	—	—	—
Respirable (1985).....	—	3(e,i)	—	—	—
Sodium azide [26628-22-8] (1996)					
as Sodium azide.....	—	—	—	C 0.29,A4	—
as Hydrazoic acid vapor.....	—	—	C 0.11,A4	—	—
Sodium bisulfite					
[7631-90-5] (1996).....	—	5,A4	—	—	—
Sodium 2,4-dichloro-phenoxyethyl					
sulfate, <i>see</i> Sesone					
Sodium fluoroacetate					
[62-74-8] — Skin (1994).....	—	0.05	—	—	—
Sodium hydroxide [1310-73-2] (1977).....	—	—	—	C 2	—
Sodium metabisulfite					
[7681-57-4] (1996).....	—	5,A4	—	—	—
Starch [9005-25-8] (1996).....	—	10,A4	—	—	—
Stearates ⁽¹⁾ (1996).....	—	10,A4	—	—	—
Stibine [7803-52-3] (1986).....	0.1	0.51	—	—	—
• Stoddard solvent					
[8052-41-3] (1987).....	100	525	—	—	—
Strontium chromate					
[7789-06-2], as Cr (1992).....	—	0.0005,A2	—	—	—
Strychnine [57-24-9] (1986).....	—	0.15	—	—	—
• Styrene, monomer					
[100-42-5] (1997).....	20,A4	85,A4	40,A4	170,A4	
Subtilisins [1395-21-7; 9014-01-1]					
(Proteolytic enzymes					
as 100% pure crystalline					
enzyme) (1977).....	—	—	—	C 0.00006(m)	—
Sucrose [57-50-1] (1996).....	—	10,A4	—	—	—
Sulfometuron methyl					
[74222-97-2] (1994).....	—	5,A4	—	—	—
◀ Sulfotep [3689-24-5]—					
Skin (1996).....	—	0.2,A4	—	—	—
Sulfur dioxide [7446-09-5] (1996)					
13,A4		2,A4	5.2,A4	5,A4	
Sulfur hexafluoride					
[2551-62-4] (1986).....	1000	5970	—	—	—
Sulfuric acid [7664-93-9] (1996)	—	1,A2☆	—	3,A2☆	—
Sulfur monochloride					
[10025-67-9] (1986).....	—	—	C 1	C 5.5	—
Sulfur pentafluoride					
[5714-22-7] (1986).....	—	—	C 0.01	C 0.10	—
Sulfur tetrafluoride					
[7783-60-0] (1986).....	—	—	C 0.1	C 0.44	—
Sulfuryl fluoride					
[2699-79-8] (1976).....	5	21	10	42	—
Sulprofos [35400-43-2] (1996)...	—	1,A4	—	—	—
Synthetic vitreous fibers					
• Continuous filament					
glass fibers (1997).....	—	1 f/cc,(h)A4	—	—	—
• Continuous filament					
glass fibers (1997).....	—	5,(i)A4	—	—	—
• Glass wool fibers (1997).....	—	1 f/cc,(h)A3	—	—	—
• Rock wool fibers (1997).....	—	1 f/cc,(h)A3	—	—	—
• Slag wool fibers (1997).....	—	1 f/cc,(h)A3	—	—	—
• Special purpose					
glass fibers (1997).....	—	1 f/cc,(h)A3	—	—	—
Systox, <i>see</i> Demeton					
2,4,5-T [93-76-5] (1996).....	—	10,A4	—	—	—
Talc (containing no asbestos					
fibers) [14807-96-6] (1996).....	—	2,(e,i)A4	—	—	—
■ Talc (containing asbestos					
fibers) (1985).....	—	Use asbestos TLV—TWA ⁽ⁿ⁾	—	—	—
Tantalum [7440-25-7], metal and oxide					
[1314-61-0] dusts, as Ta (1988)	—	5	—	—	—
TEDP, <i>see</i> Sulfotep					

ADOPTED VALUES					
Substance	[CAS #]	TWA ppm ^{a)} mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)} mg/m ^{3b)}		
Tellurium [13494-80-9] and compounds, as Te (1977).....	—	0.1	—	—	—
Tellurium hexafluoride [7783-80-4], as Te (1977).....	0.02	0.10	—	—	—
◀ Temephos [3383-96-8] (1986)....	—	10	—	—	—
◀ TEPP [107-49-3]—Skin (1986) ...	0.004	0.047	—	—	—
Terephthalic acid [100-21-0] (1993)	—	10	—	—	—
Terphenyls [26140-60-3] (1980)...	—	—	C 0.53	C 5	—
1,1,1,2-Tetrachloro-2,2- difluoroethane [76-11-9] (1986).....	500	4170	—	—	—
1,1,2,2-Tetrachloro-1,2- difluoroethane [76-12-0] (1986).....	500	4170	—	—	—
‡ 1,1,2,2-Tetrachloroethane [79-34-5]—Skin (1996).....	1,(A4)	6.9,(A4)	—	—	—
Tetrachloroethylene, <i>see</i> Perchloroethylene					
Tetrachloromethane, <i>see</i> Carbon tetrachloride					
Tetrachloronaphthalene [1335-88-2] (1986).....	—	2	—	—	—
• Tetraethyl lead [78-00-2], as Pb—Skin (1996).....	—	0.1,(0)A4	—	—	—
Tetrahydrofuran [109-99-9] (1976).....	200	590	250	737	—
• Tetramethyl lead [75-74-1], as Pb — Skin (1986).....	—	0.15(0)	—	—	—
Tetramethyl succinonitrile [3333-52-6]—Skin (1986).....	0.5	2.8	—	—	—
Tetranitromethane [509-14-8] (1996).....	0.005,A3	0.04,A3	—	—	—
Tetrasodium pyrophosphate [7722-88-5] (1980).....	—	5	—	—	—
Tetryl [479-45-8] (1986).....	—	1.5	—	—	—
Thallium, elemental [7440-28-0], and soluble compounds, as Tl—Skin (1977).....	—	0.1	—	—	—
4,4'-Thiobis(6-tert-butyl-m-cresol) [96-69-5] (1996).....	—	10,A4	—	—	—
Thioglycolic acid [68-11-1]—Skin (1978).....	1	3.8	—	—	—
Thionyl chloride [7719-09-7] (1986).....	—	—	C 1	C 4.9	—
Thiram [137-26-8] (1996).....	—	1,A4	—	—	—
Tin [7440-31-5] Metal (1982).....	—	2	—	—	—
Oxide & inorganic compounds, except SnH ₄ , as Sn (1982).....	—	2	—	—	—
Organic compounds, as Sn—Skin (1996).....	—	0.1,A4	—	0.2,A4	—
• Titanium dioxide [13463-67-7] (1996).....	—	10,A4	—	—	—
■ o-Tolidine [119-93-7]— Skin (1996).....	—	A3	—	—	—
Toluene [108-88-3]— Skin (1996).....	50,A4	188,A4	—	—	—
• Toluene-2,4-diisocyanate (TDI) [584-84-9] (1996).....	0.005,A4	0.036,A4	0.02,A4	0.14,A4	—
•• o-Toluidine [95-53-4]— Skin (1996).....	2,A3	8.8,A3	—	—	—
◀ m-Toluidine [108-44-1]— Skin (1996).....	2,A4	8.8,A4	—	—	—
••p-Toluidine [106-49-0]— Skin (1996).....	2,A3	8.8,A3	—	—	—
Toluol, <i>see</i> Toluene					
Toxaphene, <i>see</i> Chlorinated camphene					
Tributyl phosphate [126-73-8] (1986).....	0.2	2.2	—	—	—
Trichloroacetic acid [76-03-9] (1996).....	1,A4	6.7,A4	—	—	—

ADOPTED VALUES					
Substance	[CAS #]	TWA ppm ^{a)} mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)} mg/m ^{3b)}		
1,2,4-Trichlorobenzene [120-82-1] (1978).....	—	—	C 5	C 37	—
1,1,1-Trichloroethane, <i>see</i> Methyl chloroform					
■ 1,1,2-Trichloroethane [79-00-5]—Skin (1996).....	10,A4	55,A4	—	—	—
•• Trichloroethylene [79-01-6] (1993).....	50,A5	269,A5	100,A5	537,A5	—
Trichlorofluoromethane [75-69-4] (1996).....	—	—	C 1000,A4	C 5620,A4	—
Trichloromethane, <i>see</i> Chloroform					
Trichloronaphthalene [1321-65-9]—Skin (1986).....	—	5	—	—	—
Trichloronitromethane, <i>see</i> Chloropicrin					
■ 1,2,3-Trichloropropane [96-18-4]—Skin (1996).....	10,A3	60,A3	—	—	—
1,1,2-Trichloro-1,2,2-trifluoroethane [76-13-1] (1996).....	1000,A4	7670,A4	1250,A4	9590,A4	—
Tricyclohexyltin hydroxide, <i>see</i> Cyhexatin					
Tridymite, <i>see</i> Silica—Crystalline					
Triethanolamine [102-71-6] (1993)	—	5	—	—	—
Triethylamine [121-44-8]— Skin (1995).....	1,A4	4.1,A4	3,A4	12,A4	—
Trifluorobromomethane [75-63-8] (1986).....	1000	6090	—	—	—
* 1,3,5-Triglycidyl-s-triazinetriene [2451-62-9] (1997).....	—	0.05	—	—	—
Trimellitic anhydride [552-30-7] (1993).....	—	—	—	C 0.04	36
Trimethylamine [75-50-3] (1992) .	5	12	15	—	—
Trimethyl benzene [25551-13-7] (1987).....	25	123	—	—	—
Trimethyl phosphite [121-45-9] (1986).....	2	10	—	—	—
2,4,6-Trinitrophenol, <i>see</i> Picric acid					
2,4,6-Trinitrophenylmethylnitramine, <i>see</i> Tetryl					
• 2,4,6-Trinitrotoluene (TNT) [118-96-7]—Skin (1997).....	—	0.1	—	—	—
Triorthocresyl phosphate [78-30-8]—Skin (1996).....	—	0.1,A4	—	—	—
Triphenyl amine [603-34-9] (1980).....	—	5	—	—	—
Triphenyl phosphate [115-86-6] (1996).....	—	3,A4	—	—	—
Tripoli, <i>see</i> Silica—Crystalline					
Tungsten [7440-33-7], as W Insoluble compounds (1976) .	—	5	—	10	—
Soluble compounds (1976).....	—	1	—	3	—
Turpentine [8006-64-2] (1987)....	100	556	—	—	—
• Uranium (natural) [7440-61-1] Soluble & insoluble compounds, as U (1996)	—	0.2,A1	—	0.6,A1	—
n-Valeraldehyde [110-62-3] (1978).....	50	176	—	—	—
◀ Vanadium pentoxide [1314-62-1], as V ₂ O ₅ ; respirable dust or fume (1996).....	—	0.05,A4	—	—	—
Vegetable oil mists ^(P) (1972) ..	—	10	—	—	—
• Vinyl acetate [108-05-4] (1993).....	10,A3	35,A3	15,A3	53,A3	—
Vinyl benzene, <i>see</i> Styrene					
• Vinyl bromide [593-60-2] (1980) .	5,A2	22,A2	—	—	—
• Vinyl chloride [75-01-4] (1980) ...	5,A1	13,A1	—	—	—
Vinyl cyanide, <i>see</i> Acrylonitrile					
4-Vinyl cyclohexene [100-40-3] (1996).....	0.1,A3	0.44,A3	—	—	—
■ Vinyl cyclohexene dioxide [106-87-6]—Skin (1996).....	0.1,A3	0.57,A3	—	—	—

ADOPTED VALUES					
Substance	[CAS #]	TWA ppm ^{a)}	TWA mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)}	STEL/CEILING (C) mg/m ^{3b)}
†• Vinylidene chloride [75-35-4] (1996).....	5,(A3)	20,(A3)	20,(A3)	79,(A3)	483,A4
Vinyl toluene [25013-15-4] (1996)	50,A4	242,A4	100,A4	483,A4	
• VM & P Naphtha [8032-32-4] (1996).....	300,A3	1370,A3	—	—	—
Warfarin [81-81-2] (1987).....	—	0.1	—	—	—
• Welding fumes (NOC ^{d)}) (1977).....	—	5,B2	—	—	—
• Wood dust (certain hard woods as beech & oak) (1996).....	—	1,A1	—	—	—
• Soft wood (1981).....	—	5	—	10	—
◀ Xylene [1330-20-7; 95-47-6; 108-38-3; 106-42-3] (o-, m-, p-isomers) (1996).....	100,A4	434,A4	150,A4	651,A4	—
m-Xylene α,α'-diamine [1477-55-0]—Skin (1977).....	—	—	—	C 0.1	—
◀ Xylidine (mixed isomers) [1300-73-8]—Skin (1996).....	0.5,A3	2.5,A3	—	—	—
Yttrium [7440-65-5] metal & compounds, as Y (1988).....	—	1	—	—	—
Zinc chloride fume [7646-85-7] (1976).....	—	1	—	2	—
• Zinc chromates [13530-65-9; 11103-86-9; 37300-23-5], as Cr (1988).....	—	0.01,A1	—	—	—
Zinc oxide [1314-13-2] Fume (1976).....	—	5	—	10	—
Dust (1976).....	—	10 ^(e)	—	—	—
Zirconium [7440-67-7] and compounds, as Zr (1996).....	—	5,A4	—	10,A4	—

☆A2 designation refers to Sulfuric acid contained in strong inorganic acid mists.

NOTICE OF INTENDED CHANGES (for 1997)

These substances, with their corresponding values, comprise those for which either a limit has been proposed for the first time, for which a change in the "Adopted" listing has been proposed, or for which retention on the Notice of Intended Changes has been proposed. In all cases, the proposed limits should be considered trial limits that will remain in the listing for a period of at least one year. If, after one year no evidence comes to light that questions the appropriateness of the values herein, the values will be reconsidered for the "Adopted" list. Documentation is available for each of these substances and their proposed values.

Substance	[CAS #]	TWA ppm ^{a)}	TWA mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)}	STEL/CEILING (C) mg/m ^{3b)}
† Acrolein [107-02-8]—Skin.....	—	—	C 0.1,A4	C 0.23,A4	—
† Allyl glycidyl ether [106-92-3] ..	1,A4	4.7,A4	—	—	—
† Asbestos, all forms [1332-21-4].....	—	0.1 f/cc, ^(f) A1	—	—	—
† Asphalt (Petroleum; Bitumen) fume [8052-42-4], as cyclohexane- extractable inhalable particulate ..	—	0.5	—	—	—
n-Butanol [71-36-3].....	—	—	C 25	C 76	—

ADOPTED VALUES					
Substance	[CAS #]	TWA ppm ^{a)}	TWA mg/m ^{3b)}	STEL/CEILING (C) ppm ^{a)}	STEL/CEILING (C) mg/m ^{3b)}
† n-Butyl acetate [123-86-4].....	150	713	200	950	—
Coal dust	—	—	—	—	—
Bituminous.....	—	0.9,(g,i)A4	—	—	—
Anthracite.....	—	0.4,(g,i)A4	—	—	—
Copper [7440-50-8] and Inorganic compounds, as Cu Fume and respirable particulate.....	—	0.05 ⁽ⁱ⁾	—	—	—
Inhalable particulate, dusts & mists.....	—	1 ⁽ⁱ⁾	—	—	—
† Crotonaldehyde [4170-30-3]—Skin	—	—	C 0.3,A3	C 0.86,A3	—
Diesel exhaust, particulate ($<1 \mu\text{m}$).....	—	0.15,A2	—	—	—
† Diesel fuel/Kerosene—Skin ..	—	100,A3	—	—	—
† Diethyl ketone [96-22-0].....	200	705	300	1057	—
† Ethyl butyl ketone [106-35-4] ..	50	234	75	350	—
Ethyl cyanoacrylate [7085-85-0].....	0.2	1	—	—	—
† Glutaraldehyde [111-30-8]	—	—	C 0.05,A4	C 0.2,A4	—
† n-Hexane [110-54-3]—Skin	50	176	—	—	—
† 1-Hexene [592-41-6].....	30	103	—	—	—
† Methyl n-butyl ketone [591-78-6]—Skin.....	5	20	10	40	—
† Methyl 2-cyanoacrylate [137-05-3].....	0.2	1	—	—	—
† Methyl vinyl ketone [78-94-4]—Skin.....	—	—	C 0.2	C 0.6	—
† Nickel	—	—	—	—	—
† Elemental/Metal [7440-02-0]	—	1.5, ⁽ⁱ⁾ A5	—	—	—
† Soluble compounds, as Ni	—	—	0.1, ⁽ⁱ⁾ A4	—	—
† Insoluble compounds, as Ni.....	—	0.2, ⁽ⁱ⁾ A1	—	—	—
Nickel carbonyl [13463-39-3], as Ni	0.05	0.12	—	—	—
† Nickel subsulfide [12035-72-2], as Ni.....	—	0.1, ⁽ⁱ⁾ A1	—	—	—
Nickel sulfide roasting, Delete listing; included in listing for Nickel fume & dust, as Ni	—	—	—	—	—
Oil mist, mineral.....	—	5 ^(k)	—	—	—
Oil mist, mineral, sum total of 15 polynuclear aromatic hydrocarbons (PAHs) listed as carcinogens by the U.S. National Toxicology Program (NTP).....	—	0.005, ^(k) A1	—	—	—
† Ozone [10028-15-6]	—	—	—	—	—
Heavy work.....	0.05,A4	0.1,A4	—	—	—
Moderate work.....	0.08,A4	0.16,A4	—	—	—
Light work.....	0.10,A4	0.2,A4	—	—	—
Light, moderate, or heavy workloads.....	0.2☆,A4	0.4☆,A4	—	—	—
Pentane (all isomers).....	600	1770	—	—	—
Propane [74-98-6].....	2500	4508	—	—	—
Synthetic vitreous fibers	—	—	—	—	—
Refractory ceramic fibers.....	—	A2◇	—	—	—
† 1,1,2,2-Tetrachloroethane [79-34-5]—Skin.....	1,A3	6.9,A3	—	—	—
† Vinylidene chloride [75-35-4].	5,A4	20,A4	20,A4	79,A4	—

☆ ≤ 2 hours

◇ The TLV for Refractory ceramic fibers is under review.

ADOPTED APPENDICES

APPENDIX A: Carcinogenicity

The Chemical Substances TLV Committee has been aware of the increasing public concern over chemicals or industrial processes that cause or contribute to increased risk of cancer in workers. More sophisticated methods of bioassay, as well as the use of sophisticated mathematical models that extrapolate the levels of risk among workers, have led to differing interpretations as to which chemicals or processes should be categorized as human carcinogens and what the maximum exposure levels should be. The goal of the Committee has been to synthesize the available information in a manner that will be useful to practicing industrial hygienists, without overburdening them with needless details. The categories for carcinogenicity are:

- A1 — *Confirmed Human Carcinogen*: The agent is carcinogenic to humans based on the weight of evidence from epidemiologic studies of, or convincing clinical evidence in, exposed humans.
- A2 — *Suspected Human Carcinogen*: The agent is carcinogenic in experimental animals at dose levels, by route(s) of administration, at site(s), of histologic type(s), or by mechanism(s) that are considered relevant to worker exposure. Available epidemiologic studies are conflicting or insufficient to confirm an increased risk of cancer in exposed humans.
- A3 — *Animal Carcinogen*: The agent is carcinogenic in experimental animals at a relatively high dose, by route(s) of administration, at site(s), of histologic type(s), or by mechanism(s) that are not considered relevant to worker exposure. Available epidemiologic studies do not confirm an increased risk of cancer in exposed humans. Available evidence suggests that the agent is not likely to cause cancer in humans except under uncommon or unlikely routes or levels of exposure.
- A4 — *Not Classifiable as a Human Carcinogen*: There are inadequate data on which to classify the agent in terms of its carcinogenicity in humans and/or animals.
- A5 — *Not Suspected as a Human Carcinogen*: The agent is not suspected to be a human carcinogen on the basis of properly conducted epidemiologic studies in humans. These studies have sufficiently long follow-up, reliable exposure histories, sufficiently high dose, and adequate statistical power to conclude that exposure to the agent does not convey a significant risk of cancer to humans. Evidence suggesting a lack of carcinogenicity in experimental animals will be considered if it is supported by other relevant data.

Substances for which no human or experimental animal carcinogenic data have been reported are assigned no carcinogenicity designation.

Exposures to carcinogens must be kept to a minimum. Workers exposed to A1 carcinogens without a TLV should be properly equipped to eliminate to the fullest extent possible all exposure to the carcinogen. For A1 carcinogens with a TLV and for A2 and A3 carcinogens, worker exposure by all routes should be carefully controlled to levels as low as possible below the TLV. Refer to the "Guidelines for the Classification of Occupational Carcinogens" in the Introduction to the 6th Edition of the *Documentation of the Threshold Limit Values and Biological Exposure Indices* for a more complete description and derivation of these designations.

APPENDIX B: Substances of Variable Composition

B1. Polytetrafluoroethylene* decomposition products. Thermal decomposition of the fluorocarbon chain in air leads to the formation of oxidized products containing carbon, fluorine, and oxygen. Because these products decompose in part by hydrolysis in alkaline solution, they can be quantitatively determined in air as fluoride to provide an index of exposure. No TLVs are recommended at this time, but air concentration should be controlled as low as possible.

(*Trade names include: Algoflon, Fluon, Teflon, Tetran)

B2. Welding Fumes—Total Particulate ($NOC^{(d)}$); TLV-TWA, 5 mg/m³

Welding fumes cannot be classified simply. The composition and quantity of both are dependent on the alloy being welded and the process and electrodes used. Reliable analysis of fumes cannot be made without considering the nature of the welding process and system being examined; reactive metals and alloys such as aluminum and titanium are arc-welded in a protective, inert atmosphere such as argon. These arcs create relatively little fume, but they do create an intense radiation which can produce ozone. Similar processes are used to arc-weld steels, also creating a relatively low level of fumes. Ferrous alloys also are arc-welded in oxidizing environments that generate considerable fume and can produce carbon monoxide instead of ozone. Such fumes generally are composed of discrete particles of amorphous slags containing iron, manganese, silicon, and other metallic constituents depending on the alloy system involved. Chromium and nickel compounds are found in fumes when stainless steels are arc-welded. Some coated and flux-cored electrodes are formulated with fluorides and the fumes associated with them can contain significantly more fluorides than oxides. Because of the above factors, arc-welding fumes frequently must be tested for individual constituents that are likely to be present to determine whether specific TLVs are exceeded. Conclusions based on total concentration are generally adequate if no toxic elements are present in welding rod, metal, or metal coating and conditions are not conducive to the formation of toxic gases.

APPENDIX C: Threshold Limit Values for Mixtures

When two or more hazardous substances which act upon the same organ system are present, their combined effect, rather than that of either individually, should be given primary consideration. In the absence of information to the contrary, the effects of the different hazards should be considered as additive. That is, if the sum of

$$\frac{C_1}{T_1} + \frac{C_2}{T_2} + \dots + \frac{C_n}{T_n}$$

exceeds unity, then the threshold limit of the mixture should be considered as being exceeded. C_1 indicates the observed atmospheric concentration and T_1 the corresponding threshold limit (see Example A.1 and B.1).

Exceptions to the above rule may be made when there is a good reason to believe that the chief effects of the different harmful substances are not in fact additive, but are independent as when purely local effects on different organs of the body are produced by the various components of the mixture. In such cases, the threshold limit ordinarily is exceeded only when at least one member of the series ($C_1/T_1 + \text{or} + C_2/T_2$, etc.) itself has a value exceeding unity (see Example B.1).

Synergistic action or potentiation may occur with some combinations of atmospheric contaminants. Such cases at present must be determined individually. Potentiating or synergistic agents are not necessarily harmful by themselves. Potentiating effects of exposure to such agents by routes other than that of inhalation are also possible, e.g., imbibed alcohol and inhaled narcotic (trichloroethylene). Potentiation is characteristically exhibited at high concentrations, less probably at low.

When a given operation or process characteristically emits a number of harmful dusts, fumes, vapors or gases, it will frequently be only feasible to attempt to evaluate the hazard by measurement of a single substance. In such cases, the threshold limit used for this substance should be reduced by a suitable factor, the magnitude of which will depend on the number, toxicity, and relative quantity of the other contaminants ordinarily present.

Examples of processes that are typically associated with two or more harmful atmospheric contaminants are welding, automobile repair, blasting, painting, lacquering, certain foundry operations, diesel exhausts, etc.

Examples of TLVs for Mixtures

A. Additive effects. The following formulae apply only when the components in a mixture have similar toxicologic effects; they should not be used for mixtures with widely differing reactivities, e.g., hydrogen cyanide and

sulfur dioxide. In such case, the formula for **Independent Effects** should be used.

1. General case, where air is analyzed for each component, the TLV of mixture =

$$\frac{C_1}{T_1} + \frac{C_2}{T_2} + \frac{C_3}{T_3} + \dots = 1$$

Note: It is essential that the atmosphere be analyzed both qualitatively and quantitatively for each component present in order to evaluate compliance or noncompliance with this calculated TLV.

Example A.1: Air contains 400 ppm of acetone (TLV, 500 ppm), 150 ppm of sec-butyl acetate (TLV, 200 ppm) and 100 ppm of methyl ethyl ketone (TLV, 200 ppm).

Atmospheric concentration of mixture = 400 + 150 + 100 = 650 ppm of mixture.

$$\frac{400}{500} + \frac{150}{200} + \frac{100}{200} = 0.80 + 0.75 + 0.5 = 2.05$$

Threshold Limit is exceeded.

2. Special case when the source of contaminant is a liquid mixture and the atmospheric composition is assumed to be similar to that of the original material, e.g., on a time-weighted average exposure basis, all of the liquid (solvent) mixture eventually evaporates. When the percent composition (by weight) of the liquid mixture is known, the TLVs of the constituents must be listed in mg/m³. TLV of mixture =

$$\frac{f_a}{TLV_a} + \frac{f_b}{TLV_b} + \frac{f_c}{TLV_c} + \dots + \frac{f_n}{TLV_n} = 1$$

Note: In order to evaluate compliance with this TLV, field sampling instruments should be calibrated, in the laboratory, for response to this specific quantitative and qualitative air-vapor mixture, and also to fractional concentrations of this mixture, e.g., 1/2 the TLV; 1/10 the TLV; 2 × the TLV; 10 × the TLV; etc.)

Example A.2: Liquid contains (by weight):

50% heptane: TLV = 400 ppm or 1640 mg/m³

1 mg/m³ = 0.24 ppm

30% methyl chloroform: TLV = 350 ppm or 1910 mg/m³

1 mg/m³ = 0.18 ppm

20% perchloroethylene: TLV = 25 ppm or 170 mg/m³

1 mg/m³ = 0.15 ppm

$$\begin{aligned} \text{TLV of Mixture} &= \frac{1}{\frac{0.5}{1640} + \frac{0.3}{1910} + \frac{0.2}{170}} \\ &= \frac{1}{0.00030 + 0.00016 + 0.00118} \\ &= \frac{1}{0.00164} = 610 \text{ mg/m}^3 \end{aligned}$$

of this mixture

50% or (610)(0.5) = 305 mg/m³ is heptane

30% or (610)(0.3) = 183 mg/m³ is methyl chloroform

20% or (610)(0.2) = 122 mg/m³ is perchloroethylene

These values can be converted to ppm as follows:

heptane: 305 mg/m³ × 0.24 = 73 ppm

methyl chloroform: 183 mg/m³ × 0.18 = 33 ppm

perchloroethylene: 122 mg/m³ × 0.15 = 18 ppm

TLV of mixture = 73 + 33 + 18 = 124 ppm, or 610 mg/m³

B. Independent effects. TLV for mixture =

$$\frac{C_1}{T_1} = 1; \quad \frac{C_2}{T_2} = 1; \quad \frac{C_3}{T_3} = 1; \text{ etc.}$$

Example B.1: Air contains 0.05 mg/m³ of lead (TLV, 0.05) and 0.7 mg/m³ of sulfuric acid (TLV, 1).

$$\frac{0.05}{0.05} = 1; \quad \frac{0.7}{1} = 0.7$$

Threshold limit is not exceeded.

C. TLV for mixtures of mineral dusts. For mixtures of biologically active mineral dusts, the general formula for mixtures given in A.2 may be used.

APPENDIX D: Particle Size-Selective Sampling Criteria for Airborne Particulate Matter

For chemical substances present in inhaled air as suspensions of solid particles or droplets, the potential hazard depends on particle size as well as mass concentration because of: 1) effects of particle size on the deposition site within the respiratory tract, and 2) the tendency for many occupational diseases to be associated with material deposited in particular regions of the respiratory tract.

The Chemical Substances TLV Committee has recommended particle size-selective TLVs for crystalline silica for many years in recognition of the well established association between silicosis and respirable mass concentrations. The Committee is now re-examining other chemical substances encountered in particulate form in occupational environments with the objective of defining: 1) the size-fraction most closely associated for each substance with the health effect of concern, and 2) the mass concentration within that size fraction which should represent the TLV.

The Particle Size-Selective TLVs (PSS-TLVs) are expressed in three forms:

1. *Inhalable Particulate Mass TLVs* (IPM-TLVs) for those materials that are hazardous when deposited anywhere in the respiratory tract.
2. *Thoracic Particulate Mass TLVs* (TPM-TLVs) for those materials that are hazardous when deposited anywhere within the lung airways and the gas-exchange region.
3. *Respirable Particulate Mass TLVs* (RPM-TLVs) for those materials that are hazardous when deposited in the gas-exchange region.

The three particulate mass fractions described above are defined in quantitative terms in accordance with the following equations:^(1,2)

- A. *Inhalable Particulate Mass* consists of those particles that are captured according to the following collection efficiency regardless of sampler orientation with respect to wind direction:

$$SI(d) = 50\% \times (1 + e^{-0.06d})$$

for $0 < d \leq 100 \mu\text{m}$

where: $SI(d)$ = the collection efficiency for particles with aerodynamic diameter d in μm

- B. *Thoracic Particulate Mass* consists of those particles that are captured according to the following collection efficiency:

$$ST(d) = SI(d) [1 - F(x)]$$

$$\ln(d/T)$$

$$\text{where: } x = \frac{\ln(\Sigma)}{\ln(\Sigma)}$$

$$\Gamma = 11.64 \mu\text{m}$$

$$\Sigma = 1.5$$

$F(x)$ = the cumulative probability function of a standardized normal variable, x

- C. *Respirable Particulate Mass* consists of those particles that are captured according to the following collection efficiency:

$$SR(d) = SI(d) [1 - F(x)]$$

where $F(x)$ has the same meaning as above with $\Gamma = 4.25 \mu\text{m}$ and $\Sigma = 1.5$

The most significant difference from previous definitions is the increase in the median cut point for a respirable particulate matter sampler from 3.5 μm to 4.0 μm ; this is in accord with the International Organization for Standardization/European Standardization Committee (ISO/CEN) protocol^(3,4). At this time, no change is recommended for the measurement of respirable particulates using a 10-mm nylon cyclone at a flow rate of 1.7 liters per minute. Two analyses of available data indicate that the flow rate of 1.7 liters per minute allows the 10-mm nylon cyclone to approximate the particulate matter concentration which would be measured by an ideal respirable particulate sampler as defined herein.^(5,6)

Collection efficiencies representative of several sizes of particles in each of the respective mass fractions are shown in Tables I, II, and III. References 2 and 3 provide documentation for the respective algorithms representative of the three mass fractions.

TABLE I. Inhalable

Particle Aerodynamic Diameter (μm)	Inhalable Particulate Mass (IPM) (%)
0	100
1	97
2	94
5	87
10	77
20	65
30	58
40	54.5
50	52.5
100	50

TABLE II. Thoracic

Particle Aerodynamic Diameter (μm)	Thoracic Particulate Mass (TPM) (%)
0	100
2	94
4	89
6	80.5
8	67
10	50
12	35
14	23
16	15
18	9.5
20	6
25	2

TABLE III. Respirable

Particle Aerodynamic Diameter (μm)	Respirable Particulate Mass (RPM) (%)
0	100
1	97
2	91
3	74
4	50
5	30
6	17
7	9
8	5
10	1

References

1. American Conference of Governmental Industrial Hygienists: Particle Size-Selective Sampling in the Workplace. ACGIH, Cincinnati, OH (1985).
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4. European Standardization Committee (CEN): Size Fraction Definitions for Measurement of Airborne Particles in the Workplace. Approved for publication as prEN 481. CEN, Brussels (1992).
5. Bartley, D.L.: Letter to J. Doull, TLV Committee, July 9, 1991.
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APPENDIX B

PHYSICAL CONSTANTS OF SELECTED MATERIALS

Substance	Formula	Molecular Weight	Specific Gravity	Flash Point F		Explosive Limits (Volume Percent)	
				Closed Cup	Open Cup	Lower	Upper
Acetaldehyde	CH ₃ CHO	44.05	0.821	-17	—	3.97	57.0
Acetic Acid	CH ₃ COOH	60.05	1.049	104	110	5.40	—
Acetic Anhydride	(CH ₃ CO) ₂ O	102.09	1.082	121	130	2.67	10.13
Acetone	CH ₃ COCH ₃	58.08	0.792	0	15	2.55	12.80
Acrolein	CH ₂ :CHCHO	56.06	0.841	—	Gas	Unstable	
Acrylonitrile	CH ₂ :CHCN	53.06	0.806	—	32	3.05	17.0
Ammonia	NH ₃	17.03	0.597	—	Gas	15.50	27.0
Amyl Acetate	CH ₃ CO ₂ C ₅ H ₁₁	130.18	0.879	76	80	1.10	—
iso-Amyl Alcohol	(CH ₃) ₂ CHCH ₂ CH ₂ OH	88.15	0.812	109	115	1.20	—
Aniline	C ₆ H ₅ NH ₂	93.12	1.022	168	—	—	—
Arsine	AsH ₃	77.93	2.695 (A)	—	Gas	—	—
Benzene	C ₆ H ₆	78.11	0.879	12	—	1.40	7.10
Bromine	Br ₂	159.83	3.119	—	—	—	—
Butane	CH ₃ (CH ₂) ₂ CH ₃	58.12	2.085	—	Gas	1.86	8.41
1,3-Butadiene	(CH ₂ :CH) ₂	54.09	0.621	—	Gas	2.00	11.50
n-Butanol	C ₂ H ₅ CH ₂ CH ₂ OH	74.12	0.810	84	110	1.45	11.25
2-Butanone	CH ₃ COC ₂ H ₅	72.10	0.805	30	—	1.81	9.50
(Methyl ethyl ketone)							
n-Butyl Acetate	CH ₃ CO ₂ C ₄ H ₉	116.16	0.882	72	90	1.39	7.55
Butyl "Cellosolve"	C ₄ H ₉ OCH ₂ CH ₂ OH	118.17	0.903	141	165	—	—
Carbon Dioxide	CO ₂	44.01	1.53	—	—	—	—
Carbon Disulphide	CS ₂	76.13	1.263	-22	—	1.25	50.0
Carbon Monoxide	CO	28.10	0.968	—	Gas	12.5	74.2
Carbon Tetrachloride	CCl ₄	153.84	1.595	—	Nonflammable	—	—
Cellosolve	C ₂ H ₅ O(CH ₂) ₂ OH	90.12	0.931	104	120	2.6	15.7
Cellosolve Acetate	CH ₃ CO ₂ C ₄ H ₉ O	132.16	0.975	124	135	1.71	—
Chlorine	Cl ₂	70.91	3.214	—	Gas	—	—
2-Chlorobutadiene	CH ₂ :CClCHCH ₂	88.54	0.958	—	—	—	—
Chloroform	CHCl ₃	119.39	1.478	—	Nonflammable	—	—
1-Chloro-1-nitropropane	NO ₂ CH ₂ CH ₂ Cl	139.54	1.209	144	—	—	—
Cyclohexane	C ₆ H ₁₂	84.16	0.779	1	—	1.26	7.75
Cyclohexanol	CH ₂ (CH ₂) ₄ CHOH	100.16	0.962	154	—	—	—
Cyclohexanone	CH ₂ (CH ₂) ₄ CO	98.14	0.948	147	—	—	—
Cyclohexene	CH ₂ (CH ₂) ₃ CH:CH	82.14	0.810	—	—	—	—
Cyclopropane	CH ₂ CH ₂ CH ₂	42.08	0.720	—	Gas	2.40	10.40
o-Dichlorobenzene	Cl ₂ C ₆ H ₄	147.01	1.305	151	165	—	—
Dichlorodifluoromethane	CCl ₂ F ₂	120.92	1.486	—	Nonflammable	—	—
1,1-Dichloroethane	CH ₃ CHCl ₂	98.97	1.175	—	—	—	—
1,2-Dichloroethane	ClCH ₂ CH ₂ Cl	98.97	1.257	56	65	6.2	15.9
(Ethylene dichloride)							
1,2-Dichloroethylene	ClCH=CHCl	96.95	1.291	43	—	9.7	12.8
Dichloroethylether	ClCH ₂ CHClC ₂ H ₅	143.02	1.222	131	180	—	—
Dichloromethane	H ₂ CCl ₂	84.94	1.336	—	—	—	—
Dichloromonofluoromethane	HCCl ₂ F	102.93	1.426	—	—	—	—
1,1-Dichloro-1-nitroethane	H ₃ C ₂ Cl ₂ NO ₂	143.97	1.692	—	168	—	—
1,2-Dichloropropane	CH ₃ CHClCH ₂ Cl	112.99	1.159	59	65	3.4	14.5
Dichlorotetrafluoroethane	CClF ₂ CClF ₂	170.93	1.433	—	Nonflammable	—	—
Dimethylaniline	(CH ₃) ₂ NC ₆ H ₅	121.18	0.956	145	170	—	—
Dimethylsulfate	(CH ₃) ₂ SO ₄	126.13	1.332	182	240	—	—
Dioxane	O(CH ₂) ₄ O	88.10	1.034	—	35	—	—
Ethyl Acetate	CH ₃ CO ₂ C ₂ H ₅	88.10	0.901	24	30	2.18	11.4
Ethyl Alcohol	C ₂ H ₅ OH	46.07	0.789	55	—	3.28	18.95
Ethyl Benzene	C ₆ H ₅ C ₂ H ₅	106.16	0.867	59	75	—	—
Ethyl Bromide	C ₂ H ₅ Br	109.98	1.430	—	—	6.75	11.25
Ethyl Chloride	C ₂ H ₅ Cl	64.52	0.921	-58	-45	3.6	14.80
Ethylene Chlorohydrin	ClCH ₂ CH ₂ OH	80.52	1.213	—	140	—	—

PHYSICAL CONSTANTS OF SELECTED MATERIALS (con't)

Substance	Formula	Molecular Weight	Specific Gravity	Flash Point F		Explosive Limits (Volume Percent)	
				Closed Cup	Open Cup	Lower	Upper
Ethylenediamine	$\text{NH}_2\text{CH}_2\text{CH}_2\text{NH}_2$	60.10	0.899	—	—	—	—
Ethylene Oxide	$\text{CH}_2\text{CH}_2\text{O}$	44.05	0.887	—	—	3.0	80.0
Ethyl Ether	$(\text{C}_2\text{H}_5)_2\text{O}$	74.12	0.713	—	—	—	—
Ethyl Formate	$\text{HCO}_2\text{C}_2\text{H}_5$	74.08	0.917	—	—	2.75	16.40
Ethyl Silicate	$(\text{C}_2\text{H}_5)_4\text{SiO}_4$	208.30	0.933	—	125	—	—
Formaldehyde	HCHO	30.03	0.815	—	Gas	7.0	73.0
Gasoline	$\text{CH}_n\text{H}(2n + 2)$	86.0	0.660	-50	—	1.3	6.0
Heptane	$\text{CH}_3(\text{CH}_2)_5\text{CH}_3$	100.20	0.684	25	—	1.1	6.7
Hexane	$\text{CH}_3(\text{CH}_2)_4\text{CH}_3$	86.17	0.660	-7	—	1.18	7.4
Hydrogen Chloride	HCl	36.47	1.268 (A)	—	—	—	—
Hydrogen Cyanide	HCN	27.03	0.688	—	Gas	5.6	40.0
Hydrogen Fluoride	HF	20.01	0.987	—	Gas	—	—
Hydrogen Selenide	H_2Se	80.98	2.12	—	Gas	—	—
Hydrogen Sulfide	H_2S	34.08	1.189 (A)	—	Gas	4.3	45.5
Iodine	I_2	253.82	4.93	—	—	—	—
Isophorone	$(\text{CH}_3)_3\text{C}(\text{CH}_2)_2\text{CCHCO}$	138.20	0.923	—	205	—	—
Mesityl Oxide	$(\text{CH}_3)_2\text{:CHCOCH}_3$	98.14	0.857	87	—	—	—
Methanol	CH_3OH	32.04	0.792	54	60	6.72	36.5
Methyl Acetate	$\text{CH}_3\text{CO}_2\text{CH}_3$	74.08	0.928	15	20	3.15	15.60
Methyl Bromide	CH_3Br	94.95	1.732	—	—	13.5	14.5
Methyl Butanone (Isopropyl butane)	$\text{CH}_3\text{COCH}(\text{CH}_3)_2$	86.13	0.803	—	—	—	—
Methyl Cellosolve	$\text{HOCH}_2\text{CH}_2\text{OCH}_3$	76.06	0.965	107	115	—	—
Methyl Cellosolve Acetate	$\text{CH}_3\text{OCH}_2\text{CH}_2\text{OOCCH}_3$	118.13	1.007	132	140	—	—
Methyl Chloride	CH_3Cl	50.49	1.785	—	Gas	8.25	18.70
Methyl Cyclohexane	$\text{CH}_3(\text{CH}_2)_5\text{H}$	98.18	0.769	25	—	1.15	—
Methyl Cyclohexanol	$\text{CH}_3(\text{CH}_2)_4\text{CH}_2\text{OH}$	114.18	0.934	154	—	—	—
Methyl Cyclohexanone	$\text{CH}_3\text{C}_6\text{H}_{10}\text{CO}$	122.17	0.925	118	—	—	—
Methyl Formate	HCO_2CH_3	60.05	0.974	-2	—	4.5	20.0
Methyl Isobutyl Ketone	$\text{CH}_3\text{COC}_4\text{H}_9$	100.16	0.801	73	—	—	—
Monochlorobenzene	$\text{C}_6\text{H}_5\text{Cl}$	112.56	1.107	90	—	—	—
Monofluorotrichloromethane	Cl_3CF	137.38	1.494	Nonflammable		—	—
Mononitrotoluene	$\text{CH}_3\text{C}_6\text{H}_4\text{NO}_2$	137.13	1.163	223	—	—	—
Naphtha (coal tar)	C_{10}H_8	106.16	0.85	100-110	—	—	—
Nickel Carbonyl	$\text{Ni}(\text{CO})_4$	170.73	1.31	—	—	—	—
Nitrobenzene	$\text{C}_6\text{H}_5\text{NO}_2$	123.11	1.205	190	—	1.8 (200 F)	—
Nitroethane	$\text{CH}_3\text{CH}_2\text{NO}_2$	75.07	1.052	82	106	—	—
Nitrogen Oxides	NO	30.0	1.0367 (A)	—	—	—	—
	N_2O	44.02	1.53	—	—	—	—
	N_2O_3	76.02	1.447	—	—	—	—
	NO_2	46.01	1.448	—	—	—	—
	N_2O_5	108.02	1.642	—	—	—	—
Nitroglycerine	$\text{C}_3\text{H}_5(\text{ONO}_2)_3$	227.09	1.601	—	—	—	—
Nitromethane	CH_3NO_2	61.04	1.130	95	112	—	—
2-Nitropropane	$\text{CH}_3\text{CHNO}_2\text{CH}_3$	89.09	1.003	—	103	—	—
Octane	$\text{CH}_3(\text{CH}_2)_6\text{CH}_3$	114.22	0.703	56	—	0.95	3.2
Ozone	O_3	48.0	1.658 (A)	—	—	—	—
Pentane	$\text{CH}_3(\text{CH}_2)_3\text{CH}_3$	72.15	0.626	-40	—	1.4	7.8
Pentanone (Methylpropanone)	$\text{CH}_3\text{COCH}_2\text{C}_2\text{H}_5$	86.13	0.816	45	60	1.55	8.15
Phosgene	O:C:Cl_2	98.92	1.392	—	—	—	—
Phosphine	PH_3	34.0	1.146 (A)	—	205	—	—
Phosphorus Trichloride	PCl_3	137.35	1.574	—	—	—	—
iso-Propanol	$(\text{CH}_3)_2\text{CHOH}$	60.09	0.785	53	60	2.02	11.80
Propane	$\text{CH}_3\text{CH}_2\text{CH}_3$	44.09	1.554	—	Gas	2.12	9.35
Propyl Acetate	$\text{CH}_3\text{CO}_2\text{CH}_2\text{C}_2\text{H}_5$	102.13	0.886	43	60	1.77	8.0
iso-Propyl Ether	$(\text{CH}_3)_2\text{CH}_2\text{O}$	102.17	0.725	-18	-15	—	—
Stibine	SbH_3	124.78	4.344 (A)	—	—	—	—
Styrene Monomer	$\text{C}_6\text{H}_5\text{HC:CH}_2$	104.14	0.903	90	—	1.1	6.1

PHYSICAL CONSTANTS OF SELECTED MATERIALS (con't)

Substance	Formula	Molecular Weight	Specific Gravity	Flash Point F		Explosive Limits (Volume Percent)	
				Closed Cup	Open Cup	Lower	Upper
Sulfur Chloride, Mono	S_2Cl_2	135.03	1.678	245	None	—	—
Di	SCl_2	102.97	1.621	—	—	—	—
Tetra	SCl_4	173.89	—	—	—	—	—
Sulfur Dioxide	SO_2	64.07	2.264 (A)	—	Gas	—	—
1,1,2,2, Tetrachloroethane	$Cl_2CHCHCl_2$	167.86	1.588	—	—	—	—
Tetrachloroethylene	$Cl_2C:CCl_2$	165.85	1.624	Nonflammable		—	—
Toluene	$C_6H_5CH_3$	92.13	0.866	40	45	1.27	6.75
Toluidine	$CH_3C_6H_4NH_2$	107.15	0.999	188	205	—	—
Trichloroethylene	$ClCHCCl_2$	131.40	1.466	Nonflammable		—	—
Turpentine (Turpene)	$C_{10}H_{16}$	136.23	—	95	—	0.8	—
Vinyl Chloride (Chloroethane)	C_2H_5Cl	62.50	0.908	—	Gas	4.0	21.70
Xylene	$C_6H_4(CH_3)_2$	106.16	0.881	63	75	1.0	6.0

SOLVENT DRYING TIME

SOLVENT	Dry Time Relation	Boiling Range		Weight per Gal. Lbs.
		Deg. C	Deg. F.	
Ethyl Ether, C.P.	1.0	34-35	93-95	5.98
Petrolene	1.8	61-96	142-205	5.83
Carbon Tetrachloride	1.9	76	169	13.30
Acetone	2.0	55-58	133-136	6.35
Methyl Acetate	2.2	56-62	133-144	7.79
Ethyl Acetate 85-88%	2.5	74-77	165-171	7.37
Trichlorethylene	2.5	87	189	12.20
Benzol (Industrial)	2.6	79-81	174-178	7.38
Methyl Ethyl Ketone	2.7	77-82	171-180	6.95
Isopropyl Acetate 8%	2.7	84-93	183-199	7.26
Ethylene Dichloride	3.0	84	183	10.49
Solvol 19/27	3.7	86-123	187-254	6.58
Ethylene Chloride	4.0	81-87	178-189	10.49
Propylene Dichloride	4.1	93-97	199-207	9.64
Toluol	4.1	90-122	194-252	6.17
Methanol	5.0	64-65	147-149	6.63
Toluol (Industrial)	5.0	109-111	229-232	7.19
Methyl Propyl Ketone	5.2	101-107	214-225	6.77
V. M. & P.	5.8	95-141	203-286	6.23
Perchloroethylene	6.0	121	250	13.55
Nor. Propyl Acetate	6.1	97-101	207-214	7.50
Sec. Butyl Acetate	6.5	106-135	223-275	7.13
Solox (Anhydrous)	6.5	71-78	160-172	6.80
Isobutyl Acetate 90%	7.0	106-117	223-243	7.28
Apochinner	7.0	115-143	239-289	6.31
Ethyl Alcohol, Den. No. 1	7.7	78	172	6.64
Solox	8.0	76-78	169-172	6.73
Isopropyl Alcohol 99%	8.6	79-82	174-180	6.75
Nor. Propyl Alcohol	9.1	96-98	205-208	6.73

SOLVENT DRYING TIME (con't)

SOLVENT	Dry Time Relation	Boiling Range		Weight per Gal. Lbs.
		Deg. C	Deg. F.	
Solvsol 24/34	9.4	101-168	214-334	6.80
Nor. Butyl Acetate	9.6	110-132	230-270	7.29
Diethyl Carbonate	9.6	100-130	212-266	8.14
Methyl Butyl Ketone	9.7	114-137	237-279	6.84
Xylol (Industrial)	9.7	127-144	261-291	7.17
Monochlor Benzol	10.0	130-132	266-270	9.20
Tertiary Butyl Alcohol	11.9	82-83	180-181	6.55
Sec. Butyl Alcohol	14.0	99-100	210-212	6.85
Sec. Amyl Acetate	16.9	121-144	250-291	7.21
Amyl Acetate	17.4	105-142	221-288	7.24
Isobutyl Alcohol	17.7	107-111	225-232	6.70
Methyl Cellosolve	18.0	121-126	250-259	8.07
Butyl Propionate	18.0	124-171	255-340	7.31
Pentacetate	20.0	121-155	250-311	7.19
Turpentine	20.0	155-173	311-343	12.24
Butanol	21.0	116-119	241-246	6.79
Sec. Amyl Alcohol	25.0	105-125	221-257	6.79
2-50—W Hi-Flash Naphtha	27.5	148-187	298-369	7.18
Amyl Alcohol (Fusel Oil)	32.1	126-130	259-266	6.76
Di Isopropyl Ketone	33.9	164-169	327-336	6.75
Ethyl Cellosolve	36.2	133-137	271-279	7.77
Odorless Mineral Spirits	38.6	150-201	302-394	6.52
Ethyl Lactate	40.0	119-176	246-349	8.59
Sec. Hexyl Alcohol	41.7	157	315	6.97
Solvsol 30/40	43.2	142-199	288-390	7.06
Pentanol	45.0	112-140	234-284	6.76
Hi-Solvency Mineral Spirits	46.7	152-200	306-392	6.79
No. 380 Mineral Spirits	47.0	151-196	304-385	6.57
No. 10 Mineral Spirits	55.0	154-196	309-385	6.49
Distilled Water	60.0	100	212	8.32
Apco No. 125	60.5	162-200	324-392	6.52
Cellosolve Acetate	65.0	145-166	293-331	8.15
Sec. Butyl Lactate	73.0	172	342	8.14
Sec. Hexyl Acetate	76.5	129-158	264-316	7.19
Butyl Cellosolve	88.5	163-172	325-342	7.58
Dipentene	89.2	149-215	300-419	7.10
No. 140 Thinner	91.0	185-210	365-410	6.62
Octyl Acetate	152.5	195-203	383-397	7.20
Isobutyl Lactate	156.5	168-200	334-392	8.15
Hexalin	177.5	159-162	318-324	7.89
Solvsol 40/50	270.5	191-248	376-478	7.42
Methyl Hexalin	276.5	170-190	338-374	7.66
Butyl Lactate	339.0	185-195	365-383	8.14
Excellene	384.0	162-260	324-500	6.55
Special Heavy Naphtha	403.0	202-242	396-468	6.73
Dispersol	425.0	193-242	379-468	6.59
No. 50 Kerosene	626.7	178-256	352-493	6.76
Triethylene Glycol	Over 5200.0	276-310	529-590	9.30
Dibutyl Phthalate	Over 5200.0	195-200	383-392	8.73

Dry Time Relation: Below 5 — Fast
5-15 — Medium
15-75 — Slow
75 over — Nil

CONVERSION FACTORS

to convert	into	multiply by	to convert	into	multiply by
ampere-hours	coulombs	3600.0	gallons	liters	3.785
angstrom units	inches	3.937×10^{-9}	gallons/minute	cubic feet/hour	8.0208
angstrom units	microns	1×10^{-4}	gallons/minute	cubic feet/second	0.00223
atmospheres	centimeters of mercury	76.0	gallons of water	pounds of water	8.3453
atmospheres	feet of water	33.96	grains	grams	0.0648
atmospheres	inches of mercury	29.92	grams	grains	15.4324
atmospheres	inches of water	407.52	grams	ounces (troy)	3.215×10^{-2}
atmospheres	millimeters of water	10,340	grams	ounces (avoirdupois)	3.527×10^{-2}
atmospheres	millimeters of mercury	760	grams	pounds	2.205×10^{-3}
atmospheres	pascals	101513	grams/cubic foot	milligrams/cubic meter	2288.1
atmospheres	pounds/square foot	2116.3	horsepower	BTU/minute	42.44
atmospheres	pounds/square inch	14.696	horsepower	foot-pounds/second	550
BTU	foot pounds	778	horsepower	foot-pounds/minute	33,000
BTU	horsepower-hours	3.931×10^{-4}	horsepower	kilowatts	0.7457
BTU	joules	1.055×10^3	horsepower	watts	745.7
BTU	kilowatt-hours	2.928×10^{-4}	inches	centimeters	2.540
BTU/hour	watts	0.2931	inches	miles	1.578×10^{-5}
calories	BTU	3.9685×10^{-3}	inches of mercury	atmospheres	0.03342
centimeters	feet	3.281×10^{-2}	inches of mercury	feet of water	1.134
centimeters	inches	0.3937	inches of mercury	inches of water	13.61
centimeters	kilometers	1×10^{-5}	inches of mercury	millimeters of mercury	25.4
centimeters	meters	1×10^{-2}	inches of mercury	millimeters of water	345.6
centimeters	millimeters	10.0	inches of mercury	pascals	3390
coulombs	faradays	1.036×10^{-5}	inches of mercury	pounds/square foot	76.70
cubic feet	cubic meters	0.02832	inches of mercury	pounds/square inch	0.491
cubic feet	gallons	7.48	inches of water	atmospheres	0.002456
cubic feet	liters	28.3162	inches of water	feet of water	0.08333
cubic inches	cubic centimeters	16.3872	inches of water	inches of mercury	0.0735
cubic inches	liters	0.0164	inches of water	millimeters of water	25.4
cubic meters	cubic feet	35.3145	inches of water	millimeters of mercury	1.876
cubic centimeters	cubic inches	0.06102	inches of water	pascals	249.1
cubic centimeters	pints (U.S. liquid)	2.113×10^{-3}	inches of water	pounds/square inch	0.0361
cubic feet/minute	pounds water/minute	62.43	inches of water	pounds/square foot	5.196
cubic feet/minute	cubic meters/second	4.719×10^{-4}	joules	BTU	9.480×10^{-4}
cubic feet/second	gallons/minute	448.83	joules	ergs	$1 \cdot 10^7$
days	seconds	86,400.0	kilograms	pounds	2.205
degrees (angle)	radians	1.745×10^{-2}	kilograms	slugs	0.068522
degrees/second	revolutions/minute	0.1667	kilometers	feet	3281.0
dynes	newtons	1×10^{-5}	kilometers	miles	0.6214
ergs	BTU	9.480×10^{-11}	kilometers	meters	1000.0
ergs	kilowatt-hours	2.778×10^{-14}	kilometers/hour	knots	0.5396
ergs	foot-pounds	7.3670×10^{-8}	kilopascals	pounds/square inches	0.145
faradays/second	amperes	96,500	kilowatt-hours	BTU	3413.0
feet	centimeters	30.48	kilowatts	foot-pounds/second	737.6
feet	miles (nautical)	1.645×10^{-4}	kilowatts	horsepower	1.341
feet	meters	0.3048	knots	feet/hour	6080.0
feet	miles (statute)	1.894×10^{-4}	knots	statute miles/hour	1.151
feet/minute	centimeters/second	0.5080	knots	nautical miles/hour	1.0
feet/minute	meters/second	0.00508	light years	miles	5.9×10^{12}
feet/second	miles/hour	0.6818	liters	cubic inches	61.02
feet/second	knots	0.5921	liters	cubic centimeters	1000.0
foot-pounds	BTU	1.286×10^{-3}	liters	gallons (U.S. liquid)	0.2642
foot-pounds	kilowatt-hours	3.766×10^{-7}	liters	milliliters	1000.0
gallons	cubic feet	0.1337	liters	pints (U.S. liquid)	2.113

CONVERSION FACTORS

to convert	into	multiply by	to convert	into	multiply by
meters	centimeters	100.00	pascals	newtons/square miles	1.0
meters	feet	3.281	pascals	pounds/square foot	0.2089
meters	kilometers	1×10^{-3}	pascals	pounds/square inch	1.696×10^{-4}
meters	miles (statute)	6.214×10^{-4}	pints (liquid)	gallons	0.125
meters	miles (nautical)	5.396×10^{-4}	pints (liquid)	cubic centimeters	473.2
meters	millimeters	1000.0	pints (liquid)	cubic inches	28.87
microns	meters	1×10^{-6}	pints (liquid)	quarts (liquid)	0.5
miles (nautical)	miles (statute)	1.516	pounds	ounces	16.0
miles (nautical)	kilometers	1.853	pounds	ounces (troy)	14.5833
miles (nautical)	feet	6080.27	pounds	pounds (troy)	1.21528
miles (statute)	kilometers	1.609	pounds	kilograms	0.4536
miles (statute)	feet	5280.0	quarts (dry)	cubic inches	67.20
miles (statute)	miles (nautical)	0.8684	quarts (liquid)	gallons	0.25
miles/hour	feet/minute	88.0	quarts (liquid)	cubic inches	57.75
miles/hour	feet/second	1.467	quarts (liquid)	liters	0.9463
milligram/liter	parts/million	1.0	radians	minutes	3438.0
milligrams/cubic meter	grains/cubic foot	4.37×10^{-4}	radians	degrees	57.30
milliliters	liters	1×10^{-3}	revolutions	degrees	360.0
millimeters	inches	3.937×10^{-2}	revolutions/minute	degrees/second	6.0
millimeters of mercury	atmospheres	0.001316	seconds	minutes	1.667×10^{-2}
millimeters of mercury	feet of water	0.4464	slugs	pounds	32.17
millimeters of mercury	inches of mercury	0.3937	square feet	square meters	
millimeters of mercury	inches of water	0.5357	tons (long)	pounds	2240.0
millimeters of mercury	millimeters of water	13.61	tons (long)	tons (short)	1.120
millimeters of mercury	pascals	133	tons (long)	kilograms	1016.0
millimeters of mercury	pounds/square foot	2.789	tons (short)	tons (long)	0.89287
millimeters of mercury	pounds/square inch	0.01934	tons (short)	kilograms	907.18
newtons	dynes	1×10^5	tons (short)	pounds	2000.0
newtons	pounds	0.2248	watts	BTU/hour	3.4129
ounces	pounds	6.25×10^{-2}	watts	horsepower	1.341×10^{-3}
ounces (troy)	ounces (avoirdupois)	1.09714	yards	miles (nautical)	4.934×10^{-4}
pascals	atmospheres	9.872×10^{-4}	yards	meters	0.9144
pascals	inches of water	0.00401	yards	miles (statute)	5.682×10^{-4}
pascals	millimeters of water	0.102			

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