Predicting and Evaluating Draft in Summer Cooling

Predicting and Evaluating Draft in Summer Cooling

Hideki Kubota

ABSTRACT

A scale for evaluating draft in summer cooling is derived in terms of the Predicted Percentage of Dissatisfied (PPD) by applying Fanger's Predicted Mean Vote (PMV) equation and by introducing a model of air distribution.

An analytical procedure to predict values of the maximun air velocity and also the lowest air temperature in an occupied zone is presented.

Based on these techniques, the optimum value of a product : (diameter of the outlet) × (air velocity at the outlet) is suggested for high sidewall air-supply systems.

The values of PPD for drafts analytically obtained in this paper are correlated to the values of ADPI proposed by Miller et al., and it is observed that the minimum PPD corresponds to the maximum ADPI.

INTRODUCTION

This paper presents a guide for optimum design for room air distribution in summer cooling in which the main problem to be solved will be dealing with local areas of discomfort caused by excessive air motion combined with lowered air temperature.

In 1938 Houghten et al.[1] first presented the relationship between draft and the sensory responces from human subjects. However, this relationship is not always applicable for summer cooling, since the experiments in which this relationship was observed were conducted at a temperature level of 21℃ for subjects with one clo insulation of clothing, while summer cooling temperatures are around 26℃ and summer clothing is about 0.5clo or less.

Rydberg et al[2]. derived an equation for draft temperature in 1949. As recommended by Miller[3], this scale now needs to be updated.

Miller et al[4]. proposed a single number index, Air Diffusion Performance Index (ADPI), in 1964 which enables us to evaluate three dimensional room air distribution. Miller also expanded the defintion of the ADPI by applying the new Effective Temperature in 1976[3] and the Predicted Percentage of Dissatisfied (PPD) in 1975[5] so that the new ADPI (designated as ADPI-2) is more closely related to modern comfort criteria. This method of using a single number index has made a very significant contribution and has helped make a lot of data available for practical use.

The present paper derives a scale for evaluating draft by applying fanger's Predicted Mean Vote (PMV) and the PPD techniques, and may provide another basis for the support of the ADPI-2 scale. To provide a clear basis for this scale, a model of room air distribution is established, after considering the most extreme conditions of the draft in the occupied zone which are expected to appear in the region where a discharged cooled air mass hits the subjects.

The analytical procedure described in this paper to predict both maximum air velocity and air temperature difference in the occupied zone enables us to estimate both the optimum condition for the air being supplied and the size of the outlet.

Our present discussion deals with a high sidewall air-supply system, however, the principles described can be applied to other air-supply systems.

PREDICTED PERCENTAGE OF DISSATISFIED(PPD)IN AN AIR JET

An actual system of air distribution is too complex for direct analysis but by simplifying assumptions an adequate model may be obtained.

Consider a system in which the following assumptions apply:

1. The occupied zone is divided into two regions : one is a still air region and the other a cool air jet region.
2. The still air region is maintained at 26.5℃ dry-bulb, air velocity 0.15m/s and relative humidity 50% to satisfy the comfort requirments of normally clothed sedentary human subjects (0.5 clo, 1Met).
3. Mean radiant temperature is the same all over the room and equal to the temperature in the still air region.
4. In the jet region both air velocity and temperature are represented respectively by the highest and lowest values appearing in this region.

When the occupants take position in the cool jet region, they may feel to be cool. The degree of coolness can be estimated by using the equation for the Predcted Mean Vote (PMV) proposed by Fanger. The value of PMV can be used to estimate the Predicted Percentage of Dissatisfied (PPD).

Fig.1 shows the combinations of air velocity and temperature difference between jet and still air for which the PPD remains constant. The analytical procedure for deriving this relationship is presented in Appendix A.

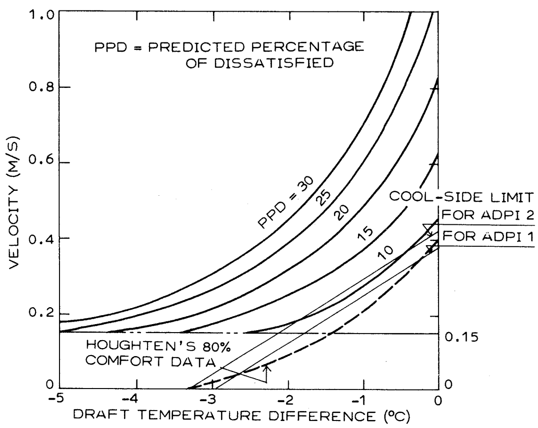

Fig.1 Predicted Percentage of Dissatisfied for subjects in cool jet region

It appears that the line representing the cool-side limit for ADPI-2 coinsides with the curve for 10% of PPD. This result could support the basis for ADPI-2.

An approximate function for lines of constant PPD value derived by trial and error is as follows:

$$PPD=(6\sqrt{V_J-0.15}+\Delta T_J)^2 \quad (\text{for } PPD<30) \tag{1}$$

where V_J = air velocity in the jet region, m/s

ΔT_J = air temperature difference between jet and still air regions, °K.

The broken line in Fig.1 indicates Houghten's 80% comfort data.

PREDICTION OF THE MAXIMUM VELOCITY AND AIR TEMPERATURE DIFFERENCE BETWEEN JET REGION AND STILL AIR REGION

The maximum air velocity and the lowest air temperature in the jet region are given respectively as centerline velocity and air temperature of air jet at a point where the centerline meets the upper boundary of the occupied zone (see **Fig.2**) . By assuming that the drop of jet is not too great, the centerline air velocity at the upper boundary can be approximated by the following equation:

$$V_J=K_P D_O V_O / X_J \tag{2}$$

where V_J = Centerline velocity of the jet at the point where the centerline meets the upper boundary of the occupied zone, m/s

K_P = constant

D_O = diameter of the outlet, m

V_O = air velocity at the outlet, m/s

X_J = horizontal distance from the outlet to the point where the centerline meets the upper boundary of the occupied zone, m.

The drop of the cool air jet was given by Koestel[7] as the equation:

$$\frac{Y}{D_0}=0.42\frac{g\beta\Delta T_0 D_0}{K_P V_0^2}\left(\frac{X}{D_0}\right)^3 \tag{3}$$

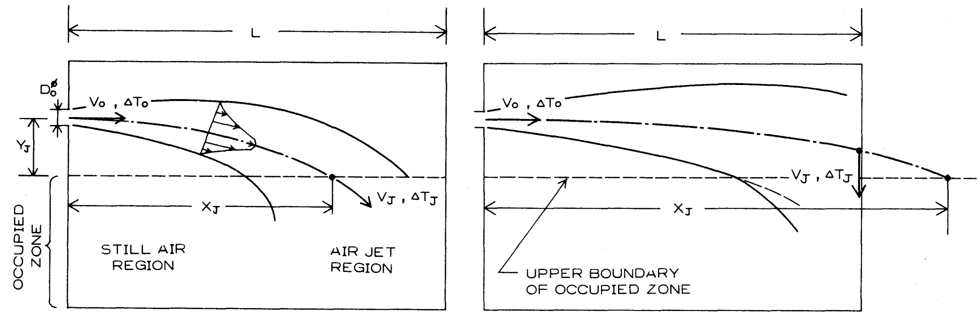

Fig.2 Schematic flow pattern in summer cooling

where Y = vertical distance from the center of the outlet to a point on the centerline of jet, m

X = horizontal distance from the outlet to a point on the centerline of jet, m

ΔT_O = air temperature difference between outlet and still air, °K

g = gravitational acceleration, m^2/s

β = volume coefficient of thermal expansion.

This equation can be rearranged into the following form :

$$\frac{D_0 V_0}{X} = 0.0246\left(\frac{q}{K_P Y}\right)^{1/3} \quad (4)$$

or

$$X = 40.7\left(\frac{K_P Y}{q}\right)^{1/3} D_0 V_0 \quad (5)$$

where q = sensible heat load of room supplied from an outlet, W

$$= c\rho \frac{\pi}{4} D_O^2 \; V_O T_O \quad (6)$$

c = heat capacity of air, J/K

ρ = density of air, Kg/m^3.

Substitute Eq.4 into Eq.2, we have,

$$V_J = 0.0246 K_P^{2/3} \left(\frac{q}{Y_J}\right)^{1/3} \quad (7)$$

where Y_J = vertical distance from the center of the outlet to the upper boundary of the occupied zone, m.

This result suggests that the maximum air velocity is independent of the conditions at the outlet : V_O, ΔT_O and D_O.

In a similar way, the lowest air temperature is given in the form of air temperature difference as follows:

$$\Delta T_J = 0.82 K_P D_O \Delta T_O / X_J \quad (8)$$

where ΔT_J = maximum air temperature difference between jet and still air region, K.

Eliminating ΔT_O from Eq.8 by applying Eq.5 and 6, we have,

$$\Delta T_J = 0.0365 K_P^{4/3} Y_J^{1/3} q^{2/3} / X_J^2 \quad (9)$$

or

$$\Delta T_J = 2.21 \times 10^{-6} K_P^{2/3} q^{4/3} / [Y_J^{1/3} (D_0 V_0)^2]. \quad (10)$$

The maximum difference ΔT_J appears to decrease with increasing $D_O V_O$.

From the results described above, the larger value of X_J(or $D_O V_O$) will be recommended, since the value ΔT_J decreases with increasing X_J, while the value of V_J does not vary with X_J. However, this recommendation is only applicable where the value of X_J is smaller than the distance to the wall perpendicular to the jet or mid-plane between opposite outlets.

When the value of X_J exceeds room length parallel to the jet, supposing the opposite wall is not located, the jet comes into the occupied region after being disturbed by the opposite wall. In this case, the maximum air velocity and air temperature difference may appear near the opposite wall, and these values are approximated as follows:

$$V_J = 0.0246 K_P^{2/3} \left(\frac{q}{Y_J}\right)^{1/3} \left(\frac{X_J}{L}\right) \tag{11}$$

$$\Delta T_J = 0.0365 K_P^{4/3}\, Y_J^{1/3}\, q^{2/3} \,/\, (L X_J) \tag{12}$$

where L = distance from outlet to the nearest wall perpendicular to jet or mid-plane between opposite outlet, m

X_J = in this case, an imaginary horizontal distance supposing the opposite wall is not located, m (see Fig.2).

As may be seen in the above equation, an increase in X_J (or $D_O V_O$) raises V_J while reducing ΔT_J. From the standpoint of PPD, it can be concluded that the value of PPD increases with X_J according to the results of the calculations being made for PPD applying the values to practical situations.

From the facts described above, we can conclude that the minimum value of PPD will be obtained when we make the value of X_J equal to the room length L.

This optimum condition is expressed as folltws.

$$(D_O V_O)_{opt} = 0.0246 \left(\frac{q}{K_P Y_J}\right)^{1/3} L \tag{13}$$

COMPARISON WITH TEST DATA

In **Table 1,** the analytical values estimated by Eq.5, 7 and 10 are compared with the test data obtained by Nelson and Stewart[8] on chilled air projected into a room from outlets of various dimensions with the same opening area. It gives good agreement between test and analytically estimated values.

TABLE 1 Comparison between experimental data by Nelson & Stwart and analytical values derived in this paper

	Original Data			Predicted Values		
Outlet Dimmension (M x M)	X_J (M)	V_J (M/S)	ΔT_J (deg °C)	X_J (M)	V_J (M/S)	ΔT_J (deg °C)
0.28 x 0.36	6	1.5	-2.4	6.3	1.5	-2.7
0.40 x 0.20	6	1.25	-1.9			
0.48 x 0.18	6	1.25	-1.9			
0.71 x 0.13	6	1.5	-2.3			
0.91 x 0.10	6	1.75	-2.7			

D_O = 0.324(M), V_O = 5.0(M/S), ΔT_O = -11.1(deg°C), q = 5.8 x 10^3(W), Y_J = 0.8(M), K_P = 6

COMPARISON WITH VALUES OF ADPI OBTAINED BY MILLER ET AL.

It is of considerable interest to correlate the analytical values of PPD with the ADPI experimental results obtained by Miller et al[9]. As indicated inAppendix B, based on flow patterns and data shown in his paper, the value of K_P is assumed to be about 3 and the drop of the cool air jet trajectory to be about 30% of the value calculated by Eq.3. This may be attributed to the effect of the ceiling surface: a kind of Coanda Effect. As we have little knowledge of this effect, we will tentatively apply the following expression as an equation for the drop of the cool air jet :

$$\frac{Y}{D_O}=0.42\frac{g\beta\Delta T_O D_O}{K_P V_O^2}\left(\frac{X}{D_O}\right)^3 A \tag{14}$$

where A = coefficient representing ceiling effect (= 0.3).

This change of equation will lead to the following equations.

$$V_J=0.0246K_P^{2/3}\left(\frac{Aq}{Y_J}\right)^{1/3}$$

$$=0.0397q^{1/3} \tag{15}$$

$$\Delta T_J=0.0365K_P^{4/3}Y_J^{1/3}q^{2/3}/(A^{1/3}X_J^2)$$

$$=0.20q^{2/3}/X_J^2 \tag{16}$$

where $K_P=3$, $A=0.3$ and $Y_J=0.64$m.

In Miller's paper, the values of ADPI were plotted as a function of the ratio $T_{0.25}/L$. The value of L (room length) was 6.1m (20ft). The synbol $T_{0.25}$ describes "the throw" of the jet which is definds as the distance from the outlet to a point in the air stream where the maximum velocity occuring in the stream cross-section has been reduced to a selected terminal velocity of 0.25m/s. Based on this definition, the following relationship is given from Eq.1.

$$K_P D_O V_O=0.25T_{0.25} \tag{17}$$

Combining Eq.2, 15 and 17, we have,

$$X_J=\frac{0.25T_{0.25}}{V_J}=\frac{0.25T_{0.25}}{0.0397q^{1/3}}=6.3q^{-1/3}T_{0.25}. \tag{18}$$

Substituting Eq.18 into 16, the following is given for ΔT_J :

$$\Delta T_J=5.1\times10^{-3}q^{4/3}/T_{0.25}^2 \tag{19}$$

The value of $T_{0.25}/L$ at $X_J=L$, where it is suggested the smallest value of PPD occurs, is given from Eq.18 as follows:

$$\frac{T_{0.25}}{L}=0.16\,q^{1/3}. \tag{20}$$

TABLE 2 Correlation between ADPI by Miller and analytically estimated values of PPD

Original Data			Predicted Values		
Room Load (W/SQ-M)	Throw $T_{0.25}/L$	ADPI (%)	PPD (%)	V_J (M/S)	ΔT_J (deg°C)
63	0.8	76	47	0.44	-3.1
	1.6	85	16	·0.44	-0.78
	1.8* 2.2	75	18	0.55	-0.52
	2.9	29	23	0.73	-0.40
	3.5	23	25	0.88	-0.33
126	0.8	40	> 70	0.56	-8.1
	0.95	67	> 70	0.56	-5.7
	1.6	79	39	0.56	-2.0
	2.1* 2.2	71	24	0.56	-1.1
	3.5	23	32	0.88	-0.67
189	1.6	72	65	0.63	-3.3
	2.2	69	39	0.63	-1.8
	2.4* 3.5	23	37	0.88	-0.97
252	1.6	67	> 70	0.70	-5.1
	2.2	65	60	0.70	-2.7
	2.6* 3.5	23	44	0.88	-1.3

* : Values of $T_{0.25}/L$ at the point where the characteristic distance of air jet X_J is equal to the characteristic room length L.

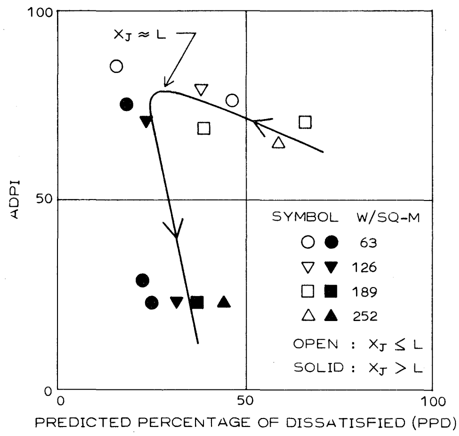

Fig.3 Correlation between PPD and ADPI

In the case: $X_J > L$

The following equations give the expressins for V_J and $\triangle T_J$.

$$V_J = \frac{K_P D_O V_O}{L} = 0.25 \frac{T_{0.25}}{L} \tag{21}$$

$$\Delta T_J = 0.0365 \frac{K_P^{4/3} Y_J^{1/3} q^{2/3}}{A^{1/3} L X_J} = 8.4 \times 10^{-4} q / \left(\frac{T_{0.25}}{L}\right) \tag{22}$$

The values of V_J and ΔT_J corresponding to $T_{0.25}/L$ as tested by Miller have been calculated and are listed in **Table 2.**

Fig.3 shows ADPI plotted as a function of the PPD value.

As may be seen from Fig.3 and Table 2, the maximum values of ADPI closely correspond to the minimum PPD and they appear to be centered around $X_J = L$.

CONCLUSION

1. A scale for evaluating draft in summer cooling was derived in terms of Predicted Percentage of Dissatisfied (PPD) by applying Fanger's Predicted Mean Vote (PMV) equation and by introducing a model of air distribution.
2. An analytical procedure to predict values of the maximum air velocity and the maximum air temperature difference between jet and still air regions in the occupied zone was presented for high sidewall air-supply systems.
3. A comparison between data obtained by Nelson et al. and the caluculated values was made and showed good agreement.
4. The values of PPD, analytically obtained in this paper, were compared with the values

of ADPI obtained by Miller et al. and it was observed that the minimum PPD corresponded approximately to the maximum ADPI.

5. For minimizing the PPD caused by drafts, the horizontal distance X_J, from the outlet to a point where the air jet centerline meets the upper boundary of the occupied zone, was suggested to be equal to the distance from the outlet to the nearest wall perpendicular to the jet.

REFERENCES

1. Houghten, F.C., Gutberlet, C. and Wilkowski, E.,"Draft Temperatures and Velocities in Relation to Skin Temperatures and Feeling of Warmth", *ASHVE TRANSACTIONS, 44,* 1938.
2. Rydberg, J. and Norback, P.,"Air Distribution and Draft."*ASHVE TRANSACTION, 55,* 1949.
3. Miller, P.L.,"Application Criteria for the Air Diffusion Performance Index (ADPI)," *ASHRAE TRANSACTION, 82,* 2, 1976.
4. Fanger, P.O., *Thermal Comfort,* McGraw-Hill, 1972.
5. Koestel, A., "Paths of Horizontally Projected Heated and Chilled Air Jet," *ASHVE TRANSACTION, 61,* 1955.
6. Nelson, D.W. and Stewart, D.J., "Air Distribution from Side Wall Outlets,"*ASHVE TRANSACTION, 44,* 1938.
7. Miller, P.L. and Nash, R.T., "A Further Analysis of Room Air Distribution Performance", *ASHRAE TRANSACTION, 77,* 1971.

APPENDIX (A)

The following equation for Predicted Mean Vote proposed by Fanger was modified by introducing linear approximation for radiant heat exchange.

$$PMV=(0.303e^{-0.036(M/A_{Du})}+0.028)[\frac{M}{A_{Du}}(1-\eta)-$$
$$3.05[5.73-0.007\frac{M}{A_{Du}}(1-\eta)-P_J]-0.42[\frac{M}{A_{Du}}(1-\eta)-58.1]-$$
$$0.0173\frac{M}{A_{Du}}(5.87-P_J)-0.0014\frac{M}{A_{Du}}(34-t_J)-$$
$$3.95f_{cl}(t_{cl}-t_{mrt})-f_{cl}h_c(t_{cl}-t_J)] \qquad (A-1)$$

where t_{cl} is determined by the equation

$$t_{cl}=[35.7-0.0275\frac{M}{A_{Du}}(1-\eta)+0.155I_{cl}f_{cl}(3.95_{mrt}+h_c t_J)]$$
$$/[1+0.155I_{cl}f_{cl}(3.95+h_c)] \qquad (A-2)$$

and h_c by

$$h_c=12.1\sqrt{V} \qquad (A-3)$$

where M = metabolic rate, W/m^2

A_{Du} = DuBois area, m^2

P_J = partial pressure of water vapour in jet region, kP_a

t_J = air temperature in jet region, ℃

t_{cl} = mean temperature of outer surface of clothed boby, ℃
t_{mrt} = mean radiant temperature in jet region, ℃
f_{cl} = ratio of the surface area of the clothed body to the surface area of the nude boby, ND
h_c = convective heat transfer coefficient, W/(m²·K)
I_{cl} = thermal resistance from the skin to outer surface of the clothed body, clo
η = external mechanical efficiency of the boby, ND

From the assumptions concerning air distribution, the conditions in the jet region are expressed as follows :

mean radiant temperature	$t_{mrt}=t_a=26.5$°C	
air temperature	$t_J=t_a+\Delta T_J$°C	(A−4)
vapor pressure	$P_J=P_a+\Delta P_J$ kPa	

where suffix "a" implies the value for still air region.

When the sensible heat ratio (SHR) of the room concerned was given, the value of ΔP_J can be expressed in the function of ΔT_J.

$$SHR=c\Delta T_O/(c\Delta T_o+\lambda\Delta W_O)=1/\ [1+\lambda W_O/(c\Delta T_O)] \qquad (A-5)$$

where ΔT_O = air temperature difference at the outlet, °K
ΔW_O = humidity ratio difference at the outlet, g/Kg
λ = heat of vaporization of water, J/Kg.

In the jet stream, where the process of vapor diffusion is similar to that of heat diffusion, we have the following relations.

$$\frac{\Delta W_J}{\Delta T_J}=\frac{\Delta W_O}{\Delta T_O} \qquad (A-6)$$

Substituting Eq. A−6 into A−5, we have,

$$\Delta W_J=\frac{c}{\lambda}\left(\frac{1}{SHR-1}-1\right)\ \Delta T_J. \qquad (A-7)$$

As the vaule of SHR for an office room is around 0.7, let SHR=0.7,

$$\Delta W_J=0.17\Delta T_J. \qquad (A-8)$$

By conversion of units from g/Kg to kPa, the value of ΔP_J is obtained :

$$\Delta P_J=0.16\Delta W_J=0.027\Delta T_J. \qquad (A-9)$$

Substituting Eq. A−2, A−4 and A−9 into Eq. A−1, and solving the Eq. A−1 for ΔT_J

TABLE 3 PMV vs. PPD

PMV	0	-0.49	-0.67	-0.83	-0.97	-1.08	-1.28
PPD(%)	5	10	15	20	25	30	40

	PMV
Neutral	0
Slightly Cool	-1
Cool	-2
Cold	-3

gives the following expressions :

$$\Delta T_J=[PMV-A_1(A_2-A_3+A_4P_a+A_5t_a-f_{cl}[A_9+A_{10}(3.95+h_c)t_a])]/A_{11} \tag{A-10}$$

where $A_1=0.303e^{-0.036(M/A_{Du})}+0.028$

$$A_2=\frac{M}{A_{Du}}(1-\eta)-0.42[\frac{M}{A_{Du}}(1-\eta)-58.1]$$

$$A_3=3.05[5.73-0.007\frac{M}{A_{Du}}(1-\eta)+0.0173\times5.87\frac{M}{A_{Du}}+0.0014\times34\frac{M}{A_{Du}}$$

$$A_4=3.05+0.0173\frac{M}{A_{Du}} \qquad A_5=0.0014\frac{M}{A_{Du}}$$

$$A_6=35.7-0.0275\frac{M}{A_{Du}}(1-\eta) \qquad A_7=0.155I_{cl}f_{cl}$$

$$A_8=1+0.155I_{cl}f_{cl}(3.95+h_c) \qquad A_9=(3.95+h_c)A_6/A_8$$

$$A_{10}=(3.95+h_c)A_7/A_8-1 \qquad A_{11}=A_1[0.027A_4+A_5-A_{10}f_{cl}h_c]$$

Using Eq. A−10, it is possible to calculate combinations of ΔT_J and V_J that correspond to a certain value of PPD. The relationship between PPD and PMV is listed in **Table 3.**

APPENDIX (B)

The value of K_P is assumed by using the following equation for isothermal free jets.

$$V_c=K_PD_OV_O/X \tag{B-1}$$

$$K_P=V_cX/(D_OV_O) \tag{B-2}$$

From the definition of the throw, when the value of X equals $T_{0.25}$ the value of V_c is 0.25m/s, then we have,

$$K_P=0.25T_{0.25}/(D_OV_O) \tag{B-3}$$

The diameter of the outlet and the air velocity at the outlet are as follow,

$$\frac{\pi}{4}D^2_O=S=0.0929\ m^2\ (24''\times6'') \tag{B-4}$$

$$D_O=\sqrt{\frac{4}{\pi}}\ S=0.344\ m^2 \tag{B-5}$$

$$V_O=G/S \tag{B-6}$$

where S=area of the outlet, m^2

G=air flow rate through the outlet, m^3/s.

Miller's paper shows the values of $T_{0.25}/L$ for values of G.

Applying these values for the equations described above, we can get the value of K_P to be around 3.

The value of coefficient "A" representing the ceiling effect, is defined as follows :

$$A = \frac{Y/D_0}{0.42 \dfrac{g\beta \Delta T_0 D_0}{K_P V_0^2}} \qquad (B-7)$$

This equation can be rearranged in the following form in the same way as Eq.5.

$$A = (40.7)^3 \frac{K_P Y}{q} \left(\frac{D_0 V_0}{X} \right)^3 \qquad (B-8)$$

Miller displays the flow patterns for a room load of 20 Btuh/sq-ft(63W/sq-m) with changing flow rates. We can obtain two values of X for the drop of Y (=0.64m) ; about 1.7m for 0.6cfm/sq-ft and 2.8m for 1.0 cfm/sq-ft.

Based on these values, we can assume the value of A to be around 0.3.

(Received May 18, 1981)