Air Distribution in Rooms, (ROOMVENT 2000)
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A STRATEGIC APPROACH FOR THE ROOM AIR CONDITIONING DESIGN

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ABSTRACT

This paper discusses the application of a new strategy approach for the room air conditioning. The basis of the classification is different aims or ideas of the temperature, gas, particle, humidity distributions and room air flow patterns that can be created within a room. A certain strategy can be applied by using different system combinations of room air distribution, exhaust, heating and cooling methods and their control. The realization of an ideal strategy is also dependent on the operating parameters and internal sources. Separating the ideal strategies from the practical room air conditioning solutions will help the evaluation of the present room air distribution methods in different operating conditions. The differences of the strategies are demonstrated in the examples.

KEYWORDS

Ventilation strategies, Room air conditioning, Air distribution, Ventilation efficiency

INTRODUCTION

At present there is no unambiguous classification for the room air conditioning strategies or terminology. Traditionally the room air conditioning classification has been based on the room air distribution methods. The most used division has been the division into mixing and displacement, while the other methods have been varied.(ASHRAE 1997, Tapola 1987) In German VDI (1994) guidelines the division has been made based on the resulting air flow pattern within the room rather than distribution methods. Etheridge (1996) and Sandberg suggested the air distribution methods to be classified as jet controlled or thermally controlled, which raises the important question how well the room air flow patterns are controlled by the air distribution method.

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AN EXPERIMENTAL AND NUMERICAL STUDY OF THE RELATIONSHIP BETWEEN VENTILATION EFFICIENCY AND AIR SUPPLY/EXHAUST SYSTEM OF AN OFFICE ROOM

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ABSTRACT

Experiments with full size model and CFD simulations of the relationship between ventilation efficiency and air supply/exhaust system of an office room are carried out. The age of air and air exchange efficiency of the occupied zone (εm) have been measured by tracer gas ethod and calculated by CFD analysis. The relationship between vertical temperature difference and ventilation efficiency and the layout of the supply and exhaust are analyzed.The main results are as follows.

(1) When the warm air is supplied from the floor, the value of εm increases regardless of the exhaust position, and this arrangement is suitable for heating.
(2) In the case of cooling with low airflow rate supplied from the floor, it is better to install the exhaust inlet on the ceiling.
(3) The most effective arrangement for both the heating and the cooling is the vertical air supply system; i.e. the air is supplied from the ceiling and exhausted to the floor or supplied from the floor and exhausted to the ceiling.

KEYWORDS

Ventilation efficiency, Air conditioning system, Model experiment, Numerical simulation, Computational fluid dynamics(CFD)

INTRODUCTION

To evaluate the indoor air quality of air-conditioned office building, carbon dioxide, temperature, humidity, air velocity, carbon monoxide and dust are usually used as indices. Specially, CO2 has been used as a typical index of indoor air quality. However, when the concentration of CO2 does not reach the limiting value, volatile organic compounds (VOC's) and other pollutants such as HCHO may have a significant influence on the health

of room occupants.

Usually, perfect mixing is assumed in the design of air-conditioning system, but in reality contaminants are unevenly distributed in the room, and the required ventilation rate is not achieved in the occupied zone. The study of air movement in a room requires the evaluation of the indoor air parameters (e.g. temperature, velocity, and concentration) throughout the space and not just at a reference point in the space. In this paper, the results from a mock-up experiment and CFD simulations of an office room are presented. The age of air and air exchange efficiency have been measured and calculated.

MODEL EXPERIMENT FOR THE VENTILATION EFFICIENCY

Experomental Room

The full size experimental room (5,000mm Dx3,600mm Wx2,450mm H) that was used in this study is shown in Figure 1. The model room is made from thermally insulated panels and was installed in the laboratory at Sanken Setsubi Kogyo Co.,Ltd. The layout and the air temperature of the supply outlet and exhaust inlet can be controlled. Heating panels are arranged on 4 places of the wall, and the heat generated from occupants bodies (100W) was simulated by heated cylinders.

Experimental Method

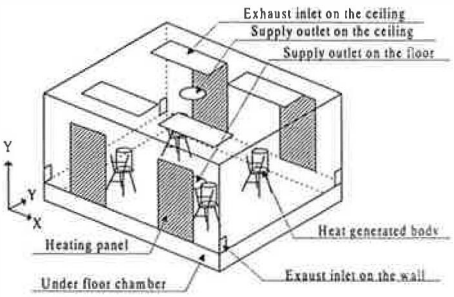
Tracer gas (SF6) is injected in the supply duct at a fixed rate, and the change in concentration at each measurement point is recorded until the concentration at each point almost becomes constant (step up method). After that, the injection of tracer gas is stopped and the concentration decay is measured (step down method).

To measure the concentration of the tracer gas, 5 sets of multi-gas analyzers (BK1303) were used. The number of measuring points of the tracer gas was 27 points inside of room and in the supply outlet and exhaust inlet.

Experimental Conditions

Table 3 shows the experimental conditions. Three kinds of supply outlets (horizontal supply from the ceiling, vertical supply from the ceiling and vertical supply from the floor), three values of ventilation rate (1.5, 3.0 and 6.0m³/min), three values of temperature difference between supply and exhaust air (ΔT=10.0, 5.0, 0.0 K) have been used in the experiments. Experimental Case A is isothermal, Case B is heating and Case C is cooling condition. The numbers after A, B, C (1 - 6) represent different air supply and exhaust positions.

Although the air supply temperature in the heating mode was maintained constant, changes in the labo-



Figur 1 Full size experimental chamber

Table 1 Diffinitions of ventilation efficiency

1. Local age of air : τp

[StepUp method] [Step Down method]

$$\tau_p = \int_0^\infty \left\{1 - \frac{Cp(t)}{Cp(tc)}\right\} dt \quad \cdots(1) \qquad \tau_p = \int_0^\infty \left\{\frac{Cp(t)}{Cp(tc)}\right\} dt \quad \cdots(2)$$

Cp(t):gas concentrarion DT the time t(min).
After the injection is start or stop
Cp(tc):Standard gas concentration
=Gas injection rate(cc/min)/ventiration air flow(m³/min)

2.Local mean air change efficiency : εp

$$\varepsilon_p = \frac{\tau_n}{\tau_p} \cdots\cdots(3) \qquad \tau_n = \frac{V}{Q} \cdots\cdots(4)$$

τn:Fnimnal time consatant (min), V:Air volume of the room(m³)
Q:ventilation air flow rate (m³/min)

3. Local mean air exchange efficiency of the occupied zone : εm
Local mean air exchange efficiency of the brezthing height :εbh

$$\varepsilon_m = \sum_{n=1}^{n} \varepsilon p(n) \times V(n) / V_m \qquad \varepsilon_{bh} = \sum_{k=1}^{k} \varepsilon p(k) \times A(k) / A_{bh}$$

V(n) : Air volume of measuring points under a height of 1800mm (m³)
Vm: Air volume of measuring points under a height of 1800mm (m³)
A(k):Area of height of 1200mm (m²)
Abh:Area of breathing zone (m²)
n,k: number of the measuring point

ratory conditions affected the heat loss from the chamber. As a result, the temperature difference between the supply air and exhaust air was not strictly constant in the case of tests B. During measurement, the difference in temperature between the supply and exhaust air is maintained at a constant value but this was only achieved after running the test for up to 8 hours. For the cooling tests, panel heaters were installed on the wall and the temperature difference between the supply and exhaust air was kept constant to give a constant cooling load. The actual ventilation rate for the chamber has been calculated from tracer gas concentration measurement by constant gas injection in the supply outlet using the step up method. This provides an accurate measurement to the airflow rate.

Evaluation Method

The distribution of the ventilation efficiency is evaluated from the local air exchange efficiency (εp) at each point that was calculated from the tracer gas concentration history. In this paper, the occupied zone is defined to be the region from the floor to a height of 1800mm in the room. The mean local air exchange efficiency of the occupied zone (εm) is used to assess the air distribution system. The breathing zone is assumed to be at a height of 1200mm from the floor. The mean local air exchange efficiency at the breathing zone is defined as εbh. The definitions of these terms are given in Table 2.

CALUCILATION OF THE VENTILATION EFFICIENCY BY CFD

Each experimental condition was simulated using the CFD code STREAM, Software Cradle co.ltd.(1990), with a standard k-ε model. The airflow distribution was analyzed, and the local air ventilation efficiency was cal-

Table 3 Calculation method of the ventilation efficiency by CFD

Turbulence model	:Standards k-εmodel
Spatial derivative	:Quick scheme for convection term, first-order upwind scheme for others
Supply outlet	:Supply outlet on the ceiling,supply outlet on the floor
Exaust inlet	:Exaust inlet on the ceiling,exaust inlet on the wall
Heating loads	:Heating panel
Calucilatin of εp	:SVE3
The number of computational cells	:37(x)x33(y)x35(z) supply from the ceiling or the wall, 50(x)x45(y)x35(z) supply from the floor

Table 2 Experimenral and CFD conditions and the Resurts of εm

		Experimental condition											CFD condition							
	Case	Supply/Exhaust			air volume	Co	Ts	Ti	ΔT	ε_m		ε_{bh}		air volume	Vs[m/s]		Vr	Ts	Heater	Results	
		Supply		Exhaust	[m³/min]	[ppm]	[℃]	[℃]	[℃]	UP	DN	UP	DN	[m3/min]	Vz	V_H	[m/s]	[℃]	[W]	ε_m	ε_{bh}
Ventilation	A1	ceiling	Horizontal	ceiling	1.49	20.2	11.8	12.0	0.2	1.2	1.1	1.3	1.1	1.5	0.44	1.111	0.02	22.0	-	0.8	0.8
	A2	ceiling	Horizontal	wall	1.44	20.8	8.7	9.1	0.4	1.2	1.1	1.1	1.1	1.5	0.44	1.111	0.05	22.0	-	1.0	1.0
	A3	ceiling	Vertical	ceiling	1.59	28.9	9.1	10.5	1.4	1.0	1.2	1.0	1.3	1.5	0.88	0.28	0.02	22.0	-	1.1	1.0
	A4	ceiling	Vertical	wall	1.37	22.0	9.4	10.2	0.8	1.0	1.3	1.0	1.3	1.5	0.88	0.28	0.05	22.0	-	0.9	0.8
	A5	floor	-	ceiling	1.38	21.7	8.2	9.4	1.2	0.9	1.1	0.9	1.1	1.5	0.1~2.0	-	0.02	22.0	-	0.9	0.8
	A6	floor	-	wall	1.35	22.2	8.7	9.9	1.2	0.9	0.9	0.9	0.9	1.5	0.1~2.0	-	0.05	22.0	-	1.1	0.9
Heating	B1	ceiling	Horizontal	ceiling	1.37	21.8	31.1	18.0	13.1	0.4	0.3	0.3	0.3	1.5	0.44	1.11	0.02	32.0	-	0.3	0.3
	B2	ceiling	Horizontal	wall	1.59	28.9	32.4	18.9	13.5	0.8	0.8	0.7	0.7	1.5	0.44	1.11	0.05	32.0	-	0.8	0.7
	B3	ceiling	Vertical	ceiling	1.40	21.4	32.9	18.6	14.3	0.4	0.3	0.4	0.3	1.5	0.88	0.28	0.02	32.0	-	0.3	0.3
	B4	ceiling	Vertical	wall	1.60	18.8	32.9	19.9	13.0	1.0	0.9	1.0	0.9	1.5	0.88	0.28	0.05	32.0	-	0.7	0.7
	B5	floor	-	ceiling	1.54	19.5	32.0	20.1	11.9	0.9	0.9	0.9	0.9	1.5	0.1~2.0	-	0.02	32.0	-	0.9	0.8
	B6	floor	-	wall	1.55	19.4	33.0	22.9	10.1	0.8	0.9	0.8	0.9	1.5	0.1~2.0	-	0.05	32.0	-	1.1	0.9
cooling	C1	ceiling	Horizontal	ceiling	1.55	24.3	20.5	27.8	-7.3	1.2	0.9	1.2	1.0	1.5	0.44	1.11	0.02	16.0	239.0	1.0	1.0
	C2	ceiling	Horizontal	wall	1.53	22.1	16.0	26.1	-10.1	1.1	1.1	1.2	1.1	1.5	0.44	1.11	0.05	16.0	239.0	1.0	1.0
	C3	ceiling	Vertical	ceiling	1.54	21.8	16.0	27.6	-11.6	1.1	1.0	1.2	1.1	1.5	0.88	0.28	0.02	16.0	239.0	1.0	1.0
	C4	ceiling	Vertical	wall	1.60	20.7	16.1	26.4	-10.3	1.2	1.1	1.2	1.1	1.5	0.88	0.28	0.05	16.0	239.0	0.9	0.9
	C5	floor	-	ceiling	1.50	24.2	23.0	33.8	-10.8	1.7	1.5	1.6	1.5	1.5	0.1~2.0	-	0.02	19.0	239.0	1.4	1.3
	C6	floor	-	wall	1.49	24.3	22.1	27.4	-5.3	1.5	1.2	1.5	1.3	1.5	0.1~2.0	-	0.05	19.0	239.0	0.9	1.0
	C5-DU	floor	-	ceiling	2.99	21.7	16.8	31.3	-14.5	1.2	1.1	1.2	1.2	3.0	0.20~4.0	-	0.04	17.7	540.4	1.1	1.1
	C5-DU	floor	-	wall	3.04	21.6	19.4	28.2	-8.8	1.2	1.1	1.2	1.0	3.0	0.20~4.0	-	0.10	17.7	540.4	1.1	1.1
	C6-HU	floor 4	-	ceiling	3.03	21.9	17.5	30.8	-13.3	1.7	1.6	0.9	1.1	3.0	0.05~1.0	-	0.04	17.7	540.4	1.5	1.3
	C6-HU	floor 4	-	wall	3.00	22.0	17.4	23.9	-6.5	0.9	0.7	0.4	0.4	3.0	0.05~1.0	-	0.04	17.7	540.4	0.5	0.3

Co:standard gas comcentration,Ts:supply air temperature,Ti:room air temperature, DT:Ts-Ti, UP:Step up method, DN: step doun method, Vs: supply air belocity, Vz: vertical supply air velocity,VH:horizontal supply air velocity,Vr:exaust air velocity,

culated from the value of SVE3, S.Kato,S.Murakami (1986). The conditions used in the calculations are given in Table 3. The room symmetry has been taken into consideration in the CFD simulation by solving the flow for 1/4 of the room. The boundary conditions used in the CFD simulation and some results are given in Table 2. The local mean air exchange efficiency of the occupied zone (εp) represent the weighted averages of all the computational cells from the floor to a height of 1800mm. The local mean air exchange efficiency at the breathing zone (εbh) is the weighted average of of all the computational cells at a height of 1200mm.

EXPERIMENTAL AND CFD RESULTS

Ventilation Efficiency for The Heating Conditions (CASE B)

Figure 2 shows the experimental(step down method) and CFD values of εp and εm at X-Z central section of the room. Figure 2 also shows the velocity contour lines from the CFD calculation.

(1) Horizontal supply from the ceiling

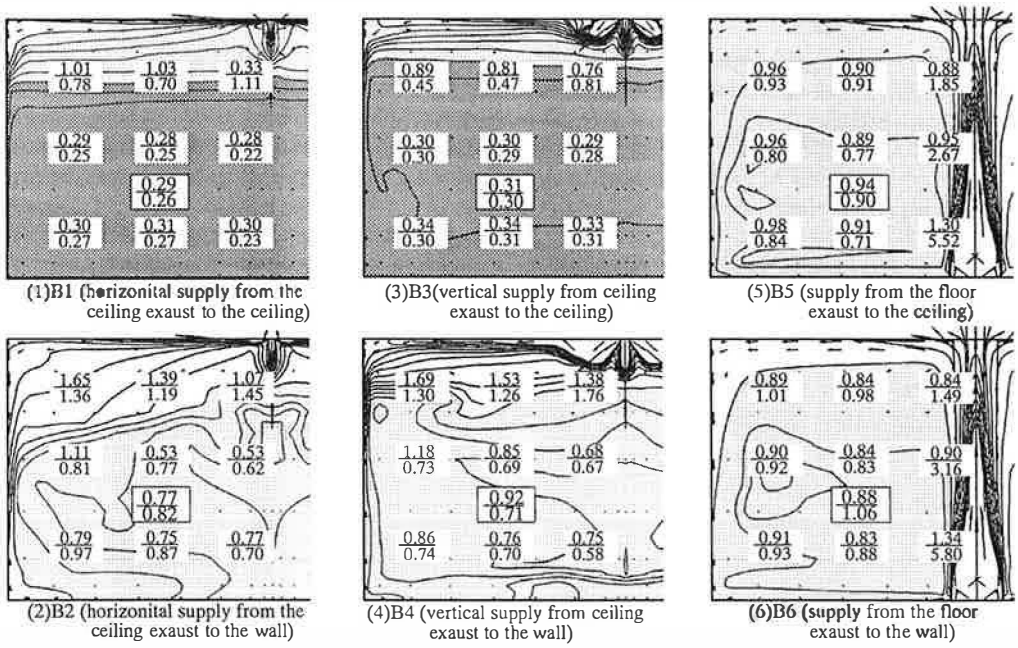
In the case of B1 (supply from the ceiling and exhaust to the ceiling), warm supply jet does not descent to the occupied zone, and exhausts back to the exhaust inlet on the ceiling as shown in Figures 3 (1), (2). Therefore, the value of εp at the occupied zone becomes very low, and εm becomes about 0.3 to 0.4 for both experiment and CFD. In the case of B2 (horizontal supply from the ceiling and exhaust to the lower wall), supply air does not descent to the occupied zone, but the exhaust outlet is arranged on the lower part of the wall, the value of εm is 0.8 which is greater than in the case of B1.

(2) Vertical supply from the ceiling

The value of εm is about 0.3 to 0.4 which is the same value as that for a horizontal supply from the ceiling, as shown in Figures 3 (3), (4). In this case, the air velocity at the supply outlet is small compared to the recommended velocity used for heating.

(3) Vertical supply from the floor (Figure 3 (5), (6))

The experimental value of εp in the case of floor supply has a uniform distribution with a value of 0.8 to 1.0,



(1)B1 (horizonital supply from the ceiling exaust to the ceiling)

(3)B3(vertical supply from ceiling exaust to the ceiling)

(5)B5 (supply from the floor exaust to the ceiling)

(2)B2 (horizontal supply from the ceiling exaust to the wall)

(4)B4 (vertical supply from ceiling exaust to the wall)

(6)B6 (supply from the floor exaust to the wall)

Figure 2 Local mean air exchange efficiency for the heating conditions

except in the neighborhood of the supply outlet, and the value of εm becomes 0.8 to 0.9. The CFD value of εp is different from the experimental value in the area of the jet, but the value of εm becomes around 1.0 and that is almost equal to the experimental value.

Ventilation Efficiency for The Cooling Conditions (CASE C)

Figure 3 shows the experimental(step down method) and CFD values of εp and εm at X-Z central section of the room. Figure 3 also shows the velocity contour lines from the CFD calculation.

(1) Horizontal supply from the ceiling

In the case of C1 and C2, cool air from the supply outlet flows along the ceiling and then spreads over the whole room as shown in Figure 4 (1), (2). Experimental values of εp show a uniform distribution for any arrangement of exhaust inlet.

(2) Vertical supply from the ceiling (Figure 4 (4), (5))

In the case of C3 and C4 the jet from the supply outlet descends to the floor and spread over the whole room. But such arrangement of the supply outlet is not common, because a cool supply air descends directory to the occupants which causes discomfort due to draft. There are no differences in the values of εm between the

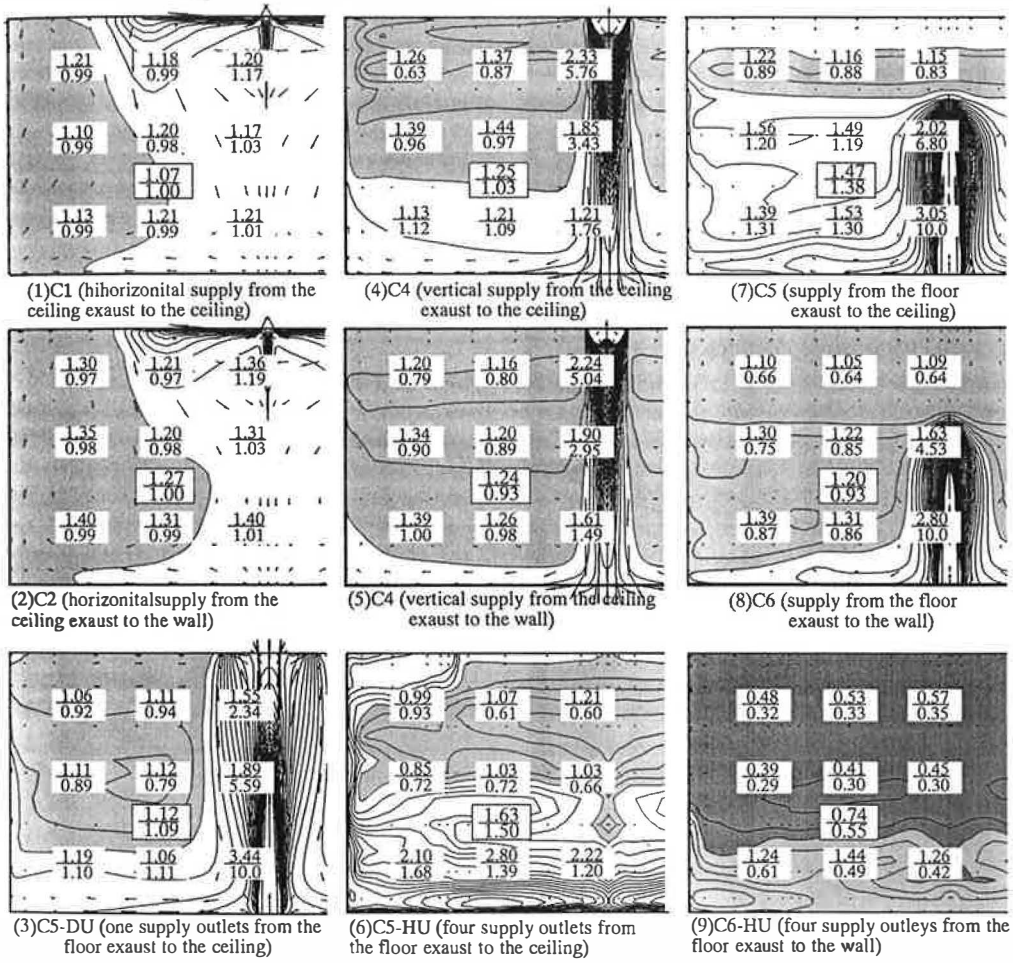


Figure 3 Local mean air exchange efficiency for the cooling conditions

horizontal supply and the vertical supply.

(3) Vertical supply from the floor (Figure 4(3) (6) (7) TO (9))

In the case C5 (exhaust to the ceiling), the supply air spreads to the upper area of the room, and εm has a high value in the range of 1.4 to 1.6 in both the experimental and CFD cases. But, in the case of C6 (exhaust to the lower wall inlet) the value of εm becomes lower than that for case C5, because the airflow from the supply outlet is discharged by the exhaust inlet before it spreads to the whole room i.e. short-circuiting. In the case of C5-DU (supply air velocity is double), the jet from the supply outlet ascends to the ceiling and the value of εm becomes in the range of 1.1 to 1.2. In the case of C5-HU(4 supply outlets, airflow velocity is half of C5, exhaust to ceiling), the air velocity from the supply outlet decreases, so that the projected height of the supply jet becomes lower and the air diffuses horizontally in an area close to the floor. In the case of C6-HU (exhaust to the lower part of wall), the air from supply outlet is exhausted to the inlet immediately , so that the value of εm becomes much lower than that for the case of the exhaust on the ceiling. In the case of the floor supply outlet, εp from the CFD is higher than the experimental value in the area of the jet, but it is lower in other regions.

Comparison Between Experiment and CFD Analysis

The airflow from the CFD calculation can be used to compare the differences between various arrangements of inlet-outlet under the same conditions independent of the outside conditions, but the experiments are influenced by the outdoor environment. The results of εp from the CFD calculation and the experiments do not agree well in the neighborhood of the supply outlet and the wall surface. The boundary conditions of the supply outlet cannot be reproduced fully in the CFD calculations. The down flow generated by heat transfer near the wall in the experiments causes the difference between the CFD and the experimental results. But, the values of εm from the CFD and the experiments are in agreement. Hence the ventilation efficiency of the occupied zone can be evaluated by εm obtained from the CFD calculation. The boundary condition of the supply outlet is considered in a future publication.

RELATIONSHIP BETWEEN AIRFLOW RATE AND VENTILATION EFFICIENCY FOR COOLING CONDITIONS

Air Supply from The Ceiling

The supply airflow rate from the ceiling was 1.5m³/min (C1 to C4), 3.0m³/min (C1-D to C4-D) and 6.0m³/min (C1-F to C4-F). The results from the CFD and the experiments are shown in Figure 4. It is shown that the airflow rate increases but the air exchange efficiency in the occupied zone (ε m) shows a tendency to decrease. The value of εbh and εm are about the same. When the supply airflow rate is 1.5m³/min, the value of εm becomes

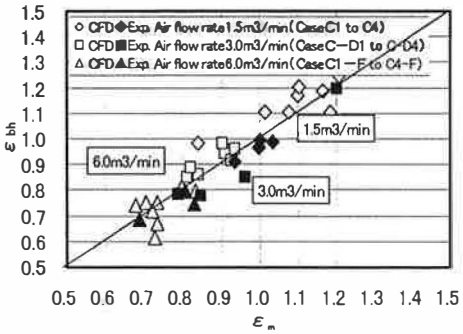


Figure 4 Relationship between the air flow rate and the ventilation efficiency of the occupied zone

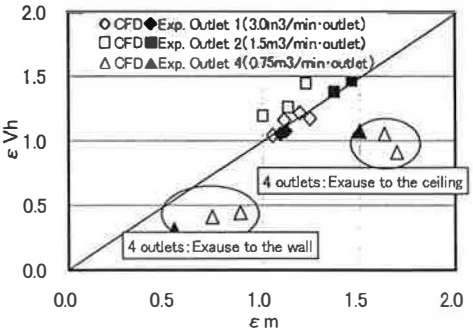


Figure 5 Relationship between the number of floor outlets and the ventilation efficiency of the occupied zone

1.0 to 1.2 in both the experiment and CFD, and becomes close to the perfect mixing value for all cases of supply outlet and exhaust inlet arrangement. In the case of 3.0m³/min, the value of each εm, except for the case of C2-D, decreases to 0.8 to 1.0. As for the value of εm for 6.0m³/min, it is less than 0.8 for most cases. εm has been normalized by the nominal time constant (reciprocal of the air change rate), so that the increase in ventilation rate does not increase the ventilation efficiency. For example, in the case of a horizontal supply outlet in the ceiling, as the ventilation rate increases, the air jet remains attached to the ceiling and walls. Which results into short-circuiting of the supply air to the exhaust, which is installed in the ceiling or the wall.

Difference in The Airflow Rate for Floor Supply Outlets

Here, the total ventilation rate was set at 3.0m³/min regardless of the number of air supply outlets, i.e. for one supply outlet (3.0m³/min × 1, C5-DU, C6-DU), for 2 supply outlets (1.5m³/min × 2, C5-D, C6-D) and for 4 supply outlets (0.75m³/min × 4, C5-HU, C6-HU), as given in Table 2. In the case of air supply from one outlet, both εm and εbh have high values in the range of 1.2 to 1.4. In the case of air supply from two supply outlets, both εm and εbh have values of about 1.0. When the number of supply outlets is increased and arranged uniformly in the room, the velocity at the supply outlet decreases, so that the projected height of the supply jet becomes lower. Therefore, the room flow in case of C5-HU (supply from the floor and exhaust to the ceiling), becomes similar to the piston flow, and εm becomes high. But in the case of C6-HU (supply from the floor and exhaust to the wall), εm decreases because the airflow from the floor is short circuited to the exhausted in the wall before the supply air reaches the occupied zone. The value of εbh is about 0.2 to 0.5 lower than εm. In this case, εm includes the high value of εp near the floor level, and the value of εm is overestimated. Therefore, in this case, the supply air can not reach the breathing zone, and εbh is better than εm for the evaluation of the floor supply system.

RELATIONSHIO BETWEEN TEMPERATURE DIFFERENCE AND THE VENTILATION EFFICIENCY

Figure 6 shows the relationship between εm and ΔT for cases A to C i.e. a constant air flow rate of 1.5m³/min.

In the case of a horizontal supply from the ceiling and exhaust to the ceiling (Figure 7 (1)), the value of εm becomes around 1.0 when ΔT is less than 0 K (isothermal and cooling conditions), but when ΔT is more than 5 K, the supply air remains close to the ceiling and therefore εm has a very low value, down to 0.3.

In the case of a horizontal supply from the ceiling and exhaust to the wall (Figure 7 (2)), the influence of ΔT is small in comparison with the other cases, and the value of εm is within the range of 0.8 to 1.2 for both cooling and heating.

In the case of a vertical supply from the ceiling and exhaust to the ceiling (Figure 7 (3)), the value of εm is greater than 1.0 when ΔT is less than 0 K (in the case of isothermal and cooling), but it becomes less than 0.5 when ΔT is more than 10 K because the warm air supplied from the ceiling remains at high level and does not reach the occupied zone

In the case of a vertical supply from the ceiling and exhaust to the wall (Figure 7 (4)), the value of εm from the CFD calculation is about 1.0 for the heating and cooling cases, but the value of εm from the experiments is higher in the heating case (ΔT=5K) and isothermal conditions. This is because in the experiments, there was heat losses from the chamber to the laboratory, which was not considered in the CFD simulation, and there was a thermal boundary layer in the room that causes the airflow to resemble a piston flow.

In the case of a vertical supply from the floor and exhaust to the ceiling (Figure 7 (5)), the value of εm becomes about 1.0 in the heating and isothermal cases, and in the cooling case it increases when ΔT increases. εm exceeds 1.5 when ΔT=-10K.

In the case of a vertical supply from the floor and exhaust to the wall (Figure 7 (6)), the value of εm becomes about 1.0 in the heating and isothermal cases. In the cooling case εm is greater than 1.0 whenΔT is between -5K to 0 K for both CFD and experimental results, but it is about 1.0 when ΔT is lower than -7.5K. WhenΔT is within the range of 0 to -5K, the buoyancy force acting on the supply jet causes the jet to return back to the occupied zone before it reaches the ceiling, hence εm is large.

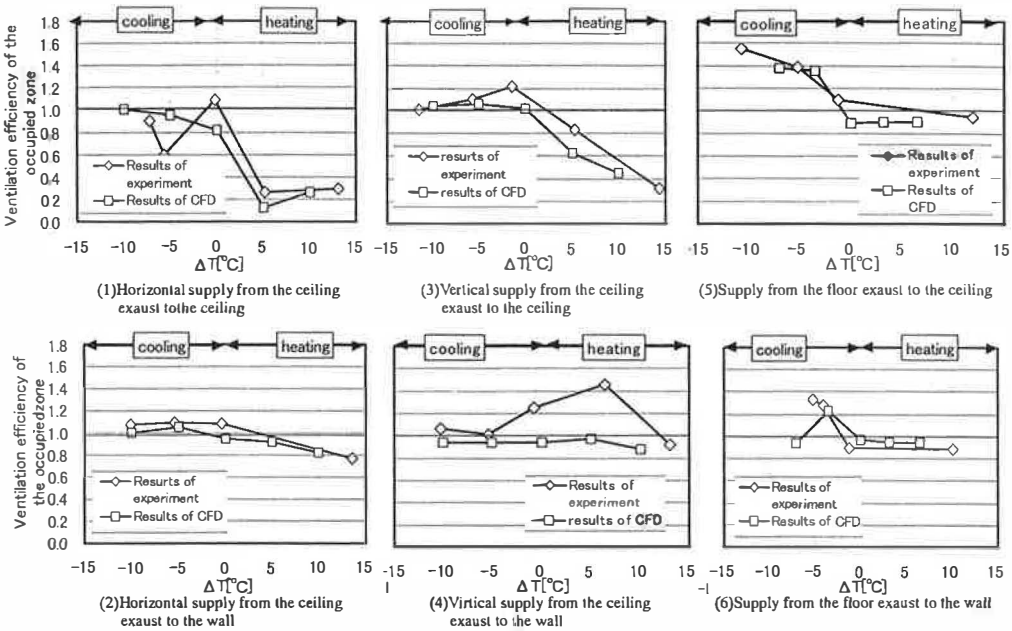


Figure6 Relationship between temperture difference and ventilation efficiency

CONCLUSION

(1) In the case of the isothermal conditions, the values of εm is within the range of 0.9 to 1.2 and there are only a small influence of the inlet-outlet arrangement.
(2) When the warm air is supplied from the ceiling and exhausted back to the ceiling, the value of εm becomes very low due to short circuiting. In this case, it is more efficient to install the exhaust inlet on the lower part of the wall. When the warm air is supplied from the floor, the value of εm increases regardless of the exhaust position, and this arrangement is suitable for heating.
(3) As the ceiling supply flow rate increases εm decreases because the high velocity jet adheres to the room surfaces and does not provide good mixing with room air.
(4) In the case of cooling with low airflow rate supplied from the floor, it is better to install the exhaust inlet on the ceiling.
(5) The most effective arrangement for both the heating and the cooling is the vertical air supply system; i.e. the air is supplied from the ceiling and exhausted to the floor or supplied from the floor and exhausted to the ceiling.
(6) There are some cases when the distribution of the value of εp was different between the experimental and CFD values, but there are not so great differences in the values of εm. Therefor, it is appropriate to evaluate the ventilation efficiency of the room by the value of εm obtained from the CFD method.

References

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