Cooled Ceilings/ Displacement Ventilation Hybrid Air conditioning System ------ Design Criteria

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1 ABSTRACT

For realizing comfort and ventilation efficiency simultaneously using a cooled ceiling (CC)/ displacement ventilation (DV) hybrid air conditioning system (CC/DV), Vertical distribution of temperature, local ventilation efficiency, and thermal comfort were studied by conducting a series of experiments to determine the design conditions. The characteristics of the thermal environment were measured in a laboratory, which was set up to simulate an office room in summer. The local air change efficiency was measured by tracer gases method and a subjective experiment was also carried out. The interaction between the major system parameters of room cooling load (P), CC cooling output ratio (η), ventilation airflow rate (Q), temperature gradient (dt) and the relevant constraints were studied.

The results of these experiments showed the temperature gradient in the occupied zone, an important factor of DV, adversely affects thermal comfort. As a result, the temperature gradient in the occupied zone should be in a range of 2-2.5 deg. C to realize comfort and ventilation efficiency simultaneously. A diagram for the system design was developed using the relationship among the above factors, which provides a useful and practical design tool.

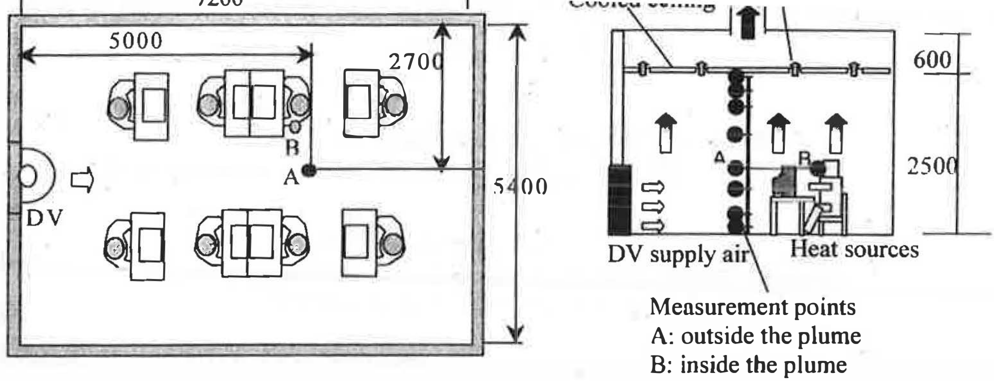
2 KEYWORDS

Displacement Ventilation, Cooled Ceiling, Thermal Comfort, Ventilation Efficiency, Temperature Gradient

3 INTRODUCTION

Recently, the displacement ventilation system (DV) which is popular in Northern Europe has been watched by air distribution procedures in Japan due to its advantage of ventilation efficiency. However, the cooling capacity of DV system is limited because it causes increase in the temperature gradient in the occupied zone which can reduces the thermal comfort. For this reason, it is becoming difficult to apply this system to higher thermal load space especially office rooms, where the inner thermal load is increasing. The combination of DV and CC seems to be representing an ideal solution because it eliminates discomfort resulting from excessive temperature gradients without undermining ventilation efficiency. Yet concerns remain that the downward airflow from the cooled ceiling could destroy the temperature stratification layer, and the superiority of ventilation efficiency be lost.

Although some experimental and theoretical studies regarding the DV system have been carried out by researchers in Europe and the United States, the relation or relevant constraints between the system parameters such as temperature gradient, cooling load and ventilating flow rate have not been clear yet. And there is a noticeable lack of reliable strength data for the system design. This paper elucidates the critical elements and their mutual relationship in the design of a cooled ceiling & DV hybrid air conditioning



(Plan) (Elevation)

Fig. 1 Arrangement of the test room

Tab.1 Measurement Method

Air temp. distribution	Thermocouple(E type) ,PC	0.1,0.3,0.6,1.1,1.6,2.1,2.3m above floor level
Temp. of wall, floor Surface	Thermocouple(E type) ,PC	1 points/each side
Temp. of ceiling surface	Thermocouple(E type) ,PC Infrared camera	6 points over all ceiling surface
Air velocity	Thermal probe	
Humidity Globe temp. PMV	Amenity meter	1.1m above floor level (center of room)
Tracer gas	Gas monitor	0.1,0.6,1.0(A,B),1.1,1.6m above floor level, air inlet/ outlet

Tab. 2 Experimental conditions (Exp.1)

DV supply air temp. [deg. C]	DV air flow rate [CMH]	Air exchange rate [1/h]	Cooled ceilings surface temp. [deg. C]	Thermal load density [w/m^2]
22	off, 360～1000	3.8～10.5	off, 16～24	15～95

system. In addition, it verifies ventilation efficiency, evaluates thermal comfort by human occupants, and proposes practical design criteria and conditions.

4. EXPERIMENT SETTING

The experiments were performed in a test room simulating as an office with the dimensions 7.2x5.4x2.5m(L x W x H). A cooled ceiling (70% coverage) and the DV system were installed in the room (see Fig.1). The room was situated inside a big warehouse (without window) and was well insulated (with 50mm glass wool plus 50mm Polyethylene form on wall, 100mm Polyethylene form on floor), so the influence from solar radiation and outdoor temperature variations was negligible.

To simulate the heat sources in the room, 'sitting person' simulator was used, which made of a circle duct with a line type electrical heater tied around the inside (60w/body) to ensure even surface temperature.

The surface square of the person simulator is

Tab. 3 Experimental conditions (Exp.2)

Case No	DV air flow rate [CMH]	Temp. gradient [deg.C/m]	Air conditioning Mode	Case No	DV air flow rate [CMH]	Temp. gradient [deg.C/m]	Air conditioning Mode
2-1	600	0.2	Mixing	2-7	800	1.6	CC/DV
2-2	600	2.2	CC/DV	2-8	800	2.6	CC/DV
2-3	600	2.9	CC/DV	2-9	800	3.4	CC/DV
2-4	600	3.3	CC/DV	2-10	800	3.6	CC/DV
2-5	600	4.0	CC/DV	2-11	800	4.2	CC/DV
2-6	600	4.2	CC/DV				

Tab. 4 Experimental conditions (Exp. 3)

Case No	Air temp. [℃]	Air humidity [%]	Ceiling surface temp. [deg.C]	DV air flow rate [CMH]	DV supply air temp. [deg.C]	Temp. gradient [deg.C/m]	Mode
3-1	27.8	46.9	27.1	645	19.2	4.4	DV only
3-2	26.2	47.1	24.0	739	18.9	3.8	CC/DV
3-3	26.4	41.2	25.8	741	19.8	3.3	DV only
3-4	25.7	42.2	22.1	575	20.5	2.4	CC/DV
3-5	25.9	49.2	22.1	680	21.7	2.2	CC/DV
3-6	24.9	44.3	20.1	393	21.9	1.2	CC/DV

Tab. 5 Category scales for subjective experiment

A. Comfort sensation	B. Sensation of vertical air temp. difference
- 3. Very uncomfortable - 2. Uncomfortable - 1. Slightly comfortable 0. Comfortable	- 2. Felt - 1. Slightly felt 0. no

about 1.56m^2 (representing the average value of Japanese person), and the outside was painted a light color in order for the emission coefficient to approximate that of a person. The lighting (ceiling-suspended fluorescent lamps: 18w/m^2) and office equipment (personal computers, etc.) also were set as normal office, but some bulbs were used for adjustment of the inner thermal load conditions.

5 EXPERIMENT METHOD

The experiments consisted of three parts: i.e. 1) measurement of thermal environment (Exp.1), 2) measurement of local ventilation efficiency (Exp.2), and 3) thermal comfort evaluation experiments involving subjects (Exp.3). The main items of measurement are shown in Tab.1. The experimental conditions are shown in Tab.2, Tab.3, Tab.4 respectively.

In Exp.1, the supply air temperature was fixed at 22 deg.C. The ventilation airflow rate and thermal load, surface temperature of cooled ceiling panels were adjusted as the experimental conditions. The cooling output of cooled ceilings was calculated from radiative and convective heat transfer at the cooled ceiling panel. The DV cooling output was calculated by the heat balance between supply air and exhaust air (latent heat load was insignificant).

In Exp.2, ventilation efficiency was measured by the step-down method using SF6 tracer gases. The measurement positions are shown in Fig. 1. Some cases were carried out to verify the difference of local air exchange efficiency between inside the plume and outside the plume, which situated in a person simulator.

In Exp.3, the subjects, totaled 14 college

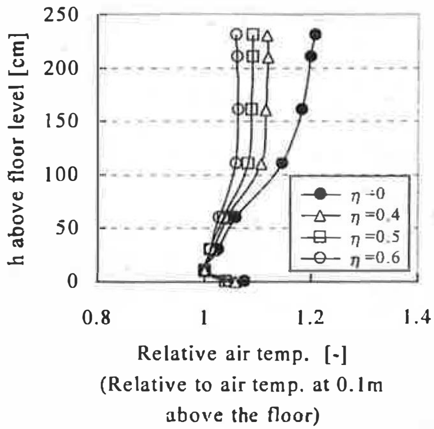


Fig. 2 Vertical air temperature distribution by CC cooling output ratio η

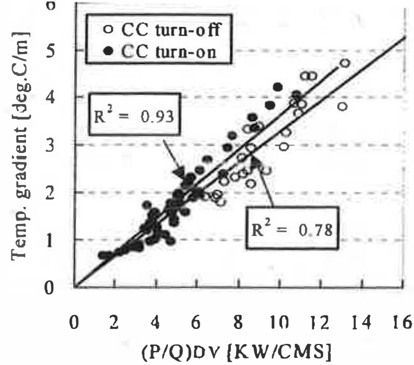


Fig. 3 Temperature gradient by $(P/Q)_{DV}$ when CC is turned on/turned off

students (male 13 and female 1) enrolled in the subjective experiment during the summer season. Every time,8 subjects were exposed to the thermal environment under the CC/DV system in the room for 60min, with clothing insulation of about 0.6 clo., and light sedentary activities. The category scales are shown in Tab.5. The votes of thermal comfort sensation and vertical temperature difference sensation were obtained at every 10 minutes during test period.

6 RESULTS and DISCUSSION

The main experiment results are described below.

6-1 The effects of CC Cooling Output Ratio η on the temperature gradient

In order to verify the effects of the cooled ceiling on temperature gradient, the cooling output ratio of the CC to total cooling capacity (η=Pcc/P) was taken as a parameter. Fig.2 shows the vertical air temperature profile with η. From the typical air temperature distribution of an office shown in Fig.2, we can see that the major air temperature difference is in occupied zone.

Fig.2 also shows that the temperature difference in upper part of the room become small significantly when CC is turned on. The temperature gradient in the occupied zone also decrease as η increase. It should be noted that the height of the stratification border decreased, but then maintained the same height, i.e. it remained to close to the breath zone level (about 1.0m above the floor) when CC was used with DV.
However, according to measurement of velocity distribution within the experimental conditions, there was no downward current.
In common sense, we know that when only DV was used, the temperature gradient is a function of the temperature difference between the supply air and exhaust air (te-ts). Due to practical considerations, we use the ratio of cooling load moved by DV to air flow rate $(P/Q)_{DV}$ as a parameter instead of (te-ts), and their correlation is shown in Fig.3. The temperature gradient in the occupied zone has a good correlation with $(P/Q)_{DV}$.
When the ceiling cooling has turned on, the temperature gradient is slightly steeper compared with that of cases using DV only. This is because, the temperature rise in the upper part of the room is restricted by the effect of cool radiation from the cooled

Tab. 6 Local air exchange efficiency εp

Location/Case-No	3-1	3-2	3-3	3-4	3-5	3-6	3-7	3-8	3-9	3-10	3-11
FL+ 10cm	120	166	223	233	270	226	167	203	215	259	234
FL+ 60cm	117	—	—	198	198	203	177	183	169	—	—
FL+100cm(B: inside the plume)	117	131	160	164	—	184	240	179	143	167	193
FL+100cm(A: outside the plume)	117	—	—	184	176	178	155	153	152	—	—
FL+110cm	114	117	103	105	110	128	123	120	126	110	141
FL+160cm	—	116	112	—	111	—	—	—	—	118	119
exhaust	100	100	100	100	100	100	100	100	100	100	100

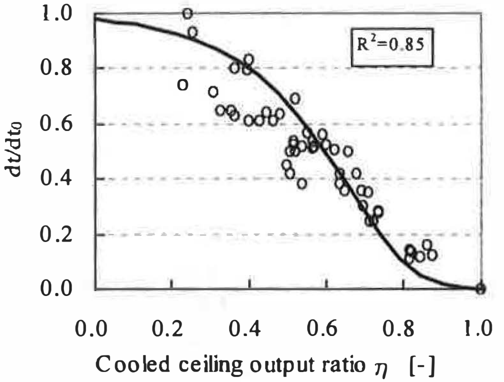


Fig.4 Dimensionless temperature gradient with CC output ratio η

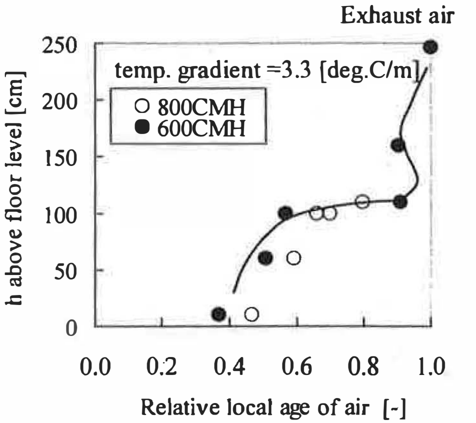


Fig.5 Distribution of Relative local age of Air(Relative to exhaust air)

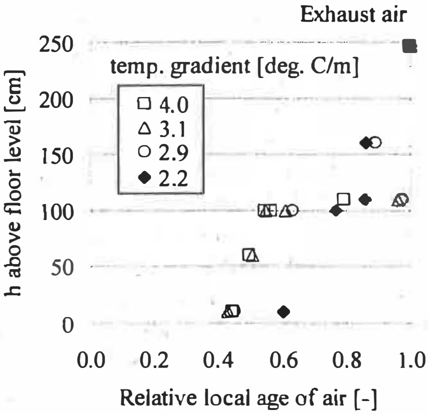


Fig. 6 Distribution of Relative local age of air (air flow rate=600CMH)

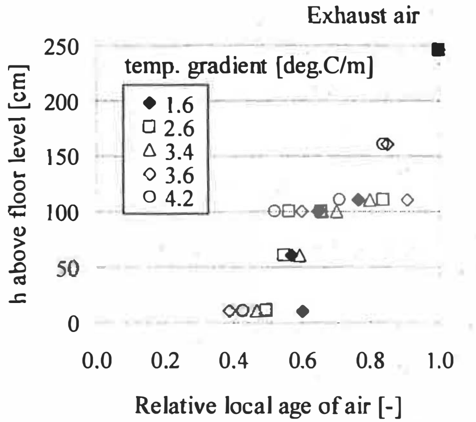


Fig. 7 Distribution of Relative local age of air (air flow rate=800CMH)

ceilings when cooled ceiling is used (as shown in Fig.2). As a result, the temperature differences in the occupied zone become a relatively greater portion of the overall temperature difference between supply and exhaust air.

To verify the effect of the CC cooling output ratio η, we used a dimensionless temperature (dt/dt0) where dt0 is the temperature gradient when only DV was used. As shown in Fig.4, the dimensionless temperature gradient (dt/dt_0) rises as a function of η. It indicates the effects of h on how much the temperature gradient decrease with the increase of η compared with the case of using only DV.

6-2 The effects of temperature gradients on Local air change efficiency

The local age of air (τp) and local air exchange efficiency (εp) were used to evaluate the local ventilation efficiency of the CC/DV system in this study. Based on the experiment results, the thermal stratification layer was formed except for in the case of mixed ventilation (case 2-1).
The distribution of the relative local age of air (refer to exhaust air) indicates that the clean zone and polluted zone are formed (see Fig.5). Although the ventilation airflow rates are different, the borders of stratification of these cases are kept at the same level (about 1.0m above the floor). The same tendency is observed for different temperature gradients, as shown in Fig.6 and Fig.7.

Fig.6 and Fig.7 present the effect of the temperature gradient on the vertical distribution of the relative local age of air. The later decrease (meaning higher local ventilation efficiency) with an increase in temperature gradient; however there is no significant difference when the temperature gradient over 3 deg. C. On the other hand, the relative local age of air is increased conspicuously when temperature gradient dropped down to 2.2 deg. C.
It should be pointed out that although thermal comfort is much improved by using the hybrid air-conditioning system in which the DV is combined with cooled ceiling, it may also kill the temperature gradient which has an important impact on ventilation efficiency. It means that the benefits of the DV system may be lost when the CC cooling output ratio (η) is too much.
Therefore, a certain level of temperature gradient (2℃ or above in this particular case) must be maintained to exploit the

efficiency.
The study also focused on the difference of local air exchange efficiency between the inside and outside of the plume. Measurements were carried out in some experimental cases. The local age of air was measured at point A (outside the person simulator plume) and point B (inside the person simulator plume). Both the points A and B situated at 1m above the floor (see Fig.1), i.e. within the breath zone. The results are shown in Fig.8 and Table 6. A difference appeared when the temperature gradient is lower (1.6 deg.C and 2.6 deg.C): the local age of air decreased with a decrease in temperature gradient. However the difference is insignificant when temperature gradient is greater than 3.0 deg.C. It has considered being due to the balance of stratification between the plume and surroundings. It means that the advantages to the local ventilation efficiency in the plume only occur when the buoyancy of the surroundings due to the thermal load is not stronger.
Another reason was assumed to that the height of the measurement points was close to the border between the clean zone and polluted zone, so the distribution of concentration was unsteady. More detail measurement is needed

6-3 The effects of the temperature gradient on thermal comfort

Each mean vote, based on responses from every subject group (8 subjects) at the end of the test period, is shown in Fig.9 as a function of temperature gradient. It is noted that the votes of sensation concerning vertical difference temperature rise to about 3 deg. C/m of temperature gradient, followed by a sharp fall. Also, as shown in Fig.9, the votes of thermal comfort sensation decreased gradually up to about 2.2 deg. C, above which it abruptly.
It suggests that although sensation concerning vertical temperature difference is sensitive to a range of temperature gradient

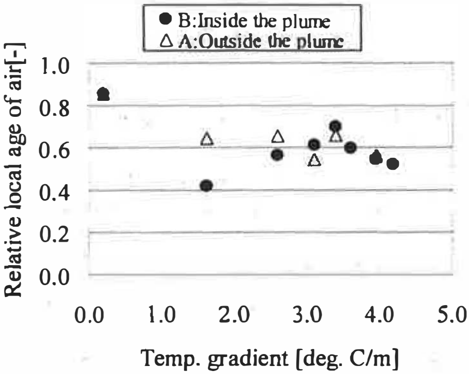


Fig.8 Difference of relative local age of air between inside and outside the plume (h=1m above the floor)

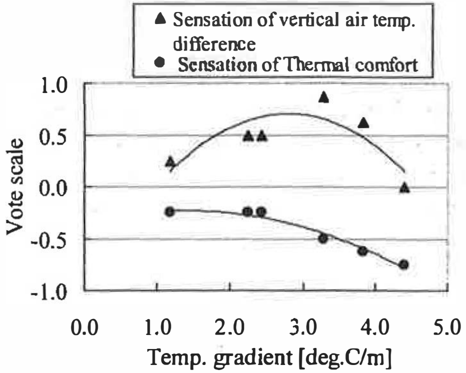


Fig.9 Thermal sensation by temperature gradient

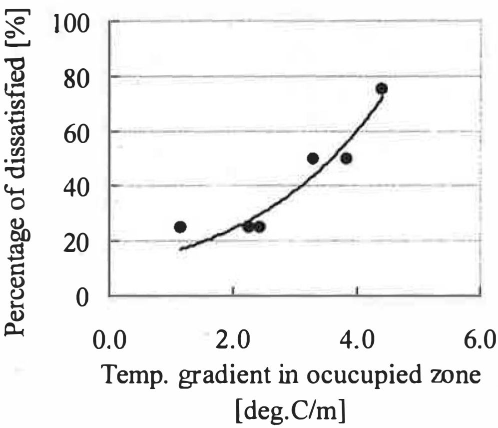


Fig.10 Percentage of dissatisfied by temperature gradient

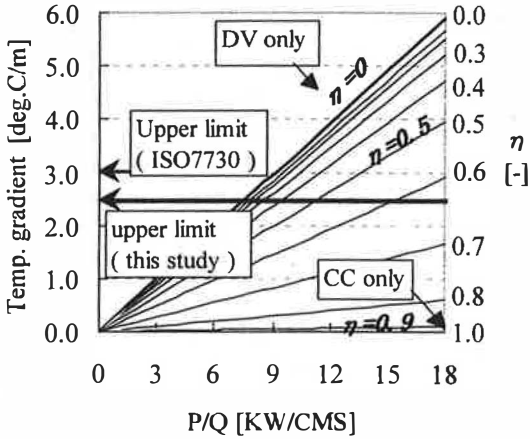


Fig.11 Design diagram for the CC/DV air conditioning system

(up to about 3 deg. C/m based on this study), causes discomfort abrupt over 2.2deg.C/m. As shown in Fig.10, the percentage of dissatisfied indicated the same result. It can be concluded that the acceptable upper limit of the temperature gradient should be lower than 2 .5 deg.C/m for a sedentary occupant, and this is within the comfort criteria from ISO7730(<3.0 deg.C/m).

7 CONCLUSIONS

When using the DV independently in an ordinary office, the maximum load processing capacity is 9 [kW/CMS] according to ISO 7730-1994, "standard on comfort " (vertical temperature difference in occupied zone < 3 deg. C). This is only sufficient for a cooling load density of about 15 W/m2 when the minimum DV airflow quantity is supplied (only the minimum outdoor-air quantity is insured). The cooled ceiling/DV system can maintain the benefits of both thermal comfort and ventilation efficiency when it applies to the higher thermal load space, but the temperature gradient must be within a range of 2.0 to -2.5 deg. C. Based on the results of the above experiments, we determined the relationship among P/Q, dt and η, and developed a design diagram for the CC/DV hybrid air

Fig.11, the cooling capacity of the CC/DV system is within a range of 7.5-18 [KW/CMS] when at the temperature gradient upper line of 2.5 deg. C/m, with the cooled ceiling output ratio within a range of 0- 65%.

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Ventilation efficiency index :

Local age of air : (by step - down method)

$$\tau_p = \int_0^\infty \frac{c_p(t)}{c_p(0)} dt$$

Local air exchange efficiency :

$$\varepsilon_p = \frac{\tau_n}{\tau_p}$$

$c_p(t)$: Concentration contamination at point p at time t.

τ_n : Nominal time constant ($= \frac{V}{Q}$).

but it is calculated based on the meaurement results in thi study.

(= local age of air at exhaust)

Symbols :

P : Total room cooling load. [KW]

P_{cc} : Thermal load removed by cooled ceilings. [KW]

Q : Ventilation airflow rate. [CMS]

$(P/Q)_{DV}$: The ratio of P to Q , where P is the Thermal load removed by DV. [KW/CMS]

dt: : Temperature gradient (between 1.1m and 0.1m above the floor). [deg.C/m]

dt_o : Temperature gradient in case of useing DV only. [deg.C/m]

η : Ratio of cooling output of Cooled ceiling to total of that. ($= \frac{P_{cc}}{P}$)