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Investigation of the mechanism for air supply with horisontal displacement flow at occupied zone containing concentrated heat source

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SUMMARY

In this paper, the characteristic of the flowage of air and heat distribution in the air conditioned room with horizontal air supply system placed in occupied zone is studied. By the way of theoretical analysis with model test, It approaches the affect of performance between the horizontal cold supply jet and natural convective heat jet formed by heat source, and the optimum coordinative state of these two jets flow is analyzed. corresponding with this state, the occupied zone gets the least heat gain. Morover, The authors find out some main influence factors which affect the air flowage and heat distribution in the room, and put forward an empirical formula for calculating the ratio number which shows how much heat would get into the occupied zone.

Investigation of the Mechanism for Air Supply With Horizontal Displacement Flow at Occupied Zone Containing Concentrated Heat Source

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Introduction:

It's understood that low level air supply has the characteristic of higher efficiency and lower energy consumption. The air supply with horizontal dislacement flow is one type of the low level air supply system, but till now there are few deign data and it's theoretical investigation are also not enough. For the purpose of further investgation of the mechanism of this type air supply and it's application characteristics, we carried out it's experimental investigations. Air supply installations we used are swirling outlets and low velocity perforated cabinet type air outlets (all made by ourslf). Because of the lower performance of swicling outlet, we didn't conclude it in this paper. The chief purpose of this article is to proceed an overall analysis and conclusion to the latter type of air outlet.

Air flow movement in room with horitontal air supply at occupied zone

From Figure 1 we can see that the horizontal mechanical cold supply jet and the upward natural convective hot jet originated by the heat source form cross movement. At the interface between cold and hot jets occurs intensive turbulent heat exchange. In result of the mutual actions of two jets formed air movement pattern and heat distribution state in the room.

The degree of mutual influence between horizontal supply jet and convective jet of heat source varies with the changes of heat emission from heat source and intensity of supply air flow. Intensity of supply air flow rate and velocity and heat emission of the heat source in practice represent the amount of kinetic energy of supply jet and hot jet. Considering the heat gain in the occupied zone of air conditioned room, too much or too small air supply will all cause the increasing of heat gain in occupied zone.

When air supply is too much, the supply jet will pierce deeper into the hot jet. The mixing of supply jet and hot jet in the interface is intensive. As a result of heat exchange of cold and hot jets, heat gain in occupied zone from the heat source is increased.

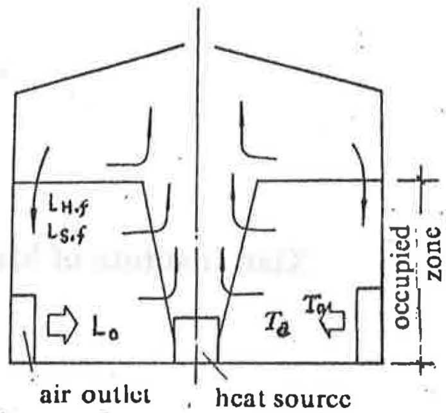

Fig. 1 Air supply sketch map

When air supply is too small, it cann't satisfy the amount of entrainment of the hot jet. In result, the high level hot air will feed back to compensate the deficiency of supply air. At this time, the convective heat exchange between two jets is weakened. But the local heat gain in occupied zone will still be increased because it gains a large amount of feed back heat. So to a known heat emission from a heat source, there exists a certain intermediate air supply between too much and too small air supplies. Using this intermediate air supply will minimize the heat gain of the occupied zone in air conditioped room. We define this coordinative condition between supply jet and hot jet "Optimum condition".

Heat exchange proccess of horizontal air supply at occupied zone

In air conditioned room with horizontal supply at occupied zone, there exist mainly two types of heat exchange:

The one is the convective heat exchange between cold and hot jet and between room air flow and surfaces of heat source and surrounding walls of enclosing construction. Heat gain of the occupoied zone in an air conditioned room contains not only covection heat Q_c and radiation heat Q_r, but also the feed back heat from the high level space Q_f.

Then the heat gain in the occupied zone:

$$Q_o = Q_c + Q_r + Q_f \tag{1}$$

Convection heat gain in the occupied zone:

$$Q_c = \int_{h_0}^{1.8} q_c 2\pi r dz + Q_c' \tag{2}$$

Where q_c expresses heat emission passing through out hot jet surface area in unit time.

From reference [1] we'll have

$$q_c = \rho_1 v_1 l_1 dI / dz \tag{3}$$

where $\rho_1 v_1$ are density and cross sectional mean velocity of supply jet on the interface respectively;

l_1 is the length of mixing travel;

h_o is the height of heat source;

I is the enthalpy of ascending hot jet;

Q_c' is the convective heat exchange between supply jet flow and the side surface of heat source.

Prandtl's mixing travel length theorem considers that there exists a certain proportion between the mixing length l_1 and piercing depth h produced by supply jet to hot jet.

In this article we define $l_1 = C_x h$ where C_x is non-dimensional empirical coefficient. Piercing depth

$$h = 1.9do \sqrt{\frac{\rho_1 v_1^2}{\rho_e v_z^2}} \tag{4}$$

From refernce [4] we have

$$v_z = k_1 Q_C^{1/3} z^{1/3} + b_1 \tag{5}$$

where do is the equivatent diameter of cabinet outlet, ρ_1, v_1 are mentioned as above, v_z is the ascending velocity of heat jet, ρ_e is the exhaust air density instead of ascending hot jet density ρ_z, k_1 and b_1 are empirical coefficients which are defind in this article. Q_C is the convective heat transfer of heat source.

We take from hot jet a differential element with height d_z. By analyzing the heat balance of this differential element, we obtained that enthalpy I_z on a certain level is the function of convection heat emission of heat source Q_C and level z expressed as follows:

$$I_z = \varphi(Q_C, z) \tag{6}$$

From reference [2], Q_c' may be expressed by following formula:

$$Q_c' = \varphi'(v_1, \rho_1, Q_C, t_a, F_{h,s}) \tag{7}$$

where t_a is the mean temperature in occupied zone, $F_{h,s}$ is side surface of heat source.

Substitute formulas (3) (4) (5) (6) (7) into (2), we obtain a series of factors which affect Q_c:

$$Q_c = F_1(\rho_1 v_1^2, \rho_e v_z^2, Q_C, d_e, d_o, t_a, F_{h,s}) \tag{8}$$

where d_e is the equivalent diameter of the top of heat source. The other symbols have been defined as mentioned above.

Radiant heat gain Q_r in occupied zone

$$Q_r = \varepsilon F \sigma_b (T_H^4 - T_W^4)$$

where ε is the blackness of room wall surface, σ_b is the absorption rate of room wall surface, T_H,T_w are surface temperature of heat source and room wall temperature respectively ^{0}k.

For a certain enclosure, the values of ε, F, δ_b are definite. Surface temperature of heat source $T_H = f(\rho_1 v_1 Q_C)$ and wall temperature of enclosure T_w are relating to temperature of occupied zone t_a and air velocity near walls (this velocity is the function of v_1). F is the surfaces of enclosure in occupied zone. Then exists the following formula:

$$Q_r = F_2(\rho_1, v_1, Q_C, t_a) \tag{9}$$

Feedback heat Q_f from high level space

Total feed back air from high level space equats to the sum of feedback air of high level space caused by entrainment of hot jet and horizontal supply jet. While the total entrainment $L_{H,en}$ of hot jet equals to the flow rate L_z required in forming the hot jet.

From reference [3] we know $L_{H,en} = L_z = 2\pi v_z c^2 z^2$, the feedback flow rate from high level space hot air, which is entrained by hot jet and enters into the occupied zone (the space under 1.8^m level) is only:

$$L_{H,f} = L_{z1.8m} - L$$

where $L_{z1.8^m}$ is the flow rate of hot jet in 1.8^m level, L is the supply air flow rate..

Air entrained by circular supply jet:

$$Le'n = 0.32(\rho_0 / \rho_e)^{1/2} x / d_0 L$$

Referring to upper mentioned formula, we may consider that the entrained air by perforated cabinet type air outlet supply jet L_{en} is the function of densities ρ_o,ρ_e, non–dimensional distance x / d_0 and air supply L, thus we have $L_{en} = f(\rho_o, \rho_e, x / d_o, L)$.

If we approximately consider that all entrained air by supply jet L_{en} is the feed back air $L_{s.f}$ from upper part above the occupied zone, then:

$$L_{s,f} = L_{en} = f(\rho_o, \rho_e, x / d_o, L)$$

While the feedback heat from high level space may be expressed:

$$Q_f = \rho_e L_{H,f} C_p (t_e - t_a) + \rho_e L_{s,f} C_p (t_e - t_o)$$

Where t_e and t_o are the temperature of exhaust air and supply air, C_p is the specific heat of air.

That is, the influence factors of Q_f is:

$$Q_f = f_3[v_z, L, \rho_o, \rho_e, d_o, x, C_p(t_e - t_a), C_p(t_e - t_o)] \tag{10}$$

summerizing the formulas (8) (9) (10), we'll know that the factors which affect the total heal gain Q_o in occupied zone, May be defined:

$$Q_o = f[\rho_o, d_o, v_o, x, v_1, L, \rho_e v_z^2, C_p(t_e - t_a), C_p(t_a - t_o), Q_C, F_{H,S}] \tag{10'}$$

To non–isothermal jet, air velocity $v_1 = \varphi(v_o, x / d_o, Ar_o)$, in which x is the distance of air outlet to the heat source, Ar_o is the function of buoyancy force $g\beta(t_n - t_o)$, v_o and T_a. Now we'll add the section area a = H. l_y^* of room in formula (10)', and adopt $L_o = L / H.\, l_y$ instead of L, adopt $4h_o d_e$ instead of $F_{H,S}$. Thus we obtain:

$$Q_o = f[\rho_o, d_o, v_o, x, g\beta(t_a - t_o), L_o, \rho v_z^2, C_p(t_e - t_a), C_p(t_e - t_o)\, H, d_e,$$
$$Q_C, 4h_o d_e] \tag{11}$$

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* (Note, H, l_y are the height and width of room respectively)

Applying the pi–theorem, we proceed dimensional analysis. The upper mentioned physical quantities refer to four fundamental dimensions [L] [T] [M] [Q].

For convenience in investigation, consider [Q] as a independent dimension (see refernce 8). therefore, we use ρ_o,d_o,v_o as fundamental quatities, By adopting suitable indexes, we obtain non dimensional expression as follows:

$$\frac{Q_o}{\rho_o\frac{\pi}{4}d_o^2 v_o C_p(t_e-t_o)}=\psi\left(1,1,1,\frac{x}{d_o},\frac{\beta g(t_a-t_o)}{v_o^2\cdot d_o^{-1}},\frac{L_o}{v_o},\frac{\rho_e V_x^2}{\rho_o v_o^2},\frac{H}{d_o},\frac{d_e}{d_o},\right.$$
$$\left.\frac{C_p(t_e-t_a)}{C_p(t_e-t_o)},\frac{Q_C}{\rho_o\frac{\pi}{4}d_o^2 v_o C_p(t_e-t_o)},\frac{h_o}{d_o}\cdot\frac{d_e}{d_o}\right) \tag{12}$$

When heat balance in air condifioned room is reached the total heat emission from heat source.

$$Q=\rho_o\frac{\pi}{4}d_o^2 V_o C_p(t_e-t_o)$$

Then $\dfrac{Q_o}{\rho_o\frac{\pi}{4}d_o^2 V_o C_p(t_e-t_a)}=\alpha$, Where α is the room heat distribution coefficient .

While $$\frac{C_p(t_e-t_a)}{C_p(t_e-t_o)}=1-\frac{t_a-t_o}{t_e-t_o}=1-\alpha$$

For heat source with low surface temperature, the radiation heat emission is approximately 10% (conclusion from author's investigations on floor air supply),Then the convection heat emission of heat source is $Q_C=90\%Q$.

We have $$\frac{Q_C}{\rho_o\frac{\pi}{4}d_o^2 V_o C_p(t_p-t_o)}=\frac{90\%Q_o}{Q_o}=90\%.$$

Then expression [12] changes to:

$$\alpha=\varphi\left(\frac{x}{d_o},Ax_o,\frac{L_o}{v_o},\frac{\rho_e V_x^2}{\rho_o v_o^2},\frac{H}{d_o},\frac{d_e}{d_o},\frac{h_o}{d_o},\frac{d_e}{d_o}\right) \tag{12'}$$

As x/d_o is relating to $v_o v_x$, So in horizontal air supply we may first assume v_o by using formla [14] in this article. As a result, we can cancel term x/d_o in [12'],We can also cancel "height of heat source" h_o and "height of room" H as they are the typical height . Then formula [12'] may be further simplified as follows:

$$\alpha=\psi'\left(Ar_o;\frac{L_o}{V_o},\frac{\rho_e V_x^2}{\rho_o V_o^2},\frac{d_e}{d_o}\right) \tag{13}$$

Conditions for the experiments and analysis of the experiment resulls:

Conditions for the experiments:

Model test of this article is carried out inside a model simulating a ventilated and air conditioned room of medium height without any moisture emission. We use swirling air outlet and perforated cabinet type air outlet (outside dimensions of later type air outlet are $0.6\times0.4\times1.8^m$ and $0.6\times0.4\times1.2^m$. The description of whole testing equipment and instruments see reference [4] in model test, the flow rates are in the range of 600–2800m^3/h. The heat emissions of heat sources are 120 W/m^2, 140W/m^2 and 160 W/m^2. Verified by calculatings, the Reynolds number Re of supply air flow in the model is Re> 2340, i.e. in the turbulent state. Near by heat source the heat current satisfied GrRr> 2×10^7, i.e. in the turbulent flow self–modeling zone, The flow may be considered as having analogg simulation.

Air distribution in the room with side air outlets at occupied zone:

From experiments associated with smoke display we discovered the following charecteristics ;

(1) Focus of supply jet curves to the ascending direction of the hot jet (i.e. to the direction of the exhaust inlets). The velocity distribution of supply jet in vertical direction is not symmetrical. See the result in Fig2.

Either the decreasing of air supply or the increasing of heat emission of heat source will increase the curvature of the main streamline of supply jet.

(2) In horizontal direction the supply jet of perforated cabinet type air Oullets flows With angle of diffusion $\alpha = 59°$. About 1 m apart from the outlet, jet will occupy in horizontal direction the whole width of the model space. The horizontal cross sectional velocity of the jet shows normal distribution. See Fig.2

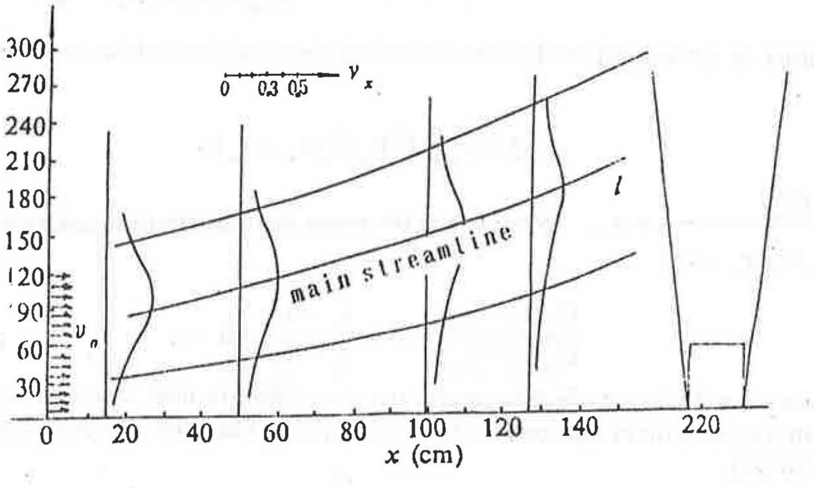

Fig.2 Velocitya distribution of horizontal Supply jet

By experiments in the model we find under certain conditions the relation of main streamline velocity of supply jet and mean velocity on the width centerline to the non-dimensional distance x/d_o and Archimedes number Ar_o of air outlet is as follows:

$$v_x / v_o = KAr_o^{m(x/d_o)^n} \qquad (14)$$

Values of coefficients k.m.n. see Table1. The result of remarkable examination shows that the regression equation we obtained is remarkable.

The mutual influence between suppply jet and hot jet.

The mutual influence between supply jet and hot jet is different when the heat emision of heat source and flow rate of supply air are different.

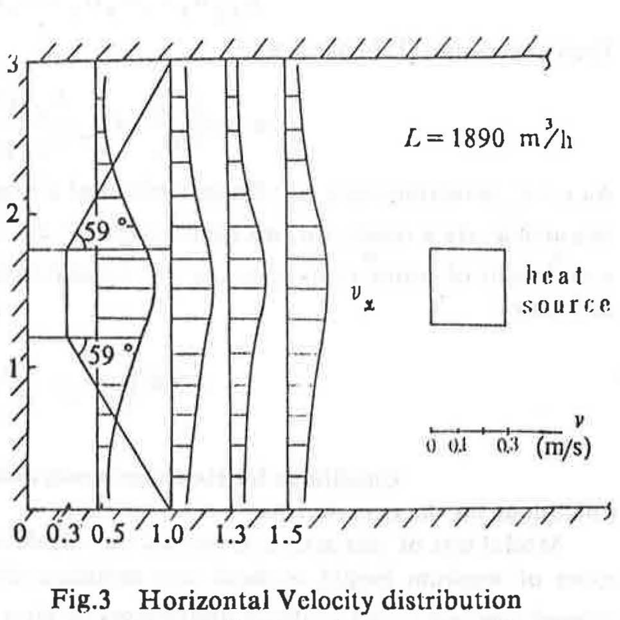

Fig.3 Horizontal Velocity distribution of supply jet

It's specific expressions:

(1) with both too much or too small supply air, axial temperatures of hot jet on the same level are all rether low. With the increasing of supply air, axial temperatures of hot jet increase gradually from low to high. It will reach it's maximum when a certain air supply is reached. And then it will again decrease gradually. Air supply $1553 m^3/h$ responds maximum temperature. See Table 2.

Table 1 Performance of perforated cabinet type air outlets with horizontal air supply

condition of application / regression coeffient	$H_0 = 1.2^m$		$H_0 = 1.8^m$
	$d_o / \sqrt{F_n} = 0.169$, $q = 160W/m^2$		$d_o / \sqrt{F_n} = 0.194$, $q = 160W/m^2$
	main streamline	mean in jet width	mean in jet width
K	0.113	0.166	0.373
m	−0.169	0.066	0.257
n	−0.495	−0.248	−0.629
correlation coefficient R	0.91	0.92	0.925
remarkable examination F	29.76	24.64	65.19
$F_{0.01}$	6.93	6.93	5.72

Table 2 change of axial temperatures of hot jet

axial temperature(℃) / air supply m^3/h / level(m)	$890m^3/h$	1181	1553	1780	2065	2206
1.2	30.6	34.1	36.3	35	33.2	33.9
1.6	30.4	30.7	31.8	30	28.9	30.4
2.0	26.3	28.6	30.1	27.5	28.2	24.8
2.4	25.5	26.1	27.7	26.5	26.7	24.7
heat distribution coefficient α	0.622	0.351	0.304	0.344	0.400	0.420
remark	level is based on the floor level$H_p = 1.2^m$, $q = 120W/m^2$					

(2) The change of intensity of heat source not only affects the flow state of hot jet, but also affects the velocity field of supply jet.

From expeniments in modle, the axial velocity of hot jet formed by low surface temperature heat source is:

$$v_z = 0.0293 Q_c^{1/3} Z^{1/2} + 0.1906 \quad \textit{(obtained frome xperiment)} \qquad (15)$$

So that when intensity of heat source Q_c is increased, the axial velocity of hot jet is also increased.

Fig 4, Fig 5, express respectively the changes of axial temperatures of hot jet and non−dimensional velocities of supply jet when intensities of heat source are changed.

When intensity of heat source is increased, the axial temperatures of hot jet are increased, while the decreasing of non−dimensional velocity v_x / v_o of supply jet slows down.

Because the hot jet will entrain the ambient air near heat source and force the ambient air move to heat source with the same direction of supply jet, so the entrainment of hot jet will help to slow down the decreasing of supply jet velocity. The increasing of heat source emission and the strengthening of entrainment of hot jet will surely slow down the decreasing of supply jet velocity.

Moreover, in order to avoid the feedback of high level space hot air to compensate the increasing demand of air entrained by hot jet, we must increase the air supply. From the point of view of room heat distribution, when heat emission of heat source is increased, the optimum air flow corresponding to minimum room heat distribution coefficient will also be increased.

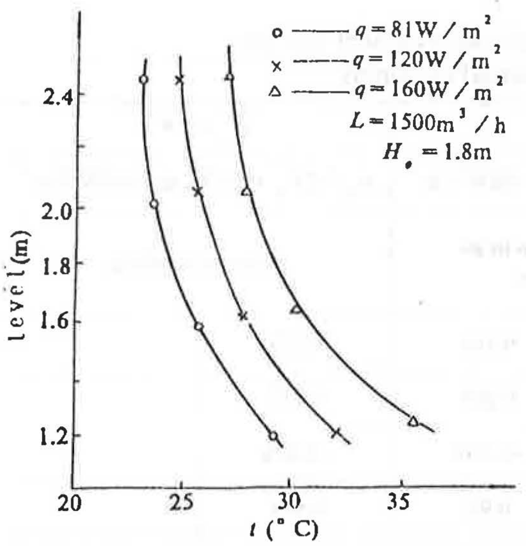

Fig 4 Axial temperature distribution of hot jet

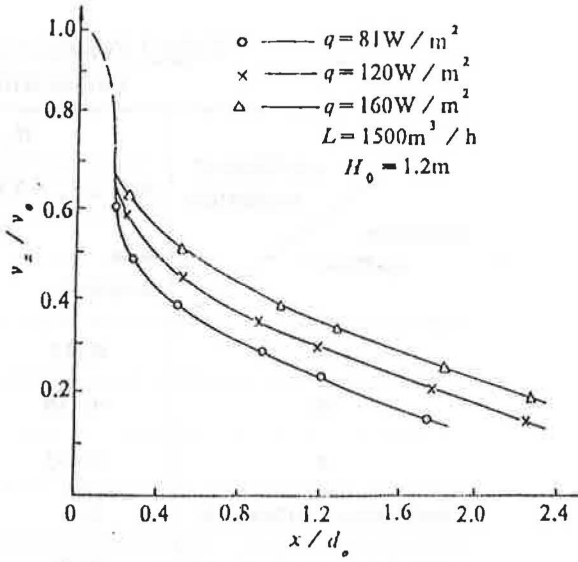

Fig 5 Main streamline velocity distribution of supply jet

The determination of optimum coordinative conditions

(1) Determination of optimum air flow rate:

By experiments in model, we obtained optimum air flow (i.e. the air flow corresponding to minimum heat distribution coefficient) under different heat emissions of heat source when the type of air outlet and the distance outlet to heat source are defined. Referring to reference [4], we can obtain the following empirical formla for calculating the optimum air flow L_j at occupied zone.

$$L_j = 0.97 k_1 k_2 (AB)^{0.833} (Q_o / T)^{0.33} \tag{16}$$

Where k_1 is the coefficient depending on types of air outlet: for perforated cabinet type air outlet with $H_o = 1.2^m$, $k_1 = 1.55$, for perforated cabinet type air outlet with $H_o = 1.8^m$, $k_1 = 1.64$, A, B are length of two sides of the rectongular heat source, T is ambient temperature, Q_o is the convection heat emission of heat source (Kw), k_2 is the function of heat source dimensions and calculating height above the heat source (can be found in reference 4).

Values of optimum air flow under different conditions calculated by using the preceding formula are listed in Table 3. From this table, we find that the calculating values and testing values of optimum air flow are very near.

Table 3 Values of optimum air flow

types of air outlet	heat source		cptimum air flow L_j (m³ / h)	
	q(W / m²)	Q_o (Kw)	testing	calculating
perforated cabinet type $H_o = 1.2^m$	120	1.62	1550	1561
	160	2.16	1740	1718
perforafed cabinet type $H_o = 1.8^m$	100	1.75	1500	1554
	140	1.89	1760	1738
	160	2.16	1850	1816

(2) The characteristics of supply jet velocity distribution under optimum coordinative conditions:

We define that supply jet has entered into it's terminal zone when the mean velocity v on the centerline of the width of supply jet is < 0.15, and also consider that the vertical section 30cm from the boundary of heat source meeting the flowing current of supply jet as interface.

Results of tests for supply jet velocity field under optimum condition show that for $H_o = 1.8^m$ perforated cabinet type air outlet supply system, supply jet forms the jet's end near the interface. Near interface under the influence of hot jet entrainment, the flow velocity increases a little. While for H_o = 1.2^m perforated cabinet type air outlet supply system under optimum condition, velocity near interface decreases to lower than 0.15m / s.

From upper mentioned, we understand that opposite horizontal displacement air supply under optimum condition has entered into the terminal zone at the interface. The supply air current may flow into near the heat source, but it will not impact the ascending hot air current formed by heat source.

Heat distribution in air conditioned room with horizontal displacement air supply at occupied zone:

From preceding analysis, we know that the room heat distribution coefficient for occupied zone with opposite air flow supply can be expressed by:

$$\alpha = \psi'\left(Ar_o, \frac{L_o}{v_o} \cdot \frac{\rho_e v_z^2}{\rho_o v_o^2}, \frac{d_e}{d_o} \right)$$

Referring to reference [4], we take following formula to regress curves of Fig.6 and 7.

$$\alpha\left(\frac{\rho_e v_v^2}{\rho_o v_o^2}\right)^n = 1 - e^{\{a_0 + a_1 l_n Ar_o (v_o / L'_o)^{0.5} + a_2 [l_n Ar_o (v_o / L'_o)^{0.5}]^2\}} \tag{17}$$

where: L'_o is the air flow from one side $m^3 / m^2.s$

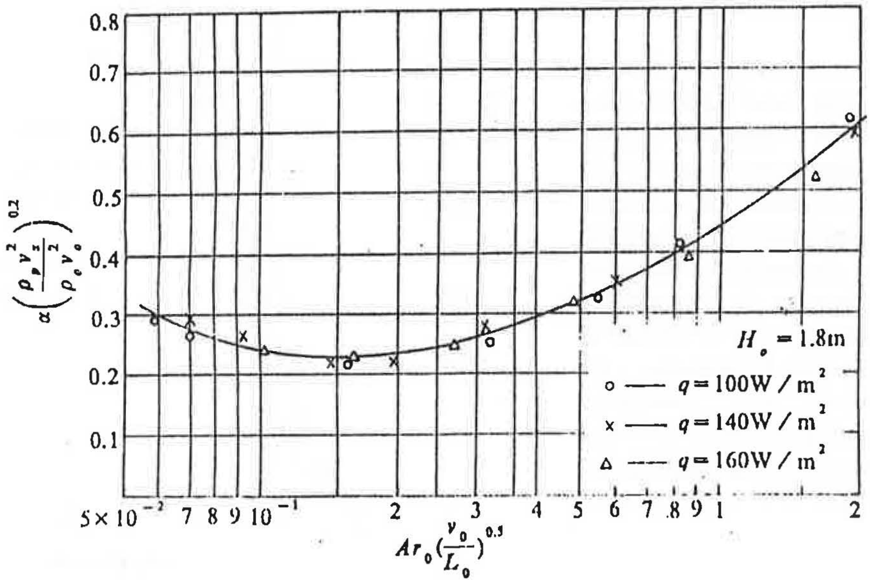

Fig.6 Heat distribution

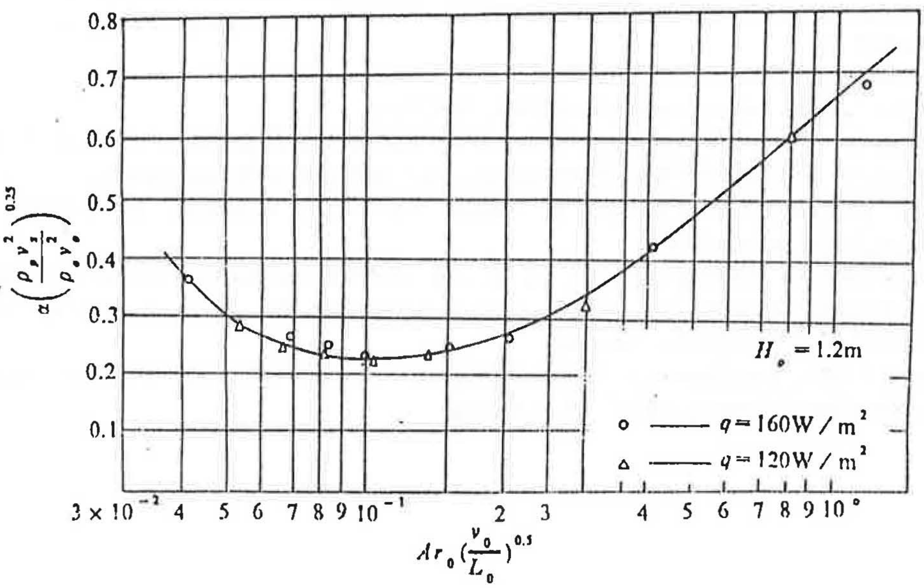

Fig 7 Heat distribution

By computing on computer, we obtain the values of coefficients n, a_0,a_1,a_2, see Table 4.

Table 4 regression coefficients

regression coefficient / condition	n	a_0	a_1	a_2
$H_o = 1.8$	0.2	−0.569	−0.353	−0.097
$H_o = 1.2$	0.25	−0.897	0.60	−0.142

And finally, let us look up the vertical temperature distribution in room with this air supply sytem in Fig 8, from which we can find out the vertical temperature gradient at occupied zone with any air flow is not greater than 2℃, Thus we can say that the horizontal displacement air supply system with perforeted cabinet outlet is able to ensure the requirement for comfort.

The result of tests shows that the temperature distributions in an horizontal directions are more uniform.

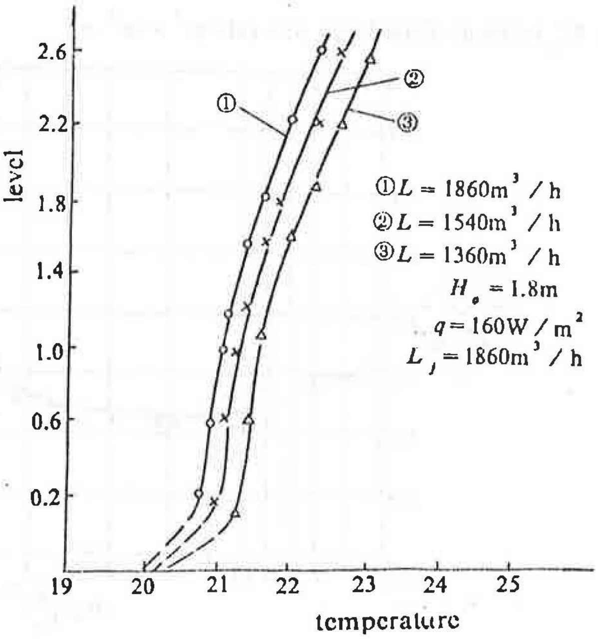

Fig 8 vertical temperature distribution in room

Conclusions:

This article investigates the air flow current and heat distribution characteristic in air conditioned room with horizontal displacement air supply at occupied zone. It analyzes the mutual influence between supply jet and hot jet, This article theoretically discusses factors which affect the air flow pattern and heat distibutions in air conditioned room with horizontal displacement air supply at occupied zone. Empieical formulas for calculating axial velocities of supply jet and room heat distribution coefficients are defined by experiments.

Reference

1. P.E. Andlefu, The Heat and Moisture Distribution of Industrial Work–shop, 1958, U.S.S.R.
2. Industrial Ventilation, 1980, in china.
3. Gas Combusion and its application 1988,. in China.
4. Ma Ren Min, The air flow distribution from the floor influencing the thermal distribution inside a room. 1987 RooM VENT–87 papers session 2a, Stockholm, Sweden.
5. A.F.C. Sherrah, Air conditioning system desigh for buildings, MCGRAN–HILL Book compang(uk) Limited 1982.
6. Ma Ren Min, A primary research on the heat distribution pattern inside room causing by Air flow through annular slot around a heat source on the floor, papers session 4b, Room VENT–87 1987 Stolkholm, sweden.
7. Eimund Skoret, Displacement Ventilation, papers session 5, Room VENT–87. Stockhlom.
8. Yang Shiming, "Heat transfer". 1980, China.