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CHARACTERISTICS OF DIFFUSER AIR JETS AND AIRFLOW IN THE OCCUPIED REGIONS OF MECHANICALLY VENTILATED ROOMS—A LITERATURE REVIEW

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ABSTRACT

In this paper, previous research findings on the characteristics of diffuser air jets and airflow in the occupied regions as well as their relationships are reviewed and summarized. Results from different sources are compared. It was found that it is possible to predict velocity, turbulence, and temperature characteristics in the occupied region based on the characteristics of diffuser air jets. However, methods that can quantify the effect of internal obstructions such as office partitions and furniture and the location of heat sources need to be developed.

INTRODUCTION

Proper air distribution is important to achieve satisfactory thermal comfort and air quality in the occupied regions of ventilated rooms. Airflow patterns and distributions of velocity, temperature, and turbulence intensity in a ventilated room are primarily determined by the momentum and trajectory of the diffuser air jet and its mixing with and entrainment of room air. Therefore, understanding the characteristics of diffuser air jets and their relations to the airflow conditions in the occupied regions is essential to the selection of diffuser type, sizing, and location. The objective of this paper is to review and summarize the recent research findings in the studies of diffuser air jets and the characteristics of airflow in the occupied region and their relationships and to identify research needs in this area.

CLASSIFICATION OF DIFFUSER AIR JETS

Diffuser air jets have different characteristics when they are supplied from different types of diffusers, or under different conditions (initial air temperature, room geometry and size, supply direction, etc.). When the temperature of the supplying air jet is equal to the temperature of the air in the room, the jet is called an *isothermal jet*. When there is temperature difference between the incoming air jet and the room air, the jet is called a *nonisothermal jet*. When the jet is discharging into a large open space (not influenced by walls and ceilings), the jet is called a *free jet*, whereas when the incoming air jet is attached to a ceiling or wall, it is called an *attached jet*, ceiling jet (attached to a ceiling), or wall jet (attached to a wall). Furthermore, if the incoming air jet is affected by the reverse flow in the room caused by the jet itself, it is called a *confined jet*.

Depending on the types of diffusers, diffuser air jets can be classified as follows:

1. Compact jets: from grilles, nozzles or other round openings, square openings, or rectangular openings with small aspect ratios. These jets are considered axisymmetric.
2. Linear jets: from slots or rectangular openings with large aspect ratios. These jets have two-dimensional characteristics.
3. Radial jets: from diffusers where the axial flow from a cylindrical chamber is deflected in all 360 degrees.
4. Incomplete radial jets: from grilles having diverging vanes.
5. Conical jets: from a cone-type or regulated multi-diffuser.
6. Swirling jets: from diffusers that can form vortexes.

CHARACTERISTICS OF THE DIFFUSER AIR JETS

Isothermal Free Jets

Intensive studies have been performed on velocity decay of isothermal free jets (e.g., Nottage et al. 1952a,

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1952b, 1952c; Tuve 1953; Koestel et al. 1950; Grimitlyn 1970; Shepelev 1961; Baturin 1972; Jackman 1973; Nielsen and Moller 1987). Diffuser air jets can usually be divided into four zones. Zone 1, the initial zone, and zone 2, the transition zone, are usually very short. Zone 3 is of most importance from an engineering point of view. The velocity decay of the centerline of linear jets in this zone can be described by Equation 1 and the velocity decay of compact, incomplete radial, and radial jets can be described by Equation 2:

$$\frac{V_x}{V_0} = K_1\sqrt{\frac{H_0}{x}} \tag{1}$$

$$\frac{V_x}{V_0} = \frac{K_1\sqrt{A_0}}{x} \tag{2}$$

where

V_x = centerline velocity at x;
V_0 = average velocity at discharge;
H_0 = effective width of diffuser;
K_1 = centerline velocity decay constant;
x = distance to the diffuser face on the jet centerline;
A_0 = effective area of diffuser, $A_0 = C_d A_c$;
C_d = discharge coefficient (usually between 0.65 and 0.90);
A_c = diffuser free area.

Zone 4 is a terminal zone where the residual velocity decays quickly into large-scale turbulence. Within a few diameters, the axial velocity subsides to the range below 50 fpm. Although this zone was studied by several researchers (Madison and Eliot 1946; Weinhold 1969), its characteristics are still not well understood.

Figure 1 shows the velocity decay of four typical jets with diffuser area A_0 = 0.4 ft^2 described by Equations 1 and 2 at the following typical values: for linear jets, K_1 = 2.2; aspect ratio = 100; H_0 = 0.063 ft; for compact jets, K_1 = 5.7; for radial jets, K_1 = 1.1; for incomplete radial jets, K_1 = 2.0. At the same effective area of the diffuser, the velocity decay described by Equation 1 (linear jet) is slower than that by Equation 2 at greater values of x when the K_1 values are similar. Because the velocity decay of Equation 1 varies with the square root of x, there is little change in velocity decay when increasing x at a greater value. The velocity decay of a compact jet is much slower than that of radial jets because, within a given distance, compact jets usually experience less shear stress than radial jets due to their shapes. Since K_1 values are directly proportional to velocity decay and the key factor determining the velocity decay, accurate measurement of K_1 of diffusers is crucial in designing ventilation systems.

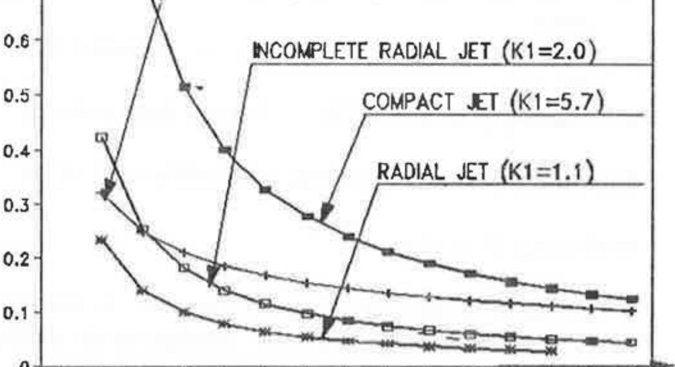

Figure 1 *Velocity decay of the centerline of isothermal jets.*

Nonisothermal Free Jets

Velocity Decay and Throw For vertically projected nonisothermal linear jets, it was found that the calculation of velocity decay and throw can be done using the following equations (Shepelev 1961; Grimitlyn 1970):

$$\frac{V_x}{V_0} = K_1\sqrt{\frac{H_0}{x}}K_n \tag{3}$$

$$K_n = [1 \pm \frac{1.8K_2}{K_1^2}A_r(\frac{x}{H_0})^{3/2}]^{1/3} \tag{4}$$

where K_2 = temperature decay constant of the jets. The "±" sign should be positive when the buoyant force is in the same direction as the initial force, and negative when they are not.

For other vertically projected nonisothermal jets (compact, incomplete radial, and conical), the following equations can be used (Shepelev 1961; Grimitlyn 1970):

$$\frac{V_x}{V_0} = \frac{K_1\sqrt{A_0}}{x}K_n \tag{5}$$

$$K_n = [1 \pm \frac{2.5K_2}{K_1^2}Ar_0(\frac{x}{\sqrt{A_0}})^2]^{1/3} \tag{6}$$

The velocity decay for sill-mounted grilles can be described by the following equations (Jackman 1971; Regenscheit 1970):

$$\frac{V_x}{V_0} = \sqrt{5.4\frac{H_0}{x} + 2.15Ar_0(\frac{x}{H_0} - 5.4)} \tag{7}$$

$$Ar_0 = \frac{g\,\Delta t_0 H_0}{TV_0^2} \tag{8}$$

where

x = distance from grille measured vertically up to the ceiling and then horizontally across the room from the wall/ceiling corner;

Δt_0 = temperature of supply air minus temperature of return air;

T = mean absolute temperature of air.

The velocity decay rates described in Equations 3, 5, and 7 are compared in Figure 2 at the following conditions: $\Delta t_0 = 20°R$; $T = 527°R$; $V_0 = 600$ fpm; $A_0 = 0.4$ ft^2; for linear jets, $K_1 = 1.8$, $K_2 = 1.7$, and $H_0 = 0.063$; for compact jets, $K_1 = 5.3$ and $K_2 = 3.4$. All vertical jets are considered to be directed upward.

From Figure 2 we can see that, although the forms of Equations 3 and 7 for cooled linear jets are different, there is not much difference in the actual velocity decay rates, therefore, either form can be used when calculating the velocity decay. There is much difference, however, in the velocity decay rates between compact jets and linear jets. The velocity decay rate of the linear jet is most rapid. Velocity decay of heated jets is much slower than that of cooled jets because of the gravitational force (Figure 2).

Temperature Decay Temperature decay of the jets can be calculated by the following equations (Grimitlyn 1970; Shepelev 1961): For linear jets, horizontally or vertically projected, use Equation 9 or 10, respectively; for compact, incomplete radial, and conical jets, horizontally or vertically projected, use Equation 11 or 12, respectively; for radial jets, use Equation 11.

$$\frac{T_x - T_r}{T_0 - T_r} = K_2 \sqrt{\frac{H_0}{x}} \qquad (9)$$

$$\frac{T_x - T_r}{T_0 - T_r} = K_2 \sqrt{\frac{H_0}{x}} \frac{1}{K_n} \qquad (10)$$

$$\frac{T_x - T_r}{T_0 - T_r} = K_2 \frac{\sqrt{A_0}}{x} \qquad (11)$$

$$\frac{T_x - T_r}{T_0 - T_r} = K_2 \frac{\sqrt{A_0}}{x} \frac{1}{K_n} \qquad (12)$$

where

T_x = centerline temperature of the jet at the distance of x;

T_r = return temperature;

T_0 = supply temperature at the diffuser.

The curves of temperature decay described by these equations are shown in Figure 3. Again, it is assumed that $\Delta t_0 = 20°R$; $T = 527°R$; $V_0 = 600$ fps; and $A_0 = 0.4$ ft^2; for linear jets, $K_1 = 1.8$, $K_2 = 1.7$, and $H_0 = 0.063$; for compact jets, $K_1 = 5.3$ and $K_2 = 3.4$; for radial jets, $K_2 = 1.7$. Vertical jets are all considered to be directed upward. It is interesting to note that the temperature decay rates of radial jet and vertically projected linear jets (cooled) are most rapid and they follow almost the same curve, though the temperature decay equations for these two jets are different (Equations 10 and 11). Actually, these two kinds of jets can use the same equation. From Figure 3, it can be seen that vertically projected cooled jets can achieve more rapid temperature decay rates than horizontal jets because of the gravitational force. Compact jets show a slower rate of temperature decay, and horizontally projected linear jets give a slower rate of temperature decay at greater values of x because of their two-dimensional characteristics.

Trajectory and Drop of Nonisothermal Jets For horizontally projected compact free jets, based on the analytical studies, the trajectory can be described by Equation 14 (Abramovich 1960; Shepelev 1978; Nosovitsky and Posokhin 1966; Omelchuk 1966; Filney and Nosovitsky 1967):

$$\frac{z}{\sqrt{A_o}} = \psi \frac{K_2}{K_1^2} Ar_0 \left(\frac{x}{\sqrt{A_o}}\right)^3 \qquad (13)$$

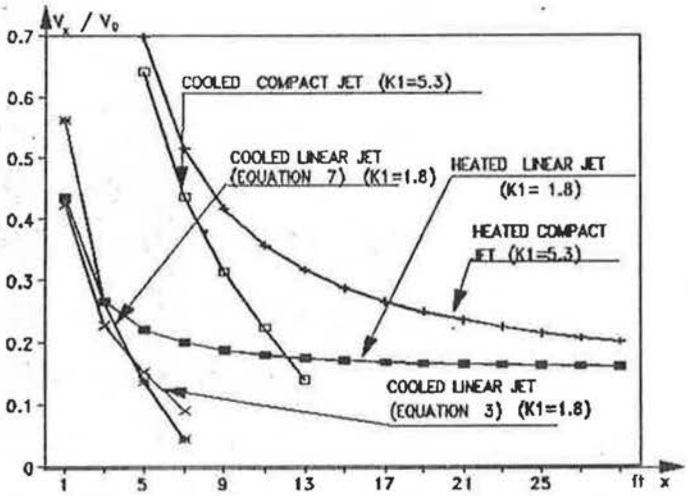

Figure 2 *Velocity decay of the centerline of nonisothermal jets.*

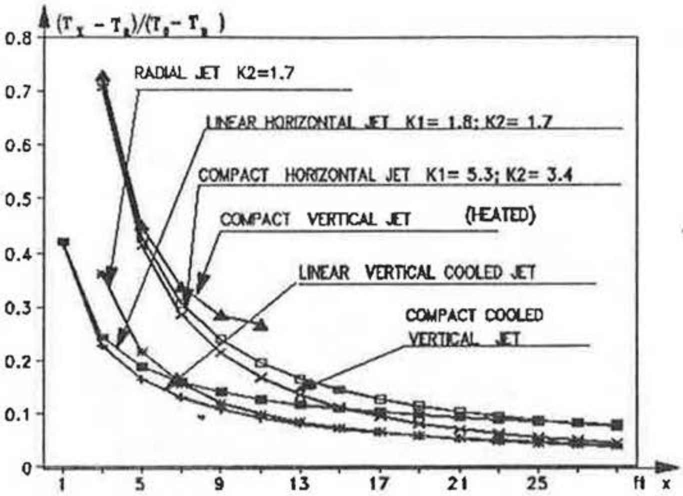

Figure 3 *Temperature decay of diffuser air jets.*

where

Ψ = coefficient, determined by diffuser type, size, etc.;
z = distance below the ceiling at which the maximum velocity in the air jet occurs.

This equation has been verified by experiments (Stein 1953; Koestel 1955; Grimitlyn 1969; Jackman 1970).

For inclined jets, an additional term can be added to the trajectory equation (Shepelev 1961):

$$\frac{z}{\sqrt{A_o}} = \frac{x}{\sqrt{A_o}}\, tg(\alpha_o) \pm \psi \frac{K_2}{K_1^2} Ar_0 \left(\frac{x}{\sqrt{A_o}}\right)^3 \qquad (14)$$

where

α_0 = angle of an inclined jet.

Experiments were conducted for inclined jets with different types of nozzles and grilles (Grimitlyn et al. 1987) to investigate the coefficients of the trajectory equations. It was found that the mean value of the coefficient Ψ obtained from experimental data was 0.47 ± 0.06. The accuracy of these values was suggested to be sufficient for designing the trajectory of inclined ventilation jets at an angle of $\alpha_o \leq \pm 45°$.

Effects of Ceilings/Walls

Isothermal ceiling jets were found to attach to the ceiling and flow along it if the initial jet axis was close to the ceiling due to the "Coanda' effect (Nottage et al. 1952a, 1952b, 1952c; Tuve 1953). The spread of the jet in the traversing direction was found to be reduced when the axis of a long jet was too close to the ceiling and parallel to it. The angle of divergence of the jet perpendicular to the wall was slightly less than one-half the angle of a free conical jet (Tuve 1953).

It was found that if an edge of the nozzle was in contact with the plane (the ceiling), so long as the axis of the nozzle formed an angle less than 40° to 45° with the plane, the jet would cling to the plane and spread over it. But if the edge of the jet was shifted away from the plane, air entrainment would occur on all sides of the jets, and the jet did not cling anymore (Baturin 1972). If the jets attached to a ceiling or a wall, K_1 became larger than the free jets. The values of K_1 were approximately those of the free jets multiplied by 1.4 (Nottage et al. 1952a, 1952b, 1952c; Miller 1990). When the temperature of the attached air jet is lower than the temperature of the ambient air, this jet will remain attached to the ceiling until the downward buoyancy force becomes greater than the upward static pressure ("Coanda" force). At this point, the jet separates from the ceiling and begins curving downward.

In designing a cooling ventilation system, careful calculation is needed to keep cold air from dropping directly into the occupied region. The distance to the diffuser face from the separation point of the jet (X_s) was found to be inversely proportional to the Archimedes number at the diffuser face (Equation 16). For slot diffusers, $a_1 = 1.6 \cdot A_0^{0.5}$, $b_1 = 0.5$ (Kirkpatrick et al. 1991), and $a_1 = 2.5H_0$ and $b_1 = 2/3$ (Rodahl 1977). For several different types of diffusers, $a_1 = 0.63 \cdot (K_1A_0)^{0.5}$, $b_1 = 1/2$ (Anderson et al. 1991), and $a_1 = 1.5 \cdot K_1 \cdot A_0^{0.5}$, $b_1 = 1/2$ (Nielsen and Moller 1987). It was also found that a cold air jet traveling a distance of about 60% of the room length before separating was regarded as sufficient to avoid uncomfortable drafts in the occupied zones.

According to theoretical analysis and experiments (Grimitlyn 1970), the separation distance of jets could be expressed by the following equations. For compact and incomplete radial jets, Equation 17 could be used; for linear and radial jets, Equations 18 and 19 should be used.

$$X_s = a_1\left(\frac{1}{Ar_0}\right)^{b_1} \qquad (15)$$

$$X_s = \frac{0.55\, K_1 \sqrt{A_0}}{\sqrt{K_2 Ar_0}} \qquad (16)$$

$$X_s = \frac{0.4\, K_1^{4/3} H_0}{(K_2 Ar_0)^{2/3}} \qquad (17)$$

$$X_s = \frac{0.45\, K_1 \sqrt{A_0}}{\sqrt{K_2 Ar_0}} \qquad (18)$$

It is interesting to note that Equation 18 is similar to Equation 16 where $b_1 = 2/3$ (Rodahl 1977), and Equations 17 and 18 are similar to Equation 16 where $b_1 = 0.5$ (Kirkpatrick et al. 1991; Anderson et al. 1991; Nielsen and Moller 1987). It should be pointed out that the constants of the equations are quite different even if they have similar forms.

Effect of Confinement

The influence of the reverse flow in the room on the centerline velocities, V_{xc}, and the temperature differential, Δt_{xc}, can be expressed by the coefficient K_c (Baharev and Troyanovsky 1958; Grimitlyn and Pozin 1973):

$$V_{xc} = V_x K_c \qquad (19)$$

$$\Delta t_{xc} = \Delta t_x \frac{1}{K_c} \qquad (20)$$

where

Δt_x = temperature difference between the centerline of the jet and the occupied region.

The values of K_c are available from graphs (Grimitlyn and Pozin 1973).

AIRFLOW AND TEMPERATURE CHARACTERISTICS IN OCCUPIED REGIONS

Prediction of Mean Air Velocity in Occupied Region

Jet Momentum Number The jet momentum number (Jm), jet momentum divided by room volume (Equation 21), of diffusers was reported to be proportional to air velocities at a level one foot above the floor (V_f) for high-aspect-ratio slot diffusers and perforated tubes located at the center of the ceiling (Ogilvie and Barber 1989):

$$Jm = \frac{Q_i U_i}{g V_l} \qquad (21)$$

where

Q_i = incoming volume flow rate;
g = gravity;
U_i = incoming air velocity;
V_l = room volume.

A good linear relationship between the jet momentum number and the mean air velocity at 1 ft above the floor (V_f, fpm) was confirmed through isothermal experiments in a full-scale room measuring 30 ft × 23 ft × 9 ft (Ogilvie and Barber 1989) (Table 1, Equation 23). It was concluded that a value of Jm of 7.5×10^{-4} or greater was needed to maintain the incoming air jet attached to the ceiling and provide energy for mixing. Experiments by Randall (1980) in a full-scale room measuring 12 ft × 25 ft × 7.5 ft under nonisothermal conditions (temperature difference between −2° and −13°F, jet momentum number 2 to 161×10^{-4}) revealed a best-fit curve (expressed in jet momentum number, Jm, Table 1, Equation 24). The difference between Equations 23 and 24 was not significant if Jm was between 4×10^{-4} and 40×10^{-4} as for Equation 23. But the difference became larger when Jm increased.

Momentum of Jet The room average velocity, V_a, was found to be highly correlated with the jet momentum (M) of the diffuser per unit room volume (V_l) (Miller 1976). Regression was done using Equation 25 (Table 1), based on the 286 tests from 5 different diffusers and 2 different room sizes, 20 × 12 × 9 ft, and 10 × 12 × 9 ft. The constants a and b were found to be dependent on diffusers. The a's varied from 0.604 to 2.02 and the b's from 0.325 to 0.543. It was suggested that each diffuser use its own regression equation for prediction of room average velocity. It was observed that there was larger uncertainty when the average room velocity was low.

The mean air velocity in occupied regions (V_R, fpm) has been correlated with the momentum of the jet (M, lb_f) and room geometry (Table 1, Equations 26 through 29) (Jackman 1970, 1971, 1973). It can be seen that at the same momentum, linear diffusion would result in lower room air velocity.

Jet Initial Velocities The room average velocity, V_a, was found to be linearly varied with the diffuser outlet velocities (Miller and Nevins 1972). This relation was also found to be dependent on the diffuser types and it was

TABLE 1
Equations for Predicting Mean Air Velocity in Occupied Region

No	Equations	Error	Conditions
1	$V_f = 21.3 + 31496Jm$ (23)	$R^2 = .95$	$4x10^{-4} <= Jm <= 40x10^{-4}$, linear diffusers, perforated tubes, ceiling level (Ogilvie et al. 1989)
2	$V_f = 995 x Jm^{.359}$ (24)	$R^2 = 0.85$	$2x10^{-4} <= Jm <= 161x10^{-4}$(Randall 1980)
3	$V_a = a(\frac{M}{V_l})^b$ (25)	——	286 tests from five types of Diffusers(High sidewall diffusers, light troffers, cones, slots, and sill grill) (Miller 1976)
4	$V_R = 515.0\sqrt{\frac{M}{B*H}}$ (26)	——	sidewall-grilles; room: 30 x 16 x 10 ft; grill location: 0.66 ft below ceiling (Jackman 1970)
5	$V_R = 261.0(\frac{M}{BH})^{0.33}$ (27)	——	sill-grilles; Location: 3.3 ft high along the sidewall (Jackman 1971)
6	$V_R = 115.6M^{0.6}$ (28)	——	circular ceiling diffuser (Jackman 1973)
7	$V_R = 71.5M^{0.4}$ (29)	——	linear ceiling diffuser (Jackman 1973)
8	$V_R = 1.11kV_{01}[1 + (1 + \frac{V_{02}^2}{V_{01}^2})^{0.5}]$ (30)	$\sigma/V_R = 0.026$	Two side-wall slots (Fissore 1991)

indicated that room volume would affect the slopes of the regression lines. The mean air velocity in the occupied regions, V_R, was correlated to the initial diffuser velocities (V_{01}, V_{02}) (Table 1, Equation 30). The test model room measured 8.2 × 14 × 9 ft. Slots were located 1.6 ft above the floor on the sidewall. The Reynolds number was from 1,500 to 6,000 and the Archimedes number wa up to 0.03 (Fissore and Liebecq 1990, 1991).

Turbulent Characteristics and Air Distribution

Standard Deviation The standard deviation (SD) of local air velocity fluctuations was found to have a linear correlation with the local mean air velocity, $\bar{U}$, to a certain extent (Thorshauge 1982; Sandberg 1987). But different studies resulted in different regressions (Hanzawa et al. 1987, results of 20 typically ventilated spaces; Kovanen et al. 1987, field measurement in 24 offices and dwellings). At a level close to the floor, the linear relationship became weaker. Therefore, measurement of both the mean velocity and the standard deviation is usually needed to describe the airflow characteristics more accurately.

Energy Distribution and Turbulence Intensity Turbulence intensity has been found to have a significant effect on human thermal comfort (Fanger et al. 1988; Melikov et al. 1987). Turbulence in a ventilated room is generated mainly in the diffuser air jet region due to the interaction of the jet with the room air and with the surface of the ceiling and walls or floor (Zhang et al. 1992). This turbulence is then transported to the occupied region by convective flow, but, at the same time, the turbulence is also damped due to the viscous effect. Therefore, turbulent kinetic energy in the occupied region is usually significantly smaller than in the jet region. However, the turbulence intensity in the occupied region may be high due to the low mean velocity there. Depending on the diffuser types, diffuser velocities, and the relative location between the jet and the occupied region, the average turbulence intensity may vary from 10% to 40% (Sandberg 1987; Zhang et al. 1992).

Turbulent kinetic energy within the room covers a wide range of frequencies, usually from 0.05 to 100 Hz. But the turbulent kinetic energy of airflow in the occupied region is distributed mainly in the low-frequency range (f < 10 Hz) (Zhang et al. 1992; Sandberg 1987). Hanzawa et al. (1987) used a linear function to correlate turbulence intensity with the local mean air velocity, but the correlation coefficients were low, ranging from 0.293 to 0.503 (Kovanen et al. 1987).

Velocity and Temperature Distribution Over Space The velocities at all measuring points were found to be normally distributed (Thorshauge 1982; Zhang 1991). The standard deviation of velocity distribution over the occupied zone (SDEV) was found to be influenced mainly by the mean air velocity V_R, and temperature distribution was found very close to Gaussian distribution (Fissore and Liebecq 1991). Equations 31 and 32 could be applied when one or two slot diffusers were used, and error could be estimated by $\sigma/V_R = 0.0053$:

$$\mathrm{SDEV} = 0.33\, V_R \quad (22)$$

$$\mathrm{SDEV} = 0.30\, V_R \quad (23)$$

The distribution of temperature differences, Δt, between the temperatures in the occupied region ($T_{o,r}$) and the temperature of the jet at the point where it enters the occupied region (T_x) was found to be close to the Gaussian assumption (Grimitlyn et al. 1986). The standard deviation of this distribution ($\sigma_{\Delta t}$) was found to be proportional to the temperature difference (Δt_x) between T_x and the mean value of the temperature in the occupied region, $\sigma_{\Delta t} = K_t\, \Delta t_x$. K_t was found to be about 0.8 for compact and conical ceiling jets, and about 1.5 for radial ceiling jets and for compact sidewall jets.

Internal heat load may have a significant effect on the room air distribution (Christianson et al. 1990). The buoyancy effect would cause the jet to fall more quickly and then cause a higher spatial average of mean air velocity in the occupied region. The heat load was also found to contribute to turbulence production and increase turbulent kinetic energy and turbulence intensity in the occupied region (Zhang 1991). Internal obstructions was found to affect the flow pattern and air distribution, as well as the mean air velocities in the occupied region (Zhang 1991).

Discussion

It can be summarized from the above analysis that the key factor affecting the mean air velocity in the occupied region was considered to be the momentum of the incoming jet. Room size and geometry and diffuser types and locations were also considered to be important, but it seems difficult to quantify all of them. Internal heat load and obstructions may also affect the mean air velocity of the occupied region, but few experimental results of these factors were built into the prediction models. For the existing models, good prediction results could be expected if the ventilation conditions are close to the test conditions under which these equations were generated. A data base is needed to correlate the mean air velocity in the occupied regions with jet momentum, etc., under various ventilation conditions. With this data base, it will be possible to predict the mean air velocity in the occupied regions at the design stage.

Since turbulence in the occupied region is transported mainly from the jet region where it is generated, a relationship exists between the turbulence in the occupied region and the flow characteristics of the diffuser air jets, room geometry, and internal obstructions. Therefore, it is possible to develop a method that can use the jet characteristics to predict the turbulence intensity as well as the mean velocity and temperature in the occupied regions.

CONCLUSIONS

1. It is possible to predict the velocity and temperature characteristics of the jets at the point where they enter the occupied region using the current jet theory. However, further research is needed to relate the jet characteristics to the airflow and temperature in the occupied region. It is especially important to quantify the effects of room geometry, obstructions, and room partitions in order to have a better understanding of room air motion and predict the thermal comfort conditions in occupied regions during the design stage.
2. The mean air velocity in the occupied region has been correlated with the jet momentum, jet momentum number, and room sizes. The effects of diffuser types and location, internal heat loads, and obstructions need to be studied and built into prediction models. A data base built on various ventilation systems and room geometries will help to produce consistent prediction models.
3. A relationship exists between the airflow turbulence in the occupied region and the characteristics of diffuser air jets. Research is needed to develop a method that can predict the turbulence intensity as well as mean velocity and temperature in the occupied regions.

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