arise due to excessive room air temperature variations (horizontally, vertically, or both), excessive air motion (draft), failure to deliver or distribute the air according to the load requirements at the different locations, or too rapid fluctuation of room temperature or air motion (gusts).

With reference to permissible room air motion it is not possible to establish a specific standard covering the entire complex problem of air distribution. Velocities less than 15 fpm generally cause a feeling of air stagnation, whereas velocities higher than 65 fpm may result in a sensation of draft. Air velocities of 25 to 35 fpm in the occupied zone are most satisfactory, but air motion of 20 to 50 fpm will usually be acceptable, with the lower values used in cooling applications, and the higher values in heating applications. The permissible air motion depends to some degree on the geographic location of the air-conditioning installation. In addition the noise level created by the introduction of supply air should be kept within acceptable limits, and streaking or smudging of walls or ceilings should be prevented.

Reference should be made to Chapter 6, Physiological Principles, for information on effective temperature and comfort lines, and to Chapter 25, Sound Control, for acceptable room noise levels and noise generated by air outlets. Material in Chapter 51, Control of the Industrial Environment, and Chapter 50, Process and Product Air Conditioning, deals with the health, safety, and efficiency of workers, and with temperature and humidity requirements for products and manufacturing processes, respectively.

PRINCIPLES OF AIR DISTRIBUTION

Conditioned air is supplied to air outlets at temperatures and velocities which differ greatly from those in the occupied zone of the room. Proper air distribution, therefore, calls for (1) entrainment of room air by the primary air stream outside of the zone of occupancy in order that air motion and temperature differences will be reduced to acceptable limits before the air enters the occupied zone, and (2) counteraction of the natural convection and radiation effects within the room.

The complexity of practical room air distribution problems is due to innumerable variations in building construction, system design, and operating requirements, and makes the exploration of the basic character of air distribution imperative. The theory of room air distribution is not complete but a considerable fund of knowledge supported by experimental guidance is available for the solution of many air distribution problems. In particular, research at the ASHAE Research Laboratory and at Case Institute of Technology have advanced considerably the fundamental knowledge on ventilating jets in air distribution. Also, a comprehensive survey of the literature relevant to air distribution problems was made at the ASHAE Research Laboratory and is an indispensable and timesaving tool for every researcher in air distribution. (Refer to Bibliography: ASHAE Research Laboratory.)

The cooperative research program in air distribution at Case Institute of Technology was started in 1937. The early research dealt with methods of measurements and the available instruments; next the basic flow patterns of free round and rectangular jets in an open space were studied. A third phase was concerned with diffusion rates and characteristics of free jets from the more complex types of outlets, such as grilles with guide vanes, narrow slots, perforated panels, plaques, and round ceiling diffusers. The latest phase has dealt with air movement produced by non-isothermal jets, especially in confined spaces, and has included experimental work in cold-wall rooms during heating and warm-wall rooms during cooling. (Refer to Bibliography: Cooperative Research at Case Institute of Technology.)

VENTILATING JETS IN AIR DISTRIBUTION

A surprising similarity between the shape of jets exists at a short distance from the outlet face, whether the outlet is round, rectangular, grille-like, or a perforated panel. The jet discharged from a round opening forms an expanding cone, whereas jets from rectangular outlets rapidly pass from rectangular to elliptical cross-sectional shape at a short distance from the outlet face, and then to circular shape, at a rate depending primarily on the aspect ratio. Even in the case of wide-angle grilles and angular outlets, the similarities are such as to permit the same analysis of performance.

For many conditions of jet discharge, therefore, it is possible to analyze jet performance, and to determine (1) the angle of divergence of the jet boundary, (2) the velocity patterns along the jet axis, (3) the velocity profile at any cross-section in the zone of maximum engineering importance, and (4) the entrainment ratios in the same zone.[1]

In using the data presented in this section, however, the following must be kept in mind:

1. The method of finding jet velocities is based upon several approximations and, therefore, the two recommended equations must be used with caution for extreme axial and radial distances.
2. The characteristics of the low-velocity regions of ventilating jets are not yet well understood. Neither for axial nor radial jets are the effects at various Reynolds number fully known.
3. The quantitative treatment of the forces that govern room air distribution problems has been limited, and non-isothermal conditions involving buoyant forces have not yet been fully explored.

For equations describing the performance of non-isothermal jets resulting from the air distribution research at Case Institute of Technology see References 2 and 3. (Refer also to Bibliography: ASHAE Cooperative Research at Kansas State College.)

4. Most investigations have been concerned with *free* jets, whereas air streams in practical room air distribution are not free streams but are influenced by walls, ceilings, floors, and obstructions. (Refer to sections on Effect of Walls and Ceilings and Room Air Motion.)

Angle of Divergence

The angle of divergence is very definite close to the outlet face, but the boundary contours are somewhat billowy and are easily affected by external influences. Here, as in air distribution generally, room air movement is replete with local eddies, vortices, and surges, which are manifestations of unbalance in the forces acting within the air stream. These internal forces govern the air motion, yet they are extremely delicate.[4]

Measured angles of divergence (spread) for discharge into large open spaces have usually ranged from 20 to 24 deg with an average of 22 deg. Coalescing jets for closely spaced multiple outlets expand at somewhat smaller angles, averaging 18 deg, and jets discharging into relatively small spaces show even smaller angles of expansion.[5] Tests indicate that in cases where the outlet area itself is small compared to the dimensions of the space normal to the jet, the jet may be considered free as long as

$$X \leq 1.5\sqrt{A_R}$$

where

X = distance from face of outlet, feet.
A_R = cross-sectional area of the confined space, square feet.

Four Zones in Jet Expansion

In analysing the performance of jets, four major zones can be distinguished. They may be roughly defined in terms of the maximum or centerline velocity existing at the cross-section being considered:

Zone 1: A short zone, extending about 4 diameters or widths from the outlet face (or vena contracta for orifice discharge), in which the maximum velocity of the air stream remains practically unchanged.

Zone 2: A transition zone, extending to about 8 diameters for round outlets, or for rectangular outlets of small aspect ratio, over most of which maximum velocities vary inversely as the square root of the distance from the outlet. For rectangular outlets of large aspect ratio, this zone is elongated and extends from about 4 widths to a distance approximately equal to the width multiplied by four times the aspect ratio.

Zone 3: A long zone, of major engineering importance, in in which the maximum velocity varies inversely as the distance from the outlet. This zone is often called the zone of fully *established turbulent flow* and may be 25 to 100 diameters long (or equivalent diameters of equal areas), depending on the shape and area of the outlet, the initial velocity, and the dimensions of the space into which the outlet discharges.

Zone 4: A terminal zone in which, in the case of confined spaces, the maximum velocity decreases at an increasing rate, or, in the case of large spaces free from wall effects, the maximum velocity decreases rapidly in a few diameters to the velocity range below 50 fpm which is usually regarded as *still air*.

Centerline Velocity in Zone 3

Research has shown that maximum or centerline velocities in Zone 3 of straight flow isothermal jets can be determined with good engineering accuracy from

$$\frac{V_x}{V_0} = \frac{KD_0}{X} = \frac{K'\sqrt{A_0}}{X} \qquad (1)$$

$$V_x = \frac{K'V_0\sqrt{A_0}}{X} = \frac{K'Q}{X\sqrt{A_0}} \qquad (2)$$

$$= \frac{K'Q}{X\sqrt{A_c \times C_d \times R_{fa}}} \qquad (2a)$$

Table 1. . . . Recommended Values of the Centerline Velocity Constant K or K'

(See Equation 1)

Type of Outlet	K: V_0 = 500 to 1000	K: V_0 = 2000 to 10,000	K′: V_0 = 500 to 1000	K′: V_0 = 2000 to 10,000
Free Openings				
Round or Square	5.0	6.2	5.7	7.0
Rectangular, large aspect ratio (<40)	4.3	5.3	4.9	6.0
Annular slots axial or radial[a]	—	—	3.9	4.8
Grilles and Grids				
Free area 40% or more	4.1	5.0	4.7	5.7
Perforated Panels				
Free area 3 to 5%	2.7	3.3	3.0	3.7
Free area 10 to 20%	3.5	4.3	4.0	4.9

[a] For radial slots use X/H instead of $X/(A)^{1/2}$. H is the height or width of the slot.

Note: K and K' are indexes of loss in axial kinetic energy. Interpolate as required. Departures from maximum value indicate losses in first and second zones when compared with the jet from a rounded-entrance, circular nozzle.

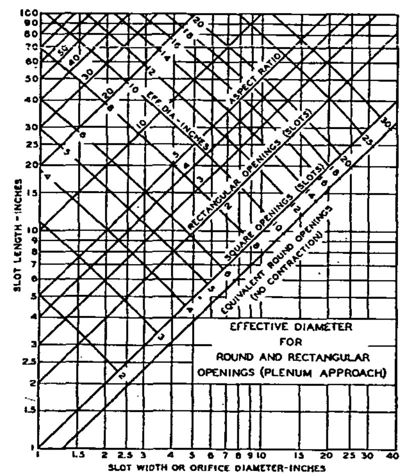

Fig. 1. . . Effective Diameters for Round and Rectangular Openings (Plenum Approach)

where

V_x = centerline velocity, feet per minute.
$V_0 = \dfrac{V_c}{C_d \times R_{fa}}$ = average initial velocity at discharge from open-end duct or across contracted stream at vena contracta of orifice or multiple-opening outlet, feet per minute.
V_c = nominal velocity of discharge based on the core area, feet per minute.
C_d = coefficient of discharge (usually between 0.65 and 0.90).
R_{fa} = ratio of free area to gross (core) area.
X = distance from face of outlet, feet.
K and K' = porportionality constants, with $K' = 1.13\,K$.
D_0 = effective or equivalent diameter of stream at discharge from open-end duct or at a contracted section, feet.
$A_0 = A_c \times C_d \times R_{fa}$ = effective area of stream at discharge from an open-end duct or at a contracted section, square feet.
A_c = measured gross (core) area of outlet, square feet.
Q = discharge from outlet, cubic feet per minute.

Equation 1 is nondimensional and requires only that consistent units be used, as in the above nomenclature. Values of K and K' are listed in Table 1.[1, 6]

Low velocity test results, in the range $V_x < 150$ fpm, indicate that the normal values of K and K' should be reduced about 20 percent for $V_x = 50$ fpm, as used in later Equation 4b for throw. Fig. 1 gives the effective diameter D_0 in inches for single openings and includes the coefficient of discharge.[7] Similar values for grilles are available in the literature or can