CHAPTER 42

DESIGN OF VENTILATION SYSTEMS

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INTRODUCTION

Not too many years ago the design of ventilation systems was an art, but with the accumulation of knowledge today it has "come of age" and may be classified as an engineering science. Various "rule-of-thumb" methods have been replaced with new rules based on theory, supported by experimentation, and validated by experience. When properly designed and installed, a good ventilation system often can add to productivity and the general well-being of workers. In some cases, a system with adequate collection devices can recover enough valuable materials to pay for itself in a reasonably short time.

It should be emphasized that in industrial hygiene the primary purpose for designing a ventilation system is to protect the health and well-being of workers.

The design of any ventilation system should include consideration of materials that will withstand normal mechanical abuse inherent to the environment in which it is operated. Frequently, extra design capacity in fans, control equipment and motors which allows for future expansion of the system at minimum cost is desirable.

There are generally two types of ventilation systems: (1) general, and (2) local exhaust. For the purpose of this chapter make-up air is considered part of local exhaust ventilation.

GENERAL VENTILATION

General ventilation refers to the commonly encountered process of flushing a working environment with a constant supply of fresh air. General ventilation differs from local ventilation in that it is a dilution process rather than strictly an exhausting process. General ventilation may be accomplished by natural infiltration of air, or with the aid of some type of air moving device.

Office Buildings

For office buildings to maintain comfortable work conditions, the proper environmental atmosphere must be achieved. This usually occurs as a result of a properly designed general ventilation system. Whereas the design criteria for local exhaust systems are relatively clear-cut, this is not necessarily the case for general systems. Here design is based on human comfort requirements, noise considerations and ease of distribution of fresh air.

Worker Comfort. The comfort of office workers is subject to the following environmental conditions:

1. Air temperature;
2. Humidity;
3. Radiant heat;
4. Concentrations of tobacco smoke;
5. Concentration of body odors; and
6. Air movement.

Before designing any general ventilation system, the aforementioned conditions must be measured or estimated. Then a fresh air flowrate can be calculated which will reduce undesirable air qualities to tolerable levels.

Comfort Zone. Various indices have been devised by investigators to describe a "comfortable" environment. A comfort chart has been developed by the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. which uses wet- and dry-bulb air temperatures as parameters (See Figure 42-1). The comfort zone is that combination of environmental conditions which is thermally neutral to the human body. Specifically, the comfortable temperature range for office workers is 68 — 74°F during the winter months and 75-82°F in summer with moderate humidities. A summary of minimum ventilation requirements for various conditions is included in Table 42-1.

Odor Control. Air exchange rates are dictated in part by the concentration of body and tobacco odors in a room. These concentrations are affected by air supply, space allowed per person, odor absorbing capacity of the air conditioning process, temperature and relative humidity.

Conditioning of Air. Comfort conditions can be met by the proper selection of air conditioning equipment. Such equipment can maintain proper conditions of temperature, humidity, and odor levels when comfort indices have been determined. The subject of air heating and conditioning is beyond the scope of this chapter; however, the reader is urged to consult the references listed at the end of this chapter, specifically 5, 6 and 7, for more information.

To summarize briefly, the criteria for design calculations for general office ventilation systems include the determination of worker comfort, zone parameters (temperature, humidity), tolerable odor levels, space allowed per person in room and the odor-absorbing capacity of the air conditioning unit to be installed. In addition, it should be stressed that fan noise is an extremely important

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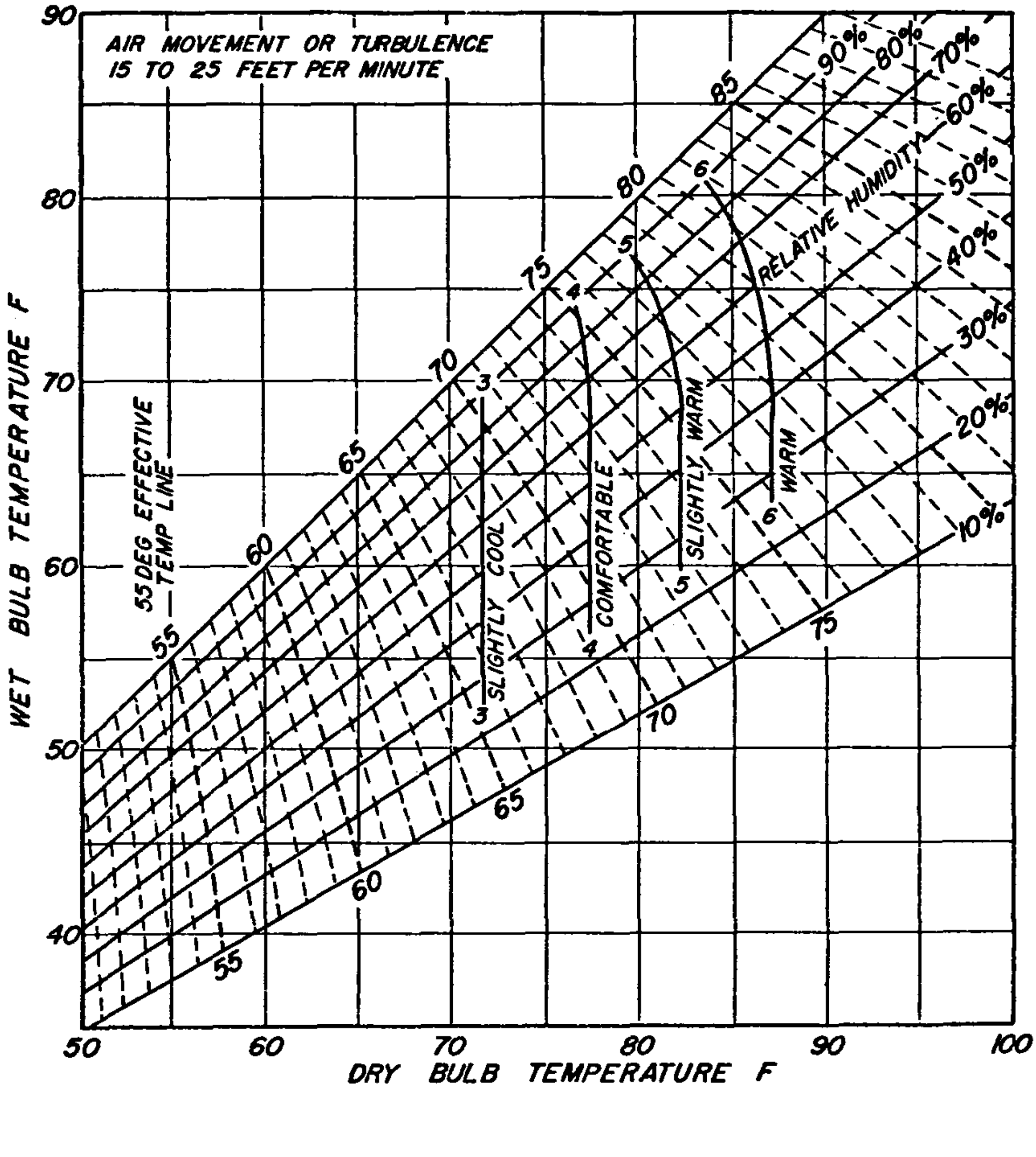


American Conference of Governmental Industrial Hygienists — Committee on Industrial Ventilation: Industrial Ventilation — A Manual of Recommended Practice, 12th Edition. Lansing, Michigan, 1972.

Figure 42-1. Comfort Chart for Still Air (from: Industrial Ventilation Manual). Notes: 1. Effective Temperature (dashed) lines indicate sensation of warmth immediately after entering conditioned space. 2. Solid lines 3, 4, 5, and 6 indicate sensations experienced after three hour occupancy. 3. Both sets of curves apply to people at rest and normally clothed.

TABLE 42-1.
Summary of Minimum Outdoor Air Requirements for Ventilation Under Various Conditions

Type of occupants	Air space per person (cubic feet)	Requirements based on primary* impressions (cfm per person)	Requirements based on impressions of occupants† (cfm per person)
Heating season with or without recirculation. Air not conditioned.			
Sedentary adults of average socio-economic status	100	25	23
	200	16	11
	300	12	....
	500	7	>5
Laborers	200	23	....
Grade school children of average class	100	29	....
	200	21	15
	300	17	....
	500	11	....
Grade school children (low income)	200	38	....
Grade school children (medium income)	200	18	....
Grade school children (high income)	100	22	....
Heating season. Air humidified by means of centrifugal humidifier. Total air circulation 30 cfm per person.			
Sedentary adults	200	12	....
Summer season. Air cooled and dehumidified by means of a spray dehumidifier. Total air circulation 30 cfm per person.			
Sedentary adults	200	>4‡	6‡

*Impressions upon entering room from relatively clean air at threshold odor intensity.
†Corresponding to an air quality of fair to good.
‡Values provisionally restricted to the conditions of the tests.
"The Industrial Environment and Its Control," J. M. DallaValle, p. 105, Pitman Publishing Corporation, New York, N.Y., 1948.

design consideration of office ventilation systems. Usually a forward-curved blade fan is chosen over the more efficient backward-curved blade fan simply because of noise considerations. More detailed information on fans and noise problems are given in a later section of this chapter and in several references[1,2,3,4,5].

Industrial Buildings

General ventilation systems employed in industrial buildings are of two types — natural and mechanical.

Natural Ventilation. The two forces which are responsible for natural ventilation are wind and thermal head. Realizing this, in the past, architects devised sawtooth and monitor type roofs to achieve maximum ventilation and lighting, although with greater use of mechanical ventilation, these building designs are slowly becoming outdated. More recently, windows such as the double hung sash and center-pin-swing-hinge type have been utilized to achieve maximum natural air flow. In buildings such as warehouses, powerhouses and pumprooms, where few people are employed, wall or roof openings generally provide enough fresh air for good ventilation. Mushroom, gooseneck, or louvered penthouse roof ventilators are reasonably effective supply ports regardless of wind direction.

Mechanical Ventilation. Although general ventilation by natural means is the most economical, it is limited in usefulness. Ventilation by mechanical devices (i.e., fans) is seldom limited, and, when used in conjunction with ductwork, air can be distributed to all parts of the building. Equipment and design considerations for general ventilation systems of this type are discussed in the following section and later in this chapter.

Design Considerations. General ventilation systems are used in industry in conjunction with local exhaust ventilation systems to achieve maximum effectiveness (for additional discussion see Chapter 39). Besides providing for a comfortable atmosphere in which to work, general ventilation systems may be employed to control vapors within acceptable limits from organic liquids of low-level toxicity. This is successfully accomplished by dilution. Table 42-2 lists the dilution air volumes for several commonly used solvents. Threshold Limit Values (TLV) represent guides to allowable toxic material concentrations in air. When the maximum allowable concentrations of the contaminant are known and the generation rate has been estimated, the quantity of dilution air re-

TABLE 42-2.
Dilution Air Volumes for Vapors
(based on 1971 TLV Values which are shown as ppm in parentheses)

Liquid	Cu. ft. of air (STP) required for dilution to TLV*	
	Per Pint Evaporation	Per Pound Evaporation
Acetone (1000)	5,500	6,650
n-Amyl acetate (100)	27,200	29,800
Isoamyl alcohol (100)	37,200	43,900
Benzol (25)	Not Recommended	
n-Butanol (butyl alcohol) (100)	44,000	52,200
n-Butyl acetate (150)	20,400	22,200
Butyl cellosolve (50)	61,600	65,600
Carbon disulfide (20)	Not Recommended	
Carbon tetrachloride (10)	Not Recommended	
Cellosolve (2-Ethoxyethanol) (200)**	20,800	21,500
Cellosolve acetate (2-ethoxyethyl-acetate) (100)	29,700	29,300
Chloroform (50)**	Not Recommended	
1-2 Dichloroethane (50)** (ethylene dichloride)	Not Recommended	
1-2 Dichloroethylene (200)	26,900	20,000
Dioxane (100)	47,300	43,900
Ethyl acetate (400)	10,300	11,000
Ethyl alcohol (1000)	6,900	8,400
Ethyl ether (400)	9,630	13,100
Gasoline	Requires special consideration	
Methyl acetate (200)	25,000	26,100
Methyl alcohol (200)	49,100	60,500
Methyl butyl ketone (100)	33,500	38,700
Methyl cellosolve (25)	Not Recommended	
Methyl cellosolve acetate (25)	Not Recommended	
Methyl ethyl ketone (200)	22,500	26,900
Methyl isobutyl ketone (100)	32,300	38,700
Methyl propyl ketone (200)	19,000	22,400
Naptha (coal tar) (100)	30,000-38,000	40,000-50,000
Naptha (petroleum) (500)	Requires special consideration	
Nitrobenzene (1)	Not Recommended	
n-Propyl acetate (200)	17,500	18,900
Isopropyl alcohol (400)	13,200	16,100
Isopropyl ether (500)**	5,700	7,570
Stoddard solvent (200)	15,000-17,500	20,000-25,000
1,1,2,2-Tetrachloroethane (5)	Not Recommended	
Tetrachloroethylene (100)	39,600	23,400
Toluol (Toluene) (200)**	19,000	21,000
Trichloroethylene (100)	45,000	29,400
Xylol (xylene) (100)	33,000	36,400

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

**See Notice of Intended Changes in TLV List for 1971.

The K value is merely a safety factor between 3 and 10 (usually 6) which is multiplied by the dilution air quantities to assure air concentrations well below the TLV.

"Industrial Ventilation — A Manual of Recommended Practice" 12th Edition, American Conference of Governmental Industrial Hygienists, Committee on Industrial Ventilation, Lansing, Michigan, 1972.

quired may be calculated using the equations given in Chapter 39. Although this method of ventilation may be used effectively to deal with low toxicity gases and vapors, it is not advisable to treat particulate contaminants or toxic vapors or gases with general ventilation. Whenever possible, local exhaust systems should be used to minimize the total amount of hazardous material released.

LOCAL EXHAUST VENTILATION

Local exhaust systems are primarily concerned with contaminant control at the point of emission and/or dispersion (for additional information see Chapter 41). As mentioned earlier, local exhaust systems usually complement (rather than replace) general ventilation systems. The components of all local exhaust systems are similar, but the total design of each system is unique. The components include a hood, ductwork, an air moving device, an air cleaning device, and special fittings. The processes to which these components are applied are numerous and varied. Therefore the size, shape and material of construction of each component will vary with the contaminated air being handled.

Hood Design

The design of any local exhaust system begins with the proper selection of an exhaust hood. Over the years, many types of hood designs have evolved, with only one purpose in mind — to confine or capture the contaminant with a minimum rate of air flow into the hood. In most instances, the more complete the hood enclosure, the more economical and effective the installation will be.

Exhaust hoods are designed to work in one of two ways: (1) they can induce an air movement which draws the contaminant into the hood or (2) they can enclose the contaminant source and induce an air movement which prevents the contaminant from escaping the enclosure. In either case, a certain air velocity in front of the hood is required for effective removal of contamination. This required air velocity in front of the hood must be determined before the exhaust system can be designed.

Unfortunately, the determination of required air velocity is not subject to direct and exact evaluation. In the past, three methods have been used to approximate a required velocity: (1) evaluation of and comparison with existing operations, (2) experimental tests, and (3) calculations based on theoretical air requirements. These methods and practical experience enable the design engineer to estimate a required air velocity in most cases. Tables 41-1 and 42-3 are helpful for estimating required control velocities.

Hood types. Exhaust hoods can be categorized as enclosures, receiving hoods, or exterior hoods.

Enclosures, such as paint-spray booths, surround the point of emission either completely or partially. They are the most effective hoods to use, but they are seldom utilized for any manual operations where workers must also be enclosed.

Receiving hoods are used on processes where contaminants may be conveniently "thrown" into the hood. For example, inertial forces carry air contaminants from a grinding wheel into a hood located in the pathway of the particles. If the hood cannot be located directly in the path of the escaping particles, baffles or shields may be placed across the line of throw of the particles to destroy their kinetic energy. Then, lower air velocities will suffice to capture and carry them into the hood.

Unlike enclosures and receiving hoods, exterior hoods must capture air contaminants that are generated from a point *outside* the hood. Exterior hoods require the most air to control a given process, are most sensitive to external conditions, and thus are the most difficult to design.

Hood Design Considerations. Before designing a hood, several principles should be considered. Some of the most important ones are listed below:

a. An attempt should be made to minimize or eliminate all air motion in the area of the contaminant source. This will reduce the amount of air needed to be exhausted and subsequently reduce system power and equipment requirements.
b. Air currents which necessarily exist should be utilized by the hood whenever possible.
c. The hood should enclose the process as much as possible without endangering workers' safety.
d. When enclosure is impractical, the hood should be located as close to the contaminant source as possible. The air velocity created by an exhaust hood varies inversely with the *square* of the distance for all but long, slot-type hoods.
e. The hood should be located so that the contaminant is removed *away* from the breathing zone of the worker.
f. The use of flanges and baffles should be considered. Flanges can increase hood effectiveness and may reduce air requirements by 25%.[1]
g. Use of a hood larger than required should be considered. Large hoods can reduce danger of "spills" by diluting them rapidly to safe levels. It has also been shown that small hoods require higher capture velocities to be as effective as large hoods.

Exhaust Duct Design

The design of an exhaust duct system is the second stage of a total ventilation system design. Initially, a rough duct layout should be prepared which shows branches, expansions, contractions, elbows, air moving and air cleaning devices. Using this as a basis, pressure drop calculations can be made and duct sizing can be estimated.

Transport Velocity. At this stage of design, the required exhaust rate for each hood has been determined. The problem now is to determine the minimum transport-velocity — i.e., the air velocity required to move the contaminant through the duct system. Information pertaining to transport velocities may be obtained by the following methods: (1) by reference to data published in the literature (See Tables 42-4 and 42-5), (2) by actual laboratory tests with the material to be conveyed, or (3) by theoretical considerations involving particle size, density and shape.

TABLE 42-3

Minimum Control Velocities and Exhaust Rates for Typical Specific Operations

Where both control velocity and exhaust rate are given, the air volume exhausted shall be based on the method which requires the larger volume.

Operation	Control Velocity, fpm	Control Velocity Basis	Exhaust Rate, cfm	Exhaust Rate Basis
Abrasive blasting				
Cabinets	500	Openings in enclosure	—	
Rooms	60-100	Downdraft in room	—	
Bagging				
Paper bags	100	Openings in enclosure	—	
Cloth bags	200	Openings in enclosure	—	
Pulverized sand	400	Point of origin	—	
Barrel filling	100	Point of origin	100	Per sq. ft. barrel top, semi-enclosure
Bin and hopper	150-200	Openings in enclosure	0.5	Per cu. ft. bin volume
Belt conveyors				
Transfer point				
Belt speed <200 fpm	150	Openings in enclosure	350	Per ft. belt width
>200 fpm	200	Openings in enclosure	500	Per ft. belt width
Belt wiper			200	Per ft. belt width
Bottle washing	150-250	Face of booth or enclosure openings	—	
Bucket elevators	—		100-200	Per sq. ft. casing cross section Tight casing required
Core sanding lathe	100	Point of origin	—	
Foundry screens				
Cylindrical	400	Openings in enclosure	100	Per sq. ft. circular cross section
Flat deck	150-200	Openings in enclosure	25-50	Per sq. ft. screen area
Foundry shakeout				
Enclosure	200	Openings in enclosure	200	Per sq. ft. grate area
Side draft	—		350-400	Cool castings per sq. ft. grate area
Downdraft			400-500	Hot castings (Per sq. ft. grate area)
			250	Cool castings (Per sq. ft. grate area)
			600	Hot castings (Per sq. ft. grate area)
Furnaces, melting				
Aluminum	150-200	Openings in enclosure	—	
Brass	200-250	Openings in enclosure	—	
Granite cutting				
Hand tool	200	Point of origin	—	
Surfacing machine	1500	Point of origin	—	
All tools	1500	Face of enclosing hood		
Grinding				
General	—	See applicable American National Standards	200	Per sq. ft. plan area of bench downdraft grille
Disc and portable	—		400	Per sq. ft. plan area of floor downdraft grille
Swing frame	150	Face of booth		
Kitchen range	100-150	Face of canopy	—	
Laboratory hood	100-150	Face of hood, door open. Less for "air supplied" hoods	—	
Metallizing				
Toxic material	200	Face of booth	Additional respiratory protection required	
Nontoxic	125	Face of booth	—	
Nontoxic	200	Point of origin	—	
Mixer	100-200	Openings in enclosure	—	

TABLE 42-3 Continued
Minimum Control Velocities and Exhaust Rates
for Typical Specific Operations

Where both control velocity and exhaust rate are given, the air volume exhausted shall be based on the method which requires the larger volume.

Operation	Control Velocity fpm	Control Velocity Basis	Exhaust Rate, cfm	Exhaust Rate Basis
Packaging machines	100-400	Openings in enclosure	25	Per sq. ft. plan area of enclosure
	50-150	Face of booth	—	
	75-150	Downdraft	—	
Paint spray	100-200	Face of booth	—	
Pharmaceutical coating pans	100-200	At opening of pan	—	
Quartz fusing	150-200	Face of booth	—	
Rubber calendar rolls	75-100	Openings in enclosure	—	
Silver soldering	100	Point of origin	—	
Steam kettles	150	Face of canopy	—	
Tanks				
Open surface	50-150	See applicable American National Standards		
Closed	150	Manhole or inspection opening		
Welding, arc	100-200	Point of origin	—	
	100	Face of booth	—	
Woodworking		See applicable American National Standards		

TABLE 42-4
Classification of Transport Velocities
for Dust Collection

Material	Minimum Transport Velocity, fpm
Very fine, light dusts	2000
Fine, dry dusts and powders	3000
Average industrial dusts	3500
Coarse dusts	4000-4500
Heavy or moist dust loading	4500 and up

The minimum transport velocity is not used for duct design; rather, a design velocity is estimated which includes a safety factor based on practical considerations. These include considerations for material buildup, duct damage, corrosion, duct leakage, etc. As shown in Tables 42-4 and 42-5, transport velocities for dust-laden air vary from 2000 fpm to 4500 fpm or higher.[8]

Balance Methods. After the preliminary duct layout has been made, the duct system pressure losses can be calculated. Two methods are used to "balance" the system — that is, adjust the duct design so that the total system will function properly. Each method has advantages and disadvantages as described below.

The first is known as the "Static Pressure Balance" method. Some texts[3] refer to this as "Air Balance without Blast Gate Adjustment" because it is a procedure for achieving desired air flow without the use of dampers or blast gates. At each junction of two air streams the static suction necessary to produce the required flow in one stream must match the static suction needed to produce the required flow in the other stream. Because there are no blast gates for workers to tamper with, this method is usually selected for use where highly toxic materials are to be controlled.

The other method is "Balance with Blast Gates." This type of system uses adjustable blast gates to balance the system and thus achieve the desired air flow at each hood. Calculations begin at the branch of greatest resistance. Pressure drops are calculated through the various sections of the main, on up to the fan. This design method is theoretically superior to the "Static Pressure Balance" method in that it is flexible enough to allow air volume changes without duct redesign. However, if blast gates are tampered with by unauthorized personnel, ducts may become plugged and the exhaust system rendered ineffectual.

Pressure Losses. Pressure losses in an exhaust duct system occur as a result of (1) hood entry, (2) special fittings, (3) duct friction, and (4) air cleaning devices. Various methods and charts are available to aid in estimating pressure losses from these sources.[1,3] Because most charts and reference tables are based on standard air (0.075 lb./cu. ft.) corrections for altitude, temperature, and density must be made if conditions vary greatly from standard (See Table 42-6). Design calculations are based on volumes increased by the reciprocal of the density factor. System pressure losses will decrease directly as d, the density factor.

TABLE 42-5

Examples of Transport Velocities

Material, Operation, or Industry	Minimum Transport Velocity, fpm	Material, Operation, or Industry	Minimum Transport Velocity, fpm
Abrasive blasting	3500-4000	Jute	
Aluminum dust, coarse	4000	Dust	2500-3000
Asbestos carding	3000	Lint	3000
Bakelite molding powder dust	2500	Dust shaker waste	3200
Barrel filling or dumping	3500-4000	Pickerstock	3000
Belt conveyors	3500	Lead dust	4000
Bins and hoppers	3500	with small chips	5000
Brass turnings	4000	Leather dust	3500
Bucket elevators	3500	Limestone dust	3500
Buffing and polishing		Lint	2000
Dry	3000-3500	Magnesium dust, coarse	4000
Sticky	3500-4500	Metal turnings	4000-5000
Cast iron boring dust	4000	Packaging, weighing, etc.	3000
Ceramics, general		Downdraft grille	3500
Glaze spraying	2500	Pharmaceutical coating pans	3000
Brushing	3500	Plastics dust (buffing)	3800
Fettling	3500	Plating	2000
Dry pan mixing	3500	Rubber dust	
Dry press	3500	Fine	2500
Sagger filling	3500	Coarse	4000
Clay dust	3500	Screens	
Coal (powdered) dust	4000	Cylindrical	3500
Cocoa dust	3000	Flat deck	3500
Cork (ground) dust	2500	Silica dust	3500-4500
Cotton dust	3000	Soap dust	3000
Crushers	3000 or higher	Soapstone dust	3500
Flour dust	2500	Soldering and tinning	2500
Foundry, general	3500	Spray painting	2000
Sand mixer	3500-4000	Starch dust	3000
Shakeout	3500-4000	Stone cutting and finishing	3500
Swing grinding booth exhaust	3000	Tobacco dust	3500
Tumbling mills	4000-5000	Woodworking	
Grain dust	2500-3000	Wood flour, light dry sawdust and shavings	2500
Grinding, general	3500-4500	Heavy shavings, damp sawdust	3500
Portable hand grinding	3500	Heavy wood chips, waste, green shavings	4000
		Hog waste	3000
		Wool	3000
		Zinc oxide fume	2000

Material in Tables 42-3, 42-4 & 42-5 is reproduced with permission from ANSI Z9.2 copyright 1971, by the American National Standards Institute, copies of which may be purchased from American National Standards Institute at 1430 Broadway, New York, N.Y. 10018.

Hood Entry Losses

A loss in pressure occurs when air enters a hood opening. This loss is indicated by the coefficient of entry for the hood, C_e. This coefficient represents the ratio of actual to theoretical flow; i.e., $C_e = 1.0$ for a theoretically "perfect" hood. Several examples of entry coefficients are shown in Figure 42-2.

The design equation used in determining the static suction at the hood throat is derived from the classical orifice theory. For standard air it becomes:

$$Q = 4005\, A\, C_e \sqrt{SP_h} \qquad (1)$$

where: Q = air flow rate, ft³/min.

A = area of opening, ft²

C_e = entry coefficient, dimensionless

SP_h = static suction at hood throat, in. w.g.

Static suction, SP_h, is related to hood entry loss according to the following equation:

$$SP_h = h_e + VP \qquad (2)$$

where: VP = velocity pressure in throat, in. w.g.

h_e = hood entry loss, in. w.g.

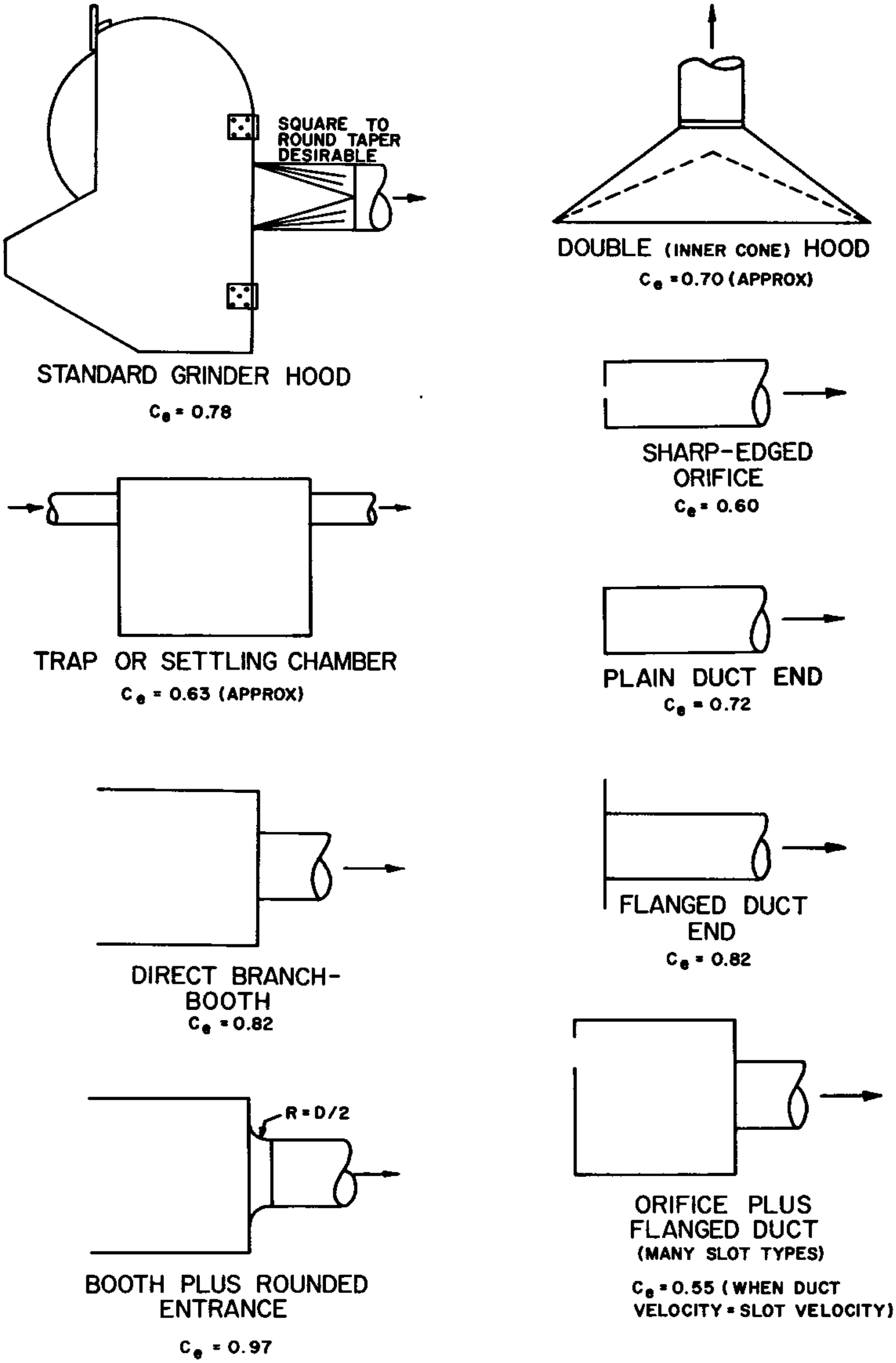
Velocity pressure (for standard air) may be calculated using the following equation:

$$VP = \left(\frac{V}{4005}\right)^2 \qquad (3)$$

where: VP = velocity pressure, in. w.g.

V = air velocity, fpm

Letting F be the fraction of the throat velocity



American Conference of Governmental Industrial Hygienists — Committee on Industrial Ventilation: Industrial Ventilation — A Manual of Recommended Practice, 12th Edition. Lansing, Michigan, 1972.

Figure 42-2. Hood Entry Loss Coefficients.

TABLE 42-6

Air Density Correction Factor, d

Altitude, ft.		—1000	Sea Level	1000	2000	3000	4000	5000	6000	7000	8000	9000	10,000
Barometer	"Hg	31.02	29.92	28.86	27.82	26.82	25.84	24.90	23.98	23.09	22.22	21.39	20.58
	"Wg	422.2	407.5	392.8	378.6	365.0	351.7	338.9	326.4	314.3	302.1	291.1	280.1
Air Temp.	−40	1.31	1.26	1.22	1.17	1.13	1.09	1.05	1.01	0.97	0.93	0.90	0.87
F	0	1.19	1.15	1.11	1.07	1.03	0.99	0.95	0.91	0.89	0.85	0.82	0.79
	40	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	1.04	1.00	0.96	0.93	0.89	0.86	0.83	0.80	0.77	0.74	0.71	0.69
	100	0.98	0.95	0.92	0.88	0.86	0.81	0.78	0.75	0.73	0.70	0.68	0.65
	150	0.90	0.87	0.84	0.81	0.78	0.75	0.72	0.69	0.67	0.65	0.62	0.60
	200	0.83	0.80	0.77	0.74	0.71	0.69	0.66	0.64	0.62	0.60	0.57	0.55
	250	0.77	0.75	0.72	0.70	0.67	0.64	0.62	0.60	0.58	0.56	0.58	0.51
	300	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	0.68	0.65	0.62	0.60	0.58	0.56	0.54	0.52	0.51	0.49	0.47	0.45
	400	0.64	0.62	0.60	0.57	0.55	0.53	0.51	0.49	0.48	0.46	0.44	0.42
	450	0.60	0.58	0.56	0.54	0.52	0.50	0.48	0.46	0.45	0.43	0.42	0.40
	500	0.57	0.55	0.53	0.51	0.49	0.47	0.45	0.44	0.43	0.41	0.39	0.38
	550	0.54	0.53	0.51	0.49	0.47	0.45	0.44	0.42	0.41	0.39	0.38	0.36
	600	0.52	0.50	0.48	0.46	0.45	0.43	0.41	0.40	0.39	0.37	0.35	0.34
	700	0.47	0.46	0.44	0.43	0.41	0.39	0.38	0.37	0.35	0.34	0.33	0.32
	800	0.44	0.42	0.40	0.39	0.37	0.36	0.35	0.33	0.32	0.31	0.30	0.29
	900	0.40	0.39	0.37	0.36	0.35	0.33	0.32	0.31	0.30	0.29	0.28	0.27
	1000	0.37	0.36	0.35	0.33	0.32	0.31	0.30	0.29	0.28	0.27	0.26	0.25

Standard Air Density, Sea Level, 70 F = 0.075 lb./ft.^{3}

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pressure loss in entry, and combining equations,

$$h_e = (F_h)\ (VP) \qquad (4)$$

Whenever a hood is made combining basic shapes, Equation 4 applies only to the parts and not to the hood as a whole.

Losses from Special Fittings

Pressure is lost when air travels through the various fittings in an exhaust system. Elbows, branch entries, enlargements and contractions are the main fittings to be considered. Pressure loss across these fittings is conveniently expressed as a fraction of the velocity pressure, VP. Tables giving pressure regain and loss values (fractions) for expansions and contractions are included, see Tables 42-7 and 42-8.[1,3]

Resistance of elbows and branch entries may also be expressed in terms of equivalent feet of straight duct (of the same diameter) that will produce the same pressure loss as the fitting. An example table (Table 42-9) is included.[1]

Duct Friction Losses

Many graphs are available which give friction losses in straight ducts. However, most graphs are based on new, clean duct. The chart included here (see Figure 42-3) allows for a typical amount of roughness, and is more practical for use in general application. Four quantities are plotted on the chart. If any two are given, the other two can be read directly from the chart.

Additional Pressure Losses

In addition to the pressure losses mentioned above, the pressure drop across collection equipment (if used) must be known in order to insure proper operation. This can vary widely, but usually data are available from the manufacturer to minimize guess work. Where data are unavailable, comparisons with known values for similar equipment should be used. Dust collection equipment is covered more extensively in Chapter 43.

Design Suggestions. A few suggestions pertaining to duct design and location are listed below:

a. Duct mains should be arranged in such a way that smaller branches enter the main near the high-suction end — closer to the fan inlet.
b. Long runs of small diameter duct should be avoided.

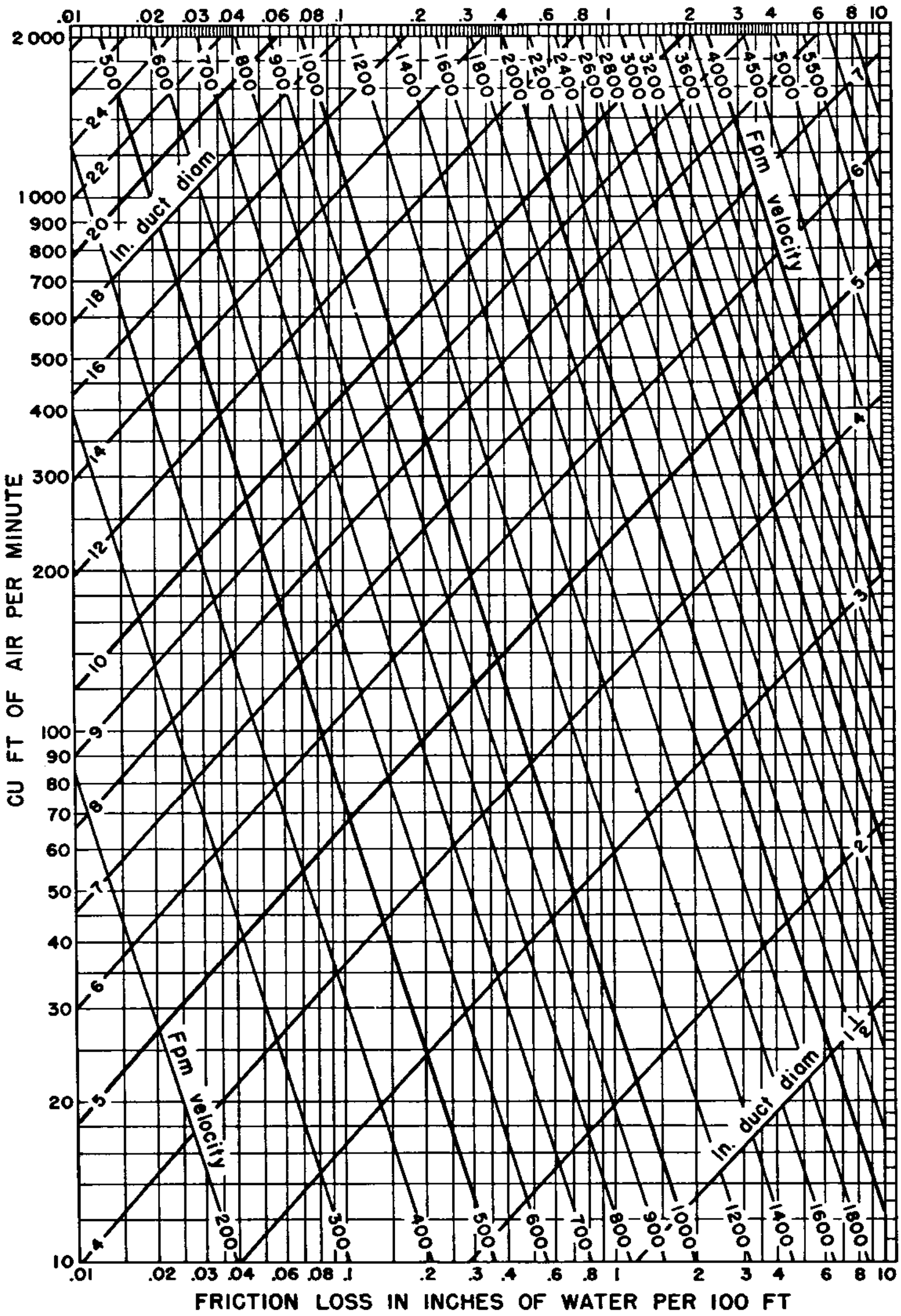
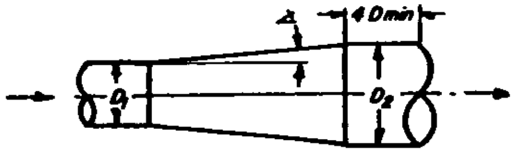
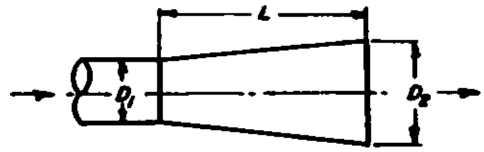


Figure 42-3. Friction of Air in Straight Ducts for Volumes of 10 to 2000 Cfm. (Based on Standard Air of 0.075 lb per cu ft density flowing through average, clean, round, galvanized metal ducts having approximately 40 joints per 100 ft.) Caution: Do not extrapolate below chart.

TABLE 42-7.
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.

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c. Extending an exhaust system to reach an isolated hood increases fan power consumption. To avoid this problem, it may be more economical to install a separate system for that hood.
d. If possible, locate the fan near the middle of an array of exhaust hoods rather than at one end.
e. If long rows of equipment are to be served, the main header duct should be located near the middle of the system to equalize runs of branch duct.
f. Ductwork should be located so that it is readily accessible for inspection, cleaning and repairs.
g. Ductwork should be out of the way of elevators, lift-trucks, cranes, etc., to avoid mechanical damage.
h. Duct cleanout areas should be provided.

AIR MOVING DEVICES

Various power-driven machines are capable of creating the required flow of air in an exhaust system. These machines are generally known as "air moving devices." Included under this general heading are fans, turbo-compressors, ejectors and positive displacement blowers.

As mentioned in Chapter 39, the air moving device manufacturer, to gain acceptance for his product, generally must earn membership in the Air Moving and Conditioning Association (AMCA). Membership is contingent upon subjecting his product to the AMCA test code for air moving devices. In addition, the manufacturer must furnish a prospective buyer of his product, certain data relative to the product and its applications. This information should include the following.

1. Classification according to static pressure limitations
2. Multirating tables — performance curves
3. Specifications — AMCA standards
4. Drive arrangement
5. Designations for rotation and discharge
6. Dimensional data
7. Materials and methods of construction
8. Sound level ratings
9. Accessories
10. Temperature limitations.

Fans are the most commonly used exhausters in the field of industrial ventilation. They are divided into two main classifications: axial flow or propeller type, and radial flow or centrifugal type. A summary of fan types is given in Chapter 39. A list of fan types appears below.

Axial Flow Fans
1. Propeller
2. Duct
3. Tube Axial
4. Vane Axial
5. Axial Centrifugal

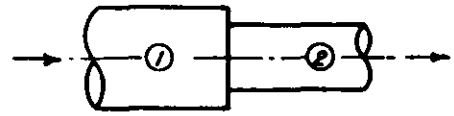
Centrifugal Fans
1. Radial Wheel
2. Forward-Curved Blade
3. Backward-Inclined Blade
4. Airfoil

TABLE 42-8.
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
0.5	0.32
0.6	0.26
0.7	0.20

A = duct area, sq ft

Note:
In calculating SP for expansion or contraction use algebraic signs:
VP is (+)
and usually
SP is (+) in discharge duct from fan
SP is (-) in inlet duct to fan

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Turbo-compressors and positive displacement blowers are used in systems having relatively low volume and high velocity and high static pressure. Turbo-compressors are typically used for industrial vacuum-cleaning systems where air must be transported through small diameter ducts at high velocities. Positive displacement blowers are used where a fixed quantity of air is required to be supplied or exhausted through an increasingly long duct, as in pneumatic conveying. Both exhausters can handle only clean, filtered air, due to the rigid design tolerances of their moving parts.

Ejectors are used in exhaust systems handling gases which are too hot, corrosive, abrasive, or sticky to be handled by a fan. Because ejectors are mechanically very inefficient, they require a much higher horsepower expenditure than equivalent fan installations.

Fan Laws and System Curves

Before selecting the proper fan, it is necessary to be familiar with the fan laws and system curves. These are listed in Table 42-10.

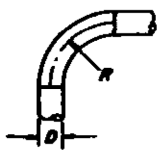
A fan or system curve shows graphically all possible combinations of volumetric flow and static pressure for a given system. Because the fan and system can each operate only at a point on their own curve, the combination can operate only where their curves intersect. See Figure 42-4) If the fan speed is changed, the operating point will move up toward the right (increased speed) or down toward the left (decreased speed) on the system curve. (See Figure 42-5)

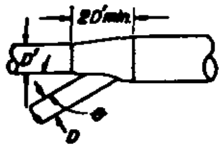
Fan Selection

A fan is chosen on the basis of its characteristics and the requirements of the system to which it will be applied. Each fan is characterized by five features: 1) volume of gas flow, 2) pressure at which this flow is produced, 3) speed of rotation, 4) power required, and 5) efficiency. These quantities are measured by the fan manufacturer with testing methods sponsored by the Air Moving and Conditioning Association or the American Society of Mechanical Engineers. Test results are plotted to provide the characteristic fan curves supplied by most fan manufacturers.

The designer chooses the fan he needs from multirating tables. Each different entry in the table has a unique performance characteristic —

TABLE 42-9.
Equivalent Resistance in Feet of Straight Pipe





Pipe D	90° Elbow * Centerline Radius			Angle of Entry	
	1.5 D	2.0 D	2.5 D	30°	45°
3"	5	3	3	2	3
4"	6	4	4	3	5
5"	9	6	5	4	6
6"	12	7	6	5	7
7"	13	9	7	6	9
8"	15	10	8	7	11
10"	20	14	11	9	14
12"	25	17	14	11	17
14"	30	21	17	13	21
16"	36	24	20	16	25
18"	41	28	23	18	28
20"	46	32	26	20	32
24"	57	40	32		
30"	74	51	41		
36"	93	64	52		
40"	105	72	59		
48"	130	89	73		

* For 60° elbows — 0.67 x loss for 90°
45° elbows — 0.5 x loss for 90°
30° elbows — 0.33 x loss for 90°

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that is, each entry describes a corresponding performance curve. Usually the fan required will have characteristics between two values given in the table. A linear interpolation is necessary to determine the right fan size, speed, horsepower, etc. needed to do the job. (See Table 42-11.)

Noise Vibration Control

The possibility of noise problems arising in exhaust systems should not be overlooked at any stage of the design process. If the system has been designed improperly, or if the wrong fan has been chosen, it is likely that a noise or vibration problem will arise. It is usually a simple matter to foresee such problems and prevent them from occurring.

The potential source of noise in any exhaust system is the fan. It is a pliable piece of equipment, is often forced to operate at high speeds, and is inherently prone to vibrate. Fan vibration is of two types: aerodynamic or mechanical.

Aerodynamic vibration varies distinctly with the volume of air drawn through the fan. This usually occurs when fans are operated at a point to the left of the peak of their static pressure curves. If the system pressure estimate is low, a smaller fan than actually needed may be specified, and forced to operate at a point other than the one for which it was selected. This type of vibration may also be caused by poor inlet connections to the fan. If possible, inlet boxes and inlet elbows should be avoided, or at least vaned to reduce fan inlet spin. (When air is forced to flow through a sharp turn as it enters the fan, it tends to load just part of the fan wheel and pulsation can occur.) Similarly, good outlet design will minimize pulsation.

Mechanical vibration cannot be foreseen by the design engineer except in cases where the structural support for the fan is inadequate. Frequently fans are supported on mounts having a natural vibration frequency near that of the fan. Under such conditions vibration is almost impossible to stop. The best support to use is an inertial mass such as a concrete pad supported by steel springs. The most common mount is the integral base — a structural steel platform built to fit

TABLE 42-10.
Fan Laws
(Q = CFM P = Pressure)

1. Variation in Fan Speed:
   Constant Air Density—Constant System
   (a) *Q*: Varies as fan speed.
   (b) *P*: Varies as square of fan speed.
   (c) Power: Varies as cube of fan speed.

2. Variation in Fan Size:
   Constant Tip Speed—Constant Air Density
   Constant Fan Proportions—Fixed Point of Rating
   (a) *Q*: Varies as square of wheel diameter.
   (b) *P*: Remains constant.
   (c) RPM: Varies inversely as wheel diameter.
   (d) Power: Varies as square of wheel diameter.

3. Variation in Fan Size:
   At Constant RPM—Constant Air Density
   Constant Fan Proportions—Fixed Point of Rating
   (a) *Q*: Varies as cube of wheel diameter.
   (b) *P*: Varies as square of wheel diameter.
   (c) Tip Speed: Varies as wheel diameter.
   (d) Power: Varies as fifth power of diameter.

4. Variation in Air Density:
   Constant Volume—Constant System
   Fixed Fan Size—Constant Fan Speed
   (a) *Q*: Constant.
   (b) *P*: Varies as density.
   (c) Power: Varies as density.

5. Variation in Air Density:
   Constant Pressure—Constant System
   Fixed Fan Size—Variable Fan Speed
   (a) *Q*: Varies inversely as square root of density.
   (b) *P*: Constant.
   (c) RPM: Varies inversely as square root of density.
   (d) Power: Varies inversely as square root of density.

6. Variation in Air Density:
   Constant Weight of Air—Constant System
   Fixed Fan Size—Variable Fan Speed
   (a) *Q*: Varies inversely as density.
   (b) *P*: Varies inversely as density.
   (c) RPM: Varies inversely as density.
   (d) Power: Varies inversely as square of density.

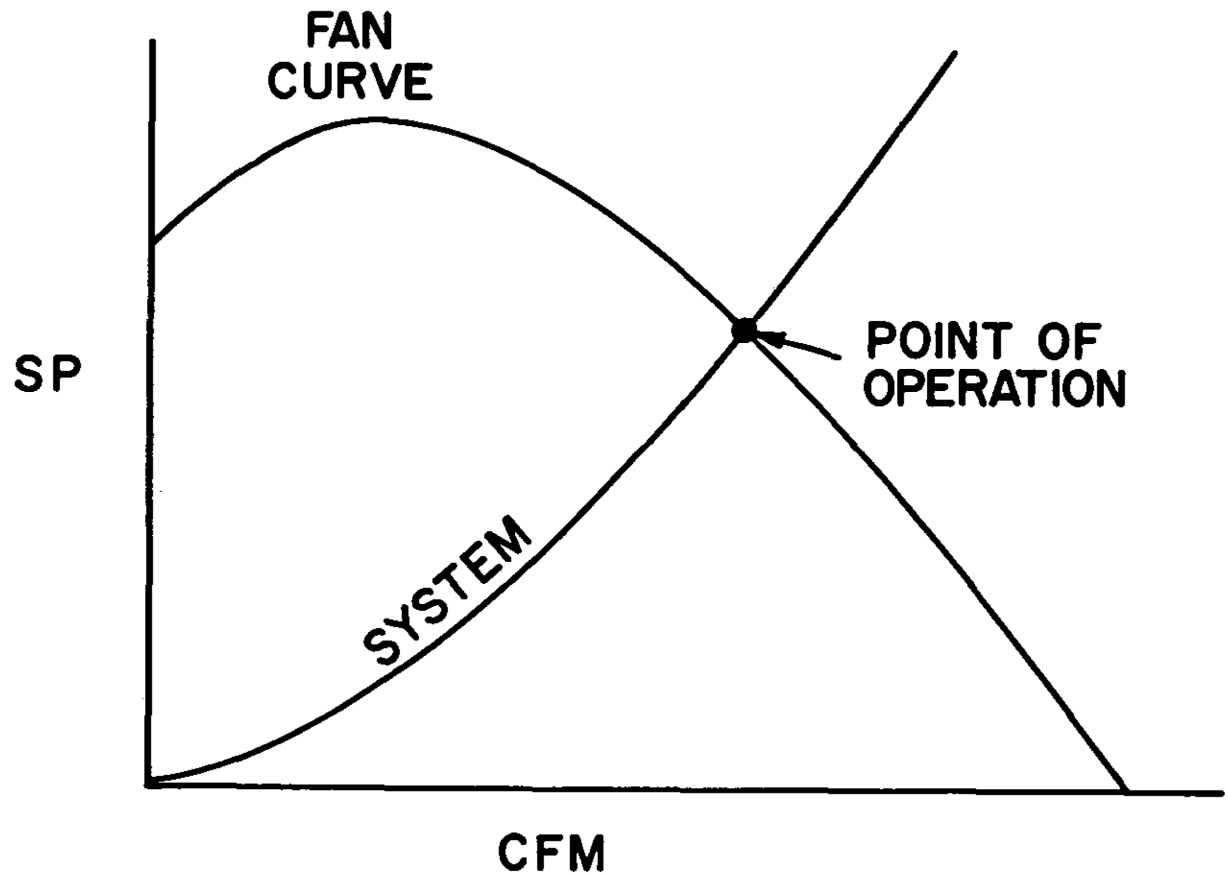

Figure 42-4. Fan and System Curves.

under the fan and motor, supported on steel spring or rubber-in-shear mounts. When the integral base is used, fan inlet and outlet vibration-elimination connections are required. In addition, a flexible conduit supplying power to the motor is essential.

The type of fan chosen for an exhaust system has a great influence on noise levels. For instance, axial flow fans are louder than centrifugal, and radial blade centrifugal fans are louder than other centrifugal types. The relatively new, airfoil-blade wheel centrifugal fans are the most quiet fans available today. This type is a modification of the backwardly-inclined-blade wheel.

Fans are now available with silencers to match the fan's aerodynamic characteristics. (Until recently fans and silencers were not designed to operate as an aerodynamic and acoustical unit.) Silencers impose additional resistance, the loss for which must be allowed in design calculations. In addition, streamlined duct transitions before and after the silencer must be considered. Silencers should always be installed on the clean air side of the system.

Finally, flexible duct-fan connections should be considered. Whenever fans are rigidly connected to ducts, the system can carry the fan's sound and vibration to remote areas. Use of duct lining can also reduce noise levels, but it is generally not used on medium to high velocity systems. Low velocity air conditioning systems usually employ linings to some degree.

MAKE-UP AIR

For a ventilation system to work effectively, the air exhausted from a room should be replaced in an amount at least equal to the exhausted volume. For best results, the supply air should exceed the exhaust volume; common practice is to allow 10% excess make-up air. The actual amount of make-up air needed depends upon the type of process involved, the amount of air exhausted, and the age and construction of the building.

The process to be exhausted may require low air exchange rates — six per hour or less. If exchange rates are small enough and the building is old and not tightly constructed, there may be no need for other make-up air. However, if the process calls for high air change rates, e.g., 60 per hour, or the building is modern and tightly sealed, then make-up air is definitely required. A negative pressure in a building is a common result of inadequate provision for make-up air.

Importance of Make-up Air. There are many reasons for providing make-up air; the most important ones are involved with the proper functioning of men and equipment. Make-up air should be provided for the following reasons:

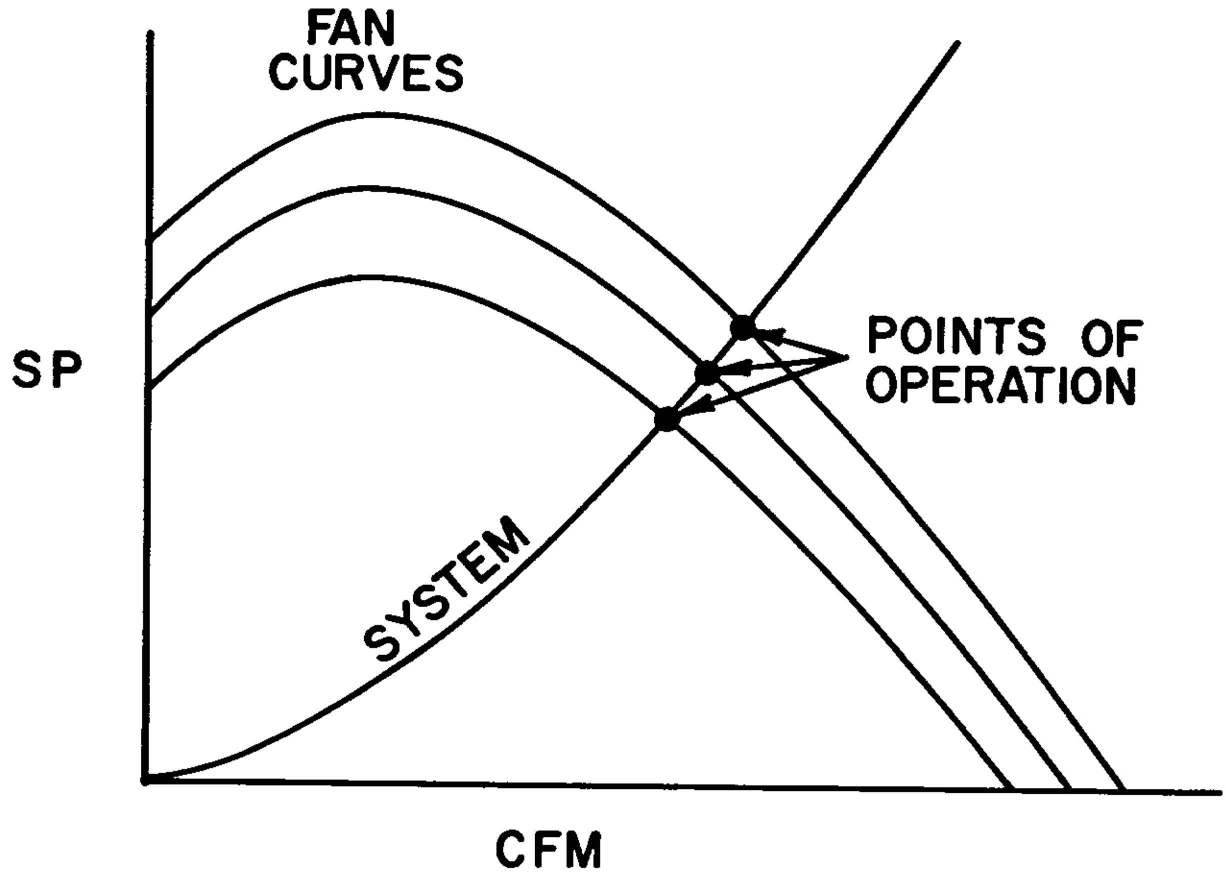

Figure 42-5. Fan and System Curves.

1. To insure that exhaust hoods operate properly. Exhaust volumes needed for proper operation of hoods may be drastically reduced under negative building pressures. If the exhaust system uses propeller fans, the flow may actually be reversed, entering instead of exhausting.
2. To insure proper operation of natural stacks. Some combustion exhaust stacks operate on natural drafts as low as 0.01 in. w.g. Under negative building pressures, flue gases, such as carbon monoxide, will not be able to leave via the stack. These gases may eventually infiltrate work areas and cause potential health hazards.
3. To eliminate high velocity cross drafts. A negative pressure as low as 0.01 to 0.02 in. w.g. may result in high velocity drafts through doors and windows. Drafts can: cause discomfort for workers; bring in or stir up dust; disrupt hood operation. Drafts also tend to cause uneven heating and adverse humidity conditions as well as poor temperature control.
4. To eliminate differential pressure on doors. A negative pressure of 0.05 to 0.10 in. w.g. is enough to make doors difficult to open. This situation is not only unpleasant for employees to deal with daily, but can also lead to injury from slamming doors.

The location and application of make-up air equipment also requires careful consideration. The following recommendations should be considered:

1. The fresh air intake should be located as far as possible from contaminant sources.
2. Exhaust stacks should be tall enough and properly located so that waste air will not re-enter the plant make-up air system.
3. Avoid the use of canopy type weather caps. By forcing air downward, weather caps on stack heads reduce the advantages gained by increasing stack height.
4. Where necessary, make-up air should be filtered to protect equipment, prevent plugging, and provide maximum heat exchange efficiency.

Equipment. The equipment used for providing and tempering make-up air is similar to or identical with that used for conventional heating and cooling systems. For heating make-up air, there are three basic equipment types: 1) heat exchangers using steam or hot water, 2) direct-fired heaters which burn gas or oil, and 3) open-flame heaters.

1. Steam heating coils are among the most common types of make-up air heaters. Moreover, if an ample steam supply is available, this type of heating may result in

TABLE 42-11.
Typical Fan Multirating Table

Volume, cfm	Outlet velocity, fpm	Velocity pressure, in. WC	1 in. SP		2 in. SP		3 in. SP		4 in. SP		5 in.SP		6 in SP		7 in. SP		8 in. SP		9 in. SP	
			rmp	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp
2,520	1,000	0.063	437	0.63	595	1.27	728	2.00	837	2.66										
3,120	1,200	0.090	459	0.85	610	1.55	735	2.30	842	3.10										
3,530	1,400	0.122	483	1.05	626	1.87	746	2.72	847	3.57	943	4.60								
4,030	1,600	0.160	513	1.33	642	2.18	759	3.17	858	4.12	950	5.21	1,030	6.29						
4,530	1,800	0.202	532	1.61	666	2.56	774	3.63	876	4.63	964	5.82	1,040	6.92	1,125	8.18				
5,040	2,000	0.250	572	2.00	688	2.97	797	4.12	890	5.30	976	6.50	1,052	7.75	1,134	8.96	1,208	10.15	1,270	11.6
5,540	2,200	0.302	603	2.36	712	3.43	816	4.66	910	5.93	999	7.38	1,068	8.60	1,145	9.93	1,210	11.18	1,279	12.8
6,040	2,400	0.360	637	2.79	746	3.99	840	5.33	926	6.73	1,017	8.17	1,088	9.50	1,160	10.88	1,230	12.25	1,288	13.9
6,550	2,600	0.422	670	3.27	762	4.62	866	6.05	954	7.83	1,032	9.08	1,095	10.50	1,171	11.98	1,245	13.50	1,298	15.1
7,060	2,800	0.489	708	3.81	795	5.32	892	6.72	963	8.78	1,050	9.97	1,125	11.60	1,188	13.06	1,257	14.70	1,310	16.4
7,560	3,000	0.560	746	4.42	833	6.05	920	7.70	993	9.32	1,068	11.00	1,142	12.75	1,210	14.28	1,277	15.98	1,328	17.8
8,060	3,200	0.638			866	6.96	943	8.71	1,020	10.40	1,097	12.10	1,168	14.02	1,228	15.50	1,292	17.36	1,340	19.1
8,560	3,400	0.721			900	7.93	964	9.80	1,053	11.48	1,120	13.30	1,188	15.35	1,248	16.93	1,310	19.00	1,360	20.9
9,070	3,600	0.808					1,010	11.00	1,078	12.70	1,148	14.65	1,213	16.70	1,270	18.42	1,335	20.75	1,380	22.6
9,570	3,800	0.900					1,038	12.25	1,108	14.15	1,170	14.90	1,240	18.80	1,292	19.46	1,355	22.35	1,405	24.4
10,080	4,000	0.998					1,162	13.60	1,138	15.40	1,200	17.35	1,270	19.70	1,320	21.70	1,380	23.15	1,430	26.4
10,580	4,200	1.100							1,168	16.90	1,230	19.05	1,283	21.50	1,348	23.50	1,405	26.10	1,450	28.4
11,100	4,400	1.210							1,198	18.58	1,258	20.55	1,322	22.50	1,373	25.40	1,430	27.95	1,478	30.6
11,600	4,600	1.310							1,232	20.30	1,290	22.50	1,355	23.80	1,405	27.40	1,450	30.15	1,500	32.9
12,100	4,800	1.450							1,270	21.00	1,321	24.40	1,383	25.65	1,432	29.60	1,482	32.40	1,528	35.2
12,600	5,000	1.570							1,301	24.20	1,355	26.40	1,410	28.80	1,462	31.80	1,513	34.60	1,555	37.8
15,120	6,000	2.230													1,622	45.90	1,670	49.00	1,702	51.5

Design of Local Exhaust Systems

"Air Pollution Engineering Manual", Public Health Service, U.S.D.H.E.W., Cincinnati, Ohio 1967, data from New York Blower Company, 1948.

the lowest fuel cost. The major drawback to steam coils is their potential to freeze and burst when the outside air is below freezing.

2. Direct fired heaters may be used where safety regulations permit (i.e., where there are no fire or explosion hazards). Here natural gas or liquified petroleum gas is burned directly in the air stream. Direct fired heating is used extensively for tempering make-up air.
3. Open flame, or indirect-fired heaters provide a heat exchange surface between the combustion chamber and the air being heated. The gaseous products of combustion are sent out through a flue. The drawback to this heating system is that condensation occurs on the heat transfer surface on every startup when using cold, outside air.

Finally, when natural infiltration can effectively provide all the required make-up air, infra-red unit heaters can be used.

Heating costs can be minimized if good engineering judgment is used in the make-up air supply design. The following recommendations should be considered:

1. Make-up air should be mixed with warmer building air before it reaches the work zone.
2. If possible, air should be delivered directly to the work zone. Make-up air should be introduced in the plant below the 8-10 foot level. In this way, the workers are constantly exposed to fresh air, and better circulation of air is achieved.
3. Sometimes it is possible to design a make-up air system that serves a dual purpose. Supply air may be used for spot cooling during warm weather, or in winter, waste heat can be recovered by cooling process equipment, motors, generators, etc. with this air.
4. Internal waste heat from a building can be recovered by using recirculated air to temper make-up air.

Supply Duct Design. The principles and methods involved in designing the supply duct are the same as explained earlier for the "Balance with Blast Gate" method. The difference is that the major portion of the ductwork is on the pressure side of the fan instead of the exhaust side. Also, only clean air will be handled by this ductwork.

Design velocities are based solely on economical factors; minimum transport velocities are not critical here. Velocities in the range of 2000 fpm are commonly used as they are most feasible.

Supply air systems are made up of rectangular ducts and branch takeoffs to save space. Light gauge construction materials are used with mechanical joints because leakage is of little consequence.

EXAMPLE PROBLEM

To assist the reader in better comprehension of this chapter, an example is presented herewith illustrating the various considerations a design engineer must give to a specific problem. This example is presented purely for illustrative purposes — no consideration has been given to the possibility that interfering machinery, trusses, etc. may alter the final design.

The problem: Design an exhaust system to control particulate residue from a 16-inch industrial disc sander. The system, shown in Figure 42-6, includes a disc sander, ductwork, fabric dust collector, and fan.

The following assumptions are derived from information presented earlier in this chapter.

1. Q=440 cfm. The air volume required to properly exhaust the disc sander.
2. v=3500 fpm. The transport velocity required for this system.
3. Hood losses=1.0 slot VP+0.25 duct VP

In addition, assume that the pressure loss across the dust collector is 2 inches of water.

It is required to design an appropriate local exhaust system for the sander and select an appropriate fan.

The first step in designing the system is to develop a systematic approach to the problem. Table 42-12, taken from the "Industrial Ventilation Manual," assists in the orderly design of a ventilation system. At the top of this table are columns numbered 1-19. The required information is inserted into the various columns as shown. An explanation of how the various data shown in the table were obtained is presented in the section of this chapter entitled "Explanation of Answer Chart."

To assist the reader in following the various steps necessary to solve this problem, please refer to column numbers and to the diagram.

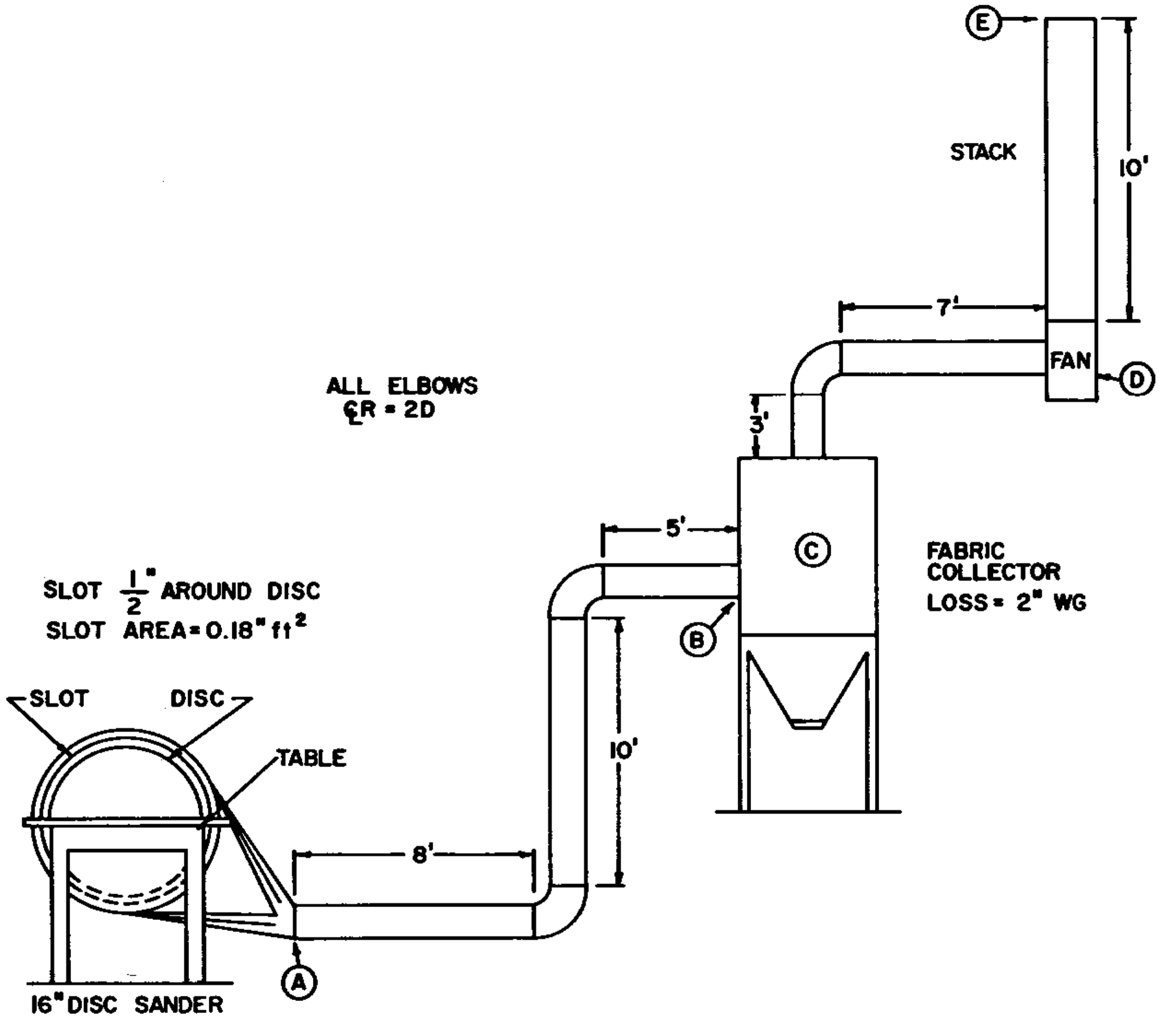

Figure 42-6. Controlling Dust from a 16" Disc Sander.

TABLE 42-12.
Answer Chart — A Worksheet for Answers to Problem

1	2	3	4	5	6	7	8		9	10	11	12	13	14	15	16	17	18	19
										Col. 7 plus Col. 9		Col. 10×Col. 11 / 100			1.00 plus Col. 14	Col. 13 times Col. 15	Col. 12 plus Col. 16		At junctio
			Air volume CFM			Length of duct in feet					Resistance in inches water gauge					Resistance in inches of water			
No. of br. or main	Dia. duct in in.	Area duct sq. ft.	in branch	in main	Vel. in FPM	straight runs	Number of elbows	entries	equiv. length	total length	per 100	of run	one VP	entry loss (VP)	hood suct. (VP)	hood suct.	static press.	gov. SP	cor-rected CFM
A-B	5	0.1364		475	3500	23′	2-90°		12′	35′	4.0	1.40	0.76	.25	1.25	0.95			
Slot		0.18			2640								0.44	1.0		0.44	2.79		
Collector												2.0					4.79		
C-D	5	0.1364		475	3500	10	1-90°		6′	16′	4.0	0.64	0.76				5.43	=SP in	
D-E	5	0.1364		475	3500	10				10	4.0	0.40	0.76				0.40	=SP out	

Column	Entry	Explanation
1	A-B	Considering section of duct from point A to B.
2	5	Example states that Q=440 cfm and v=3500 fpm. Using this information, choose duct diameter of 5″. This diameter gives Q=475 cfm at v=3500 fpm — see Column 5.
3	0.1364	The area of a 5″ circular duct is 0.1364 sq. ft.
5	475	Duct diameter=5″ and duct velocity=3500 fpm. Hence, the air volume in the duct is 475 cfm.
6	3500	Determined previously.
7	23	The length of straight pipe between points A and B is 23 feet.
8	2/90°	From A to B there are two 90° elbows.
9	12	The equivalent length of each elbow is 6 feet. (See Table 42-10.)
10	35	The total length of straight pipe equivalent from A to B is 35 feet.
11	4.0	Friction loss is read directly from Figure 42-3 knowing duct diameter and air volume.
12	1.40	To determine total resistance of run from A to B, take 1/100 of the product of Column 10 x Column 11.
13	0.76	Convert from velocity to velocity pressure $VP=\left(\frac{v}{4005}\right)^2$
14	0.25	Entry loss was given in problem statement.
15	1.25	The hood loss is 1 velocity pressure. This represents the amount of energy needed to get air to flow into the hood. NOTE: Because the 1 velocity pressure has been added on here, it will not be considered when calculations are made on the slot.
1	Slot	Considering slot opening only.
3	0.18	Given slot area in example description.
6	2640	Know slot area and air volume to be moved. Velocity can be determined from v=Q/A=475/0.18=2640 fpm.
13	0.44	Velocity pressure conversion as before.
14	1.0	This entry loss is given in problem statement.
17	2.79	Combining all duct and slot losses Column 12+Column 16+Column 16−2.79 in. w.g.
1	Collector	Considering only the collector.
12	2	Given that the pressure drop across the collector was 2 in. w.g.
17	4.79	The cumulative resistance in the system to this point.
1	C-D	Considering the duct between points C and D.
12	0.64	Resistance from duct friction is 0.64 in. w.g.

TABLE 42-12 Continued
Answer Chart — A Worksheet for Answers to Problem

Column	Entry	Explanation
17	5.43	Cumulative resistance in the system up to the fan. Quantity represents inlet static pressure.
1	D-E	Considering only the straight length of duct from the fan.
12	0.40	Duct resistance.
17	0.40	Static pressure after the fan.

It is important to note that in filling out Table 42-12, you start your design in Column No. 1 and complete the design horizontally through Column No. 17 in this particular problem.

EXPLANATION OF ANSWER CHART

Start entries in Column 1 and go across horizontally to Column 17. Columns 14, 15, and 16 need to be filled in only where air initially enters duct (i.e., through a hood). Section A-B will be considered in detail.

Fan static pressure is calculated from the following equation:

$$\text{Fan SP} = \text{SP (Fan Inlet)} + \text{SP (Fan Outlet)} - \text{VP (Fan Inlet)}$$
$$= 5.43 + 0.40 - 0.76$$
$$= 5.07 \text{ in. w.g.}$$

The fan and motor selected should be able to handle a static pressure of 5.25 to 5.5 inches of water.

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