CHAPTER 38

CONTROL OF EXPOSURES TO HEAT AND COLD

Harwood S. Belding, Ph.D.

HEAT

Introduction

At this point we expect the student to have acquired an understanding of the several elements which act to determine the stress of hot environments (Chapter 31) and of the physiological effects of heat exposures on the body (Chapter 30). The task in this section is to identify methods of control of occupational exposures to heat. These methods must be adapted to the nature of the heat stress and, if chosen properly, can be expected to ameliorate resulting physiologic strains.

Control of heat hazards has been discussed in several publications. Engineers may wish to consult more comprehensive publications issued by the American Industrial Hygiene Association[1] and the American Society of Heating, Refrigerating and Air Conditioning Engineers.[2] These offer some details not provided here, in particular on thermal control of large factory spaces by ventilation. Engineering aspects of ventilation are also given in Chapter 39. Earlier, Hertig[3] and Hertig and Belding[4] discussed methods of heat control. Wason[5] provided a comprehensive treatment of many aspects of the subject.

The ultimate goal of heat control engineering may be to create a climate of work in which true thermal comfort prevails. However, this seldom is achievable when large furnaces or sources of steam or water are present in the work area. In compromising with his ideal of providing comfort, the engineer may rationalize his shortcomings with the knowledge that man evolved as a tropical animal; he is well-endowed with physiologic mechanisms to cope with substantial levels of heat stress, particularly if he is acclimatized. It has even been suggested that some exercise of these natural mechanisms among well individuals may, as in the case with physical exercise, have beneficial effects (Chapter 30). This type of justification of hot working conditions is less warranted when jobs demand use of mental or perceptual facilities or of precise motor skills. In such cases thermal discomfort can distract attention; also, heat tolerance of physically inactive workers is less in some respects than for those whose duties require physical activity.

Analysis of the Problem and Options for Control

Before initiating control measures the engineer will wish to partition the components of the heat stress to which the worker is exposed, or in planning a new operation, is expected to be exposed. This information may be used not only as a basis for rational selection of the means of control, but also may be used, together with similar data obtained following adoption of control measures, to demonstrate the effectiveness of corrective actions that have been taken. It is evident that the heat stress for the individual worker depends on:

(1) the bodily heat production, (M), of the tasks which he performs
(2) the number and duration of exposures
(3) the heat exchanges as affected by the thermal environment of each task; namely exchanges by radiation (R), convection (C) and evaporation (E), as affected by air temperature (Ta), temperature of solid surround (Tw), air speed (v), and water vapor pressure of the skin (Pws) and the air (Pwa)
(4) thermal conditions of the rest area
(5) the clothing that is worn.

Items 1 and 2 represent elements of behavioral control; 3 and 4, environmental control; and 5 may be regarded as a combination of both.

The approach toward control may involve modification of one or more of these determinants of the heat stress. The challenge is to select specific methods for attack which will be both feasible and effective. Serious errors can result from resorting to some single pet engineering solution. Consider the consequences of ducting outside air to the task site. This air usually is blown at the worker at a temperature as warm as the upper reaches of the shed where the ducts have been installed. This will enhance cooling by evaporation of sweat, but if the air is warmer than the skin (35°C, 95°F), it will increase the convective heat load. Consideration of the trade off between needed heat loss and increased heat gain is essential. And, provided that the plant air is reasonably clean, the same goal might be achieved less expensively with portable fans. However, in such a situation the real mistake may be the failure to recognize that the heat problem derives from radiant load from a furnace and this is not decreased by air movement. This mistake has been made less frequently in recent years, but elaborate ducting across the ceilings of older plants exists as evidence of inappropriate action to control radiant heat loads.

Let us examine means and effectiveness of the five listed modifiers of the heat stress.

(1) *Decreasing the physical work of the task.* Metabolic heat can comprise a large fraction of the total heat load. However, the amount by which this factor may be reduced by control is quite limited. This is because an average sized man who is simply standing quietly while pushing buttons will produce heat at a rate of 100 kcal/hr

whereas one who is manually transferring fairly heavy materials at a steady pace will seldom have a metabolic rate higher than 300 kcal/hr and usually not more than 250 kcal/hr.[6] Obviously, control measures, such as partial mechanization, can only reduce the (M) component of these steady types of work by 100 to 200 kcal/hr; nevertheless mechanization can also help by making it possible for the worker to be more isolated from the heat source, perhaps in an air conditioned booth.

Tasks such as shovelling which involve metabolic heat production at rates as high as 500 to 600 kcal/hr require that rest be taken one-half to two-thirds of the time simply because of the physical demands of the labor. Thus, the hourly contribution of (M) to heat load will seldom exceed 300 kcal/hr. It is obvious that mechanization of such work can increase worker productivity by making possible a decrease in the time needed for rest.

(2) *Modifying the number and duration of exposures.* When the task in a hot environment involves work that is a regularly scheduled part of the job, the combined experience of workers and management will have resulted in an arrangement which makes the work tolerable most of the time for most of the workers. For example, the relief schedule for a task which involves manual transfer of hot materials may involve two workers only because of the heat, and depending on the duress, these workers may alternate at five-minute or up to hourly intervals which have been determined empirically. Under such conditions overall strain for the individual will be less if the cycles are short.[7] Where there is a standardized quota of hot work for each man, it is sometimes lumped at the beginning of the shift. This arrangement may be preferred by workers in cooler weather; however, there is evidence that the strain of such an arrangement may become excessive on hot days. The total strain, evidenced by fewer heart beats, will be less if the work is spread out.

The stress of hot jobs is dependent on vagaries of weather. A hot spell or an unusual rise in humidity may create overly stressful conditions for a few hours or days in the summer. Nonessential tasks should be postponed during such emergency periods, in accordance with a prearranged plan. Also, assignment of an extra helper can importantly reduce heat exposure of members of a working team. However, there is danger in this practice when unskilled or unacclimatized workers are utilized in this role.

Many of the critically hot exposures to heat faced in industry are incurred irregularly, as in furnace repair or emergencies, where levels of heat stress and physical effort are high and largely unpredictable, and values for the components of the stress are not readily assessable. Usually such exposures will force progressive rise in body temperature. Ideally such physiologic responses as body temperature and heart rate would be monitored and used as criteria for limiting such exposures on an *ad hoc basis.* Practically, however, the tolerance limits must be based on experience of the worker as well as of his supervisor. Fortunately, for most workers most of the time individual perception of fatigue, faintness or breathlessness may be relied upon for bringing exposures to a safe ending. The highly motivated individual, particularly the novice who desires acceptance, is at greater risk. In the same spirit, foremen should respect the opinion of an employee when he reports that he does not feel up to work in the heat at a particular time. Non-job personal factors such as low grade infection, a sleepless night or diarrhea (dehydration affects sweating) which would not affect performance on most jobs, may adversely affect heat tolerance.

Perhaps the best advice that can be offered for control of irregular exposures is (a) that formal training and indoctrination of effects of heat be provided supervisors and workers; and (b) that this include advice to the effect that each exposure should be terminated before physical distress is severe. There is abundant evidence that the physiological strain of a single exposure which raises internal body temperature to 39°C (102.2°F) is such as to contra-indicate further exposures during the same day; it may take hours for complete recovery. More work can be achieved during several shorter exposures and with less overall strain.

(3) *Modifying the thermal environment.* The environmental engineer will usually identify important sources of heat stress in a qualitative sense, without resorting to elaborate measurements. Thus, his experience will suggest that when air is static and the clothes of the workers become wet with sweat it will help to provide a fan.

Nevertheless, there are advantages in quantitative analysis of the heat stress (and where possible determination of physiological strains) on workers. The effects of various approaches to control can then be predicted and improvements in thermal conditions at the workplace can be documented for higher levels of management based on measurements made before and after action has been taken.

We cite concrete examples to illustrate how the quantitative analytic approach may be used.

CASE I. First is a case which is encountered frequently under ordinary conditions of hot weather. Let us assume a laundry where the humidity is high (Pwa = 25 mm Hg) despite the operation of a small exhaust fan on the wall. There is no high level heat source so Tw is about the same as Ta.

In the simplest situation we take Ta and Tw equal to the temperature of the skin, which, under heat stress, may be assumed to be 35°C (95°F). This means heat exchange by R and C is zero.* Let us examine the case on the basis that exposure is continuous and the average physical work is moderate (M = 200 kcal/hr). The heat load to be dissipated, Ereq, is then,

$$M + R + C = Ereq$$
$$200 + 0 + 0 = 200$$

*In this case R+C is changed by about 17 kcal/hr for each °C of deviation of Tw and Ta from Tsk.

The workers wear only shorts or shorts and halter. The air speed is low, 20 m (65 ft.) per minute. Analysis in accordance with Chapter 31 for the seminude condition yields indication of maximum cooling by evaporation of sweat, Emax.

Emax = 2.0 $v^{0.6}$ (42 − Pwa), where 42 mm Hg is Pws of completely wetted skin at 35°C; or

Emax = 2 x 6.0 (42 − 25) = 200 kcal/hr (approx.)

Nominally, a worker under these conditions would be just able to maintain bodily heat balance if he kept his skin completely wet. To do this he would have to sweat extravagantly, which means some dripping. It is easy to see why the workers wear as little clothing as possible. Wearing a long-sleeved work shirt and trousers would reduce Emax by about 40 percent, or to 120 kcal/hr. The resulting excess of heat load over Emax would result in rise of body temperature and it can be estimated that the ordinary limit of tolerance would be reached in about an hour.

When, as in this case, the heat load is itself moderate, the attack of the control engineer should be aimed at increasing Emax. In most such situations the management or the workers might find it expedient to bring in fans for spot "cooling." Note that since Emax is 0.6 power function of air speed, tripling of air movement across the skin would result in doubling of Emax. In this case an increase from 20 m/min to 60 m/min is easily achieved and it is predicted that such air speed will raise Emax to 400 kcal/hr. Sweat would be reduced to about 0.35 liters/hr and would be evaporated easily; the skin would no longer be dripping wet. It is clear that this control measure has limitations. For example, if air speed were already 60 m/min tripling would produce a wind which might disrupt operations.

A more effective permanent approach would be to replace the small exhaust fan with exhaust hoods opening over the principal source of moisture. This would work well in a dry climate, but in a humid one the make-up air from outside might have such a high Pwa as nearly to cancel the value of hoods. It is obvious that in Case I the use of mechanical air conditioning would prove expensive.

CASE II. This is selected to show how the wearing of clothing can be advantageous and the presence of high air speed a liability under very hot, dry conditions. Assume Ta = 45°C (113°F), Tw = 55°C (131°F), v = 100 m (300 ft.) per minute and Pwa, 10 mm. We use the same M as in Case I. Long-sleeved shirt and trousers are worn.*

M + R + C = Ereq

200 + 130 + 95 = 425   Emax = 610 kcal/hr

Suppose the worker wore only shorts under these circumstances. R, C and Emax would be increased:

M + R + C = Ereq

200 + 220 + 160 = 580   Emax = 1010 kcal/hr

The total heat load is increased about 155 kcal/hr. The specific cost of baring the skin would be about 0.26 liter per hour, raising the total requirement of sweating to 1.0 liter per hour, as compared with 0.71 liter per hour when wearing shirt and trousers.

Thus under conditions where Tw and Ta are above 35°C and Pwa is low, the wearing of clothing reduces heat stress and strain. In examining the above model it will be apparent that there is an optimum amount of clothing in such situations. This is the amount which reduces Emax to a value only slightly in excess of Ereq. The long-sleeved shirt and trousers happen to be just about right for this purpose under the given conditions.

With low Pwa as in a semi-arid area, a more satisfactory solution probably might be reached through installation of an evaporative cooler. In Case II, inside temperature was usually 5°C hotter than outside, due to process heat and insulation on the roof of the shed. Assuming outside Ta usually does not exceed 40°C (104°F) and Pwa is about 10 mm Hg, outside air drawn through a water spray washer in large volume theoretically could be reduced to prevailing out-of-door wet bulb temperature, namely 22°C (72°F), though in practice probably only 80 percent efficiency could be achieved. Most of the wash water could be recycled. Pwa of the conditioned air would be raised from 10 to 20 mm Hg. The temperature of the work space might actually be reduced by this means to 30°C. If so, the components of heat load for clothed workers would be reduced by 35 percent from Ereq = 425 to Ereq = 280; Emax would still permit free evaporation of sweat.

M + R + C = Ereq

200 + 130 − 50 = 280   Emax = 420 kcal/hr

CASE III. This case is chosen to illustrate the dramatic reduction in heat load achievable by provision of appropriate shielding when radiation from a furnace is substantial. Practical examples of the reduction in radiant heat load achievable by these means are provided by Lienhard, McClintock and Hughes,[8] by Haines and Hatch,[9] and by others.[5, 10] This case is chosen from the first of these references, because the situation is real, and physiological and environmental data are available. The task is that of skimming dross from molten bars of aluminum. The worker stands at the task. Manipulation of a ladle involves moderate use of shoulder and arm muscles and requires an M of about 200 kcal/hr. The environmental temperatures before the corrective action were reported as Tg = 71.7°C (161°F), Ta = 47.8°C (118°F) and Twb = 30.5°C (87°F). Air was forced from an overhead duct at 275

*The formulae used are explained in Chapter 31 and summarized here.

NUDE	kcal/hr	CLOTHED
11(Tw − 35)	R	6.6 (Tw − 35)
1.0 $v^{0.6}$ (Ta − 35)	C	0.6 $v^{0.6}$ (Ta − 35)
2.0 $v^{0.6}$ (42 − Pwa)	Emax	1.2 $v^{0.6}$ (42 − Pwa)

Tw is approximated from temperature readings of a six-inch blackened globe, Tg, using Tw = Tg + 0.24 $v^{0.5}$ (Tg − Ta); v is in m/min; "35" is assumed Tskin; "42" is Pws of completely wet skin at Tskin of 35°C.

m/min (900 fpm).* Note that the humidity was very high (Pwa = 24 mm Hg) which is characteristic of the local climate. In terms of heat load and Emax the situation was:

$$M + R + C = Ereq$$
$$200 + 870 + 220 = 1290 \qquad Emax = 630 \text{ kcal/hr}$$

It is obvious from the deficiency of evaporation and the enormous load that the workers, despite full clothing and a face shield, were able to perform this task only for a few minutes at a time. Heat exhaustion was not uncommon (and might partly be attributable to the difficult hot conditions prevailing in the nearby rest area).

Engineers undertook control of this heat exposure by interposing finished aluminum sheeting between the heat source and the worker. Infrared reflecting glass at face level permitted seeing the task and space was left for access of the arms in using the ladle. As a result of these measures it was recorded that both Tg and Ta were reduced to 43°C (110°F). The same air speed was present as before and if we assume the same Pwa we obtain:

$$M + R + C = Ereq$$
$$200 + 50 + 140 = 390 \qquad Emax = 630 \text{ kcal/hr}$$

By this action to reduce R the heat load was brought to a level that is reasonable for prolonged work, but did not completely eliminate the heat stress. The predicted requirement for sweating to maintain heat balance was reduced from the previously impossible-to-sustain level of 2.1 liters/hr to about 0.7 liter/hr. (The before and after average levels actually observed for two workers were not far from these predictions, namely 2.1 and 1.1 liters/hr. The same two subjects also showed a marked reduction in heart rate, as a result of the changes, from an average of 146 to 108 beats/min.)

The percent reduction of the radiant load can be taken as a measure of the effectiveness of the reflective shielding, and in this instance approximates 85 percent. Large errors in the estimate of R are possible at extremely high globe temperatures, but in this case it appears that the maximum relief one could expect from shielding was achieved. Haines and Hatch[8] reported smaller reductions in R of 51 to 74 percent from interposing a sheet of aluminum at eleven different work sites in a glass factory. Others[1] have shown reduction of 90 percent or more under ideal conditions not likely to prevail on the plant floor.

Control of Radiation: *Further Considerations*. While in Case III we have dealt with some aspects of control of R by shielding, the two other classical approaches of industrial hygiene engineering, namely control at the source and control at the man, offer possibilities which must be considered.

Application of insulation on a furnace wall can reduce its surface temperature and thereby the level of R. A by-product of such treatment is saving in fuel needed to maintain internal furnace temperatures. Application of a polished metallic surface to a furnace wall will also reduce R. However, a polished metallic surface will not maintain its low emissivity* if it is allowed to become dirty. A layer of grease or oil one molecule thick can change the emissivity of a polished surface from 0.1 to 0.9. And the emissivity of aluminum or gold paints for infrared is not necessarily indicated by their sheen. If the particles are smaller than about one micron they emit almost like a black body. (The same is true for fabrics coated with very fine metallic particles.)

Equal or even more effective reduction of R is achievable with non-reflective barriers through which cool water is circulated.

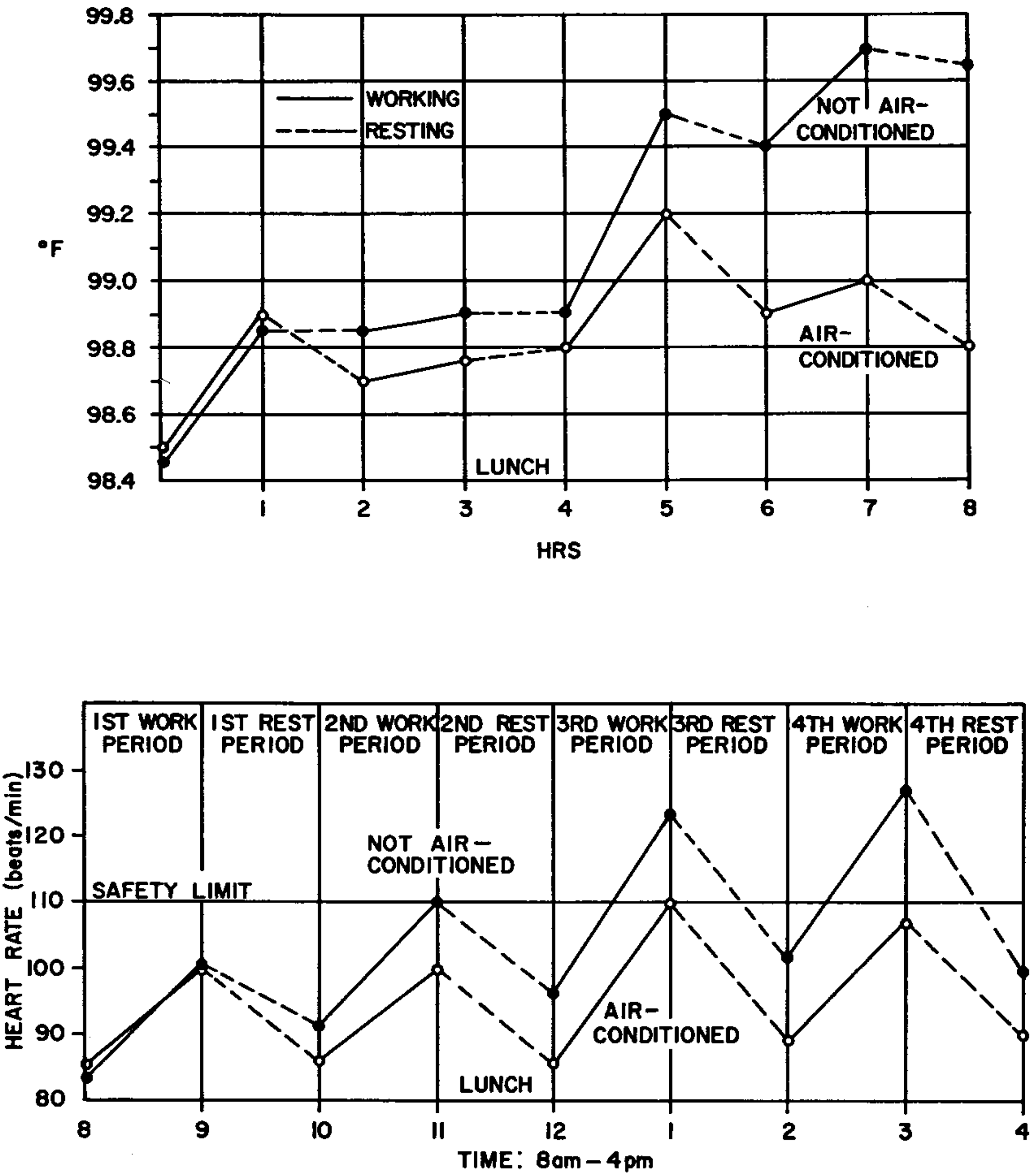
The engineer is frequently baffled in shielding by the fact that access to the heat source is required for performance of the task. We have seen various solutions to this problem. One is a curtain of metal chains which can be parted as required and which otherwise reduces emission like a fireplace screen. Another is a mechanically activated door which is opened only during ejection or manipulation of the product. And finally, remotely operated tongs may be provided, taking advantage of the fact that radiant heating from an open portal is limited to line of sight and falls off as the reciprocal of the square of the distance from the source.

(4) *Thermal conditions of the rest area.* Brouha[7] states "It is undeniable that the possibility of rest in cool surroundings reduces considerably the total cost of work in the heat." This is demonstrated by responses of heart rate and body temperature of two groups of men doing the same job with and without access to an air conditioned space for recovery (Figure 38-1). There is no solid information on the optimum thermal conditions for such areas but we have laboratory data which support setting the temperature near 25°C (77°F). This feels chilly upon first entry from the heat, but adaptation is rapid.

The placement of these areas is of some importance. The farther they are from the workplace the more likely that they will be used infrequently or that individual work periods will be lengthened in favor of prolonged rest periods.

*High speed air jets (8-inch to 12-inch diameter) are frequently used for purposes of man-cooling. These directly affect air speed over only a small portion of the body. Directed downward, the speed measured near the legs may be only 10 to 20 percent of that at the head. The head represents only 10 percent of the body surface, the legs 40 percent.

*Emissivity is the capacity to radiate relative to a black body, which has a capacity of 1.0. Bright metal surfaces are poor emitters, having emissivities of less than 0.1. Absorptivity of materials for radiant energy is equivalent to their emissivity i.e., a black body is a perfect absorber. Reflectivity is 1.0 minus the emissivity. These characteristics are dependent on spectral wavelength, which is shorter for bodies at higher temperatures. The radiation dealt with here is in the infrared range, of longer wavelength than visible light. The emissivity and absorptivity of unpolished surfaces in this range are close to those of a black body, regardless of color. Thus light-colored oil paint will emit as a black body, as will skin, regardless of its color, and as will clothing. The near side of a polished metallic shield (1) will reflect back 90 percent of the energy which impinges from a furnace and (2) will emit from its far side only 10 percent of the energy that was absorbed. To be doubly effective in this way a shield must be exposed to air on both sides.



Brouha, L.: Physiology in Industry, 2nd Edition. New York, Pergamon Press, 1967.

Figure 38-1. Average Heart Rates and Body Temperatures at Beginning and End of Successive Work and Rest Periods. Group with lower levels of responses rested in an air conditioned room.

Incidentally, the same principle applies for positioning of water fountains. When they are remote from the worker, substantial dehydration is more apt to occur. The proper temperature for drinks under hot conditions is often asked. There is no scientific answer, but most men will not willingly drink fluids that are close to body temperature. They welcome chilled water and seek chilled soft drinks and ice cream when these are available in rest areas. (Actually a liter of intake at 8°C (46°F) will contribute to body cooling by extracting 30 kcal of body heat; a half pint of frozen sherbet will remove about the same amount.) Experienced workers recognize that frequent intake of small amounts of fluid is better than large draughts.

(5) *Clothing*: (*a*) *Conventional work clothing*. Heat exposures may be controlled through selective wearing of clothing, as illustrated by Cases I and II. In Case II we illustrated effects of an ordinary work shirt and trousers in reducing heat transfer by radiation, convection and evaporation by about 40 percent from the values applicable to seminude men.[11] Design as well as thickness can be exploited. Note that air movement under clothing, that is provided for by loose fit and generous openings, will have twice as much effect on Emax as it does on C (see coefficients). On the other hand, in dry environments with high air speed, tighter fit may be employed to reduce gain by C without critical reduction of Emax. Additional thickness may be exploited for further reduction of gain by R+C and may be of particular value when alternating between extremes of heat and cold in open sheds in wintertime. In such situations long underwear may be advantageous because it acts as a "heat sponge." Thickness can also be an advantage in clothing for fire fighting.

(b) *Aluminized reflective clothing*. We have reported effects of wearing aluminum-coated clothing on heat exchanges.[11] Somewhat to our surprise our samples provided only about 60 percent reduction of radiant heat gain as compared with the 40 percent available from ordinary work clothing. At high humidities (Pwa=20 mm Hg and above) a full aluminized suit, consisting of long coat, full trousers, spats and hard hat, represented a greater handicap for prolonged use than ordinary work clothing. Subjects became overheated because the suit interferred with evaporation of their sweat. In a trial where only the front of the body was exposed to the high level radiant source, we found that an aluminized apron and similar reflective covering for the front of the legs provided nearly as much protection as the full suit and permitted necessary evaporation.

(c) *Thermally conditioned clothing*. Numerous ideas have been incorporated in special clothing for maintaining comfort in extreme heat (or cold). Some systems supply appropriately cool air from a mechanical refrigerator to points under a jacket or coveralls. When air from a remote source is used there are two problems. One is the gain of heat through the walls of the supply tubing. This problem has been solved in some cases by using porous tubing which will leak an appropriate amount of supply air to keep the wall of the tubing suitably cool. The other problem is distribution of the air through the suit. With a simple, single orifice it is difficult to cool a sufficient area of skin; cooling limited to the face or trunk is usually not enough. Provision of several orifices, though better, will create bulk and restrict mobility. In fact, the restriction of movement resulting from tethering the worker to supply line will often contraindicate this type of system.

The vortex tube source of cool air has been used successfully in some situations.[3] The device is carried on the belt. Air introduced tangentially at high velocity is forced into a vortex, which results in two separable streams of air, one cold which is distributed under the suit, the other hot which is discarded. Compressed air requirements to operate the vortex system are large.

Self-contained sources of conditioned air which can be back-packed have also been developed. One contains a liquid which is sealed into a finned container. After being cooled in a deep freeze the container is placed in the pack. A small battery-driven fan circulates air across the fins and into the suit. A single charging of this device may extend tolerance for relining furnace walls from several minutes to 30 or 60 minutes.

More sophisticated devices employ a closed fluid-filled system and a fairly elaborate network of small tubing for distribution.

The nuisance factor must be considered with all such devices. Men will not go to the trouble of donning them unless they recognize more than a marginal advantage. On the other hand, with such devices it has sometimes been possible to change hot tasks which required long rest pauses into continuous duty operations involving fewer workers.

Checklist

The emphasis of this section has been on rationalization of the options for control of heat exposures, based on consideration of all elements of specific situations. Often there are several options which may be capitalized upon simultaneously without conflict. In other cases trade-offs must be weighed. Table 38-1 is a checklist which may be helpful in considering options which have been discussed in the text.

COLD

Introduction

Protection of the body against excessive cooling has not received popular attention of occupational health workers to the extent of protection against heat, even though many workers are exposed to cold conditions. This is despite the fact man has much more innate ability to adapt to heat, attributable to his well-developed sweating mechanism. In cold, he can only resort to constriction of skin blood vessels or shivering. If nude, at rest ,and in still air the dermal vasoconstriction will avail to provide heat balance only down to about 28°C (82°F). Man's adaptation to cold has been based on his ingenuity in pro-

TABLE 38-1

CHECKLIST FOR CONTROLLING HEAT STRESS AND STRAIN

Item	Actions for Consideration
Components of Heat Stress	
M, body heat production of task	reduce physical demands of the work; powered assistance for heavy tasks
R, radiative load	interpose line-of-sight barrier furnace wall insulation metallic reflecting screen heat reflective clothing cover exposed parts of body
C, convective load	if air temperature above 35°C (95°F) reduce air temperature reduce air speed across skin wear clothing
Emax, maximum evaporative cooling by sweating	increase by decreasing humidity increasing air speed
Acute Heat Exposures	
R, C and Emax	air or fluid conditioned clothing; vortex tube
duration and timing	shorten duration each exposure; more frequent better than long to exhaustion
exposure limit	self-limited, based on formal indoctrination of workers and foremen on signs and symptoms of overstrain
recovery	air conditioned space nearby
Individual Fitness for Work in Heat	determine by medical evaluation, primarily of cardiovascular status careful break-in of unacclimatized workers water intake at frequent intervals non-job related fatigue or mild illness may temporarily contraindicate exposure (e.g. low grade infection, diarrhea, sleepless night)

viding himself with insulative clothing and heated shelter.

Clothing Requirements

The same heat balance equation that is used for heat exposures is applied also for cold. Equilibrium is achieved when $M+R+C=E$, but in cold weather R and C are minus quantities. E will reflect activity of the sweat glands as needed to balance the equation. When not overclothed, E is still present to the extent that body water diffuses through the skin (about 15 grams per hour with cooling value of 10 kcal per hour) and the lungs. The lung loss is also about 15 g per hour when inactive, but increases in proportion to ventilation of the lungs during activity. There are also small losses in warming cold inspired air,[12] which are neglected in this treatment. Minimum combined losses by E are commonly assumed to be about 25 percent of M when clothing requirements are being considered. Thus, 0.75 M is the heat available for loss by (R+C) when heat balance is being maintained without recourse to excessive sweating.

Over the thermal range of interest, Newton's Law of Cooling is applicable; this states that heat loss will be proportional to the thermal gradient divided by the insulation. In this instance:

$$(R+C)=\frac{Ts-Ta}{\text{Insulation}}, \text{ where}$$

Ts and Ta are skin surface and air temperature, respectively. In the equilibrium state, and since $(R+C)=0.75$ M, insulation requirement for a known thermal gradient may be expressed as:

$$\text{insulation required}=\frac{(Ts-Ta)}{0.75\ M}$$

The unit used for describing insulation needs or insulation value with respect to clothing of man is the clo.*[13] In this unit the insulation required is:

$$Iclo=\frac{5.55(Ts-Ta)}{0.75\ M}$$

The value 5.55 is the coefficient which applies where Ts and Ta are in °C and M is in kcal per square meter of body surface per hour.

Application of the equation is illustrated in answering the question: How much insulation is required to maintain comfort for a man seated at rest ($M=50$ kcal/m^2·hr) at 21°C (70°F)? The

*The unit was selected to represent the approximate insulation value of a business suit. One clo will maintain a thermal gradient of 0.18°C over an area of one square meter when the thermal flux is one kcal/hr.

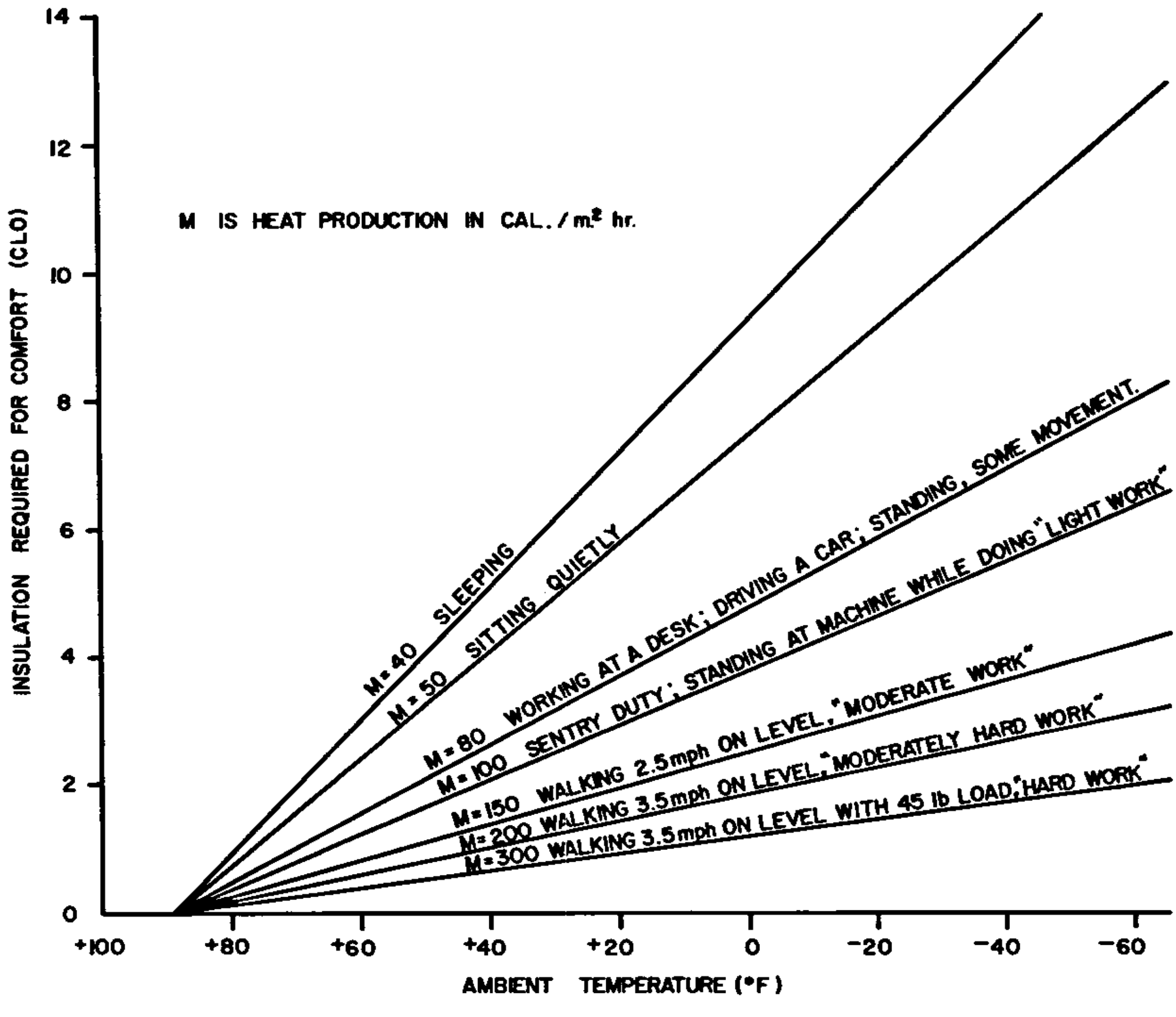
skin temperature for comfort is known to average about 33°C, so

$$Iclo = \frac{5.55(33-21)}{38} = 1.7$$

The total insulation need may be met from two sources, both of which fundamentally involve immobilization of a layer of "still air" (under practical conditions one-quarter inch of "still air" is worth about 1 clo). The first source is the film of air which always overlies the outside of the clothing, or the surface of the skin when it is bare). This insulation of air, Ia, is worth about 0.7 clo when the body is inactive and is exposed to the natural convection present in a room. Ia varies as a power function of the reciprocal of air speed: at 30 m (100 ft) per minute it is 0.5 clo, at 100 m (330 ft) per minute it is 0.3 clo. The second source of insulation must be the clothing itself, Icl. Thus, the clothing needed at 21°C would be 1 clo in still air, because Ia contributes 0.7; at 100 m per minute the clothing *per se* would have to be worth 1.4 clo.

When at rest as in the example, each added clo of insulation will protect to a 6.8°C (12°F) lower temperature. This means a total requirement for thermal equilibrium of 4.8 clo while sitting at 0°C (32°F). On the other hand when working at a fairly hard M of 200 a total of only 1.2 clo should suffice for heat balance at the temperature; i.e., 5.55(33/150) = 1.2. The requirements for various levels of work at various temperatures are shown in Figure 38.2. The disparity of requirements for work and rest gives rise to one of the biggest problems for workers who are out-of-doors for prolonged periods in cold weather. The tendency of the inexperienced is to overdress. The result is copious sweating in the body's attempt to maintain heat balance while working. The heavy clothing will not permit sufficient evaporative cooling. A substantial amount of the sweat is accumulated in the clothing and continues to evaporate during subsequent rest, thus counteracting available insulation at a time when it is most needed.

When in sunlight the net heat loss by (R + C)



Newburgh, L. H. (ed.): Physiology of Heat Regulation and the Science of Clothing. New York, Hafner, 1968 reprint of 1949 Edition.

Figure 38-2. Prediction of Total Insulation Required for Prolonged Comfort at Various Activities in the Shade as a Function of Environmental Temperature. From reference 14.

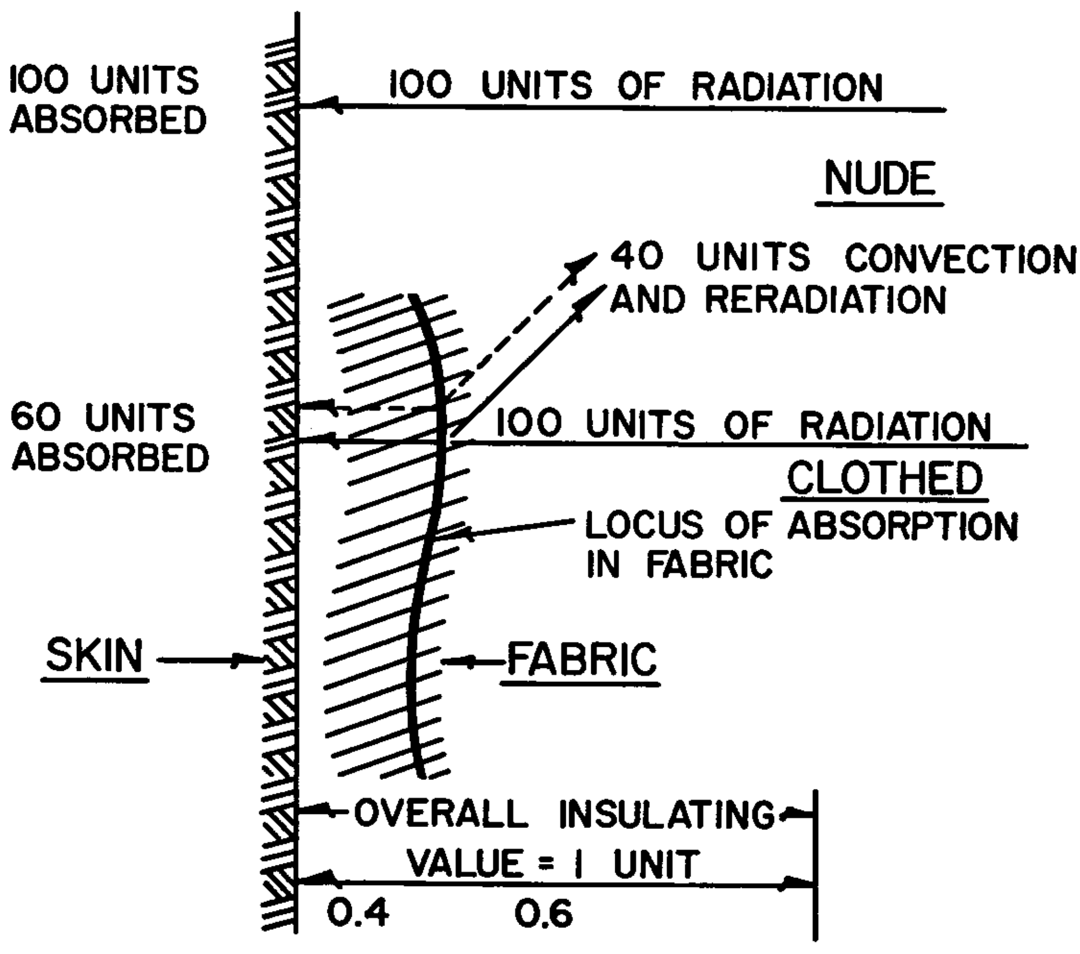
is reduced. The amount of reduction depends on the intensity of the solar energy as well as the thickness and color of the clothing. When nude the effect of sunlight may be considered the same as elevation of Ta by about 10°C (18°F). In heavy clothing the effect will be much less, probably 2 to 3°C.[13]

It has been estimated that when clothing insulation is less than adequate to maintain heat balance some 40 to 80 kcal/m² may be withdrawn in body cooling before undue discomfort develops.[14] Thus, if exposures are time-limited there is margin for error in selection of insulation.

Materials and Design of Clothing

The thermal worth of clothing may be determined by human calorimetry. A volunteer subject wears the garment assembly while engaged in constant activity in a room of fixed thermal conditions. M is estimated from measurement of oxygen consumption; Ts and internal body temperature by thermocouples; and evaporative heat loss by change in body weight. The data can be applied to finding Iclo with the equation. Alternatively, an electrically heated copper manikin can be brought into thermal equilibrium, and on the basis of knowledge of the heat input and Ts and Ta, the insulation value can be calculated.

The insulating value of clothing depends on the thickness of air which it effectively immobilizes, not on specific insulation of the materials themselves. Thus an equal thickness of steel wool and eider down will provide about the same insulation. To adjust for vagaries of weather and variables of work load cold weather outfits should be multilayered. Even so, it is difficult to raise the total protection above 3 clo without markedly interfering with body movements. The outer layer and perhaps a secondary layer should be of wind resistant fabric. Lightness of garments is achieved through use of resilient, low density materials (quilted fibers, pile, loosely woven wool or synthetics). Looseness of fit and easily adjustable closures will provide help in modifying the insulation to meet variable needs for work and rest.



Hertig, B. A., Belding, H. S.: Evaluation and control of heat hazards in: Temperature: Its Measurement and Control in Science and Industry. New York, Reinhold Publishing Corp., 1963, vol. 3, pt. 3, p. 347.

Figure 38-3. Diagrammatic Cross Section through Skin and Clothing To Show Factors Affecting Efficiency of Protection against Long-Wave Radiant Energy.

In extreme cold, there is no adequate protection for hands during periods of inactivity; this is because thruflow of warm blood is reduced to almost nothing. Mittens are better than gloves because they present less surface area for heat loss. Blood flow to the feet is similarly curtailed whenever overall Iclo falls below the requirement for heat balance. The best known foot protection is provided by insulated boots sealed inside and outside by vapor barriers. The face, which has good circulation of blood, will usually be adequately protected if a forward-projecting fur ruff is available to protect against wind. Masks are not recommended for prolonged use in extreme cold because frostbite may develop unnoticed.

A principle of clothing applicable to both hot and cold weather was illustrated in this chapter by our finding that 40 percent of radiation impinging on ordinary work clothing does not reach the skin. Actually, the effect is dependent on the ratio of insulation lying outside the locus of absorption (including Ia) to the total insulation, as shown in Figure 38-3. The same principle applies to heat absorbed or emitted from other sources. For example, the effectiveness of electrically heated garments depends on proximity of the wires to the skin.

In hot or cold weather clothing the efficiency of evaporative cooling by sweat is likewise dependent on the locus of the evaporation. If directly from the skin the efficiency is 100 percent. However, if the sweat is wicked outward and evaporated at some distance from the skin the efficiency is lowered accordingly and extra sweat must be produced to achieve the same amount of body cooling.

Further details on protection against cold appear in references (12) and (13).

References

1. *Heating and Cooling for Man in Industry*. Am. Ind. Hyg. Assoc., Akron, Ohio 48075 (1970).
2. *ASHRAE Guide and Data Book, Systems Volume*. Am. Soc. Heat. Refrig., Air Cond. Eng., Inc., New York (1970).
3. HERTIG, B. "Control of the Thermal Environment." In *Ergonomics and Physical Environmental Factors*. Int. Labour Office, Geneva, Occ. Safety and Health Series No. 21:362 (1970).
4. HERTIG, B. A. and H. S. BELDING. "Evaluation and Control of Heat Hazards." *In Temperature: Its Measurement and Control in Science and Industry, Vol. 3, Pt. 3*:347 (1963).
5. WASON, A. B. "Cooling Hot Spots in Industry." *Heating and Ventilating Ref. Sect.*, London, England (Sept. 1954).
6. "Ergonomics Guide to Assessment of Metabolic and Cardiac Costs of Physical Work." *Am. Ind. Hyg. Assoc. J. 32*:560, Akron, Ohio (1971).
7. BROUHA, L. *Physiology in Industry*. 2nd edition, Pergamon Press, New York (1967).
8. LEINHARD, W. F., R. S. McCLINTOCK and J. P. HUGHES. "Appraisal of Heat Exposures in an Aluminum Plant." *Proc. XIII Int. Congr. on Occ. Health*, 876, New York (1961).
9. HAINES, G. F., Jr. and T. HATCH. "Industrial Heat Exposures: Evaluation and Control." *Heating and Ventilating*, London, England (Nov. 1952).
10. *ASHRAE Handbook of Fundamentals*. Am. Soc. Heat. Refrig. Air Cond. Eng., Inc., New York(1967).
11. BELDING, H. S., B. A. HERTIG and M. L. RIEDESEL. "Laboratory Simulation of a Hot Industrial Job to Find Effective Heat Stress and Resulting Physiologic Strain." *Am. Ind. Hyg. Assoc. J. 21*:25, Akron, Ohio (1960).
12. WEBB, P. (Ed.) *Bioastronautics Data Book*. NASA Yellow Springs, Ohio (1964).
13. BURTON, A. C. and O. G. EDHOLM. *Man in a Cold Environment*. Edward Arnold, Ltd. London, England (1955).
14. NEWBURGH, L. H. (Ed.) *Physiology of Heat Regulation and the Science of Clothing*, Hafner, New York (1968 reprint of 1949 edition).

Preferred Reading

Heating and Cooling for Man in Industry. Am. Ind. Hyg. Assoc., Akron, Ohio (1970).

BURTON, A. C. and O. G. EDHOLM. *Man in a Cold Environment*. Edward Arnold, Ltd., London, England (1955).

CHAPTER 39

PRINCIPLES OF VENTILATION

John E. Mutchler

INTRODUCTION

Ventilation is one of the most important engineering control techniques available to the industrial hygienist for improving or maintaining the quality of the air in the occupational work environment. Broadly defined, ventilation is a method of controlling the environment with air flow. In industrial ventilation this air flow may be used for one or a combination of the following reasons:

1. Heating or cooling;
2. Removing a contaminant;
3. Diluting the concentration of a contaminant;
4. Supplying make-up air.

These basic uses of industrial ventilation can be divided into three major applications:

1. The prevention of fire and explosions;
2. The control of atmospheric contamination to healthful levels;
3. The control of heat and humidity for comfort.

All of these applications are important to the industrial hygienist, and each must be understood thoroughly in order to provide safe, healthful and comfortable working conditions. The control of heat and humidity is covered in Chapter 38. This chapter, "Principles of Ventilation," together with Chapters 40, 41 and 42, deals primarily with the control of atmospheric contamination for assuring a healthful work environment.

CLASSIFICATION OF VENTILATING SYSTEMS

The control of a potentially hazardous airborne contaminant by ventilation can be accomplished by either one or both of two methods: diluting the concentration of the contaminant before it reaches the worker's breathing zone by mixing with uncontaminated air, or capturing and removing the contaminant at or near its source or point of generation, thus preventing the release of the contaminant into the workroom. The first of these methods is termed "general ventilation" or "dilution ventilation"; the second is called "local exhaust ventilation." Dilution ventilation does not reduce or eliminate the total amount of hazardous material released into the workroom air; local exhaust ventilation prevents the release of the contaminant within the workroom. Normally, local exhaust ventilation is the preferred and more economical method for contaminant control compared with dilution ventilation.

The differences between dilution and local exhaust ventilation are not always clearly defined. Nevertheless, the following criterion is used in classifying a ventilation system as one type or the other:[1]

> A ventilation system is a dilution ventilation system if the concentration of contaminant in the exhaust air stream is not significantly higher than in the general room air; it is a local exhaust system if the concentration of material in the exhaust air stream is significantly higher than that in the general room air.

General Ventilation

The term "general ventilation" suggests that a room or an entire building is flushed by supplying and exhausting large volumes of air throughout the area. Properly used, general ventilation can be very effective for the removal of large volumes of heated air or for the removal of low concentrations of non-toxic or low toxicity contaminants from minor and decentralized sources. General or dilution ventilation is achieved by either natural or mechanical means. Often the best overall result is obtained with a combination of mechanical and natural air supply with mechanical and natural exhausters.

Natural General Ventilation. The natural means by which buildings or enclosures can be ventilated include wind and thermal convection. These two effects, usually in combination, result from natural pressure differences and air density differences respectively, causing natural displacement and infiltration of air through windows, doors, walls, floors and other openings. Obviously, if it were sufficient, natural ventilation would be much cheaper than mechanical ventilation, but wind currents and thermal convection are erratic and sometimes hard to predict. Therefore, natural ventilation is unreliable as a primary control method. Erratic wind direction alone makes this aspect of natural ventilation undependable as a primary method of solution to any critical dilution ventilation problem.

When the wind is blowing, a pressure is exerted on the upwind side and a suction is exerted on the downwind side of the building. Wind forces can be accurately predicted for a flat terrain, but the determination of wind forces within a cluster of industrial buildings defies the inherent simplicity which design parameters must exhibit to be useful and applicable to ventilation engineering problems.

The amount of air that enters a building under a natural ventilation scheme depends both upon the wind and upon thermal effects occurring within the building. Warmer air inside a building rises and leaks out of openings, cracks, and vents in the upper areas; colder air leaks into the building by the same process in the lower areas, assuming the same degree of tightness throughout.

Thermal (air density) effects are more predictable than wind forces, and these effects can be reduced by calculation to useful engineering parameters.[2,3] Air density differences are often significant in hot, dry, industrial areas such as foundries and steel mills. The combined effect of wind and temperature differences can be significant under certain conditions, and can be characterized quantitatively in some applications.

The design of modern industrial buildings has increased the problem of controlling man's working conditions. Older buildings often provided a significant amount of natural ventilation because they were tall and narrower in width than length. Heated air rose to the roof and was expelled at the top of the building, while replacement air entered the building at the lower perimeter. This type of design is still used in many hot industries such as foundries but obviously this arrangement cannot provide acceptable working conditions under all circumstances.

Mechanical General Ventilation. The modern large-area, low-height industrial plant and most multi-story buildings of masonry and glass construction, present entirely different ventilation problems. In these cases, natural ventilation forces are practically nil, and mechanical ventilation must be relied on almost completely. To this end, mechanical exhaust of contaminated air is required and mechanical air supply must be provided all year round to reach interior areas, provide adequate air distribution and prevent creation of negative pressures in the building.

In large open industrial buildings, general ventilation can be achieved by roof fans used with or without gravity ventilators. Where little or no heat is available to furnish natural ventilation or where roof head-room is low, roof fans should be used in place of gravity ventilators to provide a measure of general exhaust ventilation.

The best method of providing general ventilation in a closed building is to supply air through duct work and distribute it into the work areas in a manner that will provide both humidity and temperature control.

Local Exhaust Ventilation

A local exhaust system is one in which the contaminant being controlled is captured at or near the place where it is created or dispersed. In contrast to dilution or general ventilation, local ventilation places much more reliance on mechanical methods of controlling air flow. A local exhaust system usually includes the use of hoods or enclosures, ductwork leading to an exhaust fan, an air cleaning device for air pollution abatement and finally, discharge to the outside air. Local exhaust systems usually contain more mechanical components than general exhaust systems, offer more operational parameters to be controlled within acceptable ranges, and therefore require more maintenance.

The term "local exhaust" does not necessarily imply that the system is small. For example, the hood over a basic oxygen furnace in a steel mill is a local exhaust hood even though it may exhaust a million cubic feet per minute of air. A local exhaust system is usually superior to general ventilation if the main purpose of the ventilation is contaminant control. These advantages include the following:[2,4]

1. If the system is properly designed, the capture and control of a contaminant can be complete. Consequently, the exposure to workmen from the sources exhausted can be prevented. With general ventilation the contaminant has been diluted when the exposure occurs, and at any given workplace this dilution may be highly variable and therefore inadequate at certain times.
2. The volume rate of required exhaust is less with local ventilation; as a result, the volume of make-up air required is less. Local ventilation saves in both capital investment and heating costs.
3. The contaminant is contained in a smaller exhausted volume of air; therefore, if air pollution control is needed, it is less costly. As a first approximation, the cost of air pollution control is proportional to the volume of air handled.
4. Many local exhaust systems can be designed to capture large settleable particles or at least confine them within the hood and thus greatly reduce the labor required for good housekeeping.
5. Auxiliary equipment in the workroom is better protected from the deleterious effects of the contaminant, such as corrosion and abrasion.
6. Local exhaust systems usually require a fan of fairly high pressure characteristics to overcome pressure losses in the system. Therefore, the performance of the fan system is not likely to be adversely affected by wind direction or velocity, or inadequate make-up air, etc. This is in contrast to general ventilation which can be greatly affected by seasonal factors.

Glossary of Terms in Industrial Ventilation

The following list of terms have a special meaning within the field of industrial ventilation:[2,4,5]

Blast gate. A device for restricting airflow in a duct, usually consisting of a flat sliding plate which moves perpendicularly to the duct center line.

Capture velocity. The air velocity at a point within or in front of an exhaust hood necessary to overcome opposing air currents and particle inertia, causing the contaminated air to flow into the hood.

Coefficient of entry. The ratio of the actual rate of air flowing into an exhaust opening to the theoretical rate, calculated by assuming that the negative static pressure in the exhaust opening is completely converted to velocity pressure.

Control velocity. The air velocity required at the face of an enclosing hood to retain the contaminant within the hood.

Damper. A device for restricting the airflow in a duct, usually consisting of a flat disc mounted on a shaft which is perpendicular to the direction of airflow.

Entry loss. Loss in static pressure caused by air flowing into a duct or hood opening.

Entry loss factor. A factor derived from the coefficient of entry which, when multiplied by the velocity pressure at the hood, yields the entry loss in inches of water gauge.

Exhaust rate. The volumetric rate at which air is removed.

Grain. A unit of weight equal to 1/7000 of a pound.

Micron. A unit of length equal to 0.001 millimeter or 0.0000394 inches.

Plenum. A receiving enclosure for gases in which the static pressure at all points is relatively uniform.

Reynolds number. A dimensionless parameter computed by dividing the product of pipe diameter, average velocity and fluid density by the fluid viscosity.

Slot velocity. Linear flowrate of air through a slot.

Standard air. Dry air at 29.92 inches of mercury absolute pressure and 70°F, weighing 0.075 pound per cubic foot.

Static pressure. The pressure of a fluid exerted in all directions equal and opposite to the pressure tending to compress the fluid. In ventilation applications, static pressure is usually the difference between the absolute pressure in an exhaust system and atmospheric pressure, such that static pressure less than atmospheric pressure is termed "negative static pressure."

Still air. Air with a velocity of 25 feet per minute or less. Under practical circumstances there is always random air motion of 10 to 20 feet per minute even in rooms regarded as tightly constructed. This non-zero convection results from thermal circulation due to temperature differences, leakage in the building due to wind pressure and thermal head, and the ordinary movement of people.

Tempered air. Supply air which has been heated sufficiently to prevent cold drafts.

Total pressure. The algebraic sum of static pressure and velocity pressure.

Transition. A change in the cross-sectional shape or area of a duct or hood.

Transport (conveying) velocity. That velocity required to prevent the settling of a contaminant from an air stream, usually related to the flow of air in a duct.

Velocity pressure. The kinetic pressure due to flow, equal to that required to bring a fluid at rest to flow at a given velocity. Velocity pressure is always positive and in the direction of air flow.

FUNDAMENTALS OF VENTILATION AIR FLOW

The basic laws describing the complete motion of a fluid are complex and largely unknown. In the simple case of laminar flow the motion of the fluid may be computed analytically. However, for turbulent flow only a partial analysis can be made using the principles of fluid mechanics. The air flow in a local exhaust system is always turbulent to some degree; that is, the Reynolds number, an index of turbulence, is greater than 4,000. Therefore, the analytical solution to motion of air in exhaust systems is largely empirical and depends on experimental data.

Conservation of Mass

A basic consideration in the principles of air flow is the continuity equation or conservation of mass. This equation states that the mass rate of flow remains constant along the path taken by a fluid. Therefore, for any two points in a fluid stream:

$$A_1 v_1 \delta_1 = A_2 v_2 \delta_2$$

where A = cross sectional area, ft^2

v = velocity, ft/min.

δ = specific weight, lb/ft^3

In most applications in industrial ventilation, δ is relatively constant because the absolute pressure within a ventilation system usually varies over a very narrow range and the air remains relatively incompressible. Therefore,

$$A_1 v_1 = A_2 v_2$$

and

$$Q_1 = Q_2$$

where $Q = Av$, the volumetric rate of air flow, ft^3/min.

Conservation of Energy

The basic energy equation of a frictionless, incompressible fluid for steady flow along a single stream line is given by Bernoulli's Theorem:

$$H + \frac{P}{\delta} + \frac{v^2}{2g} = C$$

where H = the elevation above any arbitrary datum plane, ft

P = absolute pressure, lb/ft^2

δ = specific weight, lb/ft^3

v = velocity, ft/sec

g = gravitational acceleration, ft/sec^2

C = a constant, different for each streamline.

Each of the three variable terms in Bernoulli's equation has the units foot-pounds per pound of fluid, or feet of fluid, frequently referred to as elevation head, pressure head and velocity head, respectively.

When Bernoulli's equation is applied to industrial exhaust systems the elevation term, H, is usually omitted since only relatively small changes in elevation are involved. Since all streamlines originate from the atmosphere, a reservoir of nearly constant energy, the constant, C, is the same for all streamlines and the restriction of the equation to a single stream line can be removed. Furthermore, since the pressure changes in nearly all exhaust systems are at most only a few percent of the absolute pressure, the assumption of incompressibility may be made with negligible error.

Velocity Pressure

Air in motion exerts a pressure called velocity pressure. Velocity pressure maintains air velocity and is analogous to kinetic energy. It exists only when air is in motion, it acts in the direction of air flow, and it is always positive in sign. In Bernoulli's equation, the term $\frac{v^2}{2g}$ represents the velocity head. The relationship between the velocity of air and velocity pressure is:

$$v = \sqrt{2gh}$$

Where v = velocity, ft/sec
g = gravitational acceleration, ft/sec²
h = head of air, ft

When $g = 32.17$ ft/sec² and the density of air is 0.075 pound per cubic foot, then:

$$V = 4005\sqrt{VP} \quad \text{fpm}$$

where VP = velocity pressure, inches of water. Table 39-1 presents standard air velocity equivalents for velocity pressures between 0.01 and 14 inches of water.

Static Pressure

Static pressure produces initial air velocity, overcomes the resistance in a system caused by friction of the air against the duct walls, and overcomes turbulence and shock caused by a change in direction or velocity of air movement. Static pressure is analogous to potential energy and it exists even where there is no air motion. It acts equally in all directions and either tends to collapse the walls of the duct upstream from the fan or tends to expand the walls of the duct on the downstream side. Static pressure is usually negative in sign upstream from a fan and positive in sign downstream. It is measured as the difference between duct pressure and atmospheric pressure. The most common units of static pressure are inches of water.

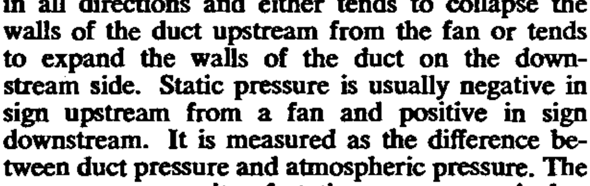
Total Pressure

The driving force for air flow is a pressure difference. Pressure is required to start and maintain flow. This pressure is called total pressure and has two components: velocity pressure and static pressure. Static pressure, velocity pressure and total pressure are all interrelated:

$$SP + VP = TP$$

Figure 39-1 shows the relationship between static, velocity and total pressure at different points in a duct system.

If gas flowing through a duct system undergoes an increase in velocity, a part of the available static pressure is used to create the additional velocity pressure necessary to accelerate the flowing gas. Conversely, if the velocity is reduced at some point in a duct system, a portion of the kinetic energy or velocity pressure at that point is converted into potential energy or static pressure. Static and velocity pressure are, therefore, mutually convertible. However this conversion is always accompanied by a net loss of total pressure due to turbulence and shock; i.e., the conversion is always less than 100% efficient.



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

Figure 39-1. Relationship Between Static Pressure, Velocity Pressure, Total Pressure.

Friction Losses

Air in motion encounters resistance along any surface confining the flowing volume. Consequently, some of the energy of the air is given up in overcoming this friction and is transformed into heat. The rougher the surface confining the flow or the higher the flow rate, the higher the frictional losses will be.

Frictional loss in a duct varies directly as the length, inversely as the diameter, and directly as the square of the velocity of air flowing through the duct. This loss can be calculated from charts[6] using the Fanning friction factor, which is an empirical function of Reynolds number, duct material and type of construction.

Dynamic Losses

Another type of energy loss encountered in air flow results from turbulence caused by a change in direction or velocity within a duct. The pressure drop in a duct system due to dynamic losses increases with the number of elbows or angles and the number of velocity changes within the system. The resulting pressure drop from these energy losses is expressed in units of "equivalent length." For example, an elbow of 12-inch diameter and 24-inch centerline radius is said to have an equivalent length of 17, meaning that the loss through the elbow will be the same as the loss through 17 feet of straight pipe with the same diameter operating under the same conditions (see *Industrial Ventilation*,[4] Fig. 6-11).

Another method of defining the losses due to turbulence and friction is to express the losses in terms of velocity pressure. For example, a loss of 0.28 VP in a transition or elbow means that the incremental pressure drop is equal to 0.28 of the velocity pressure of the air stream at that point (see *Industrial Ventilation*,[4] Figure 6-12, 6-13).

Acceleration and Hood Entrance Losses

This type of dynamic loss, h_e, is a drop in pressure caused by turbulence when air is accelerated from rest to enter a duct or opening. Turbulence losses of this type vary with the type of opening and are defined for ducts and common types of hoods in Fig. 4-8, *Industrial Ventilation*[4].

This entry loss plus the acceleration energy required to move the air at a given velocity (one VP) comprise the hood static pressure, SP_h. SP_h is expressed algebraically as:

$$SP_h = h_e + VP.$$

SP_h can be measured directly at a short distance downstream from the hood entrance. The calculation of SP_h is the first step in the design or evaluation of a local exhaust system, discussed in Chapters 41 and 42.

TABLE 39-1

Velocity Pressures for Different Velocities — Standard Air[4]

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

V = VELOCITY FPM
VP = VELOCITY PRESSURE, INCHES OF WATER

VP	V	VP	V	VP	V	VP	V	VP	V	VP	V
0.01	400	0.52	2888	1.03	4064	1.54	4970	2.05	5734	3.10	7051
0.02	556	0.53	2916	1.04	4084	1.55	4986	2.06	5748	3.20	7164
0.03	694	0.54	2943	1.05	4103	1.56	5002	2.07	5762	3.30	7275
0.04	801	0.55	2970	1.06	4123	1.57	5018	2.08	5776	3.40	7385
0.05	896	0.56	2997	1.07	4142	1.58	5034	2.09	5790	3.50	7492
0.06	981	0.57	3024	1.08	4162	1.59	5050	2.10	5804	3.60	7599
0.07	1060	0.58	3050	1.09	4181	1.60	5066	2.11	5817	3.70	7704
0.08	1133	0.59	3076	1.10	4200	1.61	5082	2.12	5831	3.80	7807
0.09	1201	0.60	3102	1.11	4219	1.62	5098	2.13	5845	3.90	7909
0.10	1266	0.61	3127	1.12	4238	1.63	5114	2.14	5859	4.00	8010
0.11	1328	0.62	3153	1.13	4257	1.64	5129	2.15	5872	4.10	8109
0.12	1387	0.63	3179	1.14	4276	1.65	5144	2.16	5886	4.20	8208
0.13	1444	0.64	3204	1.15	4295	1.66	5160	2.17	5899	4.30	8305
0.14	1498	0.65	3229	1.16	4314	1.67	5175	2.18	5913	4.40	8401
0.15	1551	0.66	3254	1.17	4332	1.68	5191	2.19	5927	4.50	8496
0.16	1602	0.67	3279	1.18	4350	1.69	5206	2.20	5940	4.60	8590
0.17	1651	0.68	3303	1.19	4368	1.70	5222	2.21	5954	4.70	8683
0.18	1699	0.69	3327	1.20	4386	1.71	5237	2.22	5967	4.80	8774
0.19	1746	0.70	3351	1.21	4405	1.72	5253	2.23	5981	4.90	8865
0.20	1791	0.71	3375	1.22	4423	1.73	5268	2.24	5994	5.00	8955
0.21	1835	0.72	3398	1.23	4442	1.74	5283	2.25	6008	5.10	9044
0.22	1879	0.73	3422	1.24	4460	1.75	5298	2.26	6021	5.20	9133
0.23	1921	0.74	3445	1.25	4478	1.76	5313	2.27	6034	5.30	9220
0.24	1962	0.75	3468	1.26	4495	1.77	5328	2.28	6047	5.40	9307
0.25	2003	0.76	3491	1.27	4513	1.78	5343	2.29	6061	5.50	9392
0.26	2042	0.77	3514	1.28	4531	1.79	5359	2.30	6074	5.60	9477
0.27	2081	0.78	3537	1.29	4549	1.80	5374	2.31	6087	5.70	9562
0.28	2119	0.79	3560	1.30	4566	1.81	5388	2.32	6100	5.80	9645
0.29	2157	0.80	3582	1.31	4583	1.82	5403	2.33	6113	5.90	9728
0.30	2193	0.81	3604	1.32	4601	1.83	5418	2.34	6128	6.00	9810
0.31	2230	0.82	3625	1.33	4619	1.84	5433	2.35	6140	6.10	9891
0.32	2260	0.83	3657	1.34	4636	1.85	5447	2.36	6153	6.20	9972
0.33	2301	0.84	3669	1.35	4653	1.86	5462	2.37	6166	6.30	10052
0.34	2335	0.85	3690	1.36	4671	1.87	5477	2.38	6179	6.40	10132
0.35	2369	0.86	3709	1.37	4688	1.88	5491	2.39	6192	6.50	10210
0.36	2403	0.87	3729	1.38	4705	1.89	5506	2.40	6205	6.60	10289
0.37	2436	0.88	3758	1.39	4722	1.90	5521	2.41	6217	6.70	10366
0.38	2469	0.89	3779	1.40	4739	1.91	5535	2.42	6230	6.80	10444
0.39	2501	0.90	3800	1.41	4756	1.92	5550	2.43	6243	6.90	10520
0.40	2533	0.91	3821	1.42	4773	1.93	5564	2.44	6256	7.00	10596
0.41	2563	0.92	3842	1.43	4790	1.94	5579	2.45	6269	7.50	10968
0.42	2595	0.93	3863	1.44	4806	1.95	5593	2.46	6282	8.00	11328
0.43	2626	0.94	3884	1.45	4823	1.96	5608	2.47	6294	8.50	11676
0.44	2656	0.95	3904	1.46	4840	1.97	5623	2.48	6307	9.00	12015
0.45	2687	0.96	3924	1.47	4856	1.98	5637	2.49	6320	9.50	12344
0.46	2716	0.97	3945	1.48	4873	1.99	5651	2.50	6332	10.00	12665
0.47	2746	0.98	3965	1.49	4889	2.00	5664	2.60	6458	11.00	13283
0.48	2775	0.99	3985	1.50	4905	2.01	5678	2.70	6581	12.00	13874
0.49	2804	1.00	4005	1.51	4921	2.02	5692	2.80	6702	13.00	14440
0.50	2832	1.01	4025	1.52	4938	2.03	5706	2.90	6820	13.61	14775
0.51	2860	1.02	4045	1.53	4954	2.04	5720	3.00	6937	14.00	14986

The coefficient of entry, C_e, is a measure of how efficiently a hood entry is able to convert static pressure to velocity pressure. The coefficient of entry is defined as the ratio of rate of flow by the hood static pressure to the theoretical flow if the hood static pressure were completely converted to velocity pressure. This term is computed as follows:

$$C_e = \frac{4005A\sqrt{VP}}{4005A\sqrt{SP_h}} = \frac{\sqrt{VP}}{\sqrt{SP_h}}$$

Pressure Drop through Ductwork

The result of the friction and dynamic losses to air flowing through ductwork is a pressure drop in the system. Bernoulli's Theorem can be restated in a simplified expression of conservation of energy as follows:[4]

$$SP_1 + VP_1 = SP_2 + VP_2 + \text{losses.}$$

Static pressure plus velocity pressure at a point upstream in direction of air flow equals the static pressure plus velocity pressure at a point downstream in direction of air flow plus friction and dynamic losses.

PROPERTIES OF AIRBORNE MATERIALS

Dusts

Dusts are solid particles generated by handling, crushing, grinding, and detonation of organic and inorganic materials such as rock, metal, coal, wood, and many other materials. Dust particles do not diffuse in air in the classical sense, but the finer particles of diameters <20 microns are carried with air currents because the settling rate is so low as to be of no practical significance. Dust particles must be about 5 microns or smaller to reach the lungs; larger particles are filtered out in the nasal passages or other parts of the breathing apparatus.

Fumes

Fumes are small, solid particles created by condensation from the gaseous state, generally after volatilization or by chemical reaction such as oxidation. Fumes are usually submicronic in size.

The outstanding characteristic of most fumes is their tendency to flocculate and coalesce. Very small spherical fume particles tend to attach themselves together in long chains or clumps of particles due to Brownian diffusion and electrostatic attraction.

Mists

Mists are suspended liquid droplets generated by condensation from the gaseous to the liquid state or by breaking up a liquid into a dispersion such as by splashing, foaming and atomizing. Fogs are similar to mists but the term is usually reserved for high concentrations of very fine droplets that are more frequently airborne.

Smoke

Smoke is the aerosol mixture which results from the incomplete combustion of carbonaceous material such as coal, oil, tar and tobacco.

Vapors

Vapors are the gaseous forms of substances which are normally in the liquid or solid state and which can be changed to these states either by increasing the pressure or decreasing the temperature alone.

Gases

Gases are normally compressible, formless fluids which occupy the space of an enclosure and which can be changed to the liquid or solid state only by the effect of increased pressure and decreased temperature or both.

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; hazardous dusts, 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.

APPLICATIONS OF DILUTION VENTILATION

When considering whether dilution or local exhaust is better, it should be remembered that dilution ventilation has four limiting factors:[4]

1. The quantity of contaminant generated must not be excessive or else the air volume necessary for dilution will be impractical.
2. Workers must be far enough away from contaminant evolution or else the contaminant must be in sufficiently low concentrations so that workers will not have an exposure above acceptable concentrations.
3. The toxicity of the contaminant must be low.
4. The evolution or generation of contaminants must be reasonably uniform.

On the basis of these factors, dilution ventilation is usually not recommended for fumes and dust because: (1) the high toxicities often encountered require excessively large quantities of dilution air; and (2) the velocity and rate of evolution are usually very high, resulting in locally high concentrations.

Dilution Ventilation for Comfort

Ventilation for heat relief includes certain aspects of air conditioning or treating of air to control simultaneously its temperature, humidity, cleanliness and distribution to meet the requirements of the conditioned space. In most residential, office and commercial ventilation the requirements are comfort for the occupants. In many industrial situations, however, comfort conditions are impractical to maintain; and the function of ventilation and air conditioning along with other control methods is to prevent acute discomfort and adverse physiological effects.

Exhaust ventilation may be used to remove heat and humidity if a source of cooler air is available. If it is possible to enclose the heat source, such as in the case of ovens or certain furnaces, a gravity or forced air stack may be all that is necessary to prevent excessive heat from entering the workroom.

Dilution Ventilation for Preventing Fires and Explosions

One function of dilution ventilation is to reduce the concentration of vapors within an enclosure to below the lower explosive limit. However, this concept must not be applied in cases where workers are exposed to the vapor. In such instances, dilution rates for health hazard control must always be applied. The reason for this distinction is fundamental and must be thoroughly understood.

Threshold Limit Values or Acceptable Concentrations for health hazard control are typically 1-3 orders of magnitude below the lower explosive limit for a given material. A table of comparative values is shown below.[4, 7]

Material	TLV	LEL	
	ppm	%	ppm
Acetone	1000	2.55	25,500
Ethanol	1000	3.28	32,800
Isopropanol	400	2.02	20,200
Toluene	100	1.27	12,700
Xylene	100	1.00	10,000

Exposure to atmospheres controlled to concentrations "below" the lower explosive limit (or some fraction thereof) could cause narcosis, severe illness or death. Therefore it is extremely important not to confuse dilution ventilation requirements for health hazard control with fire and explosion prevention.

Dilution Ventilation for Health

In general, dilution ventilation is not as satisfactory as local exhaust ventilation for primary control of health hazards. However, there are occasional circumstances in which dilution ventilation must be used because the operation or process prohibits local exhaust. Circumstances may occasionally be found in which dilution ventilation provides an adequate amount of control more economically than a local exhaust system. However, this is the exception. One should be careful, moreover, not to base economical considerations entirely upon the initial cost of the system because dilution ventilation invariably exhausts large volumes of heated air from a building. This can easily result in huge operating costs in the form of conditioned make-up air which will make the general ventilation scheme much more expensive over a period of time.

Dilution ventilation for the control of health hazards is used to best advantage in controlling the concentration of vapors from organic solvents of low toxicity. In order to successfully apply the principles of dilution to such a problem, data must be available on the rate of vapor generation or on the rate of liquid evaporation. Usually such data can be obtained from the plant if it keeps any type of adequate records on material consumption.

Rate of Air Change

The volume of a room to be ventilated and the ventilation rate are frequently related to one another by taking the ratio of the ventilation rate to the room volume to yield a "number of air changes per minute" or "number of air changes per hour." These are terms that are used quite frequently in discussions of ventilation requirements. Unfortunately, through widespread use over the years, they are more often employed incorrectly than properly.

It must be understood that ventilation requirements based on room volume alone have no validity. Calculations of the required rate of air change can only be made on the basis of a material balance for the contaminant under question. Similar calculations can be made for the rate of concentration increase or decrease; however, they require not only the air change rate but also the rate of generation of contaminant. In the design of industrial ventilation, "X number of air changes" has valid application only very rarely. The term is useful when applied to meeting rooms, offices, schools and similar spaces where the purpose of ventilation is simply the control of odor, temperature, or humidity and the only contamination of the air is from the activity of people.

Dilution ventilation requirements should always be expressed in cubic feet per minute or some other absolute unit of air flow, not in "Air Changes per Minute."

Calculating Dilution Ventilation

The concentration of a gas or vapor at any time can be expressed by a differential material balance, which, when integrated provides a rational basis for relating ventilation to the generation and removal rates of a contaminant.

Let C = concentration of gas or vapor at time t

G = rate of generation of contaminant

Q = rate of ventilation

K = design distribution constant, allowing for incomplete mixing

$Q' = Q/K$ = effective rate of ventilation, corrected for incomplete mixing

V = volume of room or enclosure

Starting with a fundamental material balance, assuming no contaminant is in the supply air,

Rate of Accumulation = Rate of Generation − Rate of Removal

$$VdC = Gdt - Q'Cdt$$

Consider the following applications:

1. *Concentration Buildup*

Rearranging the differential material balance,

$$\int_{C_1}^{C_2} \frac{dC}{G - Q'C} = \frac{1}{V}\int_{t_1}^{t_2} dt$$

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

if $C_1 = 0$ at t_1,

$$\ln\left(\frac{G - Q'C}{G}\right) = -\frac{Q'}{V}t$$

or,

$$\frac{G - Q'C}{G} = e^{-Q't/V}$$

Example A

$V = 100{,}000\ ft^3$
$C_1 = 0$
$Q = 6000$ cfm
$K = 3$
$Q' = 2000$ cfm
$G = 1.2$ cfm

How long before the concentration of the contaminant reaches 200 ppm?

Solution:

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

Example B

Using the same values as in the preceding example, what will the concentration of the contaminant be after one hour?

Solution:

$$C = \frac{G - G_e^{-\left(\frac{Q't}{V}\right)}}{Q'} = 419 \text{ ppm}$$

2. *Rate of Purging*

In the case where a volume 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:

$$VdC = -Q'Cdt$$

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

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

Example:

In the room of the examples above, 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 to 25 ppm?

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

$$\Delta t = 69.3 \text{ minutes}$$

3. *Maintaining Acceptable Concentrations at Steady State*

At steady state, $dC = 0$

$$G dt = Q'C dt$$

$$\int_{t_1}^{t_2} G dt = \int_{t_1}^{t_2} Q'C dt$$

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

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

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

$$Q = \frac{KG}{C}$$

Therefore, the rate of flow of uncontaminated dilution air required to reduce the atmospheric concentration of a hazardous material to an acceptable level can be easily calculated, if the generation rate can be determined. Usually the acceptable concentration is considered to be the Threshold Limit Value or Acceptable Eight-Hour Time Weighted Average Concentration. For liquid solvents the steady-state dilution ventilation requirement can be conveniently expressed as:

$$Q = \frac{(6.71)(10^6)(SG)(ER)(K)}{(MW)(TLV)}$$

Where: Q = actual ventilation rate, cfm
SG = specific gravity of volatile liquid
ER = evaporation rate of liquid, pints/hr
MW = molecular weight of liquid
K = design safety factor for incomplete mixing
TLV = Threshold Limit Value, ppm.

Example:
Methylene chloride is lost by evaporation from a tank at the rate of 1.5 pints per hour. How much dilution air is required to maintain the concentration below the TLV?

For methylene chloride,
TLV = 500 ppm
SG = 1.336
MW = 84.94

Assuming K = 5,

$$Q = \frac{(6.71)(10^6)(1.336)(1.5)(5)}{(84.94)(500)}$$

$$Q = 1583 \text{ cfm}$$

Specifying Dilution Ventilation

The foregoing discussion introduced the concept of a "design safety factor" (K) for calculating dilution ventilation requirements. This K factor is based upon several considerations:[5]

1. The efficiency of mixing and distribution of make-up air introduced into the room or space being ventilated.
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 judgment of any other circumstances which the industrial hygienist determines to be of importance, based upon experience and the individual problem. Included in these criteria are such considerations as:
   a. Seasonal changes in the amount of natural ventilation.
   b. Reduction in operation efficiency of mechanical air moving devices.
   c. Duration of the process, operational cycle and normal location of workers relative to sources of contamination.
   d. Location and number of points of generation of the contaminant in the workroom or area and,
   e. Other circumstances which may affect the concentration of hazardous material in the breathing zone of the workers.

The K value selected will usually vary from 3 to 10 depending on the above considerations.

Industrial Ventilation[4] (Table 2-1) lists the air volumes required to dilute the vapors of twenty-nine common organic solvents to the TLV level based upon the liquid volume solvent evaporated per unit time. These values must be multiplied by a K factor to allow for variations in uniformity of air distribution, and other considerations. Hemeon[6] includes a table of recommended dilution rates for fifty-three organic solvents. The "Ventilation Design Concentrations" in this table are not based on threshold limit values alone, but are based also on odor. All of the concentrations in this table are lower than the threshold limits, but those substances which are especially toxic or which have a very disagreeable odor have the greatest safety factors.

It must be emphasized that Threshold Limit Values are subject to revision, and the use of tables to estimate dilution values may become obsolete if the Threshold Limit Values are lowered; therefore, such a table should be used with caution, and with reference to the *latest* TLV list.

MAKE-UP AIR

All local and general exhaust ventilation systems must have air to exhaust, and by the basic consideration of conservation of mass, that air must be supplied pound for pound by a make-up air system. The supply and distribution of make-up air is often over-looked or neglected in the design of ventilation but remains fundamental to its successful operation.

Principles for Supplying Make-Up Air

1. The fresh air intake should be located away from any contaminating sources such as exhaust stacks or furnace exhausts. It is advisable to filter the fresh air to protect

the equipment and provide maximum heat exchange efficiency.

2. The air supply system must be provided with a fan, otherwise the room or building will be under a negative pressure.
3. Make-up air sources should be located to provide cross ventilation. In this way, the air can be "used twice." First, it will provide general dilution and secondly it will provide make-up air for the exhaust systems. This does not apply for spot-cooling applications where the air will be introduced directly at the work station and may vary significantly from room temperature. The air distribution pattern must be engineered carefully to provide effective area coverage without excessive cross drafts which will interfere with the workers or the existing systems.
4. Make-up air should be introduced into the "living zone" of the plant, that is, below the 8-10 foot level. In this manner, the air is used first by the people and the best results of general or dilution ventilation are obtained. This distribution also provides closer control of the ambient working temperature.
5. Make-up air temperature is usually heated or cooled to approximately the same as desired in the room being supplied. Since the air is being used for ventilation and for replacement purposes, the usual temperature range will be 65-80°F.

Recirculation of Air from Industrial Exhaust Systems

Plant Circulation. It should be apparent that if large amounts of air are exhausted from a room or building in order to remove obnoxious dusts, gases, fumes or vapors an equivalent amount of fresh, tempered air should be supplied to the room. This supplied air must be heated in cold weather, and heating costs may be large if sizable amounts of air are handled. Therefore, attempts are sometimes made to eliminate such heating costs by appropriate cleaning of the exhausted air and subsequent recirculation of the air into the room. The acceptability of such recirculation depends on the degree of health hazard associated with the particular contaminant being exhausted as well as on other factors.

It is generally accepted as good practice not to recirculate exhaust air if the contaminants therein can have an effect on the health of the worker. The reasons are:

a. Many types of air cleaners do not collect toxic contaminants efficiently enough to eliminate the health hazard. This is especially true for gases and vapors.
b. Poor maintenance of the air cleaner would result in the return of highly contaminated air to the breathing zones of the workers. One of the facts of life is that air cleaners are not generally production equipment, and are too often poorly maintained or not maintained at all.
c. Improper operation of the air cleaner through mechanical failure or through ignorance or neglect on the part of the operators can also result in the return of highly contaminated air.

In general, recirculation should be avoided unless the reasons indicating its use are truly compelling. Its use will always require an understanding of the hazardous nature of the contaminant, the knowledge of the performance of the specific air cleaner, the general ventilation scheme, and the judgment of experience.

Unplanned Recirculation. Unplanned recirculation of exhausted air can be a serious problem. This usually results from inadequately separated exhaust stacks and air inlets or insufficient discharge height, either from short stacks relative to the roof line or low "effective" stack height resulting from poorly designed weather caps on stack heads. This subject is treated in more detail in Chapter 42.

AIR MOVING DEVICES

The term "air moving device" is an inclusive one which denotes machines more commonly known as fans, blowers, exhausters, turbo compressors and ejectors. By definition, an air moving device (AMD) is a power-driven machine causing a continuous flow of air. In more practical terms, the air moving device manufacturer, to gain acceptance for his product, must earn membership to the "Air Moving and Conditioning Association" by subjecting his product to its test code for air movers. In addition, air moving devices manufacturers must furnish a prospective buyer of his product, certain data relative to the product and its applications. This subject is covered in more detail in Chapter 42.

Because fans are the most commonly used exhausters in ventilation for industrial hygiene purposes, they alone are considered in this discussion.

In the field of industrial ventilation, two major types of fans are used: axial flow types, where the air flow is parallel to the fan shaft, and centrifugal type, where air flow is perpendicular to the fan shaft. These two major fan types can be further defined by listing the various units that comprise each type. The components of an air moving device which influence its performance most are the wheel and the air inlet. This will be apparent in the list below which shows comparative components for the two major types of air moving devices.

Axial Flow Fans

1. Propeller: This is an AMD with a propeller or disc wheel within a mounting ring or plate through which the air flow is predominantly parallel to the axis of rotation. This unit is used to move large volumes of air at relatively low velocity against a low static pressure (0-½"W.G.) and low temperature. This fan is commonly used for general ventilation.

2. Duct Fan: This is a step up in the evolution of fans from the propeller fan, in that it constitutes a propeller mounted inside a section of duct. The improvements gained in this configuration are elevated temperature applications (to 600°F in belt-drive units), and higher static pressures (from 0-2" W.G.)
3. Tube-Axial: This is an axial fan without guide vanes. It is used for medium to large volumes against static pressures up to 4" W.G. and temperatures up to 600°F in belt-driven units. This fan is best suited for moving an air stream containing materials that will not collect on fan blades.
4. Vane-Axial: This is an axial fan with either inlet or discharge vanes or both, which impart greater efficiency in delivering medium to high volumes against static pressures up to 8" W.G. Higher pressures are attainable in some units with variable pitch impellers. Temperature applications up to 600°F are attainable in belt-driven units. This type of fan is commonly used for mine ventilation or industrial systems whose characteristics vary widely.
5. Axial-Centrifugal: This is an axial fan with a centrifugal wheel. The wheel is available in both backwardly inclined and airfoil design. This fan is the latest improvement in axial flow air moving devices in that it combines the high efficiency of the axial unit with the quiet operation and high static pressure level of the centrifugal fan. Although temperatures of operation are relatively low (under 200°F), static pressures up to 20" W.G. are attainable. In addition, the non-overloading feature of the backwardly inclined and airfoil design wheels is an important advantage of this fan.

Centrifugal Fans

1. Radial Wheel: This fan is the workhorse of the dust control industry. It is the least efficient centrifugal fan type, but is quite suitable for rough service, including material handling applications. Generally it it is used to handle volumes to about 100,000 CFM at static pressures to 20" W.G., and temperatures up to 1000°F.
2. Forward-Curved Blade: This fan delivers high volumes at low static pressure with relatively low noise levels. However, it is not very efficient, and therefore, has lost much of its favor to other type units.
3. Backward-Inclined Blade: Sometimes known as "power limiting" or "non-overloading," this type wheel is used more and more for general air handling. It is an efficient fan, with high top speeds and is a good choice for handling large volumes of clean air.
4. Airfoil Design: This unit is the latest development in centrifugal fan wheels. The airfoil wheel was developed to reduce noise levels; however it is also backwardly inclined and has the non-overloading feature. It, too, can deliver large volumes against high static pressures.

Fan Selection

In order to select the proper fan for a given application, the following information is required:[4]

1. Air volume to be moved;
2. Fan static pressure;
3. Nature and extent of airborne particulate (a radial-bladed centrifugal fan would be needed if the air stream contains a high concentration of particulates);
4. Direct or belt-drive (belt-drive can be changed for variation in air volume handled; direct drive is inflexible, but occupies less space and requires less maintenance);
5. Noise Level — (a function of tip speed, it is usually a limiting factor in industrial applications);
6. Special considerations such as high operating temperature, corrosiveness, flammable or explosive materials, and space limitations.

The application of specific fan types in the design of exhaust systems is described in Chapter 42, after a more thorough discussion of local exhaust ventilation in Chapter 41.

References

1. POWELL, C. H., "Dilution Ventilation," *The Industrial Environment — Its Evaluation and Control,* Public Health Service Publication 614, Second edition, 1965.
2. AMERICAN IRON AND STEEL INSTITUTE, COMMITTEE ON INDUSTRIAL HYGIENE, *Steel Mill Ventilation,* AISI, 150 East 42nd Street, New York, New York, 1965.
3. AMERICAN SOCIETY OF HEATING, REFRIGERATION AND AIR CONDITIONING ENGINEERS, INC., *ASHRAE Guide and Data Book — Fundamentals and Equipment,* ASHRAE, Inc., New York, 1963.
4. AMERICAN CONFERENCE OF GOVERNMENTAL INDUSTRIAL HYGIENISTS, COMMITTEE ON INDUSTRIAL VENTILATION, *Industrial Ventilation — A Manual of Recommended Practice,* ACGIH, P.O. Box 453, Lansing, Michigan, 12th Edition, 1972.
5. AMERICAN NATIONAL STANDARDS INSTITUTE, *Fundamentals Governing the Design and Operation of Local Exhaust Systems,* ANSI, Z-9.2 Committee, 1040 Broadway, New York, New York.
6. PERRY, J. H., editor, *Chemical Engineers Handbook,* McGraw-Hill Book Company, Inc., New York, 1960, Fourth edition.
7. AMERICAN CONFERENCE OF GOVERNMENTAL INDUSTRIAL HYGIENISTS, *Threshold Limit Values for 1972,* ACGIH, 1014 Broadway, Cincinnati, Ohio 45202, 1972.
8. HEMEON, W. C. L., *Plant and Process Ventilation,* 2nd edition, The Industrial Press, New York, 1963.

CHAPTER 40

INSTRUMENTS AND TECHNIQUES USED IN EVALUATING THE PERFORMANCE OF AIR FLOW SYSTEMS

Richard D. Fulwiler, Sc. D.

INTRODUCTION

Objectives

It is the objective of this chapter to discuss the topic of evaluation of air flow systems as well as provide insight on practical approaches to system evaluation, utilizing up-to-date instrumentation and techniques. The material presented herein is directed to the student as well as the practicing industrial hygienist and will embrace instruments and techniques in evaluating both local exhaust and general ventilation systems.

This chapter will not delve deeply into the theory of air flow, but will require the reader to have a basic understanding of the general principles of ventilation such as the principles of air flow, pressure drops through ducts, and characteristics of blowing and exhausting. These subjects are discussed in some detail in Chapter 39, "Principles of Ventilation."

Bases for System Evaluation

To assure adequacy of design and performance. Independent of the type of system being evaluated, it should be constructed in accordance with a design basis be it a sophisticated engineering approach or the sketches of a sheet metal fabricator. Once a system is completely installed, various air flow measurements should be made before any aspect of the process or area to be controlled has an opportunity to affect the air flow characteristics of the system.

Sufficient input must go into the design basis to assure adequacy of system performance in the control of occupational hazards. Factors to be considered are toxic vapors, gases or dusts; nuisance materials or conditions; and explosive or flammability hazards. Assessment of the hazard in terms of environmental monitoring in conjunction with an evaluation of the air flow system immediately after start up is one sure way to test the adequacy of the design and installation.

To assure system performance is maintained. Since in many instances air flow systems are not an integral part of the process in terms of production or output, it is essential to run periodic checks on their performance. This is especially true for systems which have dampers, blast gates, etc., as well as those which may be affected by accumulations of the material they are conveying or controlling.

Maintenance of the entire air flow system from the entry to the exhaust stack cannot be overemphasized. Various air flow measurements discussed below will prove invaluable in determining the adequacy of, or the need for, maintenance.

To determine the feasibility for expanding (adding to) the system. The performance of many well designed and installed air flow systems is rendered inadequate by irrational expansion of the system. Various air flow measurements will provide input for judicious expansion while still maintaining the performance initially designed into the system. In many instances, relatively minor modifications may be indicated by such measurements.

To establish improved design parameters for new systems. The evaluation of existing systems may provide valuable input related to the specific operating conditions and characteristics of the hazards being controlled which may lead to improved design of new systems. For example, higher conveying velocities may be indicated to prevent ducts from becoming clogged, or turbulence and eddy currents at a hood entry resulting in contaminant escape, may require improved hood design.

To assure compliance with federal, state, or other regulations. Obviously, the purpose of any air flow system is either the control of hazardous chemical or thermal stresses, or providing a comfortable work environment. To assure that the former purpose is met, some regulations (laws) include air flow performance parameters which must be met. Chapter 1 includes a discussion of the 1970 Occupational Safety and Health Act. The initial package of standards embraced by this Act includes standards requiring certain minimum air flow requirements be maintained. It is not the purpose of this chapter to discuss paragraph 1910.-94 Ventilation[1] in detail, but merely to cite examples requiring minimum air flow parameters.

Minimum exhaust volumes for grinding wheels (Table G-4 of reference 1).

Minimum maintained velocities in spray booths (Table G-10 of reference 1).

Control velocities for undisturbed locations for open surface tanks (Table G-14 of reference 1).

Minimum ventilation rate for lateral exhaust of open surface tanks (Table G-15 of reference 1).

EVALUATING AIR FLOW SYSTEMS

Introduction

As there are numerous reasons for evaluating systems, there are degrees to which they may need to be evaluated. Instruments and techniques are

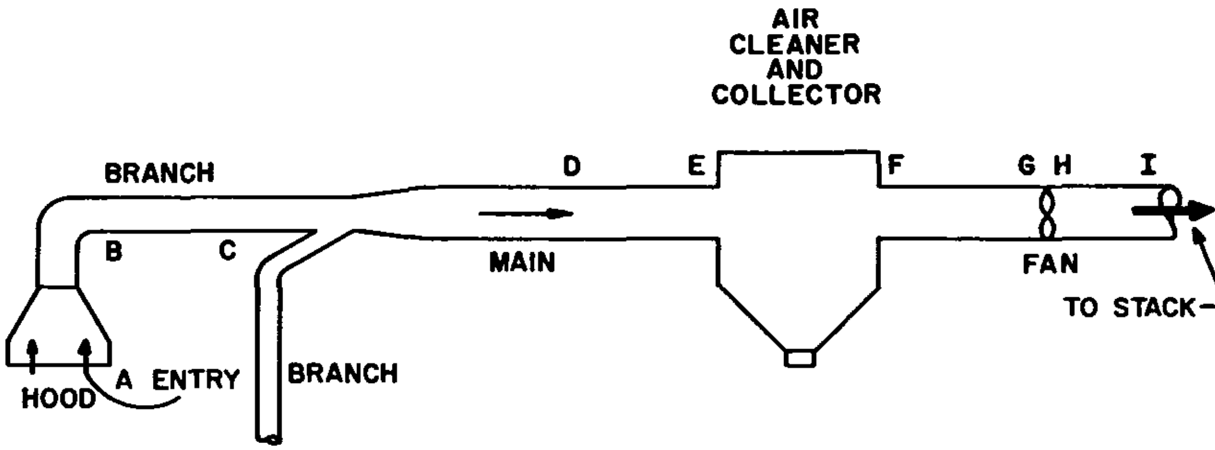
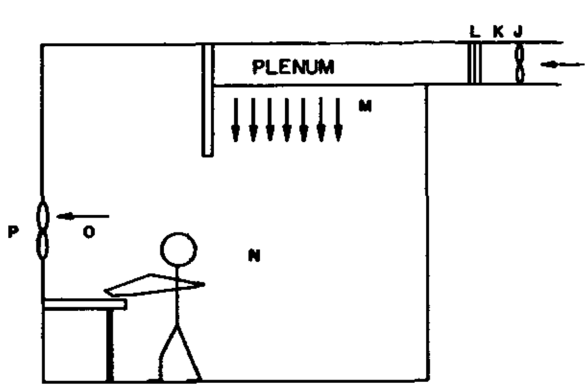


Figure 40-1. Schematic of a Local Exhaust System.

described below which may provide only a cursory evaluation of part of the system or an in-depth survey of the total system.

Figures 40-1 and 40-2 represent the two generic systems requiring air flow evaluation. Figure 40-1 is a schematic of a local exhaust system and Figure 40-2 is a schematic of a general room ventilation system[2] with the locations requiring evaluation identified by A, B, C, etc.

The instruments and techniques described in this chapter are in the order of increasing precision and accuracy. This order is reflected by the degree of interaction the instruments and techniques have with the system. The reader will note that visualization, the first approach discussed, barely interacts with the total system, whereas the final approach is actually titled "Measurements within the System." However, one should not lose sight of the fact that only by applying a combination of instruments and techniques can the total performance of any air flow system be evaluated.



American Conference of Governmental Industrial Hygienists — Committee on Industrial Ventilation: Industrial Ventilation Manual, 11th Edition. Lansing, Michigan, 1970.

Figure 40-2. Schematic of a General Room Ventilation System.

Visualization — A Cursory/Qualitative Estimate of Performance.

Purpose.

Visualization is an extremely important aspect of performance evaluation which heretofore has rarely been discussed in chapters or publications on air flow measurements. The primary application of visualization is as a cursory estimate of local exhaust system performance. Moreover, it provides an extremely effective technique for demonstrating the pluses of good design and the minuses of poor design to management and engineering personnel. Visualization techniques can also be used in training operating personnel in the proper use of the ventilation system.

Instruments/Materials.

"Smoke tubes" is a descriptive term applied to a glass tube containing titanium tetrachloride adsorbed on a granular medium. When the ends of the glass tube are broken and air passed through the tube, the moisture in the air reacts with the $TiCl_4$ to generate hydrochloric acid "smoke."

CAUTION: Direct inhalation of this "smoke" will be irritating to the respiratory system and should be avoided.

A squeeze bulb, rubber tubing, and the tubes can be purchased from numerous suppliers (Table 40-1) and are relatively inexpensive.

Titanium tetrachloride is a chemical reagent available through standard chemical supply companies. It is also available in single use glass ampules (Table 40-1).

CAUTION: These fumes and the liquid are corrosive to the skin, and irritating to the eyes and respiratory system.

Because of the nature of this material, glass ampules are recommended.

Smoke candles are available in a range of sizes and a few colors. They can be purchased in terms of the cubic feet of smoke produced or the duration of smoke evolution (Table 40-1). *Other sources of visualization media.* There are numerous other means of generating visual clouds

TABLE 40-1

Instruments/Purposes/Suppliers

Instruments	Purposes	Suppliers*
Smoke Tubes Titanium Tetra-chloride Smoke Candles	Visualization of air flow in and around exhaust hoods. Quick indication of room pressure. Demonstrate general room air flow patterns.	E. Vernon Hill Co. Mine Safety Appliances Co. National Environmental System
Rotating Vane Anemometers	Measures air velocity.	Bendix Environmental Science Division E. Vernon Hill Co.
Vane Anemometers	Measures air velocity (some have attachments for static pressure).	Alnor Instrument Co. E. Vernon Hill Co. Bachrach Instrument Co.
Heated Thermocouple Anemometers	Measures air velocity (some are applicable for non-directional air velocity).	Hastings-Raydist Co. Alnor Instrument Co.
Heated Wire Anemometers	Measures air velocity, static pressure, temperature.	Anemostat Products
Pitot Tubes	Measures total, velocity, and static pressure.	Western Precision Co. Dwyer Instruments, Inc. Meriam Instrument Co. E. Vernon Hill Co. Ellison and Co.
Manometers	Measures total, velocity, and static pressure.	Dwyer Instruments, Inc. Meriam Instrument Co. E. Vernon Hill Co.
Aneroid Gauges (Magnehelic)	Measures total, velocity, and static pressure.	Dwyer Instruments, Inc.
Transducers	Used for remote readings and when rapid changes in pressure must be maintained.	Hastings-Raydist Co.

*This represents neither a complete list nor endorsement.

to follow air flow. A "heavier than air" cloud can be generated by simply placing dry ice in an alcohol bath. A "lighter than air" cloud can be generated by blowing air through a smoldering mixture of sawdust and oil[3].

Techniques. Visualization media are best suited for the evaluation of air flow patterns and velocities at exhaust entries and supply outlets.

1. Smoke tubes

Smoke tubes can be carried with the industrial hygienist on any of his surveys or inspections. They can be used best as an immediate survey type tool in assessing the ability of a local exhaust system to capture contaminants. Smoke should be administered close to the hood entry initially, and gradually the smoke source moved away from the entry to observe the sphere of containment the exhaust system produces. Larger quantities of smoke can be generated inside of the hood or enclosure to estimate rate of clearance as well as to check for eddy currents, reverse air flows, and escapement. Small amounts of smoke can be used to estimate the force and direction of air from outlets as well as a qualitative check of the performance of return air outlets.

2. Titanium tetrachloride

Titanium tetrachloride is used best by swabbing it along the periphery of hoods as a check for eddy currents, reverse air flow, and lack of control. Once swabbed inside of a hood, the smoke will persist for several seconds and thus provide an opportunity for prolonged observation or photographs.

3. Smoke candles

Smoke candles can be used to estimate clearance rates and containment of large hoods such as paint spray booths, laboratory hoods, or other high volume exhaust systems. Minimal performance of the system must be determined before igniting a smoke candle to assure reasonable removal of the smoke. Smoke candles can be held by forceps and moved across hood face openings to estimate the air distribution at the face. Colored smoke can be introduced in ventilation systems downstream from the fan to check for leaks.

Limitations. There are two significant limitations. First, visualization is strictly qualitative and does not provide any information in terms of design or performance specifications. Second, the materials used may be hazardous or at the very least — a nuisance; thus their use in occupied areas should be somewhat restricted.

Air Velocity and Flow — Exhaust and Supply Openings

Relationship of velocity to rate of flow. The velocity to rate of flow relationship is expressed in Equation 1.

Equation 1: $Q = AV$

Where: Q = Rate of flow in cfm

V = Average linear velocity in fpm

A = Cross-sectional area of the duct or hood in ft^2

From this equation it is possible to calculate air flow rate if the velocity (V) and cross-sectional area (A) are known.

The purpose of a local exhaust system is to capture and convey the contaminant from the source through an air cleaner to the atmosphere. Precise measurements of capture velocities as well as estimates of exhaust or supply volumes can be made at the point where the air flow system interacts with the work environment. These points are identified as "A" on Figure 40-1 and "M" and "O" on Figure 40-2.

Instruments/Techniques. In using the instruments described below, the need to take multiple measurements of a given slot, hood, or diffuser must be kept in mind. Only by making a uniform traverse of the opening being evaluated will one be able to arrive at a satisfactory average velocity to use in the calculation of air flow. The reader should review Pitot traverse techniques covered in this chapter, and in greater detail in reference 4, to develop an appreciation for multiple measurements in evaluating air flow.

TABLE 40-2.

Correction factors for rotating vane anemometers.

Opening	Correction Factor*
Pressure openings, more than 4 in. wide, up to 600 sq. in. area, with free opening 70% or more of gross area, no directional vanes.	1.03
Suction opening, more than 4 in. wide, up to 600 sq. in. area, with flange 2 in. wide, free-open area 60% or more of gross area	0.85

Volume: For suction openings, cfm = (factor) (velocity) (gross area)

For pressure openings, cfm = (factor) (velocity) $\left(\frac{\text{gross area} + \text{net area}}{2}\right)$

*If the opening is covered with a grille, the instrument should touch the grille face but should not be pushed in between the bars. For a free opening without a grille, the anemometer should be held in the plane of the entrance edges of the opening. The anemometer must always be held in such a manner that the air flow through the instrument is the same direction as was used for calibration (usually from the back toward the dial face).

From "Industrial Ventilation — A Manual of Recommended Practices", Committee on Industrial Ventilation, American Conference of Governmental Industrial Hygienists, Lansing, Mich 1970.

TABLE 40-3.

Correction factors for swinging vane anemometers.

Opening	Correction Factor
Pressure	
More than 4 in. wide and up to 600 sq. in. area, free opening 70% or more of gross area, no directional vanes. Use free-open area.	0.93
Suction	
Square punched grille (use free-open area)	0.88
Bar grille (use gross area)	0.78
Strip grille (use gross area)	0.73
Free open, no grille	1.00
Volume: cfm = (factor) (area) (velocity)	

From "Operating Instructions for the Alnor Series 6000-P Velometer" Alnor Instrument Company, Chicago, Ill., 1970.

1. Rotating vane anemometers

The rotating vane anemometer is comprised of a vane or propeller on a shaft connected to gears. The air movement causes the vane to rotate. The revolutions of the vane turn the gears which register the revolutions on the dial of the instrument as linear feet. Readings are usually taken over one-minute periods, thus giving air velocity in linear feet per minute. These instruments are available in a number of sizes, however the most common vane sizes are 3, 4, and 6 inches in diameter (Table 40-1).

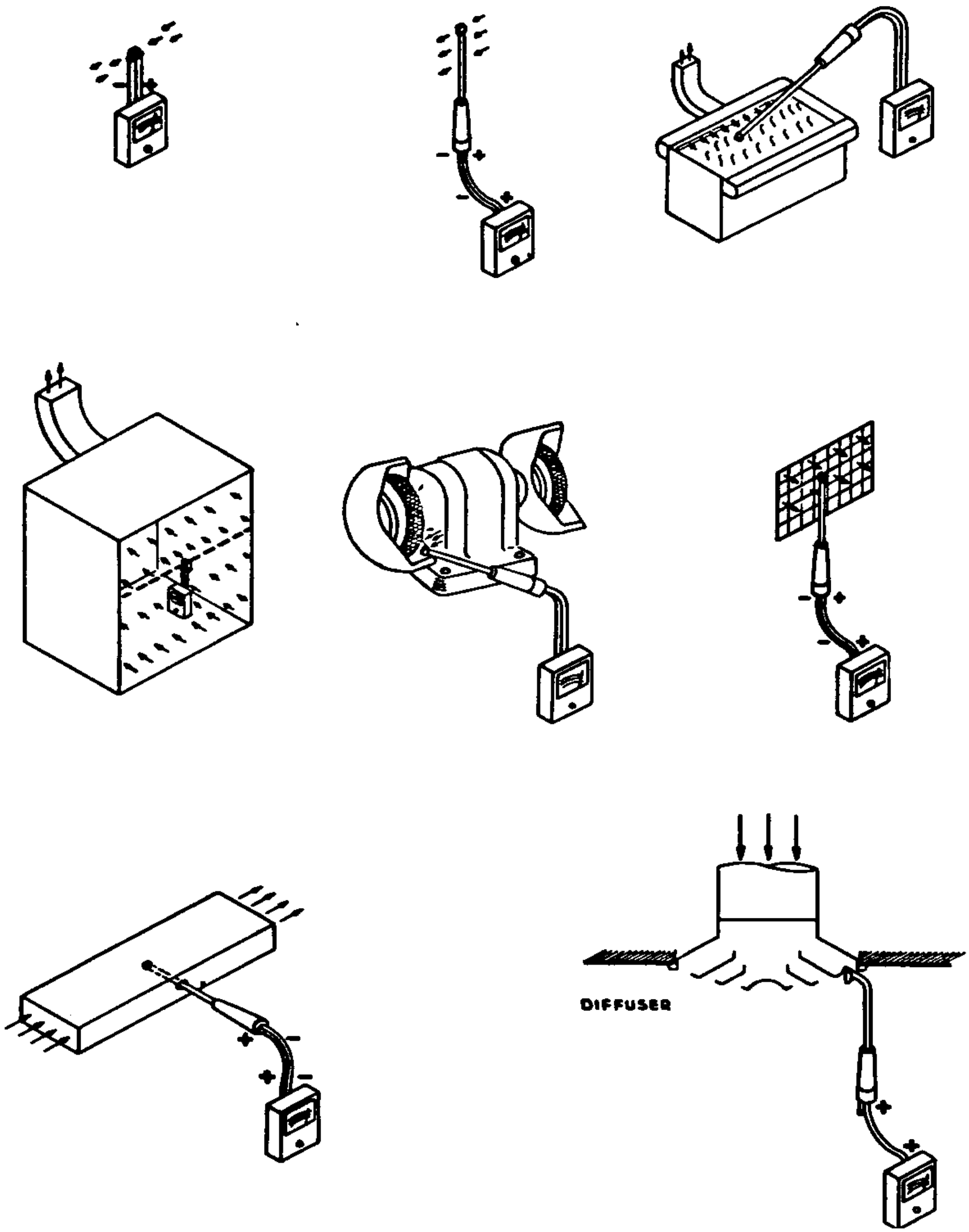
These instruments are best suited for determining air velocities and estimating air flow through large openings such as mine shafts and air supply and discharge grilles. Readings are generally obtained by traversing the opening at a uniform rate for a given period of time. Table 40-2 provides information on correction factors, techniques for taking measurements as well as the equations for calculating air flow rate.

The optimum range for these instruments is between 100 and 3000 fpm. They are best suited for use in relatively clean air and require the use of a timing device. They require frequent calibration and must be handled with care.

2. Swinging vane anemometer

The swinging vane anemometer indicates air velocity as a function of the pressure exerted by the air stream against a spring loaded swinging vane. They are quite portable and used extensively by industrial hygienists and ventilation engineers in the field (Table 40-1).

These instruments are used primarily for measuring velocities of exhaust or supply openings as shown in Figure 40-3. Fittings are available for some swinging vane devices which allow these to be used for a number of applications, such as measuring static pressures, as well as over a wide



Alnor Instrument Company, Chicago, Illinois.

Figure 40-3. Applications of a Commercially Available Swinging Vane Anemometer (Alnor Instrument Company).

range of velocities from 25 to 24,000 fpm. The optimum range of performance is 100 to 10,000 fpm.

Openings must be traversed with velocity measurements taken uniformly over the face of the the grille or hood being evaluated. Care must be taken to use probes for small openings (less than 3 ft^2) since the size of the device could represent a significant portion of the area. It is important to follow the operating instructions provided with the instrument, otherwise significant errors can be introduced. Correction factors for supply (pressure) and exhaust openings and information on calculating air flow are shown in Table 40-3.

When air temperatures vary more than 30°F from 70°F and/or the altitude is greater than 1000 ft. above sea level, readings should be corrected in accordance with the following equations:

Equation 2: $$Vt = Vr\sqrt{\frac{460+t}{530}}$$

Where: Vt = true velocity
Vr = velocity read from meter
t = air stream temperature

Equation 3: $$Vt = Vr\sqrt{\frac{1}{d}}$$

Where: $$d = \left(\frac{530}{460+t}\right)\left(\frac{B}{29.92}\right)$$

B = barometric pressure in inches of Hg

Caution must be exercised in using these devices in a dusty, moist, or corrosive atmosphere, since it can affect the readings as well as impair the performance of the instrument. They require periodic calibration but are a good choice for general and field applications.

3. Heated thermocouple anemometers

The operating principle of the heated thermocouple anemometer instruments is simply that air moving past a heated object removes heat. The amount of heat removed is proportional to the quantity of air passing which is a function of velocity. These instruments have one or more thermocouples as sensing elements which are heated by either alternating or direct current. A change in air flow causes a change in temperature of the thermocouples, resulting in a change in the direct current output. Another unheated thermocouple is in the direct current circuit to a meter. As a result of the changes in temperature, a change in voltage is developed which is read as air velocity.

These instruments are usually comprised of a single probe connected to an operating unit housing the circuitry, meter, batteries, etc., and are about the size of a cigar box. They are quite portable and commercially available (Table 40-1). The units incorporate balancing circuits which render errors due to radiant heat and ambient temperature fluctuations negligible. Since these instruments are direct reading and have a short response time (less than one minute), they are applicable for field as well as laboratory use. Some of the instruments may be used in determining non-directional velocities such as general room air movement, depending upon the type of shielding around the sensors in the probe.

The limitations are related primarily to maintaining the integrity of the probe. Heavy dust loadings or corrosive materials as well as mechanical shock could damage the delicate wires in the probe. The range of air velocities is quite wide, with some instruments having advertised ranges from 10 to 10,000 fpm. A general rule of thumb is that velocities from 10 to 50 fpm may be estimated while velocities from 100 to 2000 fpm may be measured with some precision depending on the calibration of the unit. Periodic calibration is required.

These devices, as with any probe type velometer, can be used to measure entry or exit velocities of hoods, slots, grilles, etc., by placing the probe perpendicular to the direction of air flow and recording the measurements. The greater the number of measurements distributed uniformly across the opening being measured, the better the estimate of air velocity and flow rate will be.

4. Heated wire anemometers

The heated wire anemometer devices depend upon the change in resistance of a wire with a change in temperature. The degree of temperature change is proportional to the velocity of air passing the wire. Velocity is read directly on a meter which is actuated by a change in voltage from a Wheatstone bridge circuit.

Generally, the advantages and limitations are the same as those previously described for heated thermocouple anemometers.

There are some units available which can measure temperatures ranging up to 250°F and static pressures up to 10 inches of water. More precise hot wire anemometers measure velocity exclusively. Static pressure measurements will be discussed in more detail in the section titled "Measurements within the System."

5. Other thermal anemometers

a) Heated thermometer anemometer

The principle for the heated thermometer anemometer is the same as for heated thermocouple anemometers except that two thermometers are used instead of thermocouples. It is not amenable to field use because of the fragility of the thermometers and the amount of time required for the thermometers to reach equilibrium. Its primary use is that of a laboratory type device in the calibration of air samplers requiring negligible static pressure losses in the calibration train (e.g., electrostatic precipitators).

b) Kata thermometer[7]

The Kata thermometer is a special thermometer with a large bulb, containing alcohol, and a stem with marks at 95 to 100°F. It is heated above 100°F and the time required for it to cool from 100 to 95°F is a measure of the non-directional air velocity in the room. It was designed for comfort ventilation measurement, and its surface-to-volume ratio is similar to that of the human body. The useful velocity range is 25 to 500 fpm. It has the disadvantage of being fragile and having large radiation and convection areas. A silvered bulb is necessary to minimize the effect of thermal radiation. As expected, it has a slow response time. The disadvantages are sufficient to classify this device as a poor choice for the measurement of air velocity.

MEASUREMENTS WITHIN THE SYSTEM

Precise measurements which characterize the performance of air flow systems are made within the system. Instead of measuring the velocity of air going into exhaust hoods or coming from air outlets, measurements are made *inside* the ductwork leading to the point of entry or discharge. Such locations are depicted as B, C, D, E, F, G, H and I on Figure 40-1 and J, K and L on Figure 40-2. Measurements within the system are made to determine static pressure drops associated with hood entries, ducts, across air cleaners (e.g. filters and bag houses) as well as velocity pressures. Therefore, the instruments discussed will be primarily those used to measure pressure in terms of inches of water.

Some of the anemometers previously discussed, those with relatively narrow probes and fittings, can also be used to measure air velocities within the system. Independent of the measuring device, the accuracy in determining either duct velocities or flowrate is dependent upon the location and number of measurements taken in traversing the duct. Techniques discussed under "Pitot Traverse" below can be applied to the use of most measurements requiring multiple sensing points within an air flow system.

Relationship of Velocity to Pressure Measurements

The total pressure of a system is the algebraic sum of the static pressure plus the velocity pressure. Air velocity can be computed from the velocity pressure according to equation 4.

Equation 4: $$V = 1096\sqrt{\frac{VP}{0.075d}}$$

Where: V = velocity in fpm
VP = velocity pressure in inches of water
d = density factor equal to:

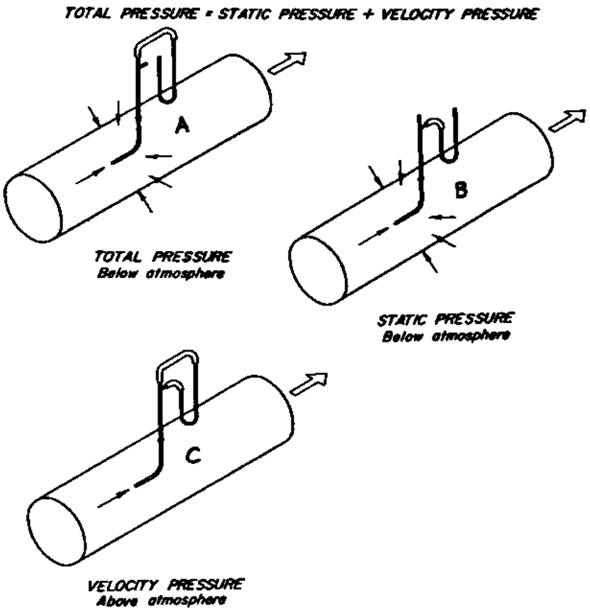
$$\frac{530}{460+t} + \frac{B}{29.92}$$

Where: B = barometric pressure in inches of mercury
t = air temperature in degrees F.

For air at standard conditions (70°F, 29.92" Hg), the density factor will equal unity. The equation then becomes:

Equation 5: $$V = 4005\sqrt{VP}$$

Table 1 of Chapter 39 can be used in converting velocity pressures at standard conditions to velocities in fpm. A "rule of thumb" in making corrections for density is that they should be made when the altitude is greater than 1000 ft. above sea level, the temperature of the air in the system is ± 30°F from standard, and the moisture content equal to or greater than 0.02 lb./lb. of dry air.



American Conference of Governmental Industrial Hygienists — Committee on Industrial Ventilation: Industrial Ventilation Manual, 11th Edition. Lansing, Michigan, 1970.

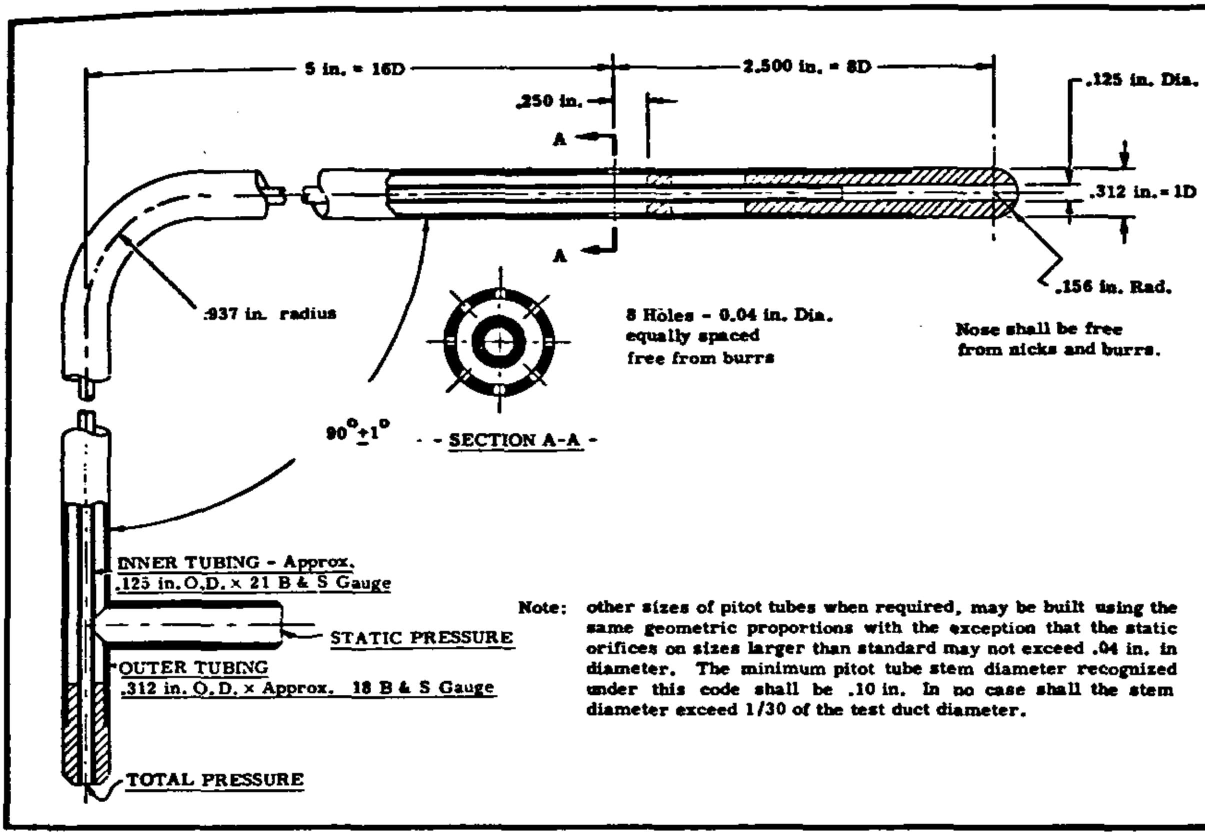
Figure 40-4. Relationship of Velocity Pressure to Static and Total Pressure for an Exhausting System.

Pitot Tube — Velocity Pressure

Equipment. The Pitot tube is the standard instrument for measuring the velocity of air in ducts. Figure 40-4 is a graphic representation showing the relationship of velocity pressure to static pressure and total pressure of an exhausting system.[4] The Pitot tube consists of two concentric tubes. The opening of the inner tube is axial to the flow and measures total pressure (A of Figure 40-4), while the larger tube with circumferential openings measures static pressure (B of Figure 40-4). The difference between the total pressure and static pressure is the velocity pressure (C of Figure 40-4).

Pitot tubes which are built to Air Moving and Conditioning Association[8] and ASHRAE[9] standards are considered primary standards and require no calibration. See Figure 40-5. Standard Pitot tubes fabricated from type 304 stainless steel are recommended because of their resistance to corrosion and use over wide temperature ranges up to 1000°F. Above this temperature, water-cooled Pitot tubes are required.

Pitot tubes are used to determine the velocity pressure contours inside of ducts. These measurements are obtained by connecting the static and total pressure taps to a manometer as shown in Figure 40-6. Inclined manometers (10:1) are normally used since they increase the accuracy and precision especially for velocities below 2000 fpm. A brief table appears in reference 4 which shows that the percent error using a carefully leveled



Air Moving & Conditioning Society: Standard 210. Arlington Heights, Illinois.

Figure 40-5. The Standard Pitot Tube. Note: Other sizes of Pitot tubes, when required, may be built using the same geometric proportions with the exception that the static orifices on sizes larger than standard may not exceed .04 in. in diameter. The minimum Pitot tube stem diameter recognized under this code shall be .10 in. In no case shall the stem diameter exceed 1/30 of the test duct diameter.

10:1 inclined manometer ranges from 0.25% at 4000 fpm to 4.0% at 1000 fpm, and up to 15% at 600 fpm. Thus, Pitot tubes used in the field are generally restricted to velocities above the 600-800 fpm range. Inclined manometers which read in fpm and inches of water are commercially available (Table 40-1). Conventional ranges are 400-12,600 fpm and 600-19,200 fpm. These manometers eliminate the need to convert VP to velocity as previously discussed.

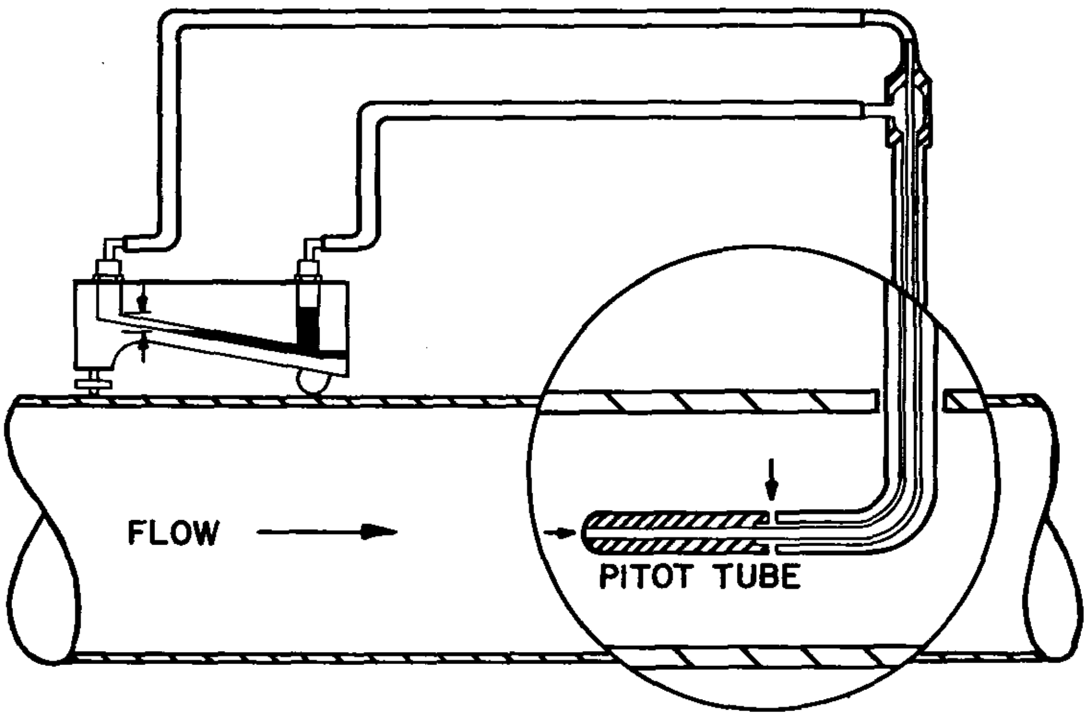
Pitot traverse. Aside from instrument error, the most significant requirement in making valid velocity or air flow measurements is the location selected for the measurements and the traverse of that location. The reason for these requirements is that air flow is not uniform in the cross section of a duct. This is especially true near such interferences as elbows, entries, etc. Therefore, for greatest accuracy, measurements should be taken at least 8.5 diameters of straight run downstream or 1.5 diameters upstream from interferences. Once the location is selected, a Pitot traverse can be conducted. Figure 40-7 shows a cutaway of both a round and rectangular duct, exemplifying the principle of measuring the VP of equal areas. Note that for round ducts it is advisable to traverse in two planes perpendicular to each other. The optimum number of measurements per plane for ducts of stated diameters is suggested below:

Duct Diameter, Inches	Number of Measurements
3-6	6
5-48	10
44 and greater	20

Reference 4 includes tables giving distances from the duct walls for the diameters stated above.

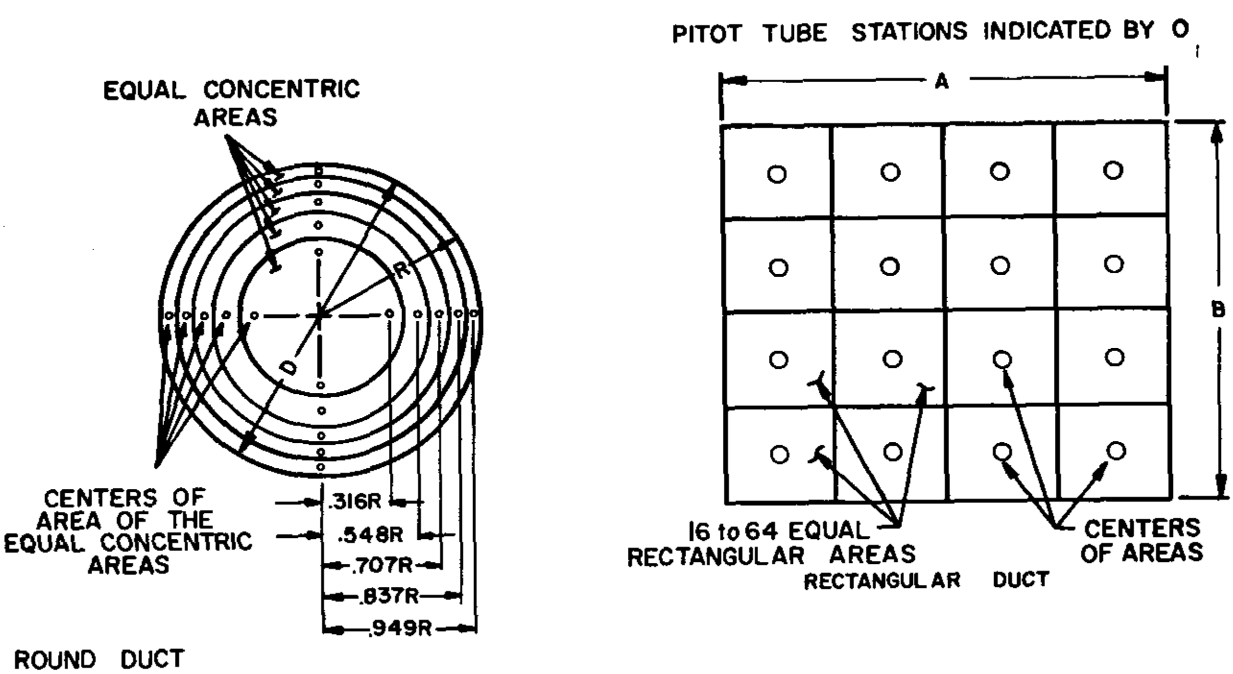
Like round ducts, rectangular ducts are traversed in terms of equal area segments. Rectangular ducts should be divided in a minimum of 16 to a maximum of 64 equal area rectangles with readings taken in the center of each rectangle as shown in Figure 40-7.

Once the readings as VP are taken and tabulated, the average velocity, in fpm, should be calculated. *Do not average the VP;* however the $\sqrt{VP}$ may be averaged and converted to velocity. From the velocity in fpm (V) and the area of the duct in ft^2 (A), flowrate (Q) can be calculated according to Equation 1.



Dwyer Instruments, Inc.: Bulletin No. H-100. Michigan City, Indiana, p. 3.

Figure 40-6. The Pitot Tube Connected to an Inclined Manometer.



Dwyer Instruments, Inc.: Bulletin No. H-100. Michigan City, Indiana, p. 3.

Figure 40-7. Traverse of a Round and Rectangular Duct Area.

In cases when accuracy is not a prime consideration, a single centerline reading can be taken at least 10 diameters of straight duct downstream from the nearest interference. This VP should be adjusted by multiplying by 0.81, or the velocity should be multiplied by 0.90.

When it is not possible to find undisturbed traverse locations 8.5 to 10 diameters downstream, alternate locations should be selected on the basis of a 5:1 ratio between downstream and upstream interferences. Depending on the situation and need for accuracy, multiple points for traverse can be selected and those points within 10% agreement averaged and used to determine velocity and air flow.

Limitations. There are fewer limitations for Pitot tubes than for other air velocity measuring devices. Whereas they can be used in corrosive or variable temperature conditions, the impact and static openings can become clogged with particulate matter. Also, as with the other instruments discussed, corrections should be made if the temperature is ± 30°F from standard, the altitude is greater than 1000 ft., and the moisture content is 0.02 lb./lb. or greater. They cannot be used to measure low velocities (less than 600 fpm) and require an inclined manometer which must be level and free from vibration. They are not applicable for use in small diameter ducts (less than 3 inches) or in orifice type openings.

Aneroid Gauges

The most common and best known of the aneroid gauges is the magnehelic gauge. Aneroid gauges can be used for total, static, and, in conjunction with a Pitot tube, velocity pressure measurements. They are small, extremely portable, and not as sensitive to vibration and leveling as liquid filled manometers. Since the inches of water pressure is a function of the location of an indicating needle on a dial, they are extremely easy to read. Magnehelic gauges are commercially available in ranges from 0 to 0.5" WG. (500-2800 fpm) to 0 to 150" WG. (2000-125,000 fpm) (Table 40-1).

The principal limitations are accuracy and calibration. Accuracy is usually below ± 2% full scale. Since they are mechanical, there is a need to calibrate these devices periodically.

Manometers

Manometers range from the simple U-tube to inclined manometers already mentioned. A range of sizes and varieties of U-tube manometers are available and they may be filled with a variety of media ranging from alcohol to mercury. Readings can be converted to inches of water simply by correcting for differences in density (e.g., 1 inch of mercury is equal to 13.61 inches of water).

When extreme accuracy is not essential or in high pressure systems, U-tube manometers will suffice. However, for accuracy and in low pressure systems, inclined manometers are required.

Static Pressure Measurements

Instrumentation and taps. Instruments used in measuring static pressure include the static leg of the pitot tube as well as any pressure measuring device connected to a hole in the side of a duct. U-tube manometers and Magnehelic gauges are quite acceptable. Whereas the exact location of the hole is not extremely critical, the type of hole is. Generally, the holes should not be located in points where there is some basis for turbulence or non-linear flow such as the heel of an elbow. Holes should be flush with the inside of the duct with no projections or burrs. Thus, holes should be drilled and not punched. The location of holes 90° apart will allow for the averaging of multiple readings to provide an improved estimate of static pressure.

Taps can range in complexity from a simple soft rubber hose held tightly against a 1/16 inch hole, to soldered pet cocks for use in high pressure applications.

Applications. Static pressure measurements at strategic points in a system provide invaluable information as to the performance. These measurements are neither difficult to obtain nor do they require expensive or delicate instrumentation.

Estimation of air flow by the throat suction method[6] provides a fairly accurate estimation of flowrate of an exhaust opening if the coefficient of entry can be determined. Coefficient of entries for various hoods are given on Figure 2 of Chapter 42. Measurements are made between one and three diameters of straight duct from the throat of the exhaust inlet (point where the hood is connected to the branch duct). It is advisable to take multiple readings 90° apart. The flowrate in cfm can then be determined according to equation 6:

Equation 6: $Q = 4005\ CeA\sqrt{SP_h}$

Where: Q = Rate of flow in cfm

Ce = Ratio of actual flow to theoretical flow (Figure 2 of Chapter 42) (Entry loss in " WG)

A = Cross-sectional area of duct in ft^2

SP_h = Average static pressure reading in inches of water

Static pressure comparisons provide a means of either continuously or periodically monitoring the performance of a system. Additional information may be required, but strategically located static pressure taps can flag malfunctioning equipment, clogged ducts, dirty or broken filters, dented exhaust hoods, and changes in fan static pressure.

The permanent installation of manometers immediately downstream from exhaust hoods controlling a hazardous material or critical process is advisable, as is the placement of such devices across a filter to determine the need for shaking, cleaning, or maintenance.

Other Measuring Devices

There are a number of other devices for measuring fluid flow, but their application is restricted to either laboratory use or the calibration of air sampling devices. Some are discussed briefly below:

Orifice meter[7]. An orifice meter is simply a restriction in a pipe between two pressure taps. There are several types of orifice meters used, but the simplest and most common is the square edged orifice. If it is properly constructed, the orifice plate will be at right angles to the flow, and the surface will be carefully smoothed to remove burrs and other irregularities. Orifice meters are seldom used as permanent flow meters in ventilation systems because of their high permanent pressure loss. They are more typically used in the ventilation laboratory for calibration purposes. Permanent head loss will vary from 40 to 90 percent of the static pressure drop across the orifice as the ratio of orifice diameter to pipe diameter varies from 0.8 to 0.3. Detailed discussions of orifices and orifice equations can be found in reference 11.

Venturi meters[7]. A Venturi meter consists of a 25° contraction to a throat, and a 7° re-expansion to the original size. This differs from the orifice meter where the changes in cross section are abrupt. The advantage of the Venturi over the standard orifice is that the permanent reduction in static pressure is small, because the velocity head in the throat is largely reconverted to static pressure by the gradual enlargement. A well designed and constructed Venturi will have a permanent static pressure loss of only 0.1 to 0.2 inches of H_2O as compared to 0.4 to 0.9 for the orifice plate. Venturi meters are used in conjunction with a manometer as an in-line flow measuring device. A more detailed explanation of the Venturi is offered in reference 11.

CALIBRATION OF INSTRUMENTATION

All too often the need for calibration is not applied to devices for measuring air flow and velocity, yet as a group, with the exception of the Pitot tube, they require periodic calibration. Generally, air flow measuring instrumentation is based on electrical or mechanical systems which are sensitive to shock. In addition, use of these instruments in corrosive or dusty atmosphere affects their reliability.

A calibration wind tunnel as shown in Figure 40-8 represents the method of choice for calibrating the devices described in this section. Reference 12 is an excellent treatment of the design and use of the calibration wind tunnel. A well designed wind tunnel must have the following components[12]:

1. *A satisfactory test section.* Since this is the location of the probe or sensing element of the device being calibrated, the gas flow must be uniform, both perpendicular and axial to the plane of flow. Streamlined entries and straight runs of duct are essential to eliminate pronounced vena contracta and turbulence.
2. *A satisfactory means of precisely metering air flow.* A meter with adequate scale graduations to give readings of $\pm 1\%$ is required. A Venturi or orifice meter represent optimum choices since they require only a single reading.
3. *A means of regulating air flow.* A wide range of flows are required. A suggested range is from 50 to 10,000 fpm. Therefore, the fan must have sufficient capacity to overcome the static pressure of the entire system at the maximum velocity required. A variable drive provides for a means of easily and precisely attaining a desired velocity.

Meters must be calibrated in a manner similar to how they are used in the field. Vane actuated devices should be set on a bracket inside a large test section with a streamlined entrance. Low velocity probe type devices may be tested through appropriate openings in the same type of tunnel. High velocity ranges of probe type devices and impact devices should be tested through appropriate openings in a circular duct at least 8.5 diameters downstream from any interference. Straighteners as shown in Figure 40-8 will reduce this requirement to 7 diameters.

NOTE: Devices must be calibrated at multiple velocities throughout their operating range.

AIR FLOW SYSTEM SURVEYS

System Start-up vs. Design Basis

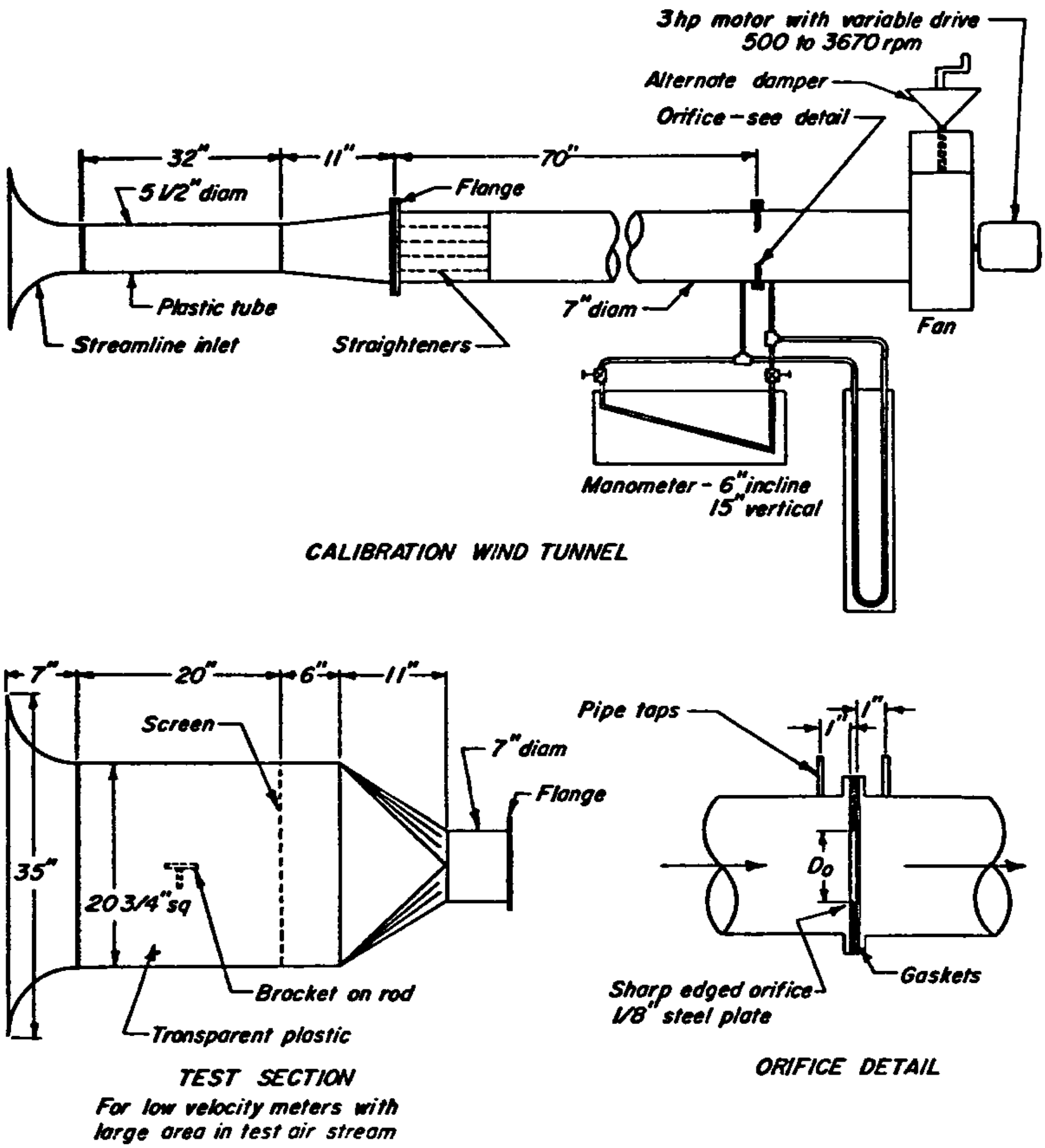
Any ventilation system, be it local exhaust for contaminant control or general for comfort, is designed in terms of removing or distributing a specified quantity of air at a specified velocity at a total system pressure which is the sum of the parts. An initial survey of the system is the only time a valid comparison can be made between the design basis and optimum system performance.

Sketch of the system. A sketch not necessarily to scale but representative of dimensions should be drawn noting such items as hoods, elbows, branchings, air cleaner, fan and stack. Supply ducts, plenums, and diffusers should be shown for general systems. Figure 40-1 and 40-2 represent gross simplifications of this concept.

The sketch should be considered as part of the permanent record on which future changes in the systems may be recorded.

Specific air flow measurements. Measurements in terms of air flow, velocity, and static pressure must be made to determine that the system is adequately balanced and performing according to the design basis. These measurements include:

1. Static pressure measurements at:
   a) hoods
   b) up and downstream of the air cleaner
   c) up and downstream of the fan
2. Air flow in cfm at:
   a) hoods (throat suction method)
   b) branches and mains (Pitot tube)
   c) up and downstream of fan (Pitot tube)
3. Supply, capture, and conveying velocities at:
   a) diffuser outlets (supply velocity)
   b) face or opening of hood (capture velocity)
   c) branches and mains (conveying velocity)
4. Fan performance
   a) fan speed in rpm
   b) horsepower (BHP) calculated using cfm (Q), total pressure (TP), and mechanical efficiency (ME) of fan.



American Conference of Governmental Industrial Hygienists — Committee on Industrial Ventilation: Industrial Ventilation Manual, 11th Edition. Lansing, Michigan, 1970.

Figure 40-8. The Calibration Wind Tunnel.

Equation 7: $$BHP = \frac{(Q)\ (TP)}{6356 \times ME}$$

The locations of the measurements must be identified on the sketch and a record kept for future comparisons. A sample form can be found in reference 4.

The measurements obtained should agree within 10% of the design basis. If not, system modification should be made until such agreement is obtained.

Other Checks. Local exhaust systems are installed for the singular purpose of removing some contaminant from the work environment. Visualization techniques using smoke tubes or candles can be most helpful in verifying the system exerts a sphere of control over a sufficient area to prevent excessive exposures to operating personnel. Air evaluation for specific contaminants is also recommended to verify the system will control contaminants to levels known to be safe. Air samples taken in the breathing zone of operating personnel will be most helpful in assessing the adequacy of contaminant control.

As with the previous measurements, photographic records of smoke tests and the results of air evaluation tests should be maintained for future reference.

System Operation vs. System Start-up

Once systems are started up and determined to perform satisfactorily, the degree of evaluation *can be reduced as long as good records of start-up or initial conditions have been made.* Experience with air flow systems clearly indicates periodic surveys are required to assure system performance is adequate. Operating personnel cannot be relied upon as an "indicator" of system performance. Also, ventilation systems are rarely an integral part of the operation in terms of quality and production, and all too often receive inadequate maintenance.

For most systems simple velocity measurements at exhaust hoods and supply ducts will provide a crude indication of system performance when compared with start-up evaluations. For local exhaust systems, the throat suction method applied to exhaust hoods and static pressure differentials for air cleaners and fans will suffice in confirming the system is performing satisfactorily.

The throat suction method will provide valid information unless:

1. The hood entry has been modified/damaged;
2. There are obstructions ahead of the point of measurement; or
3. The system has been modified.

However, a reduction in throat suction can provide valuable information, such as an indication that there has been:

1. Accumulations of material in an elbow, branch, or main, thus clogging or restricting air flow. Build-up in the elbows result from impaction, while build-ups in straight runs result from insufficient conveying velocity or overloading the system.
2. A change in blast gate setting if the system is balanced using blast gates.
3. Additional branches and hoods added to the system. "Adding on" to a system is a real temptation. It is not sound economics when it renders the entire system deficient.
4. Excessive build-up on the filter. It is best to monitor filter build-up by attaching a static pressure measuring device across the filter.
5. Reduced fan output resulting from belt slippage, damaged or worn rotor, or build-up on the fan blades.

Data Handling and Recording

The sketch of the system made at start-up or for the initial air evaluation survey and the results of the ensuing air flow survey must be recorded and filed in such a manner that future air flow surveys can be conducted in a similar manner. The periodicity of air flow surveys can only be determined by such conditions as:

1. Nature of the materials being controlled. The more hazardous the materials, the more frequently the system should be checked.
2. Nature of the system. A blast gate system will require more frequent checks than other systems.
3. The degree of maintenance. Air flow surveys can be used to indicate the need for more frequent and improved maintenance.

Reference 4 provides a sample of a diagram, check list and additional information regarding checking and testing systems.

References

1. Occupational Safety and Health Standards. *Federal Register,* Subpart G. 1910.94 Ventilation, Washington, D. C., May 29, 1971.
2. *Industrial Ventilation — A Manual of Recommended Practice.* American Conference of Governmental Industrial Hygienists, 11th ed., Section 2, Dilution Ventilation, Cincinnati, Ohio, 1970.
3. SCHULTE, H. F., E. C. HYATT, H. S. JORDAN and R. N. MITCHELL. "Evaluation of Laboratory Fume Hoods." *Am. Ind. Hygiene Quarterly,* 15:195, Chicago, Illinois, 1954.
4. *Industrial Ventilation — A Manual of Recommended Practice.* American Conference of Governmental Industrial Hygienists, 11th ed., Section 9, Testing of Ventilation Systems, Cincinnati, Ohio, 1970.
5. Operating Instructions for the Alnor Series 6000-P Velometer. Alnor Instruments Co., Chicago, Ill., 1970.
6. TUVE, G. L. and D. K. WRIGHT. "Air Flow Measurements at Intake and Discharge Openings and Grilles." *Heating, Piping and Air Conditioning* 12:501, 10 South LaSalle St., Chicago, Illinois, 1940.
7. LIPPMAN, M. and G. W. FISCHER. Ch. 7 Air-Flow Measurements. *The Industrial Environment — Its Evaluation and Control.* USPHS, Washington, D. C., 1965.
8. AMCA Standard 210-67. Air Moving and Conditioning Association, Inc., 205 W. Toughy Ave., Park Ridge, Illinois, 60068.
9. *Heating, Ventilation, and Air Conditioning Guide.* ASHRAE, ASHRAE, Inc., New York, 1963.
10. *Air Velocity Measurements.* Dwyer Bulletin H-100, Dwyer Instruments, Inc., Michigan City, Indiana, 46360, 1970.
11. BRANDT, A. D. *Industrial Health Engineering.* John Wiley and Sons, New York, 1947.
12. HAMA, G. "A Calibration Wind Tunnel for Air Measuring Instruments." *Air Engineering,* 9:18, Detroit, Michigan, 1967.

Preferred Reading

In addition to References 2, 4, 6, 9, 10, 11, and 12, the following represent selected sources which can contribute to the reader's knowledge of the subject title.

Heating and Cooling for Man in Industry, American Industrial Hygiene Association, 1st edition, Ch. 11 Testing, 1970.

ASHRAE Guide and Data Book — Systems. Ch. 38 Testing, adjusting, and balancing; and Ch. 39 Preventive Maintenance, 1970.

ASHRAE Guide and Data Book — Handbook of Fundamentals. Ch. 13 Measurements and Instruments, 1967.

"Fundamentals Governing the Design and Operation of Local Exhaust System." ANSI Z9.2 — 1971. Section 9, Operation and Maintenance; and Section 10, Checking Operation of Local Exhaust Systems. American National Standards Institute, 1430 Broadway, New York, New York 10016.

"Velometers and Other Air Velocity Measuring

Instruments." Bulletin No. 72-60-10M269. Alnor Instrument Co., 402 N. LaSalle St., Chicago, Illinois 60610.

Hastings Air Meter Bulletin. Hastings-Raydist Co., P.O. Box 1275, Hampton, Virginia 23361.

"Meriam Pitot Tubes." *Bulletin 51*. Meriam Instrument Co., 10920 Madison Ave., Cleveland, Ohio 44102.

Anemotherm Air Meter Bulletin. Anemostat Products, P. O. Box 1083, Scranton, Pa. 18501.